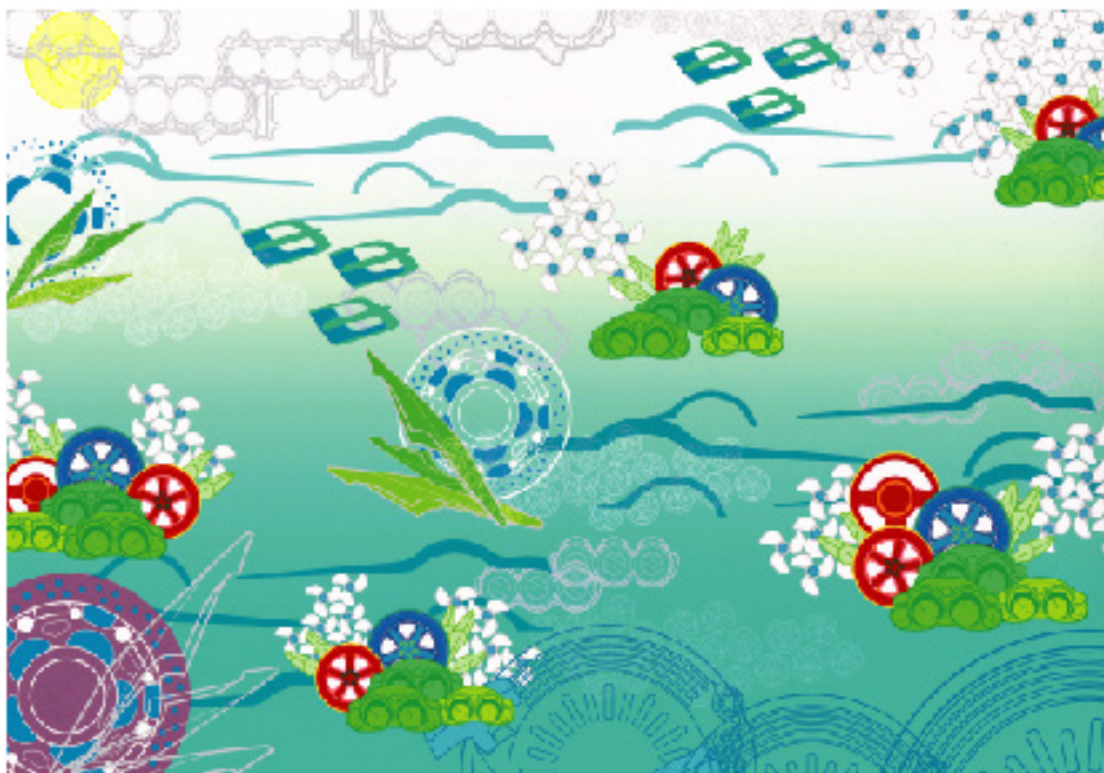




Honda R&D

Technical Review

October 2005



Comment on the Cover

In designing this cover, I wanted to convey the sense of just working from inspiration, as though I let my brain conceive whatever it would and let my hands move however they would.

The scene is filled with forms that Honda has created, like creatures with their own personalities.



Asakahigashi
R&D Center
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Development of Integrated Fuel Injection System for Small Sized Motorcycles

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ABSTRACT

Demands for reducing exhaust emission and enhancing fuel economy have been increasing in accordance with growing concerns regarding global environmental issues. An electronically controlled fuel injection system which precisely controls the air-fuel ratio has been adopted to effectively meet the demands. Technologies for miniaturizing and achieving a high-efficiency system have been established for fuel injectors, the fuel pump and ECU (Electrical Control Unit). For the fuel injector, a spray form and magnetic circuit was optimized through computer simulation. As the result, a fuel injector with good installation ability for engine displacements ranging from 50 cm³ to 900 cm³ was realized. For the fuel pump, the maximized motor efficiency along with an optimized pump rotor shape reduced current consumption by 25%. For the ECU, the development of a circuit-integrated custom IC and the standardization of a housing achieved a compact, light, inexpensive design.

1. Introduction

Following the lead of industrialized countries, governments in the ASEAN region are implementing regulations calling for enhanced motorcycle fuel efficiency and reduced emissions. Since most of the world's motorcycles are used in the ASEAN region as a vital form of transportation, it is only natural that these motorcycles have a significant impact on global environmental issues. Electronically controlled fuel injection system (hereafter fuel injection) is a very effective means of addressing environmental concerns without compromising the performance and function of the motorcycle. Honda adopted fuel injection technology⁽¹⁾ for small-sized motorcycles beginning with the 2003 Pantheon (for

the Italian market), followed by the Wave 125i (Thailand) and Dio Z4 (Japan)⁽²⁾. To expand the use of fuel injection in small-displacement motorcycles, it is essential to make a system smaller and more efficient. This will help reduce power consumption and cost. This report introduces technological measures that were taken to make the fuel injection system smaller, lighter and less expensive.

2. Issues in Adopting and Techniques for Achieving Fuel Injection

2.1. Issues in Adopting Fuel Injection

Figure 1 shows the fuel injection system of the 2003 Wave 125i. This motorcycle was launched in Thailand in July of 2003.

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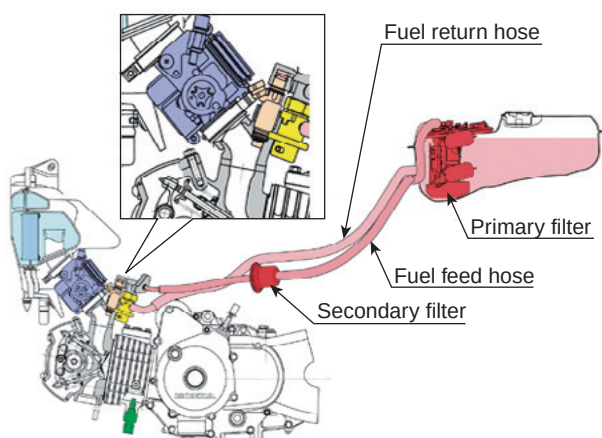


Fig. 1 Fuel supply system of 2003 Wave125i

The fuel injection system has new technologies such as the integration of the ECU with the functions of the throttle body. The system was based on automobile technology but was not optimized for small-sized motorcycles. Therefore it's unsatisfactory in terms of the layout and cost.

2.2. Techniques for Achieving Fuel Injection

The current development focuses on reducing size, weight and cost for the following four items.

- (1) A compact fuel injector^{(3), (4)}
- (2) A highly efficient fuel pump and a fuel feed pump module without a secondary filter⁽⁵⁾
- (3) A small integrated ECU (electrical control unit)⁽⁶⁾
- (4) High pressure fuel hose using small-diameter resin tubes

3. Major Specifications and Outline

Table 1 shows the major specifications of the recently developed 2005 Wave 125i compared to those of the earlier model. Figure 2 and 3 show the fuel supply system and the appearance of the motorcycle, respectively. Figure 4 shows a fuel injection control

diagram. For the fuel supply system, fuel vapor lock was prevented by revising the fuel injector position and changing the fuel supply pressure. This allowed for the elimination of the fuel return hose.

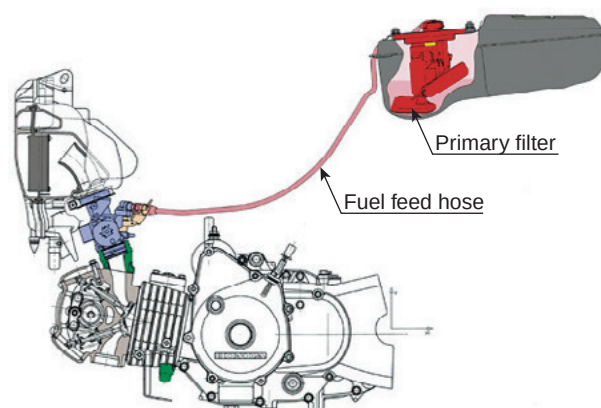


Fig. 2 Fuel supply system of 2005 Wave125i



Fig. 3 2005 Wave125i

Table 1 Major specifications

		2003 Wave125i	2005 Wave125i
Engine	Engine type	Air-cooled 4-stroke SOHC	Air-cooled 4-stroke SOHC
	Bore x Stroke	mm	52.4 x 57.8
	Displacement	cm ³	124.9
	Compression ratio	9.3	9.3
	Throttle body	Keihin GQ20	Keihin GQ22
	Max. output	kW/ r/min	6.75 / 7 500
	Max. torque	Nm/ r/min	9.74 / 5 500
Fuel system	Fuel supply system	In-tank return	In-tank return-less
	Fuel pump	Keihin	Keihin (Carbon commitator)
	Fuel pressure	kPa	250
	Fuel feed hose	FKM (φ6)	Resin (φ2.5)
	Injector	Denso C-3	Keihin type 7
Electric	ECU (Electrical Control Unit)	Keihin (Integrated to throttle body)	Keihin
	Throttle position sensor	Alps	Alps
	Manifold pressure sensor	Fuji	Fuji
	Air Temperature sensor	Shibaura	Shibaura
	Engine temperature sensor	Denso	Matsushita
	Ignition coil	MP06	MP33

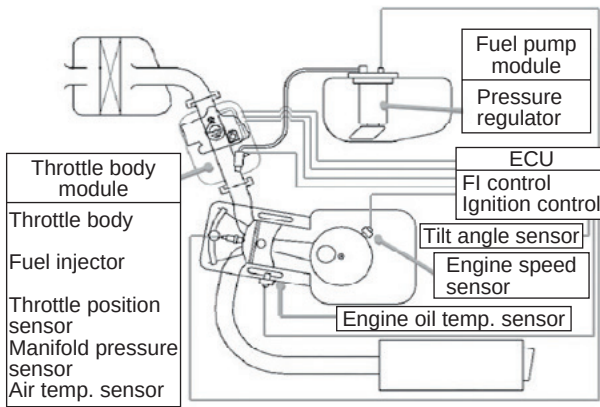


Fig. 4 Schematic of fuel injection system

4. Main Technologies

4.1. Compact Fuel Injector

The current fuel injectors for automobiles are rather large and therefore offer little flexibility for installation on small-sized motorcycles. Achieving engine performance has been a challenge with these systems. This research developed a new compact fuel injector with flexibility for installation on small-sized motorcycle engines^{(3), (4)}. Figure 5 compares its size to that of a current fuel injector. Table 2 provides their performance specifications.

In order to make the fuel injector smaller while keeping its basic performance, the construction of the fuel injector was fundamentally revised based on the simulation results of the magnetic circuits and fuel flow.

- (1) Simplification of the movable valve stopper structure
- (2) Optimization of the magnetic circuit configuration
- (3) Reduction of pressure loss in fuel passage

4.1.1. Simplification of the movable valve stopper structure

Conventionally, two types of movable valve stopper structures are common: As shown in Fig. 6(a), the first uses a non-magnetic parted C-shaped washer to create an air gap between the movable valve and the fixed iron core. The second type, shown in Fig. 6(b), uses a hard chromium plating on the movable valve to create an air gap.

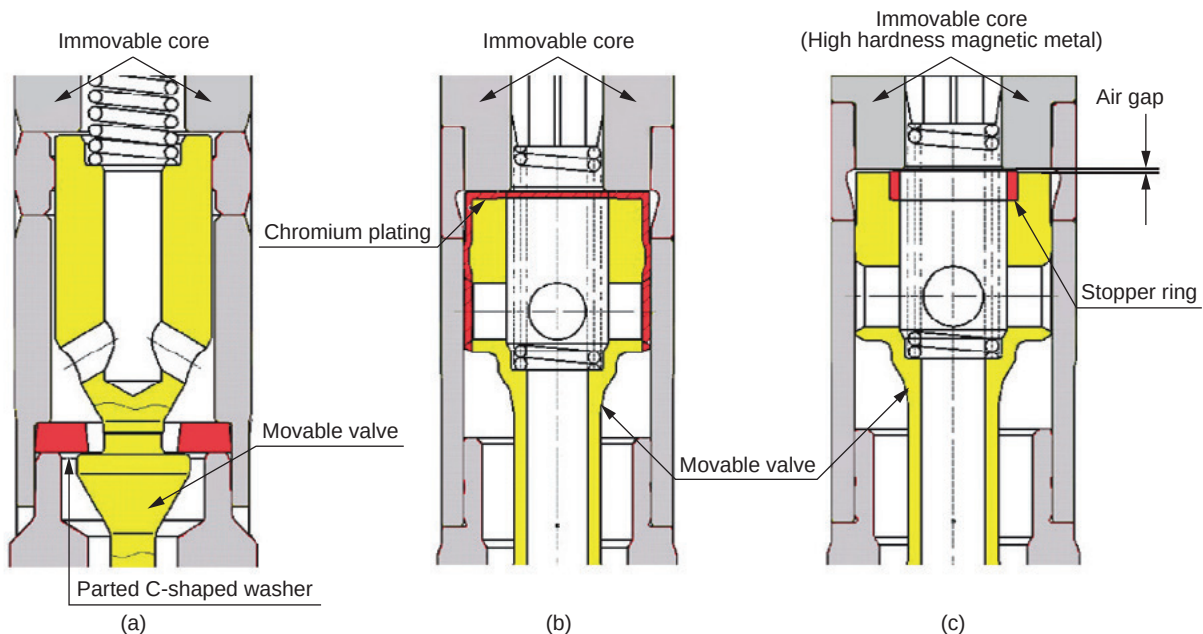


Fig. 6 Simplification of plunger stopper system

Both types are expensive. The type with the parted C-shaped washer consists of many components while the type with the hard chromium plating on the movable valve requires more surface treatment processing.

For this research a new construction was used for the movable valve stopper structure (c).

First, an air gap was formed between the movable valve and fixed iron core. A non-magnetic stopper ring, which acts as the movable valve's stroke stopper, was press-fitted into the movable valve.

In order to reduce the abrasion, a new magnetic material that is harder than the surface hardness of the stopper ring was used for the fixed iron core that touches the stopper ring.

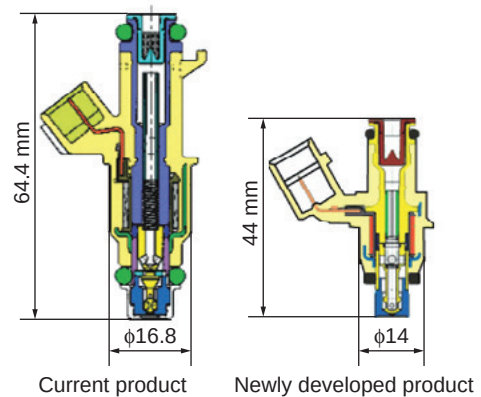


Fig. 5 Comparison of injector size

Table 2 Specifications of injector

	2003 Model	2005 Model
Number of nozzle	4	4
Dynamic flow rate (2 msec)	1.73 mm ³ /st	2.12 mm ³ /st
Static flow rate	90 cm ³ /min	78.8 cm ³ /min
Spray form	1 Jet 12 deg	1 Jet 15 deg
S. M. D	120 μm	72 μm

This resulted in a construction with unprecedented economy and simplicity.

Moreover, its durability is greater than that of the existing C-shaped washer type.

4.1.2. Optimization of the magnetic circuit configuration

An optimally compact design was created for the magnetic circuit through the use of CAE analysis.

The new design balances coil inductance reduction, surplus solenoid attraction force reduction, optimal position settings for the movable valve, restraint of the overall coil bobbin length, etc. to achieve a smaller design without compromising performance.

For example, CAE analysis indicated that the gap between the outside diameter of the fixed iron core side and that of the movable iron core side was too large. As shown in Fig. 7, this resulted in increased flux leakage. Therefore, the outside diameter gap was reduced to enhance efficiency.

4.1.3. Reduction of pressure loss in fuel passage

The configuration of the fuel channel was also revised in order to assure a fuel flow rate applicable to a range of displacements, from small to large, with the more compact design. Specifically, a center fuel channel with low channel resistance was used. The cross-sectional area of each part was designed to be more than three

times the opening area of the seat part when flow was at its maximum. This resulted in a great reduction of pressure loss as compared to existing fuel injectors, as Fig. 8 shows.

4.2. Fuel Feed Pump Module

Motorcycles in the ASEAN region, where there is such a strong need for enhanced fuel efficiency and lower costs, require further reductions in current consumption. Therefore, with a goal of achieving current consumption of less than 1A, a new low-cost fuel pump module was developed⁽⁵⁾.

By focusing on the low-flow range required for small-sized motorcycles, the development of the fuel pump module aimed at achieving the following four requirements:

- (1) Rotor blade angle optimization
- (2) Pump passage gauge optimization
- (3) Pump motor optimization
- (4) Elimination of secondary filter

Figure 9 and Table 3 compare the size and properties of the new and current pumps.

The new fuel pump (c) is about the same size as (b) but has greatly enhanced performance. The new fuel pump was able to satisfy flow requirements while reducing current consumption to the target level of less than 1A.

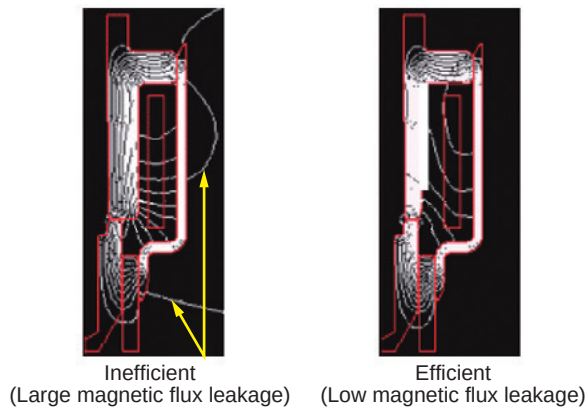


Fig. 7 Optimization of magnetic circuit

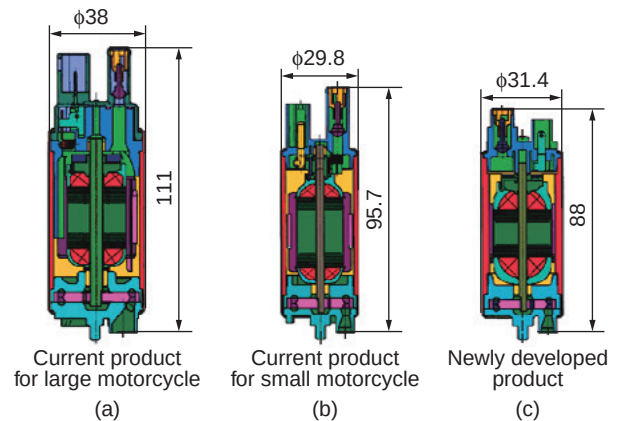


Fig. 9 Comparison of fuel feed pump size

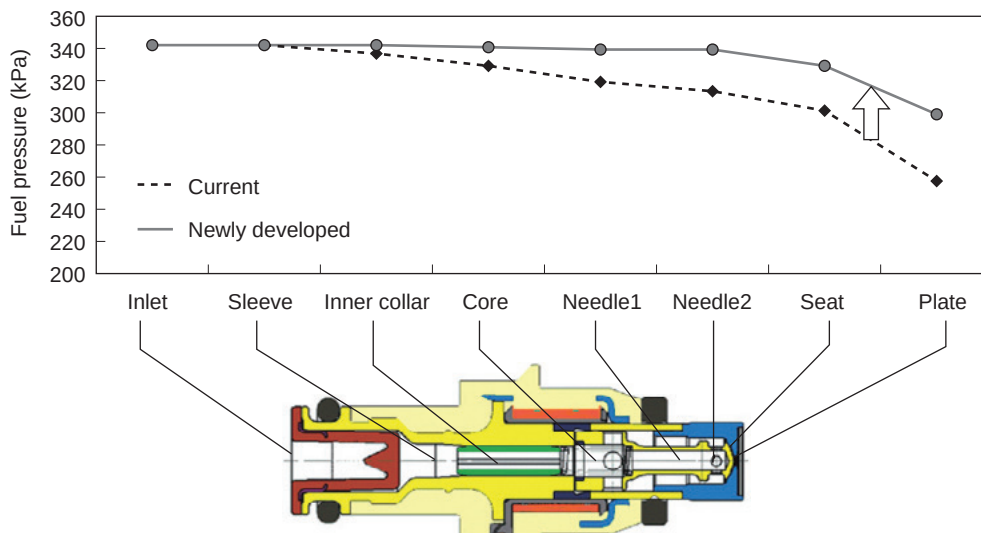


Fig. 8 Pressure loss at various locations

Table 3 Specifications of fuel pump

	2003 Model	2005 Model
Fuel pressure	250 kPa	294 kPa
Flow rate	17.7 L/h	16.5 L/h
Current	1.0 A	0.95 A
Pump type	Turbine	Turbine
Motor type	DC motor	DC motor
Magnet type	Ferrite	Neodymium

4.2.1. Rotor blade angle optimization

Enhancing pump efficiency requires increasing the flow velocity of a fluid's swirl flow.

Setting the rotor blade angle is very important in achieving increased velocity.

Rotors with various blade angles were built and their efficiencies measured, as shown in Fig. 10.

The result was that reducing the rotor blade angle from 60° (the angle of the existing type) to 45° increased pump efficiency by 3%.

4.2.2. Pump passage gauge optimization

The passage area was optimized for the set flow. In the low displacement range, pump efficiency is enhanced because bulkhead resistance declines as a result of reducing the passage cross-sectional area. This is shown in Fig. 11. If the cross-sectional area is reduced too much, the discharge flow rate itself declines, reducing pump efficiency.

The new pump uses a passage cross-sectional area of 11.1 mm² as compared to the 14.0 mm² of the existing pump. The result is that pump efficiency at low flows was increased by 3.5%.

4.2.3. Enhancing pump motor efficiency

To enhance the efficiency of the motor, a Nd-Fe-B bond magnet was used for the magnetic material in place of the ferrite of existing motors. As shown in Fig. 12, residual flux density rose 78% and motor efficiency increased by 7.6% over that of the existing motor.

4.2.4. Elimination of secondary filter

Conventionally, it is typical in fuel pump modules to place a primary filter on the fuel pump's intake side and a secondary filter on the high pressure side. With the new pump module, a high density nonwoven fabric filter was used for the primary filter and a carbon

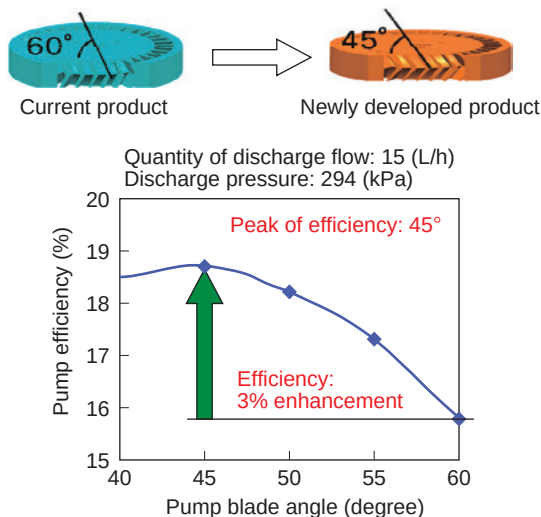


Fig. 10 Optimization of rotor blade angle

commutator was used in the pump motor. This made it possible to eliminate the secondary filter at the high pressure end. Table 4 lists the specifications of the primary filter.

4.3. Small Integrated ECU

Among small-sized motorcycles, the devices required vary depending on model, destination market and other factors. Table 5 shows the major combinations.

Since the combination of Engine Management System (EMS) devices (auxiliaries) varies according to model, the ECU's sensors, switch input circuit and device drive circuit all must be general-purpose and multi-functional. An ECU was developed through this research to accommodate the mutually opposing goals of smaller size, weight and cost with that of advanced and general purpose functionality⁽⁶⁾.

4.3.1. Development of circuit-integrated custom IC

Figure 13 shows the ECU's circuit configuration. In the fuel injection system, functions were built into the custom IC taking

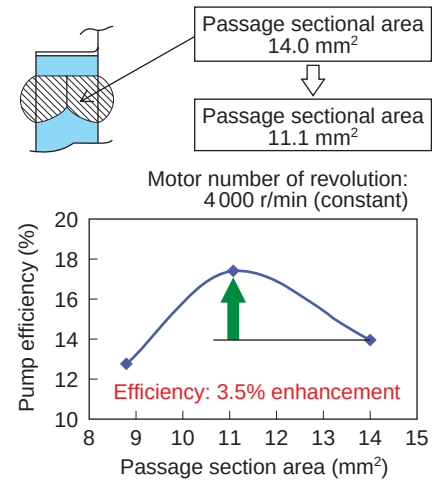


Fig. 11 Optimization of fuel passage design

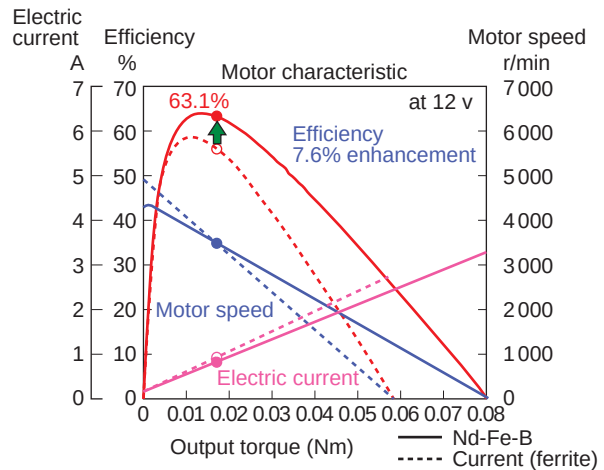


Fig. 12 Developed motor characteristic

Table 4 Specifications of fuel filter

	2003 Model	2005 Model
Filtration area	25 cm ²	65 cm ²
Filtration density	30 μm 90% min	10 μm 95% min
Max. diameter of pass	35 μm	10 μm
Life performance (Pressure loss by 3 grams in dust)	1.2 kPa	0.6 kPa

Table 5 Demands and applications for small motorcycles' ECU

Sensor/Device	Wave 125i (Thailand)	Scooter (Europe)
Throttle position sensor	Yes	Yes
Manifold pressure sensor	Yes	Yes
Air temperature sensor	Yes	Yes
Engine temperature sensor	Yes	Yes
Oxygen sensor	—	Yes
Crank pulse sensor	Yes	Yes
Speed sensor	—	Yes
Side-stand switch	—	Yes
Neutral switch	Yes	Yes
Injector	Yes	Yes
Ignition coil	Yes	Yes
Fuel pump relay	Yes	Yes
Power relay control	—	Yes
Oxygen sensor heater	—	Yes
Exhaust air induction	—	Yes
Positive crankcase ventilation	—	Yes
Fan control	—	Yes
Idle air control valve	—	Yes

into account the combination of sensor devices and added functions that would be needed to address strict exhaust emission regulations. The functions bundled in the custom IC were the basic input and output needed by the fuel injection system, a stepping motor drive circuit, the ECU's internal 5V power source and microcomputer monitoring/reset circuit.

4.3.2. Housing standardization and mounting design

The space in which the ECU can be mounted is limited on a motorcycle for structural reasons. Additionally, since these are electronic components, the environmental conditions of the mounting location must be considered.

To address these concerns, small connectors and other components were employed first in order to easily configure the layout. Furthermore, the connectors were designed to be waterproof and the substrate was filled with resin. This resin filling also helped enhance the vibration resistance of the ECU.

(1) Electronic components

By using surface-mounted electronic components that allow for high-speed automatic mounting, manual labor and process cycle time for this task were reduced. Since lead-free solder is used to connect components, components were selected that have suitable electrode surface treatment and heat resistance properties (Fig. 14).

(2) Connectors

The terminal size was reduced from the current 040 to 025 while still achieving the carrying capacity requirement. This allowed for smaller connectors. Additionally, the technique for affixing the housing to the substrate changed from a screw on type to that of a screwless snap-fit design, thus eliminating the screw-down process.

(3) Substrate

The substrate must have enough area so electronic components, having a load drive current of 1A or greater (such as the solenoid and motor), can release heat. Therefore an ordinary FR-4, through-via 4-layer laminates was used.

4.4. Small-diameter Resin Fuel Hose

To reduce costs, high pressure resin fuel hoses were used in place of rubber.

Existing resin hoses, however, are rigid and cannot bend freely. It was necessary to perform a form molding and bending process in advance for the desired form. Accomplishing this in the complicated form of a motorcycle frame has been a challenge. Moreover, existing hoses were intended for automobiles and their large flows are not appropriate for the low flows required by small-sized motorcycles.

For this research a small-diameter resin hose suitable for low flows was developed that makes it possible to arrange tubing in the complicated forms of motorcycles, without any molding and bending process involved. Detailed specifications are shown in Fig. 15.

4.4.1. Construction

Like existing resin hoses, the new small-diameter resin hose has a three-layer construction, consisting, from the inner layer, of ETFE, PA12 and EPDM. Figure 16 illustrates this construction. The innermost layer of ETFE and the outermost EPDM are protective materials that respectively withstand deteriorated gasoline and provide weather-fastness.

4.4.2. Anti-kinking

CAE-based analysis was performed to judge the flexibility of the small-diameter resin hose. As Fig. 17 shows, while existing resin hoses kink at a bending curve of R25, the new small-diameter hose does not kink even at R8. It was confirmed that the new hose has as much flexibility as existing rubber hoses.

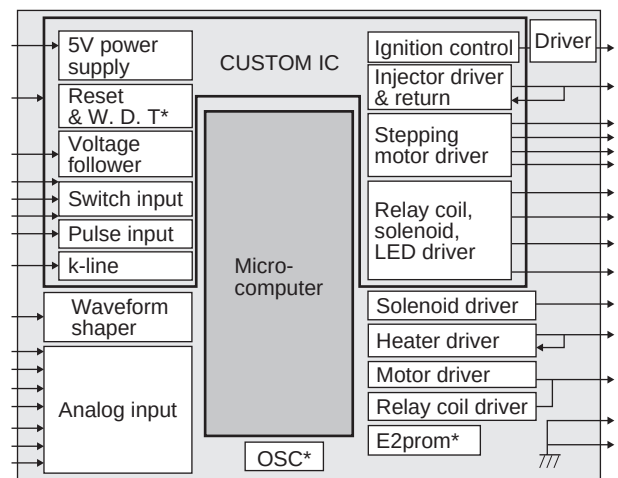


Fig. 13 ECU circuit

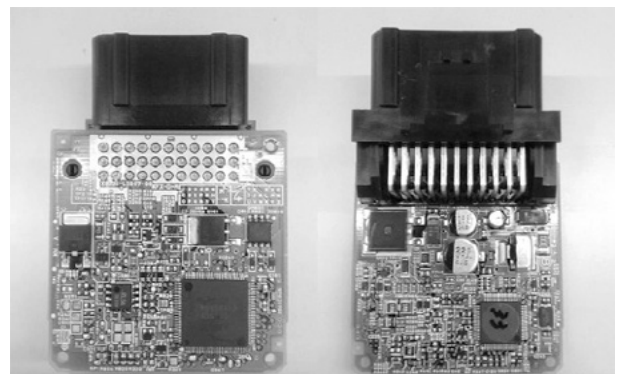


Fig. 14 ECU board for small motorcycle

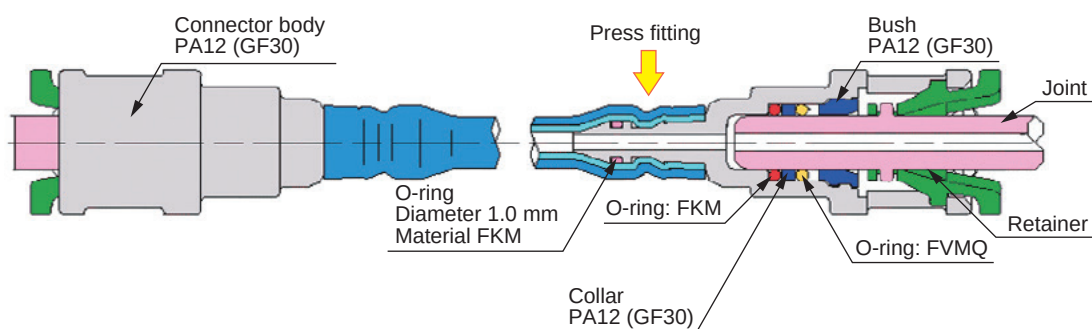


Fig. 15 Thin resin tube

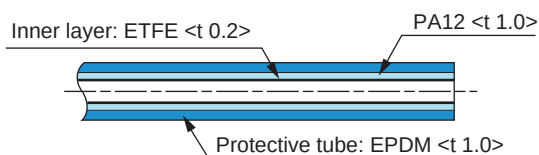


Fig. 16 Thin resin tube

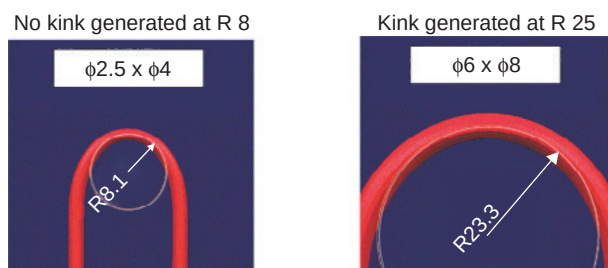


Fig. 17 Comparison of anti-kink

5. Conclusion

For this research, a small and highly efficient fuel injector, fuel pump, ECU and high pressure fuel hose were developed as the functional components of a small-sized motorcycle's fuel injection system. Using these components, a simple and inexpensive fuel injection system was built and used on the 2005 Wave 125i.

- (1) The fuel injector was among the world's smallest, but maintained as much performance as that of an ordinary fuel injector for automobiles.
- (2) By increasing overall efficiency by 25% over Honda's existing fuel pumps for small-sized motorcycles, the new fuel pump achieved a current consumption of less than 1A.
- (3) A custom IC was used for the ECU and the housing was standardized, making the ECU suitable for all small-sized motorcycles.
- (4) The development of a flexible new small-diameter resin hose made it possible to arrange tubing without molding it in the shape of the frame of the small-sized motorcycle.

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Next-generation Fuel Cell Stack for Honda FCX

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ABSTRACT

The next generation fuel cell stack employs an aromatic electrolyte membrane, which offers outstanding proton conductivity, a stamped metal separator formed with a unitized seal for superb electrical and thermal conductivity, and a simple stack structure which is obtained using the springiness of the stamped metal separators. The combination of these innovative new technologies enables the new fuel cell stack to achieve output density that ranks among the best in the world, to be half the size of its predecessor, and to operate in temperatures from -20 to $+95$ °C, leading to its enhanced environmental adaptability.

In order to maximize space efficiency in the fuel cell system box, the FCX uses two compact FC stacks. The resulting integrated module structure, with the humidifier unit, enables a more compact fuel cell system box to be achieved. Energy efficiency of the new fuel cell system has also increased over the previous model, due to the enhanced power generation efficiency of the fuel cell stack.

1. Introduction

To address energy and air pollution issues, Honda has been carrying out research and development on a clean power plant that discharges practically no CO_2 or toxic substances.

Since equipping a test vehicle with self-developed fuel cell stacks in September 1999, Honda has aimed to promote fuel cell-powered vehicles, and has worked on developing more compact fuel cell stacks with higher output so as to make them adaptable to wider areas^{(1), (2)}. The new-generation fuel cell stack was announced in October 2003, and a Honda FCX, the first electric car equipped with this fuel cell stack, was delivered to State of New York in November 2004.

This report introduces the lightweight and compact next-generation fuel cell stack which drastically reduces the number of parts used, produces high output that ranks among the best in the world, and starts even below freezing temperature.

2. Development Goals

Currently, the competition to develop proton exchange membrane fuel cells for transport vehicles, including automobiles, and for general household use has become fierce.

This type of fuel cell can be used at comparatively low temperatures in the vicinity of 80 °C, and is relatively compact. However, issues for automobile use such as the starting of the vehicle below freezing temperature, effective utilization of space, and energy efficiency still need to be addressed. The two main goals for the development of the next-generation fuel cell stack were identified as below.

- (1) Compact with high output (output density)
- (2) Environmental adaptability (performance at high temperatures/low temperatures)

* Wako Research Center

3. Fuel Cell Stack

As shown in Fig. 1, a proton exchange membrane fuel cell makes use of a polymer electrolyte membrane and directly converts chemical energy to electric energy using the electrochemical reactions between hydrogen and oxygen.

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

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

The polymer electrolyte membrane that conducts the protons is sandwiched between two electrode layers (hydrogen electrode/oxygen electrode), and this assembly in turn is sandwiched between two gas diffusion layers to form the Membrane and Electrode Assembly (MEA)⁽³⁾. The MEA is held between two separators on both sides to form a single cell. Since a voltage of only 1.23 V can be obtained from one cell theoretically, the cells are stacked as shown in Fig. 2, so that the stacking of cells one by one generates progressively higher voltages. The voltage is determined by the number of cells stacked, and the current is determined by the surface area of the electrodes. Accordingly, either the number of stacked cells has to be increased, or the surface area of the electrode has to be increased, to increase the power output.

Conventionally, fluorine electrolyte membranes were used for the stack. Because this type of membrane becomes soft at high temperatures, its use has been restricted to the range of 0 °C to 80 °C. Carbon separators have also been used in the past, which with their high electrical resistance and poor heat conduction became an obstacle in attaining high performance. A quick rise in temperature, from low temperatures, was also suppressed. Furthermore, carbon separators do not have very high strength. To use the separator as a structural member that firmly holds multiple cells within a stack from both sides, its thickness needed to be

increased. Thus, compactness of the stack could not be achieved easily.

High output of the stack was achieved by developing an aromatic electrolyte membrane with high proton conductivity, and both compactness and high output were achieved by developing a stamped metal separators having high electrical and thermal conductivity formed with a unitized seal. A simple stack structure was obtained using the springiness of the stamped metal separator formed with a unitized seal. This arrangement helped to achieve approximately a two-fold increase in compactness and high output compared to the conventional stack, as shown in Fig. 3, and enabled starts below freezing temperature. The evolution of the high power output, compactness, and lightness of the stack are shown in Fig. 4, Fig. 5 and Fig. 6, respectively.

3.1. Aromatic Electrolyte Membrane

An electrolyte membrane with a new molecular structure was developed, as shown in Fig. 7, by combining an ion exchange

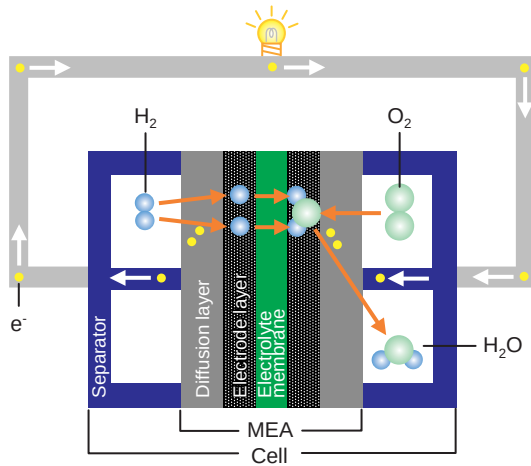


Fig. 1 Electrical generation in fuel cell

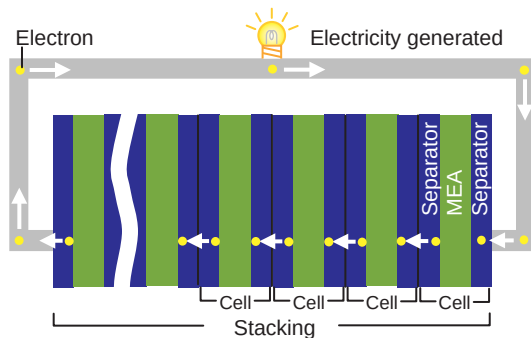


Fig. 2 Structure of fuel cell stack

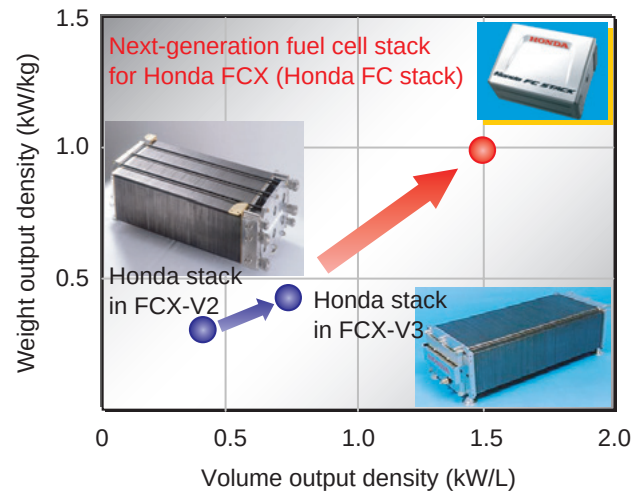


Fig. 3 Honda stack evolution

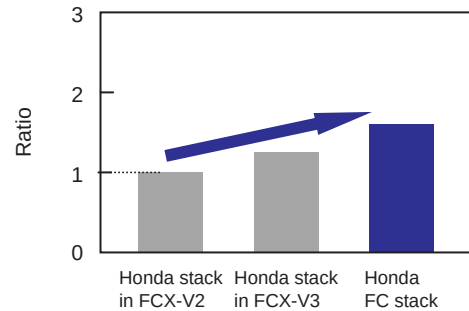


Fig. 4 Power output evolution

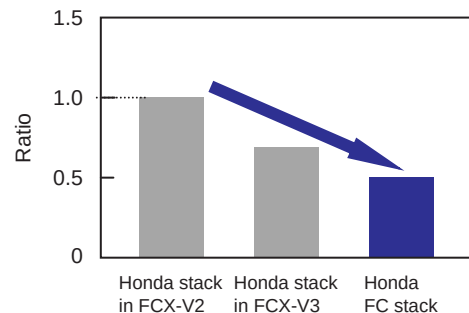


Fig. 5 Compactness evolution

substrate (sulfone substrate) with the main chains of an aromatic structure having high thermal stability. The ion exchange substrate concentration governing ion conductivity was increased, more than a comparable conventional fluorine electrolyte membrane. The results were that the membrane resistance was halved as shown in Fig. 8, and high proton conductivity was attained, even below freezing temperature. A fuel cell stack that could generate power at -20°C was therefore achieved.

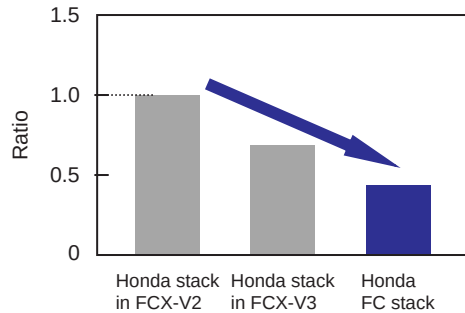


Fig. 6 Lightness evolution

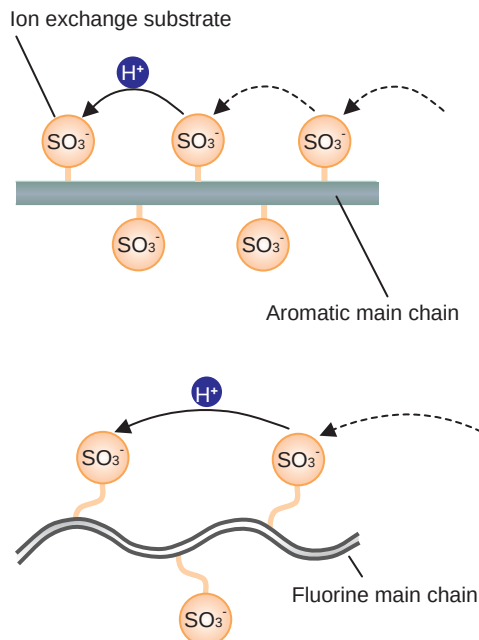


Fig. 7 Comparison of polymer structures

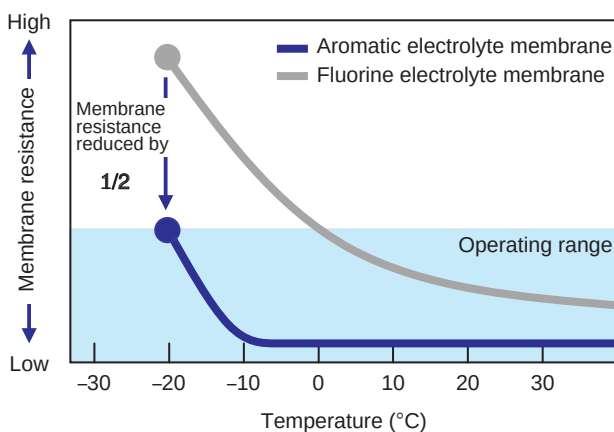


Fig. 8 Enhanced proton conductivity electrolyte membrane

The conventional fluorine electrolyte membrane becomes soft and deforms abruptly when the temperature rises above 80°C , therefore, it could not be used at temperatures above this level. The newly developed aromatic electrolyte membrane does not become soft or deform at high temperatures, due to its aromatic main chains having high strength and high thermal resistance. As shown in Fig. 9, the new membrane enabled high temperature use at 95°C , in addition to exhibiting high durability (Fig. 10). However, chinks occur between the electrode layer and the membrane in conventional membrane/electrode layer manufacturing, as shown in Fig. 11, even though the aromatic main chain shows good anti-deformity characteristics at high temperatures. An original manufacturing method was developed to address this issue, and superior adhesion has been realized between the electrolyte membrane and electrode layer.

3.2. Stamped Metal Separator Formed with Unitized Seal

The separator should have high electrical conductivity since it forms a pathway for the electrons. At the same time, thermal conductivity (temperature rise property) is also important,

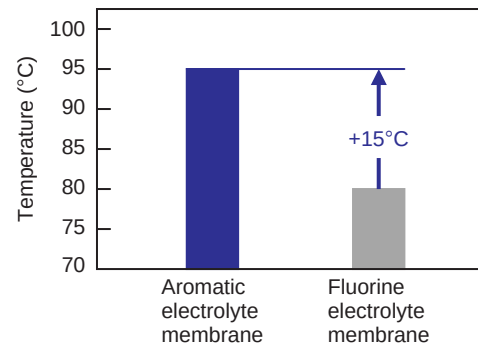


Fig. 9 Enhanced max operating temperature

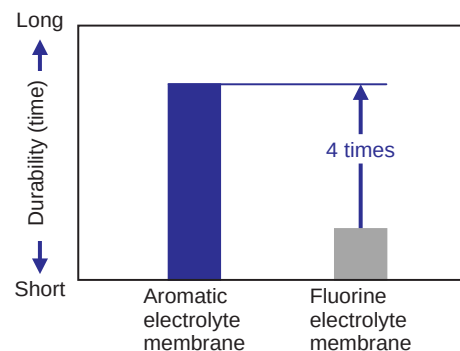


Fig. 10 Enhanced high-temperature durability (80i)

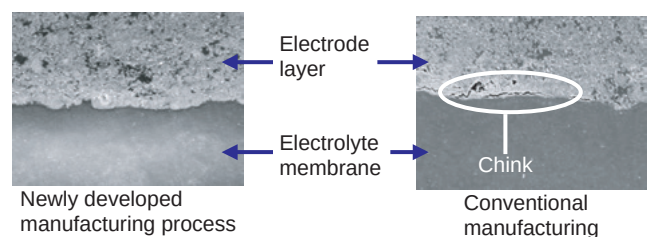


Fig. 11 Comparison of adhesion between aromatic electrolyte membrane and electrode layer

considering that the water produced, when power is generated below freezing temperature, should not freeze. Making use of the high electric and thermal conductivity of stainless steel, a stainless steel plate containing electrically conductive metallic inclusions was developed reducing the contact resistance to one-fourth the resistance of conventional carbon separators at -20°C (Fig. 12).

By using stainless steel, which has higher strength than carbon, a design with half the thickness of that of a carbon separator was possible. As a result, the thermal conductivity was enhanced five-fold (Fig. 13), enabling the entire stack to be warmed quickly and equally, curtailing the time required from the start of power generation to warm-up of the unit.

Metallic materials have been considered in techniques for enhancing both electrical and thermal conductivity. There are issues involved, however, mainly oxidation and corrosion because of the electric current. For instance, if passive treatment is used to prevent oxidation and corrosion, the resistance of the metallic surface increases and adequate performance cannot be obtained, which is why carbon separators were normally used in the past. To resolve this issue, conductive particles (electrically conductive metallic

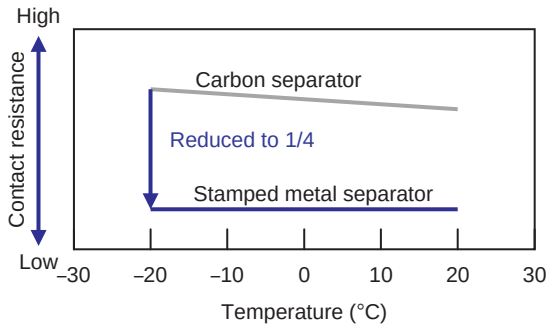


Fig. 12 Enhanced electrical conductivity of separator contact surface

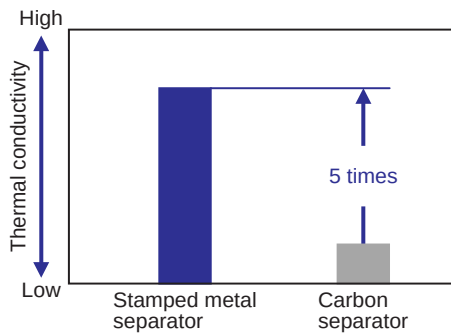


Fig. 13 Comparison of thermal conductivity

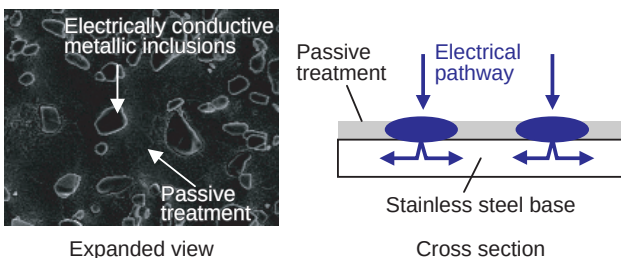


Fig. 14 Developed stainless steel

inclusions) that allow electricity to pass through easily were dispersed in the stainless steel base, as shown in Fig. 14, and the surface was subjected to passive treatment to achieve a good balance between high conductivity on the one hand, and durability and thermal conductivity on the other hand.

3.3. Stack Construction

As shown in Fig. 15, previous stack structures employed carbon separators and separate seals. A disk spring and backup plate were installed and connected by large size bolts to attain a high level of airtightness. In contrast, the fuel cell stack developed this time, as shown in Fig. 16, makes use of metallic separators with high strength and with 50% thickness of carbon separators. Moreover, the cell units have become thinner and the number of parts has been reduced, thanks to the use of unitized seals. Using the springiness of stamped metal separators, construction has been simplified effectively by merely encasing the stacked cells by panels, thereby reducing the number of parts of the entire stack to nearly half. In this way, the fuel cell stack has been made lighter and more compact, and the output density has been increased to more than two times the density of the conventional stack (Fig. 17 and Table 1).

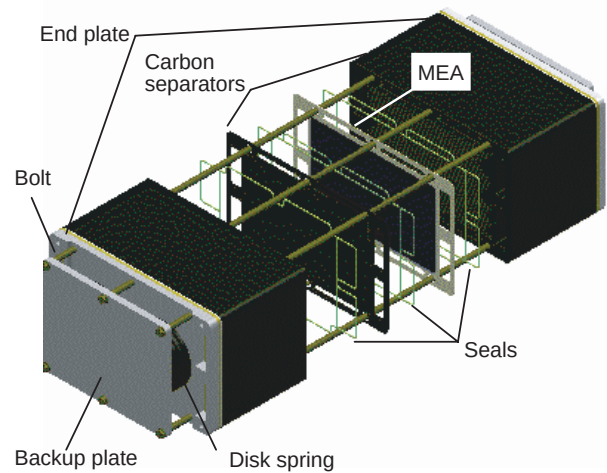


Fig. 15 Honda fuel cell stack in FCX-V3 (previous stack)

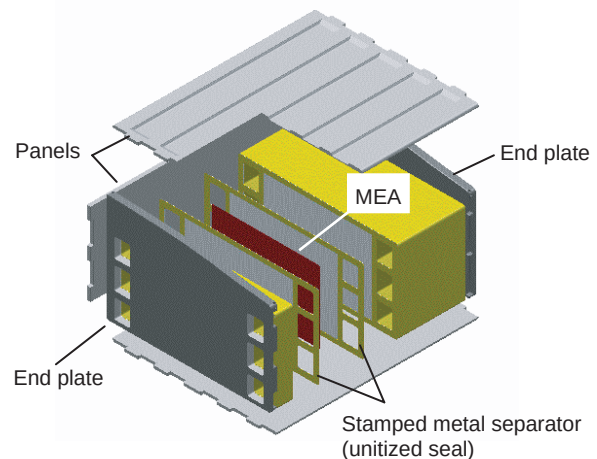


Fig. 16 Next-generation fuel cell stack for Honda FCX (Honda FC stack)

Table 1 Honda FC stack specifications

Output (kW)	50
Volume (L)	33
Weight (kg)	48
Max operating temperature (°C)	95
Cold start allowable temperature (°C)	-20



Fig. 17 Honda FC stack

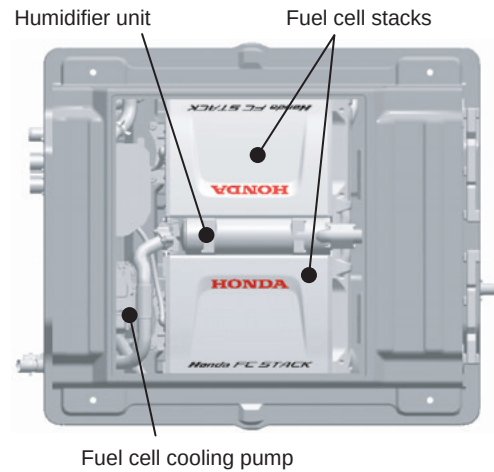


Fig. 18 Fuel cell system box configuration

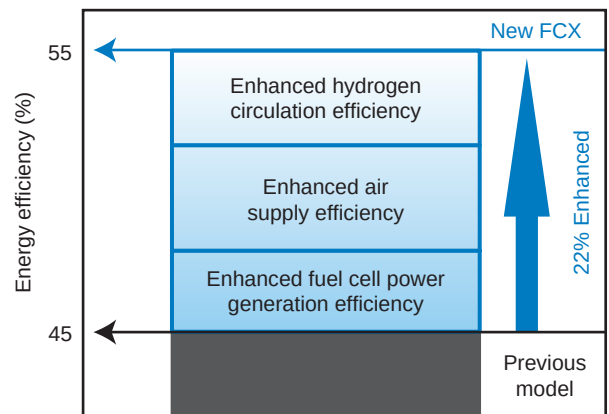


Fig. 19 Breakdown of enhanced energy efficiency (Honda data, LA-4 mode)

4. Fuel Cell Auxiliary System

The fuel cell stack was divided into two parts, with a humidifier module arranged between these two parts and integrated, which enabled the efficient layout of fuel cell auxiliaries, and attained compactness in the overall fuel cell system. The construction of the fuel cell system box fitted in the new model FCX is shown in Fig. 18.

Despite adding features to allow for low temperatures, the overall system has become lighter and more compact after rearranging the system. As a result, a fuel cell system that is about 5% smaller in volume and approximately 3% lighter in weight than that of the previous model⁽⁴⁾ was achieved.

The energy efficiency of the air supply system was enhanced by reducing the pressure losses of the air pathways, including the stack. The efficiency of the hydrogen circulation system was enhanced by reducing the power consumption through utilization of the low-pressure loss characteristics of the stack and reduction of pressure losses in the pathways, while also taking into account equal distribution of the hydrogen supply to the two fuel cell stacks.

With these changes, the energy efficiency of the entire fuel cell system, including stacks, was enhanced by 22% compared to that of the previous model (Fig. 19).

Furthermore, the start of power generation at temperatures as low as -20 °C was easily attained by optimally setting the air supply, hydrogen supply, humidification, and cooling.

5. Conclusion

The next-generation fuel cell stack fitted in the Honda FCX has achieved compactness and high output comparable to those of the highest levels worldwide, in addition to enabling starts below freezing temperature for the first time ever in fuel cell cars. Important technologies employed are summarized below.

- (1) Development of the aromatic electrolyte membrane enabled starting below freezing temperature, through increased proton conductivity, and enhanced durability at high temperatures.
- (2) Development of a stamped metal separator formed with a unitized seal, enabled the time from the start of power generation to warm-up to be reduced, because of enhanced electrical and thermal conductivity, while also enabling a significant reduction in the stack size.
- (3) The aromatic electrolyte membrane, stamped metal separator formed with a unitized seal, and simple stack structure enabled a compact design and high output to be achieved.
- (4) Utilizing two separate, compact fuel cell stacks enabled the appropriate arrangement and layout of auxiliaries, including humidifier modules to be achieved, thereby reducing the size of the fuel cell system. In addition, reviewing the operating conditions and reducing the pressure losses in the pathways enabled energy efficiency to be enhanced.

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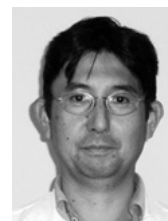
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Development of V6 ACCORD Hybrid 5 Speed Automatic Transmission System

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ABSTRACT

The 5-speed automatic transmission employed in the 2005 model year V6 Accord Hybrid was developed to enable the vehicle to display the powerful and refined driving performance of a V6 sedan, while achieving the fuel economy of a 4-cylinder compact sedan.

The Accord Hybrid represents the first combination of an IMA system and a stepped AT employing a torque converter. While its overall length has been reduced, the new AT has been designed with consideration of the high torque generated by the V6 3.0L i-VTEC engine and the IMA motor.

Expansion of the cylinder idling operating range and more effective regeneration of deceleration energy required a redesigned long torsion damper and a highly responsive lock-up clutch mechanism. The system has also been equipped with an electronic oil pump to enable smooth takeoff after idle stop, and a hydraulic system that employs a linear solenoid for direct control of all shift stages. The use of these new technologies has enabled the achievement of a superb balance of fuel economy and driving performance.

1. Introduction

Year by year, the demand for lower automotive emissions and higher fuel economy increases. In the United States, with its steep rise in gasoline prices due to the change in energy policies since September 11th 2001, the focus on low fuel consumption, and low polluting vehicles fitted with hybrid systems has been gaining steady momentum.

On December 10, 2004, Honda launched the V6 ACCORD Hybrid, in response to these market demands.

The IMA system for this ACCORD Hybrid is based on the IMA system employed in the CIVIC Hybrid but with enhanced motor generator performance. A V6 3.0L i-VTEC engine with Variable Cylinder Management (VCM) is incorporated in the system, and integrated control is carried out. This paper introduces the new

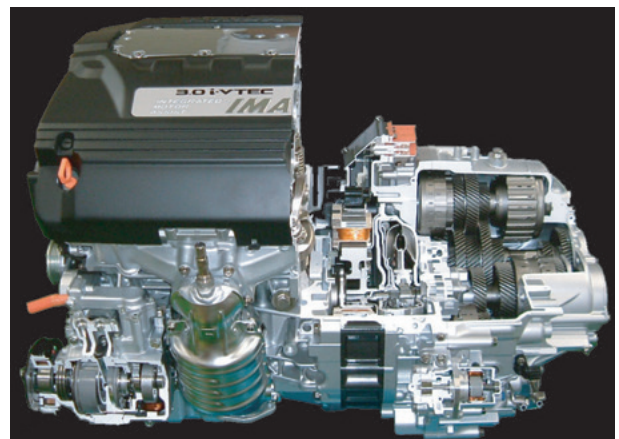


Fig. 1 ACCORD Hybrid powerplant cut model

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5-speed automatic transmission applied in this system along with its configuration, features, and performance^{(1), (2)}.

Figure 1 shows a photograph of a cut model of the power plant.

2. Configuration of 5-Speed Automatic Transmission for the ACCORD Hybrid

2.1. Basic Layout and Main Specifications

The IMA system power plant of the ACCORD Hybrid comprises the engine, motor, and transmission, and is fitted in a lateral series layout. In this development project, a dramatically shorter transmission length enabled the total increase in power plant length to be minimized, even with the addition of the IMA motor.

Adoption of a thin-walled torque converter having a flatness rate of 65%, and the addition of a thirdshaft for 3rd gear drive, reduced the transmission's total length by 60 mm compared to the 2003 model ACCORD V6 5-speed AT. Figure 2 and Table 1 give a section diagram of the 5-speed AT and its main specifications.

Along the thirdshaft are arranged the 3rd gear clutch, 3rd gear itself, and 4th gear. When driving in 3rd gear, torque is transmitted via the thirdshaft. Adding this shaft enables the length of the other shafts to be reduced, greatly contributing to a reduction in the overall length of the transmission.

In addition, moving the idle gear row from the rear end of the transmission to the center of the support bearing minimizes gear noise due to shaft flex.

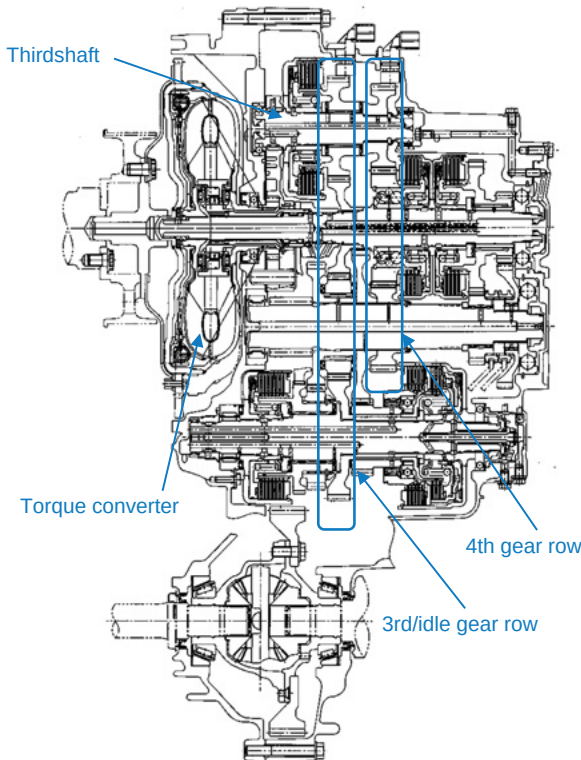


Fig. 2 Main section of ACCORD Hybrid 5AT

Table 1 Main specifications of ACCORD Hybrid 5AT

Type	4-Axis 5speed AT	
Ratio		O/A
1ST	2.685	11.747
2ND	1.565	6.849
3RD	1.023	4.476
4TH	0.729	3.190
5TH	0.530	2.321
RVS	1.888	8.263
Final	4.375	
Torque converter	$\tau_0=5.4$ $k_0=1.7$ (Long-torsion damper, Highly-responsive lock up clutch)	
Hydraulic system	All shift stage direct control Idle stop control	
Electric oil pump	Type: Trochoid rotor 3cc/rev Rating: 7.4L/min @ 343kPa	

2.2. Gear Ratio

Generally, the higher the gear ratio is, the better the highway fuel economy. With the ACCORD Hybrid VCM + IMA system, considering a balance with the cylinder idle range, the optimum gear ratio for the system is selected to achieve both driving and fuel economy performance.

Figure 3 shows the relationship between acceleration performance and mode fuel economy, while Fig. 4 shows that between the final gear ratio and mode fuel economy.

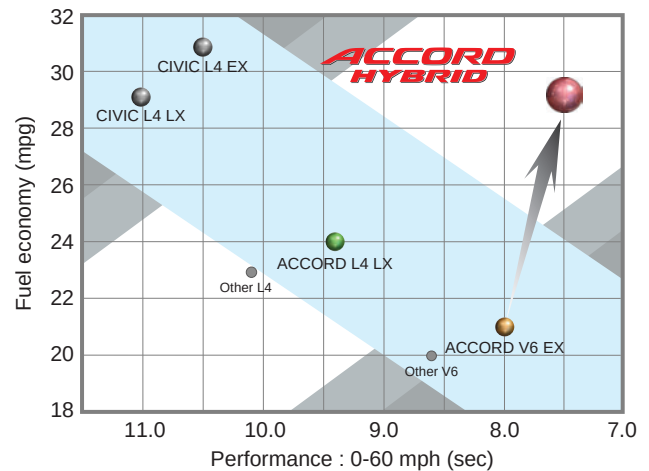


Fig. 3 Performance vs. fuel economy

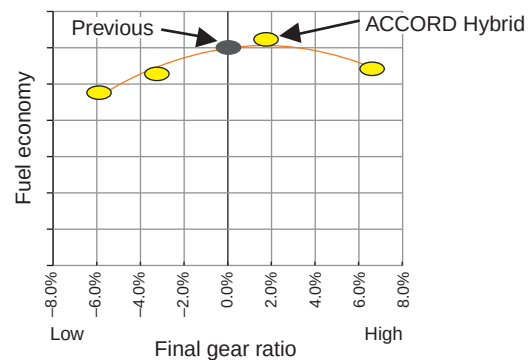


Fig. 4 Final gear ratio vs. fuel economy

2.3. Transmission Case Weight Reduction

When creating a hybrid vehicle, the addition of an IMA system increases the vehicle's weight compared to that of a conventional vehicle.

In order to minimize weight increases, CAE technology was employed for designing the new 5-speed AT system. The weight of the torque converter case and transmission case was reduced by 6%, while ensuring rigidity equal to or better than that of a conventional type AT. Figure 5 shows the results of a case stress simulation.

2.4. Electric Oil Pump

A trochoid type electric oil pump that operates during idle stop was newly developed in response to the IMA idle stop system.

With previous AT systems having only an oil pump that is driven by the engine, if an idle stop occurs, the oil pump stops and hydraulic pressure is not available. When the engine restarts there is a sudden rise in hydraulic pressure, causing a shock at start-off due to sudden clutch engagement. Therefore, in order to maintain the hydraulic pressure necessary for clutch engagement even during idle stop, an electric oil pump is employed, ensuring smooth start-off performance (Fig. 6).

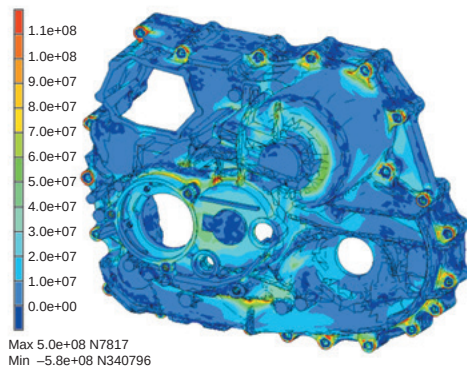


Fig. 5 Simulation of case stress

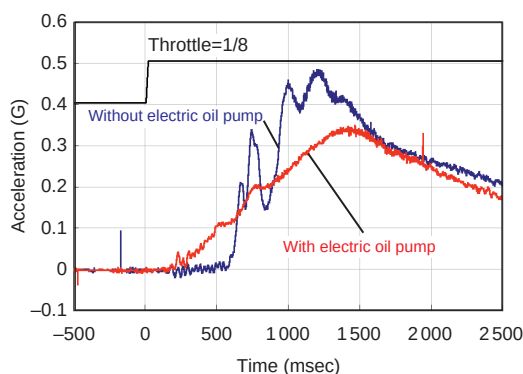


Fig. 6 IMA start characteristics

2.4.1. Electric oil pump system configuration

The electric oil pump system is composed of the pump assembly attached at the front of the transmission, and a driver unit.

The electric oil pump driver receives output torque requirement commands and sends information such as oil pump operation status, through two-way communications with the ECU. It controls the brushless motor in the electric oil pump assembly without a rotation sensor or current sensor. Figure 7 is a block diagram of the electric oil pump system.

During idling or when traveling, hydraulic pressure is supplied by the engine driven oil pump. During idle stop, pressure is supplied by the electric oil pump. To achieve this function, the hydraulic circuit configuration is such that the intake and discharge hydraulic passages are positioned parallel to the engine driven oil pump, and merge with the passage to the clutch. In addition, a check valve at the merge point with the discharge passage from the engine driven oil pump provides a simple device for switching the source of hydraulic pressure. Figure 8 shows the hydraulic circuit concept diagram.

The electric oil pump assembly is composed of the trochoid type oil pump and the brushless motor. The motor drive force is transmitted to the pump through an Oldham's coupling (Fig. 9).

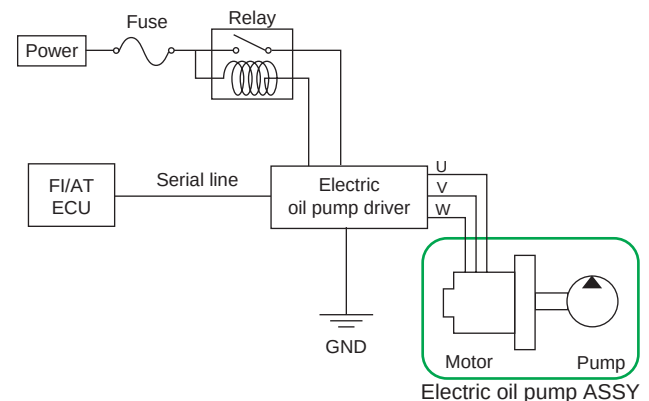


Fig. 7 Electric oil pump system

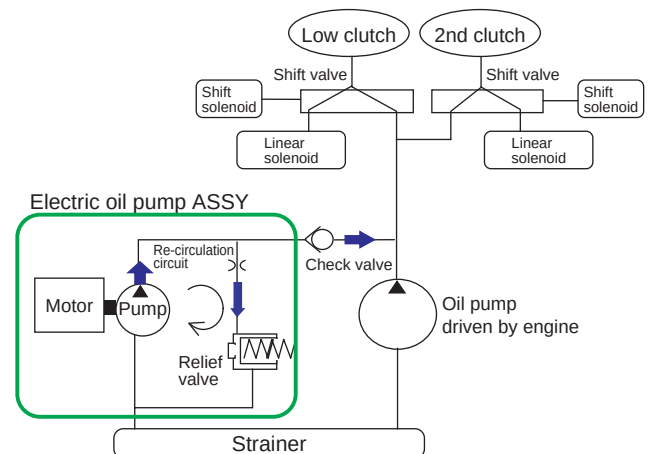


Fig. 8 Hydraulic circuit

2.4.2. Oil pump

To reduce friction, a small-diameter, wide trochoid rotor was adopted, minimizing side surface sliding resistance.

In addition, to minimize volumetric loss of efficiency at low viscosity high oil temperatures, the rotor housing is made of the same material as that of the rotor. Even at high oil temperatures, the pump side clearance can be maintained at a fixed minimum level, thereby achieving high efficiency in a compact and light-weight unit.

When the electric oil pump is activated, the discharge pressure from the engine driven oil pump closes the check valve. This blockage causes the motor speed to drop below the limit at which the current sensorless motor can turn stably. Another issue is that the motor load is initially light due to air bubbles being trapped in the pump while it's stopped, and it takes time until the motor speed stabilizes.

To address these issues, an internal recirculation circuit structure is provided in the unit. Before idle stop begins, the electric oil pump begins creating pressure and stands by at minimum speed or higher. By operating for a fixed period of time, the air bubbles inside the electric oil pump are diffused, minimizing load fluctuation when the check valve is released. With these measures, smooth starting after idle stop is possible.

2.4.3. Pump motor

The adoption of a brushless motor gives an increase in efficiency of approximately 20% over a DC brush motor of the same output.

A low noise motor was developed to drive the pump during idle stop when there is no noise from the engine or other devices. In order to reduce the noise from the bearings, which are a major source of noise from motors, the engagement dimensions were optimized and a preload spring installed, thereby minimizing bearing looseness.

These measures reduced noise in the cabin when the electric oil pump assembly is driven to less than 30 dB, the level of noise pressure in the cabin when the engine is stopped.

2.4.4. Electric oil pump driver

In order to minimize any electrical power loss due to wiring voltage drops, the electric oil pump driver is placed in the engine compartment. The location of the driver is on top of the wheelhouse, which has limited exposure to water, although a waterproof structure is also employed (Fig. 10).

The drive frequency is set outside of the audible range, cutting high frequency noise generated by the driver.

2.5. Long Torsion Damper

A new long torsion damper was developed to respond to an extended range of cylinder idle driving (Fig. 11). To attenuate torque fluctuations during cylinder idling, the torsion damper's spring constant was made 17% lower than that of the conventional unit. The resonance frequency was designed to be that of the lower limit speed of cylinder idle driving or less, thereby minimizing drive train vibration. This ensures good marketability, and at the same time contributes to enhanced fuel economy.

In addition, to reduce the shock from stepping on the throttle when the lock-up clutch is engaged, the torsion damper's operation angle is five times greater than that of a conventional unit (Fig. 12).

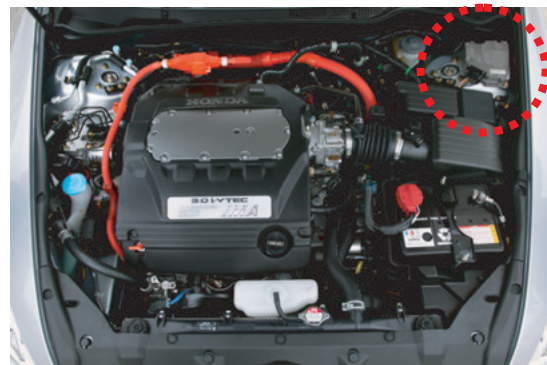


Fig. 10 Electric oil pump driver layout

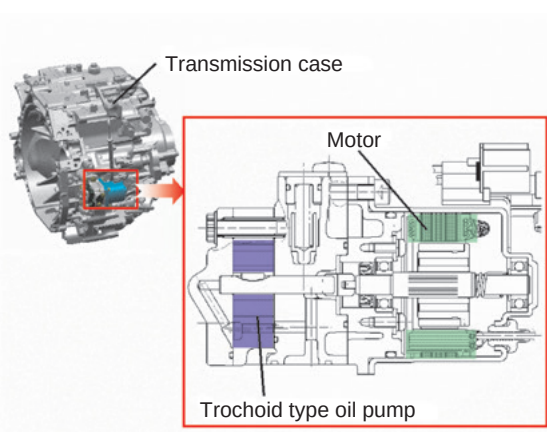


Fig. 9 Electric oil pump

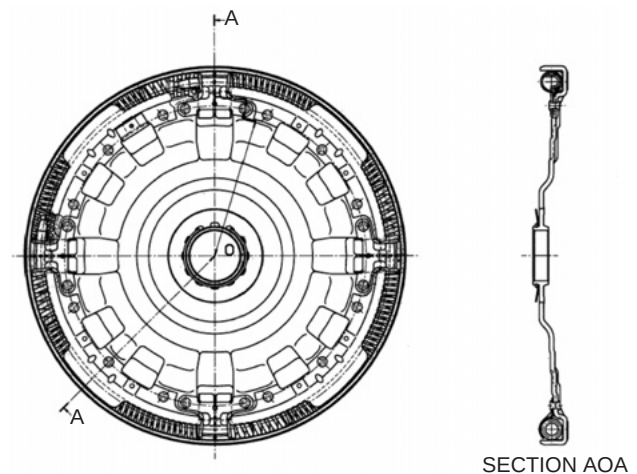


Fig. 11 Optimization of torsion damper characteristic for lock-up clutch

2.6. Highly Responsive Lock-up Clutch

As a means of increasing fuel economy when cruising or during deceleration regeneration, a highly responsive lock-up clutch, as shown in Fig. 13, was adopted.

An assistance mechanism, using a diaphragm spring, was added onto the backpressure chamber side of the lock-up clutch, making the response time one third to one fourth than that of a conventional lock-up clutch. This achieves a fuel economy increase of approximately 1.9%. (Fig. 13, 14)

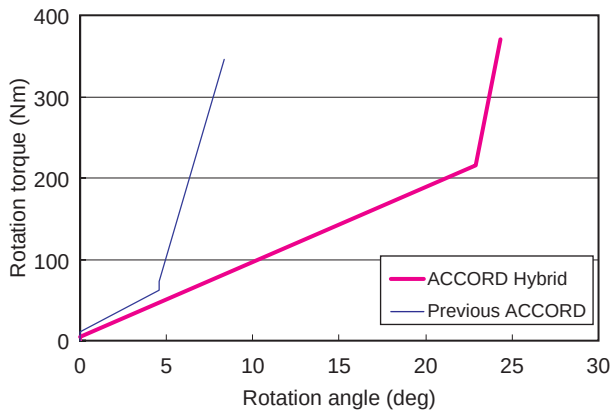


Fig. 12 Damper characteristics

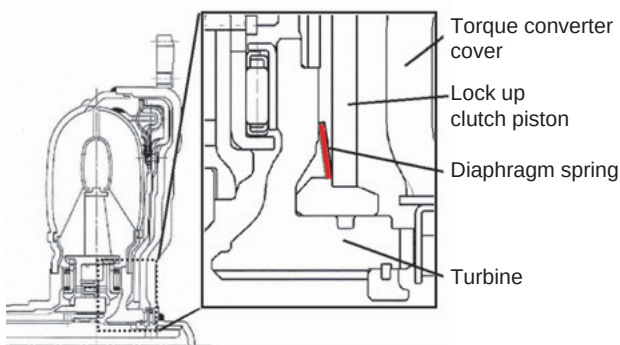


Fig. 13 Highly-responsive lock-up clutch

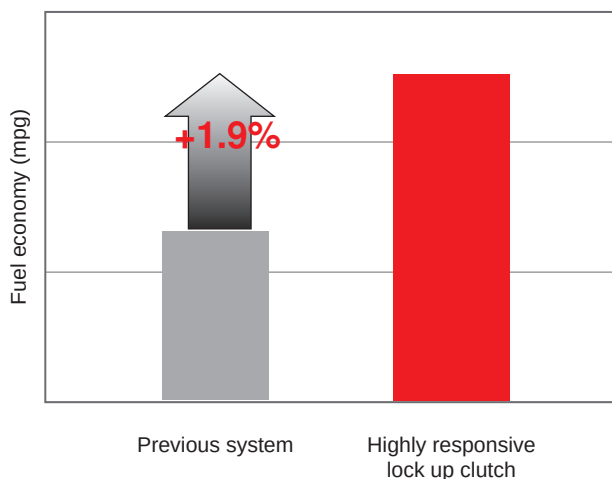


Fig. 14 Ratio of contribution to fuel economy

3. ACCORD Hybrid 5-Speed AT Controls

In addition to the idle stop, power assist, and the regeneration systems adopted in the CIVIC Hybrid, the ACCORD Hybrid introduces Drive by Wire throttle (DBW) and torque management control which work in cooperation with the cylinder idling system to increase fuel economy.

In order to achieve both low fuel consumption and high gearshift marketability, direct control was adopted for all clutches, including 1st.

3.1. 2nd Gear Start-off Control

In order to minimize wasted acceleration energy that accompanies engine revolution rise at start-off, start-off in 2nd gear is carried out with normal acceleration pedal openings. This provides a 0.1 mpg fuel economy increase. Under this condition DBW cooperative control is utilized, and sufficient start-off acceleration according to the accelerator pedal position is ensured.

If more acceleration is required, it is possible to have a smooth start-off in Low gear utilizing the advantages of direct control. An acceleration rate of 60 mph in 7.5 seconds can be achieved without spoiling the motor assist effect at start-off.

3.2. Deceleration Regeneration Control

To minimize fuel consumption during deceleration, it is necessary to increase electrical power regeneration and minimize the amount of injected fuel.

To increase electrical power regeneration during deceleration, the engine shifts to a 3-cylinder mode which reduces pumping losses. Lock-up clutch engagement is maintained and down shifting occurs such that the engine revs remain above the fuel cut maintenance revs. This allows fuel cut and energy regeneration in nearly the entire range of deceleration until idle stop, minimizing fuel consumption. Figure 15 shows operation during deceleration.

For a marketability benefit, the motor regeneration momentarily weakens at the instant of down-shift reducing the energy borne by the clutch and enhancing down-shift shock. In addition, by controlling the clutch engagement force before and after engine stop, the engine stop energy is not transmitted to the vehicle body, achieving a smooth stop feeling.

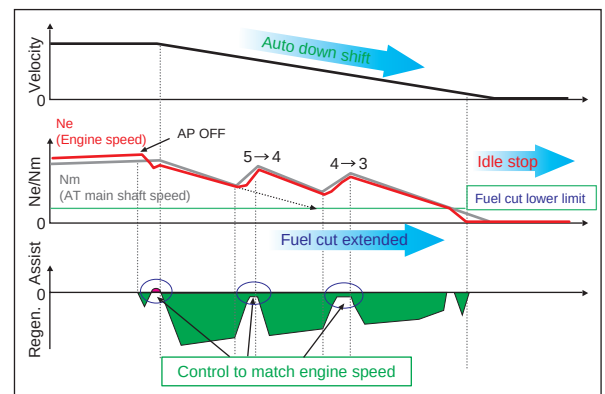


Fig. 15 Chart of deceleration regeneration operation

4. Conclusion

The ACCORD Hybrid 5-speed automatic transmission takes as its focus enhanced fuel economy by expanding the cylinder idling range with motor assist. An increase in IMA system output performance, AT idle stop measures, and maximum deceleration regeneration were achieved, without the loss of marketability.

As a result, it was possible to achieve both the powerful and refined driving performance of a V6 sedan, surpassing that of the current model, and the fuel economy of US City 29 mpg and US Highway 37 mpg, equivalent to that of a 4-cylinder compact sedan.

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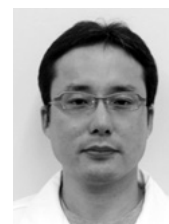
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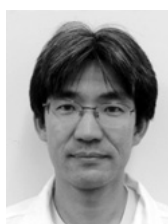
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Control Technology of Direct Electromagnetic Clutch Employed in SH-AWD System

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ABSTRACT

The SH-AWD system is a 4WD system providing continuously variable control of the torque transmitted to all four wheels to radically enhance vehicle dynamic performance. The core of the system is a direct electromagnetic clutch positioned in the rear drive unit. This new clutch was developed to enable more precise control than is possible in a conventional electromagnetic clutch. In the direct electromagnetic clutch, there is an air gap between the magnetic materials, that form the solenoid magnetic circuit. The solenoid force generated across this gap is converted into clutch thrust, without the need for the torque amplifier employed in conventional electromagnetic clutches. The fact that thrust acts directly on the clutch increases the precision of control. However, the size of the air gap changes when the thickness of the clutch changes, and this causes change in the solenoid force. A search coil is therefore used to detect changes in the size of the gap as changes in magnetic flux, enabling the current to be corrected in order to ensure that the appropriate solenoid force is consistently generated. In addition, measures to increase the accuracy of detection have enabled more precise control of torque transmission.

1. Introduction

The SH-AWD system performs variable control of the torque transmitted to the four wheels according to the driver's operation and the condition of the vehicle, thus enhancing the vehicle's dynamic performance, as represented by cornering performance and stability⁽¹⁾.

In order to satisfy the above performance factors, the electromagnetic clutch mounted on the rear drive unit must have high torque control accuracy including reduced hysteresis. The conventional electromagnetic clutch uses a ball ramp as the torque amplifier. However, the response delay due to play in rotation angle up to the operational position and the resulting drag load, affect

the torque control accuracy⁽²⁾. A direct electromagnetic clutch uses a planetary gear mechanism which enables the torque to be boosted without the above concerns occurring.

The conventional electromagnetic clutch employs a construction in which the clutch plate is held between the magnetic materials that constitute the magnetic circuit, hence the clutch plate forms part of the magnetic circuit through which magnetic flux passes. For this reason, it is not possible to use paper-based friction material, which is a non-magnetic material, in the clutch plate, and thus a ferrous clutch plate is used. As a result, it is necessary to specially treat the surface of the clutch plate in order to stabilize the inherent friction characteristics⁽³⁾. Also, because the clutch plate is a strength member that transmits torque, it is not possible to use pure iron which has superior magnetic characteristics⁽⁴⁾.

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In order to address the above issues, an air gap is provided at the magnetic material around the periphery of the solenoid, and the solenoid force generated in the magnetic material is used as the propulsion force for the clutch located outside the magnetic circuit. As a result, it is possible to use pure iron in the magnetic circuit, and to use paper-based friction material on the clutch plate.

However, when the air gap changes due to the wear of the clutch, the solenoid force also changes. Consequently, in order to accurately control the torque, it is necessary to detect changes in the air gap, and correct the solenoid force accordingly. This paper discusses the detection of the change in the air gap, solenoid correction control and the method of increasing control accuracy.

2. Direct Electromagnetic Clutch

Figure 1 shows the construction of the direct electromagnetic clutch. The yoke made of magnetic material has the main coil built-in and an air gap is provided between the yoke and the armature made of the magnetic material. When current flows through the main coil, the yoke functions as a solenoid, and a load that causes the yoke to attract the armature is generated inside. This solenoid force is transmitted to the clutch via a push cylinder, causing torque to be generated between the case and the hub.

With this construction, when the thickness of the clutch plate changes due to wear, the air gap also changes. In this case, the correlation between the current and the solenoid load changes (Fig. 2), and if no correction is performed, accurate control is not guaranteed.

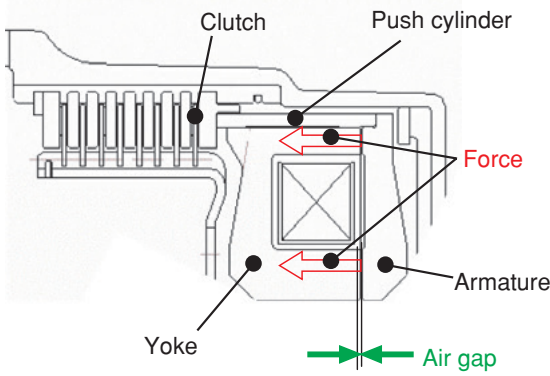


Fig. 1 Cross-section of direct electromagnetic clutch

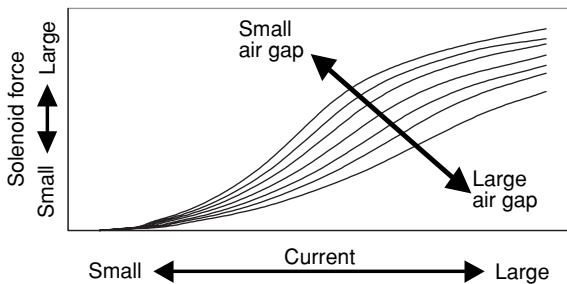


Fig. 2 Current-solenoid force characteristic

3. Air Gap Detection

The solenoid force between the armature and the yoke of the direct electromagnetic clutch is expressed by Eq. (1). As shown in Fig. 3, the solenoid force is inversely proportional to the air gap.

$$F = 10^7 S \mu_0^2 U^2 / 8\pi \delta^2 \quad (1)$$

Where,

F : Solenoid force S : Cross-sectional area of the magnetic circuit

μ_0 : Permeability of a vacuum ($4\pi \times 10^{-7}$)

U : Magnetic field strength

δ : Air gap

The cross-sectional area of the magnetic circuit is expressed by Eq. (2). As shown in Fig. 4, the solenoid force is proportional to the magnetic flux.

$$S = 8\pi F / 10^7 B^2 \quad (2)$$

Where,

B : Magnetic flux

Consequently, it is possible to detect the air gap by detecting the magnetic flux.

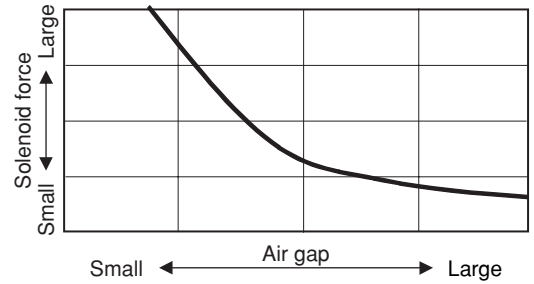


Fig. 3 Air gap-solenoid force characteristic

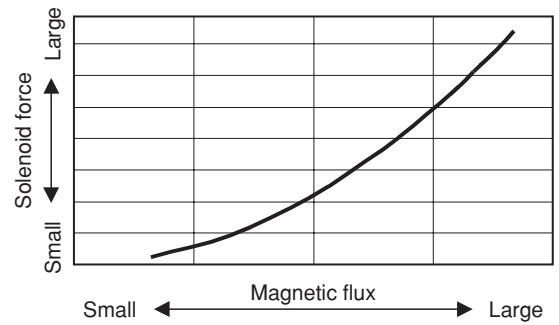


Fig. 4 Magnetic flux-solenoid force characteristic

4. Magnetic Flux Detection

A search coil is used to detect the magnetic flux (Fig. 5). The search coil is located on the inside of the yoke. This is because the search coil is integrated⁽⁵⁾ with the main coil for manufacturing ease, and it is easily affected by the magnetic flux flowing through the magnetic circuit (Fig. 6).

The detection principle uses Faraday's law, as expressed by Eq. (3).

$$e = N_s d\phi/dt \quad (3)$$

Where,

e : Search coil Voltage

N_s : Number of search coil turns

ϕ : Magnetic flux around the search coil

From this it can be seen that a voltage is generated in the search coil along with the time change of the magnetic flux.

By integrating Eq. (3) with time, Eq. (4) is obtained.

$$\phi = \int e dt / N_s \quad (4)$$

The current flowing through the main coil is fluctuated, causing the magnetic flux around the search coil to change, which results in the generation of a voltage in the search coil (Fig. 7). The integrated area in the figure indicates the right term of Eq. (4). The magnetic flux can be obtained by dividing this area by the number of turns on the search coil.

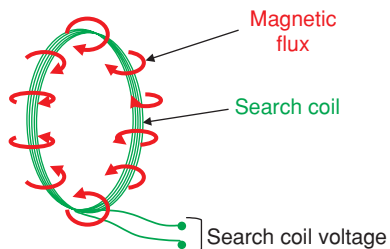


Fig. 5 Search coil concept

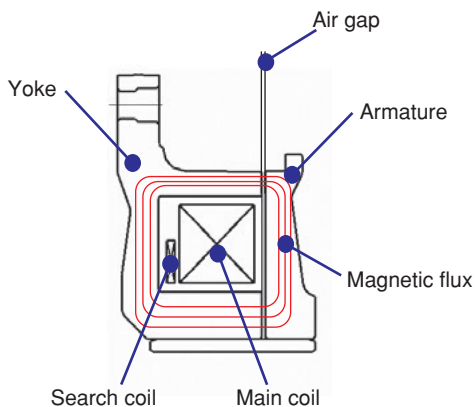


Fig. 6 Layout of magnetic circuit and search coil

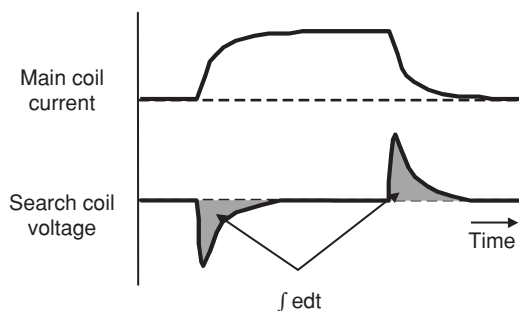


Fig.7 Voltage during DC current control

5. Solenoid force Correction Control

Next, the correction control method of the solenoid force is described. First, a 3-dimensional table representing Magnetic flux – Current – Torque (Fig. 8) is incorporated in the ECU.

Based on the detected magnetic flux, the Target torque – Current value table is selected from the 3-dimensional table. Based on this table, the current value corresponding to the target torque is obtained, and control implemented (Fig. 9).

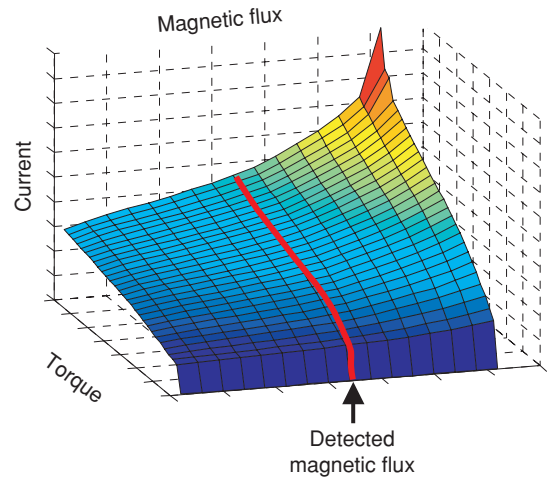


Fig. 8 Current-torque-magnetic flux table

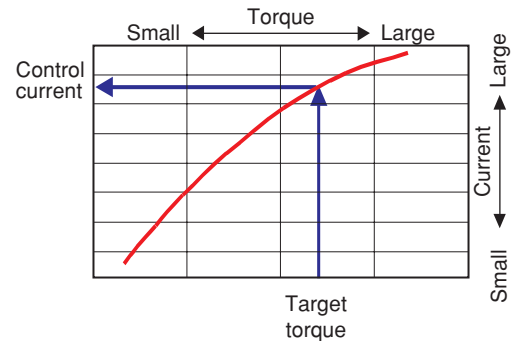


Fig. 9 Current-torque table

6. Control Accuracy Increase Technology

6.1 Effect of PWM Control

Figure 7 shows the results of measurement using direct current. The actual current control method is PWM (Pulse Width Modulation) control. While PWM control of the current is taking place, the search coil voltage oscillates due to variations in the magnetic flux, reducing its accuracy.

Therefore, the current was switched off after a PMW control of constant current, and the region in which the current falls gradually, along with the magnetic inductance, was sampled (Fig. 10). The search coil voltage changes slowly without oscillating. This enables the integrated area to be obtained stably, thus permitting increased magnetic detection accuracy.

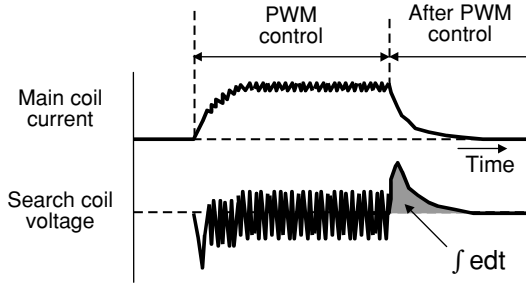


Fig. 10 Voltage during PWM current control

6.2. Effect of ECU Reference Voltage

If the sampling period when the ECU converts a search coil voltage to A/D is defined as θ , it is possible to express Eq. (4) by means of Eq. (5), and as shown in Fig. 11 the voltage is integrated.

$$\phi = \int e dt / Ns \div \theta \Sigma e / Ns \quad (5)$$

Where,

θ : Sampling period

The reference voltage is different for each individual ECU, and even for the same ECU it varies with changes in the environmental temperature. Consequently, when detecting the search coil voltage, the offset voltage between the ECU reference voltage and the search coil zero voltage is included as an error (Fig. 12). Therefore, it was decided to integrate the search coil voltage over the same period as the detection period (Fig. 13) immediately after the search coil voltage is detected, and then using Eq. (6) to subtract the resulting value from the integrated value obtained when the voltage was detected. This enables the effect of variations in the ECU reference voltage to be eliminated, and realizes increased magnetic flux

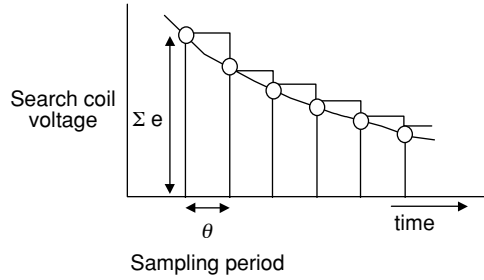


Fig. 11 Sampling period of search coil voltage

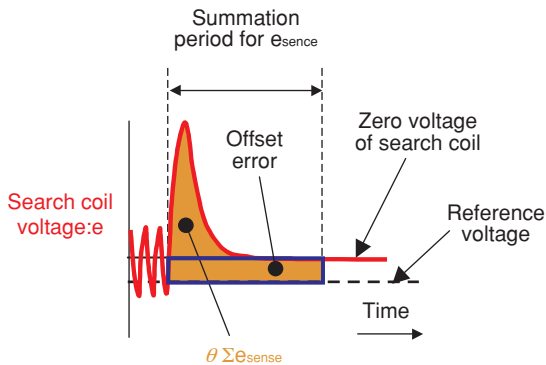


Fig. 12 Offset voltage error of search coil

detection accuracy.

$$\phi \div \theta (\Sigma e_{\text{sense}} - \Sigma e_{\text{zero}}) / Ns \quad (6)$$

Where,

e_{sense} : Integrated value of calculated voltage

e_{zero} : Integrated value of correction voltage

6.3. Effect of the Profile of the Magnetic Circuit Gap

The solenoid force characteristics versus the air gap for different magnetic circuit profiles were confirmed. For the magnetic circuit profile, three types of gaps, 30° taper type, 45° taper type, and flat were studied (Fig. 14).

It can be seen from the measured results of the solenoid force in Fig. 15 that flat types show abrupt changes in the solenoid force versus the change in the air gap, compared to the taper type. In other words, the smaller the air gap, the smaller the current determined needed for the target solenoid force. For this reason, the current resolution of the solenoid force declines, and the error in the solenoid force resulting from the current error tends to

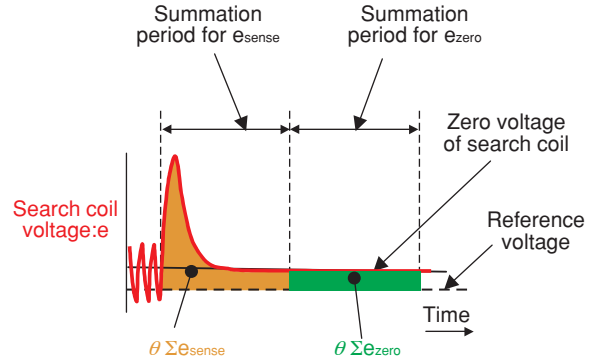


Fig. 13 Zero correction of search coil voltage

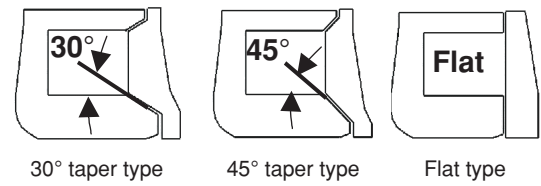


Fig. 14 Gap profile of magnetic circuit

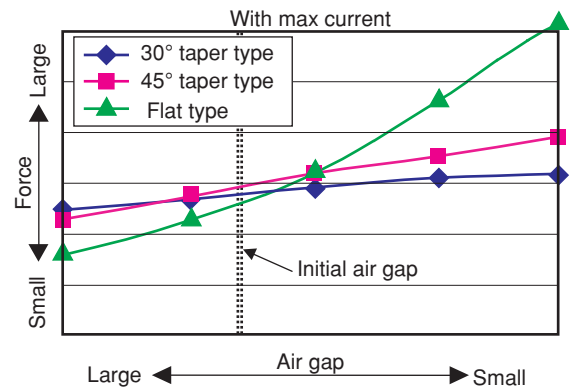


Fig. 15 Air gap-solenoid force characteristic

increase. Also, the search coil voltage when measuring the magnetic flux should be set to as high a value as possible, within the allowable range of the ECU, from the perspective of voltage resolution. However, in consideration of increases in the magnetic flux and the search coil voltage with respect to the current after the air gap is changed, it is unavoidably necessary to measure the magnetic flux using a small current from the outset. Consequently, the current resolution declines, and the magnetic flux detection error is possible to increase. This means that it is possible to increase the magnetic flux detection accuracy and the solenoid force control accuracy by selecting a suitable taper type magnetic circuit gap. Here, a taper angle of 45° which results in a large solenoid force at the initial gap was selected.

6.4. Effect of the Set Position of the Search Coil

The magnetic flux density distribution of a 45° taper type was magnetically simulated (Fig. 16). The magnetic analysis software used was ELF/MAGIC produced by ELF Co.

As a result, it can be seen that the change in the magnetic flux distribution is large in the vicinity of the air gap. It is considered that this is due to the effect of magnetic flux leaking from the air gap. Consequently, it is possible to detect the magnetic flux in a stable magnetic field created by setting the search coil at a position remote from the air gap, increasing the magnetic flux detection accuracy (Fig. 17).

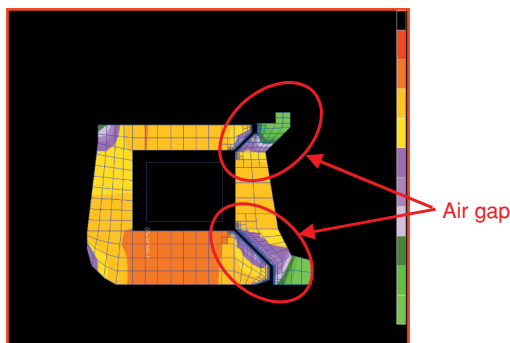


Fig. 16 Magnetic flux simulation result

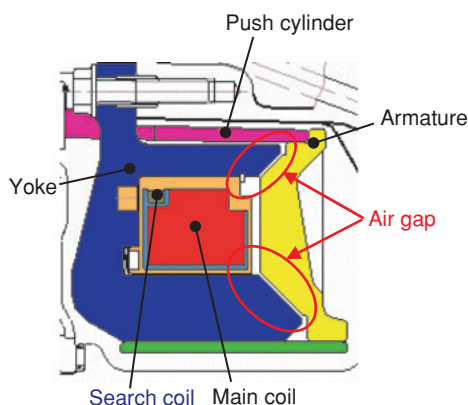


Fig. 17 Layout of search coil

7. Results

The correlation between the target torque and the actual torque was measured both with and without current correction based on magnetic flux detection. Figures 17 and 18 show the results obtained from measurements both with and without current correction. By adding correction control, more accurate correlation between the target torque and the actual torque was obtained regardless of the size of the air gap.

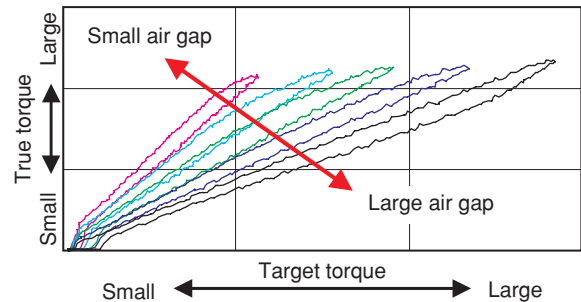


Fig. 18 Torque control result without correction

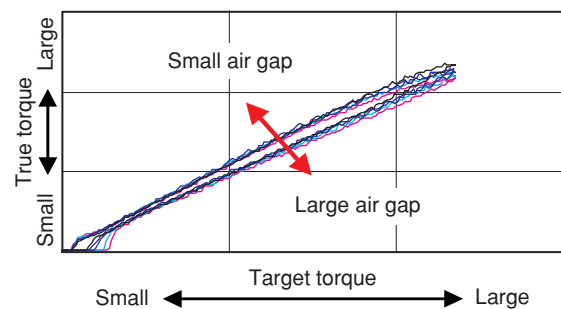


Fig. 19 Torque control result with correction

8. Conclusion

It was found that correction of the solenoid force with respect to changes in the air gap could be made by performing magnetic flux detection using a search coil and current correction control, based on the detected magnetic flux. In addition, it was verified that more accurate torque control could be realized by taking steps to increase the control accuracy. As a result, it was possible to establish direct electromagnetic clutch technology as the main technology of SH-AWD.

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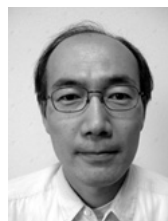
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Development of Resin-Based Multi-Piece Truck Bed

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ABSTRACT

A material system has been developed to achieve the strength and durability required of a truck while maintaining the design flexibility required to produce a multi-piece truck bed with lockable under-floor storage. This system consists of a new structural SMC, a 2-part structural adhesive, and a spatter-applied 2K urethane coating. Material targets were developed considering both the particular usage of a truck bed and the over-arching goals of light-weight and excellent appearance. Each component was formulated to balance these requirements and validated through comprehensive specification tests including static strength, chemical and temperature durability, and other usage conditions.

1. Introduction

The use of Sheet Molding Compound (SMC) as a replacement for steel parts in automotive structural applications has been the subject of investigation for well over 35 years. Until very recently, there have been few applications of structural SMC to appearance parts. Rather, most structural SMC usage has been for non-appearance parts such as grill opening reinforcements and structural inners for vertical and horizontal body panels. With advances in materials and process technology as well as the continued push for increased fuel economy, the idea of using fiber-reinforced plastics such as SMC to replace large, highly-visible steel structures has begun to gain momentum⁽¹⁾.

Honda has applied various fiber reinforced plastic parts, such as PP/glass fiber for bumper beams and non-structural class A SMC for spoilers, to mass production models. However, the application of a structural fiber reinforced plastic on a highly visible component has not previously been achieved.

This paper describes the development of a material system for a multi-piece truck bed consisting of a new structural SMC material, 2-component (2K) urethane coating, and structural adhesive. This material system balances the strength and durability required of a structural part and the good surface appearance and processability important to producing a high quality exterior part.

2. Material System Concept

Figure 1 identifies the final application areas for each component of the multi-piece truck bed material system.

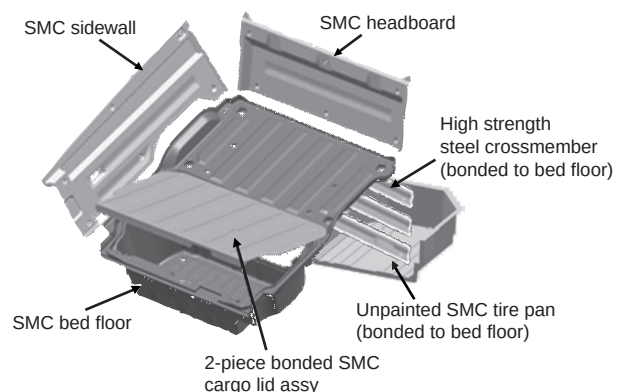


Fig. 1 Material application on multi-piece truck bed

* Honda R&D Americas

2.1. Structural SMC

Aside from the obvious benefit of weight reduction, resin-based composite materials also offer the advantage of increased design flexibility due to the ability to consolidate parts and conform to curves and deep draws not possible to replicate with steel. As Table 1 shows, within the category of resin-based structural composites, the strength and durability of SMC combined with the design flexibility it affords, gives this material the edge in applications where both aspects are critical. SRIM, while offering comparable physical properties to SMC, lacks in the area of design and processing freedom, while long fiber reinforced thermoplastics (LFRTs) are generally not competitive for part durability or dimensional tolerances.

Table 1 Relative property comparison for candidate substrate materials

Property	Material			
	Steel	SMC	SRIM	LFRT
Modulus	++	+	+	+
Temperature resistance	++	+	+	-
Surface durability	- (Rust)	++	++	-
Chemical resistance	+	++	++	+(Resin dependent)
Design flexibility	-	+	-	+
Dimensional stability	++	+	+	-
Weight	-	+	+	++

2.2. Adhesive System for Multi-Piece Design

With design requirements driving a multi-piece truck bed configuration, the need for a strong, durable method of joining parts became critical. Considering the size and usage conditions of these parts, it was determined that the joining method should include a structural adhesive, meaning a class of chemical bonding agents where the strength of the adhesive is near to or greater than the strength of the substrates bonded together.

The adhesive goal for these applications was that it needed to perform under a variety of extreme environmental conditions and loads that could be associated with a pick-up truck. To achieve this goal, two types of adhesive formulations were investigated: polyurethane and epoxy. Also, both room temperature and heat cure systems were studied for each adhesive type.

Table 2 Physical property targets for truck bed SMC Property

Property	Target value
Tensile strength	>125 MPa
Flexural strength	>265 MPa
Flexural modulus	>10 000 MPa
Compressive strength	>180 MPa
Izod impact resistance (-30°C)	>1000 J/m

Table 3 Adhesion performance targets

Test item	SMC (glass loading)	
	High	Low
Lap shear strength	> 3.4 MPa	> 1.5 MPa
Cross peel strength	> 1.2 MPa	> 0.45 MPa
3-point bend strength	> 150 N	Not applicable
Side impact	> 3.0 J	> 2.5 J

2.3. Coating System for Structural SMC

In order to understand the requirements for a truck bed coating system, the initial focus was to understand the overall functional performance of a steel or composite truck bed itself. A steel truck bed coating endures levels of denting and scratching, while the major limitation in a composite truck bed coating is scratching. After understanding these requirements, we were able to create the development goal for the first Honda truck bed coating system: A coating system that has equal or better durability than currently available truck bed coating systems combined with an overall increase in scratch resistance.

Table 4 Key coating requirements

Requirement items	Target value
5-Finger Scratch resistance	> 20 N
Sliding Friction Load (Pull Force)	> 2.7 N
DuPont Impact resistance	> 23.5 J
Weather resistance	Delta E<3.0

3. Defining Target Values

3.1. Structural SMC Property Targets

Target physical properties for the truck bed were based on part strength requirements for both the typical and extreme usage requirements of truck owners, such as heavy load hauling and exposure to the environment. Considering the design intent and demands for surface quality and fit and finish required of the overall vehicle, the targets of increased processability and appearance for the material were applied as well. General property targets are shown in Table 2.

3.2. Structural Adhesive System Targets

The adhesive target requirements were determined from existing structural adhesive specifications for aluminum/steel applications. The strength requirement was determined from 3 σ test result values of high and low glass loading SMC in fiber tear failure mode. Durability conditions covered the following conditions: initial, hot circumstance, cold circumstance, water, humidity, heat, thermocycle, and accelerated weathering. General adhesive targets are shown in Table 3.

3.3. Coating System Targets

For the coating system, target properties were created from consideration of the rigorous usage expected of a composite truck bed coating. Key modules considered in setting targets included hauling cinder blocks, shoveling payload from the bed, and dragging multiple substrates across the bed surface. Table 4 lists key coating requirements developed in this process.

4. Material Formulation Development and Performance

4.1. Effect of SMC Formulation on Physical Properties

To set the initial material formulation, Design of Experiments (DOE) based investigations were conducted both on flat panels and a study tool containing both ribs and bosses to identify the main effects of resin type (polyester <PE> versus vinyl ester <VE>), glass fiber content, and glass fiber length combinations on strength and durability. The variables considered for these tests as well as their associated levels are highlighted in Fig. 2. Key responses studied included tensile strength, flexural strength, and flexural modulus.

As expected, glass fiber content was observed to be a significant factor affecting the strength and modulus of flat panels. Additionally, the selection of resin type was also found to have first order effects on modulus. As the vinyl ester content in the formulation increased, so too did the modulus of the resulting formulation, as in Fig. 3.

A similar investigation of the strength of these formulations was then accomplished using panels containing stiffening ribs and various boss attachments.

In contrast to the flat panel results, the ribbed panels demonstrated an inverse relationship between glass fiber content and rib strength, as seen in Fig. 4.

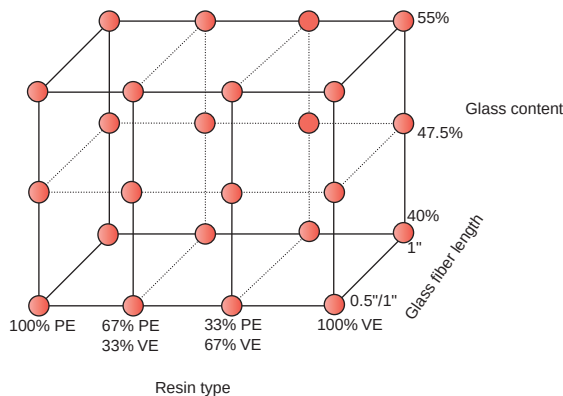


Fig. 2 Investigation matrix for material formulation

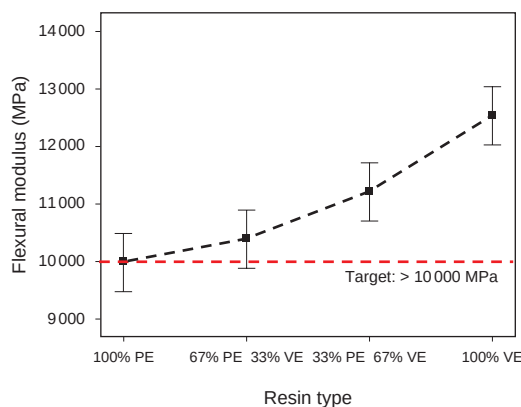


Fig. 3 Effect of resin type on flat panel modulus

Investigation into the potential cause of this relationship revealed that as glass content increased, it became increasingly challenging to completely fill the bottom of the ribs with fibers. The tips of the ribs were therefore resin-rich and in many instances also contained porosity. Without fiber reinforcement in these areas, cracks initiated quickly and led to part failure at lower loading levels than would have been predicted based on flat-panel material strength alone.

A similar trend of decreasing rib strength was seen with increasing vinyl ester resin content. Although the vinyl ester resin had higher stiffness and toughness than the polyester resin, this component was also more viscous and more of a challenge to flow into complex geometries. The results of these competing factors may be seen in Fig. 5. This figure highlights the criticality of incorporating part geometry and processing factors into the final material formulation.

4.2. SMC Material and Process Optimization

Optimization of both the final SMC formulation and processing parameters was conducted. Two design of experiments-based investigations were conducted using mass production quality study tools to identify the main processing factors affecting final part quality. Table 5 lists the primary variables under study as well as the responses evaluated.

One significant observation from these studies was that PE resin type had a large effect on part warping and surface wrinkling. Two

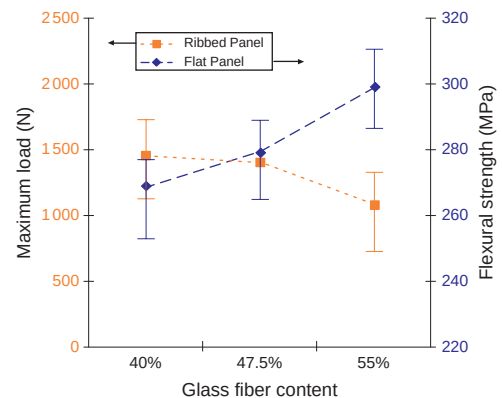


Fig. 4 Effect of glass fiber content on ribbed panel strength

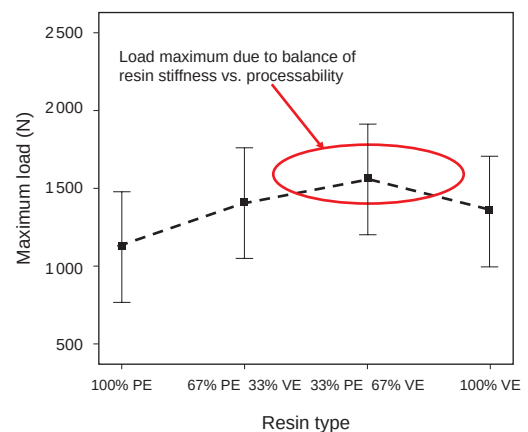


Fig. 5 Effect of resin type on ribbed panel strength

Table 5 Design summary for processing optimization

Variables	Responses
PE resin type	Part warping
Temperature	Surface wrinkling
Charge pattern coverage	Porosity
Material viscosity	Blisters
Clamp tonnage	
Vacuum	

types of thermosetting PE were considered: an ISO-type PE and a more reactive type PE. Generally, the ISO-type PE had greater warp and more surface wrinkles than the reactive-type PE. The reactive PE had higher miscibility with both filler particles and low profile additives, which are thermoplastic additives designed to reduce the overall material shrink on curing. As a result, this resin provided enhanced resistance to both warp and surface wrinkles, each of which is a result of the resin shrinking upon cure. The following two figures graphically explain this phenomenon. Fig. 6 is a schematic of the main components in an SMC resin, while Fig. 7 shows the differences between the PE resin components studied and their effect on warping.

Surface wrinkling was also affected in some part by mold temperature, mold pressure, the use of vacuum in molding, material viscosity, and the % charge pattern coverage. Porosity and blistering were shown to be dependent on molding pressure and the use of vacuum. Finally, part strength was reduced by using a small charge pattern. In this case, the material was forced to flow larger distances to fill the part, resulting in a high degree of fiber orientation and reduced strength in the direction perpendicular to flow. Overall results of the processing investigation are shown in Table 6.

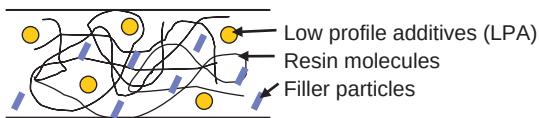


Fig. 6 Schematic representation of SMC

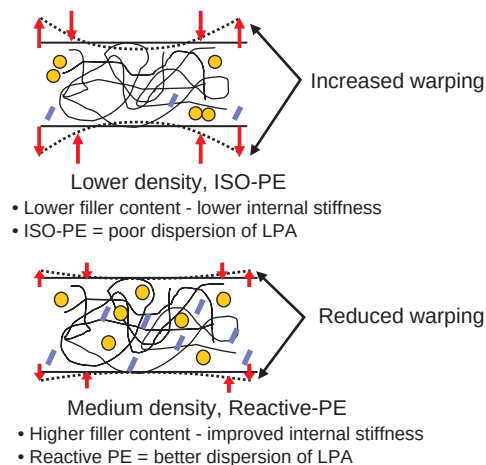


Fig. 7 Effect of PE resin type on warping

Table 6 Effect of processing on surface appearance

Condition	Blisters	Warping	Wrinkles	Porosity	Strength
Material viscosity	Higher		-		
	Lower		++		
Mold pressure	Higher		+	+	
	Lower		-	-	
Mold temperature	Higher	-	-		
	Lower	+	+		
Charge pattern coverage	Higher	+	--	-	+
	Lower	-	++	+	-
Vacuum	On	++		+	
	Off	--		-	

Table 7 Adhesive test results (High glass load SMC)

Adhesive	Initial	Hot Pull	Humidity
	Force (MPa)	Force (MPa)	Force (MPa)
Heat cure	5.8	5.3	5.4
Room temp. cure	4.1	2.4	4.2

As seen in the table above, beneficial levels for several process parameters, such as higher molding pressure, lower mold temperature and lower material viscosity were obtained from these processing studies. Identification of these factors made it possible to focus on charge pattern optimization during the final stages of process development.

4.3. Structural Adhesive Determination

In the investigation of structural adhesive type, two main areas were considered:

- (1) Processing and cure temperatures (room temperature vs. higher heat)
- (2) Base resin type (polyurethane vs. epoxy)

All formulations were based on two-component (2K) systems. This allows for faster cycle times and enhanced cross linking.

4.3.1. Room temperature cure vs. heat cure temperature adhesive

During the adhesive investigation, it was determined that the room temperature cure adhesive could not meet the target strength and mode of failure requirements. Table 7 is a brief summary of the test results with high glass loading SMC while Fig. 8 shows an initial condition lap shear sample failure mode comparison between room temperature and heat cure systems.

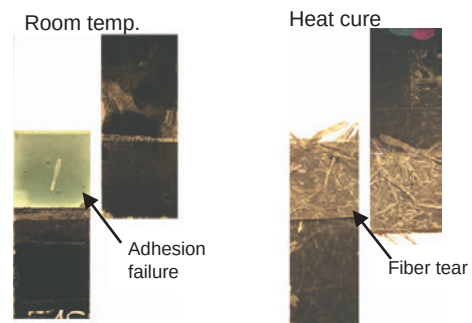


Fig. 8 Lap shear samples

The heat cure adhesive allowed for better wetting of the adhesive and increased cross-linking. As a result, the heat cure systems were able to meet both strength and mode of failure requirements and were selected as the adhesive system of choice.

4.3.2. Polyurethane vs. epoxy

After selection of the cure system was completed, specific adhesive formulation chemistry remained to be fixed. Both the heat cure polyurethane and the epoxy formulations performed very well to the target requirement. The polyurethane system, however, had a higher elongation and could therefore provide better flexibility under load. The epoxy formulations, by contrast, had very little elongation and became very brittle in the cold circumstance, starting to peel at a significantly lower displacement than the polyurethane formulation.

This displacement amount is a critical property for cold weather loading situations on bonded areas of the multi-piece truck bed. Therefore, a polyurethane structural adhesive was selected for use in the multi-piece truck bed construction.

4.4. Structural SMC Coating Determination

During the coating investigation, three main items were identified as key development areas:

- (1) Coating technology - 2K vs. 1K urethane
- (2) Coating orientation vs. scratch performance
- (3) Coefficient of friction vs. scratch performance

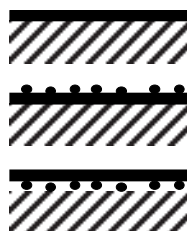
4.4.1. Coating technology

During the coating development, it was determined that the 1K urethane coating system could not meet the established requirements for physical properties, chemical resistance and weathering resistance. The 2K urethane coating consistently outperformed the 1K coating systems in these areas. An overall summary of test results is in Table 9.

The 2K urethane coating system has a higher cross-link density and as a result, an overall enhancement of tested properties compared to the 1K coating system.

4.4.2. Coating orientation

After determining that the 2K urethane provided the highest quality for the Honda truck bed coating, coating orientation was investigated to optimize scratch resistance. Based on established appearance targets, the following were considered as possible options:



1. Single Layer mono coat only
2. Mono coat layer, then spatter coat
3. Spatter coat layer, then mono coat

Using the 5-finger scratch test method, it was determined that option #3 performed the best for scratch resistance. Table 10 provides a summary of these results.

4.4.3. Friction increasing additives

Usage verification testing indicated that the surface of the truck bed did not provide enough resistance to objects sliding along the surface, such as might happen when driving with a loaded vehicle. Therefore, additional investigation was conducted to increase the

Table 8 Displacement results for three point bend tests

Adhesive resin	Displacement (mm)		
	Initial	-30°C	80°C
Polyurethane	18	14	19
Epoxy	11	10	20

Table 9 Coating properties

Key properties	2K urethane	1K urethane
Physical	++	+
Chemical	++	--
Weather	++	-

Table 10 Coating application

Paint orientation	Typical value
Mono coat only	<15N
Mono coat/Spatter coat	15N – 20N
Spatter coat/Mono coat	>20

Table 11 Friction increasing additives

Additive	Composition
Additive #1	Rubber material
Additive #2	Glass material
Additive #3	Rubber & Glass blend

friction of the coating system. Three friction additives, each of dissimilar characteristics, were selected for evaluation.

Additive #2 performed best with regard to friction value (pull force), as seen in Fig. 9.

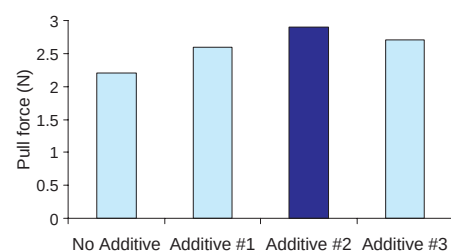


Fig. 9 Effect of additive type on friction

5. Final Material System Properties

Based on the investigations in Section 4, the authors proceeded with the development of a structural SMC formulation meeting target strength and durability conditions by focusing on a composition with at least 40% glass reinforcement and a blend of vinyl ester and polyester resins. Processing and appearance were optimized using filler content and resin ratios. Basic properties

Table 12 Basic properties of the Honda structural SMC

Property	Target value
Tensile strength	140 MPa
Flexural strength	275 MPa
Flexural modulus	11 800 MPa
Compressive strength	200 MPa
Izod impact resistance (–30°C)	1 100 J/m

Table 13 Basic structural adhesive formulation

Component	Formulation	Curing
A: Isocyanate prepolymer	• Polyol/Isocyanate prepolymer (MDI) • Fillers	$R-N=C=O + H-O-R' -$ Diisocyanate Polyol
B: Polyol/amine curative	• Polyol mixture • Sag control agent • Catalyst, fillers	

are in Table 12.

From design of experiments testing and material processing properties, the final adhesive and coating systems were developed as well. For bonding operations, a heat cure 2K polyurethane was chosen, as shown in Table 13.

For painted portions of the truck bed, a 2K urethane was selected as the Honda truck bed coating. Additive #2 was included in the paint formulation to achieve the required coefficient of friction.

6. Charge Pattern Optimization

Charge pattern optimization was conducted to finalize the relationship between molded part surface appearance and part performance in terms of static strength and fatigue durability. For the bed floor, two critical factors drove the final charge pattern: surface appearance (wrinkles) and rough road durability testing. Issues with durability testing arose in the rear mount point area due to excessive fiber flow. To enhance durability while keeping overall appearance OK, a number of different charge patterns were tried. After each stage, parts were tested on-vehicle during durability testing. Parts were also examined fiber flow orientation by sectioning them near the rear mount points and burning off the resin component to leave only the fiberglass in its actual orientation.

Similar investigations were done for each of the SMC parts making up the truck bed.

7. Conclusion

A new material system was developed for use in a near class A multipiece truck bed. This system consists of a new structural SMC, heat cure structural adhesive, and two-component urethane coating. The authors' findings are:

- (1) Combining a reactive PE resin and a tougher VE resin was necessary to balance part appearance and durability.
- (2) Glass fiber content between 40% and 50% maximized part strength.
- (3) Optimized charge pattern were used to balance appearance and performance.
- (4) A 2K heat cure polyurethane was required to bond multiple substrates.

- (5) 2K urethane coating materials outperformed their 1K counterparts.
- (6) Use of coating additives was effective in increasing sliding friction.

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Development of High-pressure Diecast Magnesium Intake Manifold

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ABSTRACT

A magnesium high-pressure diecast (Mg HPDC) intake manifold has been used in the newly developed V6 Integrated Motor Assist (IMA) engine on the ACCORD Hybrid for the North American market.

The body of the intake manifold features the same split structure used in the aluminum high-pressure diecast (Al HPDC) intake manifold fitted in the Honda ELYSION. The two parts of the body are sealed with a formed in-place gasket (FIPG). This FIPG is also used in Honda's high-pressure diecast magnesium transmission case. A modified version of a metal gasket has been used for the seal of fuel injector base.

The new magnesium high-pressure diecast (Mg HPDC) intake manifold is 44% lighter than a conventional aluminum low-pressure diecast (Al LPDC) intake manifold and 33% lighter than an Al HPDC intake manifold.

1. Introduction

One means of addressing global environmental issues, is to attain enhanced fuel economy. The application of magnesium alloys has expanded in recent years as an effective means of reducing vehicle weight.

To date, Honda has used the AZ91D magnesium alloy for engine head covers and the AM50A Mg alloy for steering wheels. Honda has also used a heat resistant magnesium alloy for the engine oil pan and manual transmission case.

As part of their continued efforts to reduce engine weight, the authors have developed a Mg HPDC intake manifold, a major component of the V6 engine.

2. Development Goals

The development goal for this project was to achieve a lightweight intake manifold capable of handling a 300 ps level high power engine. This would be done by changing the Al LPDC intake manifolds adopted for Honda's V6 engines to that of a Mg HPDC manifold using existing technology and mass production facilities.

The authors set the following targets for development, in order to achieve the weight reduction with Mg HPDC.

- (1) The targeted intake manifold must have seal performance, as well as strength and durability, equivalent to that of the mass-produced Al LPDC intake manifold.

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- (2) The target manifold must attain noise and vibration characteristics equivalent to those of the Al LPDC unit.
- (3) Casting of large, thin-walled parts using the Mg HPDC method must achieve product quality equivalent to that of Al HPDC parts.

3. Structure and Configuration

3.1. Basic Structure

As shown in Fig. 1, the intake manifold unit is composed of two parts: the intake manifold upper and the intake manifold lower. The joining surfaces are sealed using a FIPG, and are then bolted together.

An upper cover and an injector base, as well as a throttle body, are later bolted to the intake manifold unit.

In addition, a stainless steel EGR pipe is bolted inside the intake manifold, and a steel joint is inserted at the external hose connection.

3.2. Separation Structure

The Mg HPDC intake manifold unit is comprised of a plane separation area and a three-dimensional separation area, as shown in Fig. 2.

3.2.1. Port breaking structure

With the Mg HPDC intake manifold, a three-dimensional separation structure was adopted in order to create a port shape capable of handling a 3.5L 300 ps high power engine. When a plane separation structure is used, as shown in Fig. 3(1), the port is shaped like an L and the airflow is not smooth. This led to the decision to use a three-dimensional separation structure. When the port shape is shaped with a smooth curve as shown in Fig. 3(2) the air flows more easily.

3.2.2. Throttle body mounting flange separation structure

By giving the throttle body mounting flange a plane separation structure, the throttle body mounting flange, as shown in Fig. 4(1), is comprised of an intake manifold upper and intake manifold lower.

This structure forces the throttle body sealing surface to be machined together after the intake manifold upper and intake manifold lower are assembled. The result is a complex procedure, consisting of part machining, followed by assembly, followed by machining the surfaces together. To avoid such a procedure, the throttle body mounting flange side is given a three-dimensional separation structure in the outer flange area, as shown in Fig. 4(2). This allows the throttle body mounting flange surface can be machined as a single intake manifold lower part.

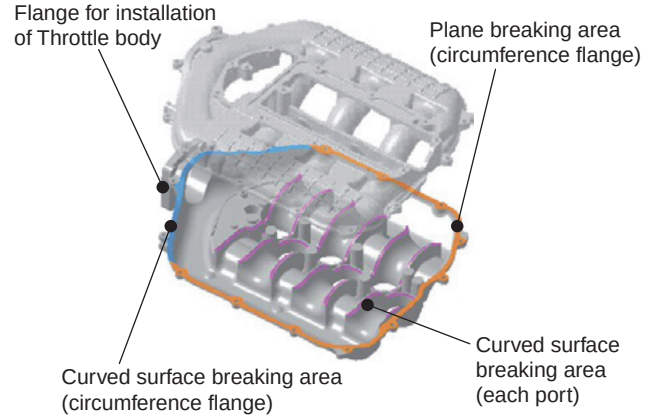


Fig. 2 Breaking structure

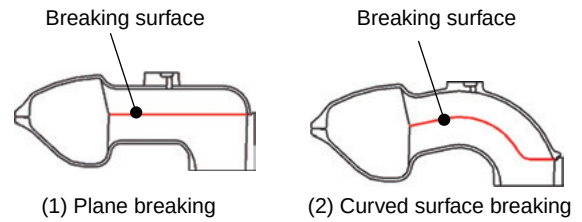


Fig. 3 Section shape of port

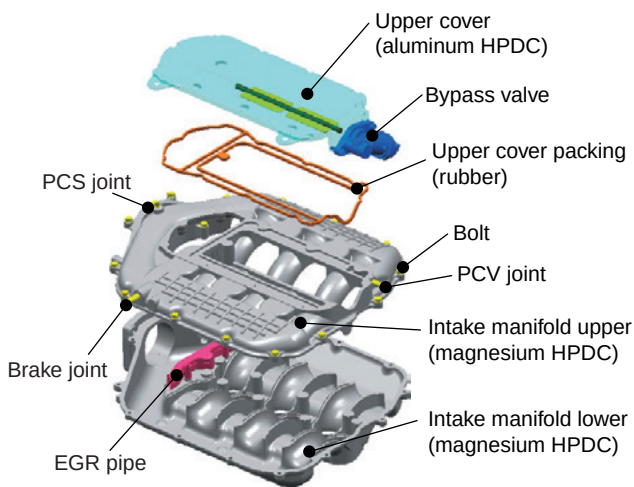


Fig. 1 Structure of magnesium HPDC Intake manifold

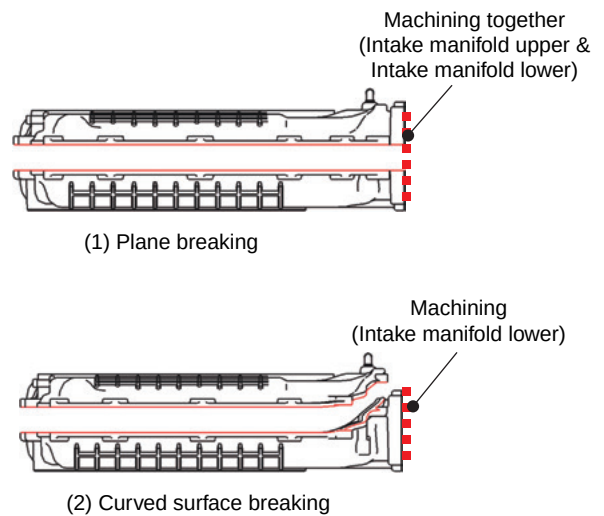


Fig. 4 Breaking pattern

3.3. Seal Structure

The Mg HPDC intake manifold has a FIPG seal between the intake manifold upper and intake manifold lower. In addition, as shown in Fig. 5, the intake manifold has connection surfaces with the fuel injector base, intake port, upper cover, and throttle body. Each connection surface employs the use of a metal gasket, rubber packing, and joint seat to ensure airtight seal within the intake manifold.

3.3.1. FIPG seal area

As shown in Fig. 2, the FIPG seal between the intake manifold upper and intake manifold lower is divided into an outer flange area and port area. There are differences between the outer flange FIPG seal surface and the port area FIPG surface, as shown in Fig. 6.

The outer flange area has a FIPG retention area (circled in Fig. 6), enabling the FIPG to stretch along the sealing surface in reaction to internal pressure changes in the intake manifold. In addition, because the port area does not use bolt tightening, a three-dimensional seal structure was chosen. This allows the seal to be maintained under any deformation condition – vertical, horizontal, and at an angle – due to pressure fluctuations within the intake manifold when the engine is running.

3.3.2. Injector base seal area

The injector base surface uses a metal gasket to ensure an airtight seal between the injector base and the intake manifold. Magnesium alloys have a lower Young's modulus than aluminum alloy, so it is believed that the injector base surface has a lower connection

rigidity. To compensate for this, the gasket seal bead has been changed from a full-bead type to a half-bead type, as shown in Fig. 7. This allows the bead to follow the seal surface movement more precisely and to ensure a seal performance equal to or better than that of the conventional part.

The half-bead type has more than twice the bead restoration ratio (bead height after the compression released / initial bead height) of the full-bead type.

3.3.3. Upper cover seal area

The upper cover surface is sealed with rubber packing material. Low temperature seal performance is ensured by using Fluoro Rubber (FKM) as the rubber packing material. Moreover, the use of a rubber packing material with a left/right symmetrical section, as shown in Fig. 8, provides a shape that will maintain seal performance in the case of any collapse.

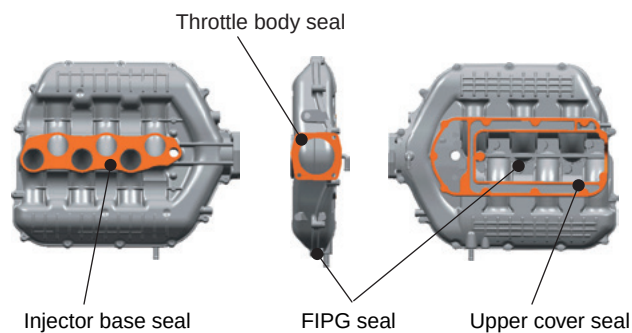


Fig. 5 Seal position

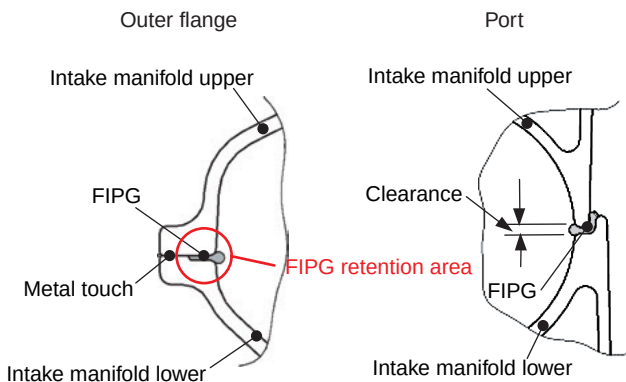


Fig. 6 Structure of FIPG seal plane

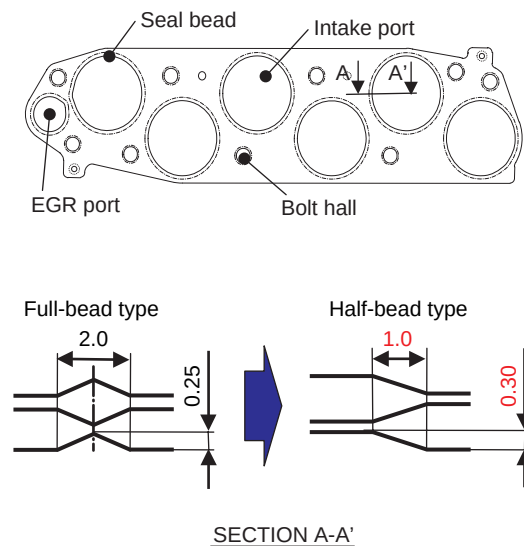


Fig. 7 Appearance and section view of Injector base gasket

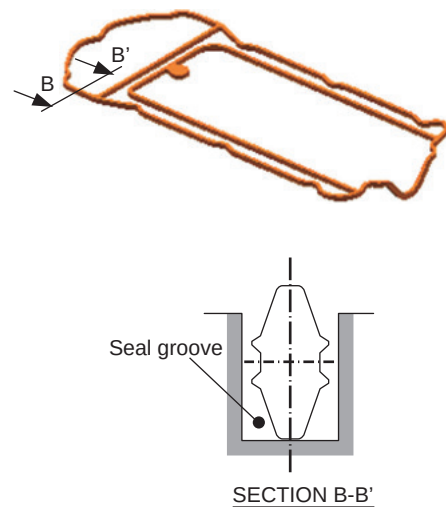


Fig. 8 Appearance and section view of Upper cover packing

3.4. Noise and Vibration Countermeasures

The Mg HPDC intake manifold has a thinner and flatter shape than the AL LPDC intake manifold, so reduced NVH characteristics were expected. From the results of the vibration analysis, surface ribs were added in a grid pattern on the flat areas of the resonance pipe to increase planar rigidity, as shown in Fig. 9. Additionally, connection bolts were added inside the intake manifold, enhancing vibration characteristics on the resonance pipe.

3.5. Resonance Conversion Valve Mechanism

With the conventional Al LPDC intake manifold, multiple hole locations were drilled and a resonance conversion valve was fitted. With the Mg HPDC intake manifold the resonance conversion valve was placed on the intake manifold upper. This is because the joining surface between the intake manifold upper and the intake manifold lower is an as-cast surface where no machining is done. Moving the resonance conversion valve away from the sealing surface avoids applying excessive stress due to flatness inconsistencies on the as-cast surface.

3.6. Corrosion Resistance

The magnesium alloy material AZ91D used for the Mg HPDC intake manifold has corrosion resistance equivalent to the aluminum ADC12 alloy used for the Al HPDC intake manifold. There is no need for coatings to prevent any corrosion. In previous models the intake manifold upper area only was painted, for appearance and commercial appeal.

In addition, as a countermeasure to prevent galvanic corrosion of the bolt connection areas, the same chrome plated bolts as were used on the magnesium HPDC transmission case were adopted.

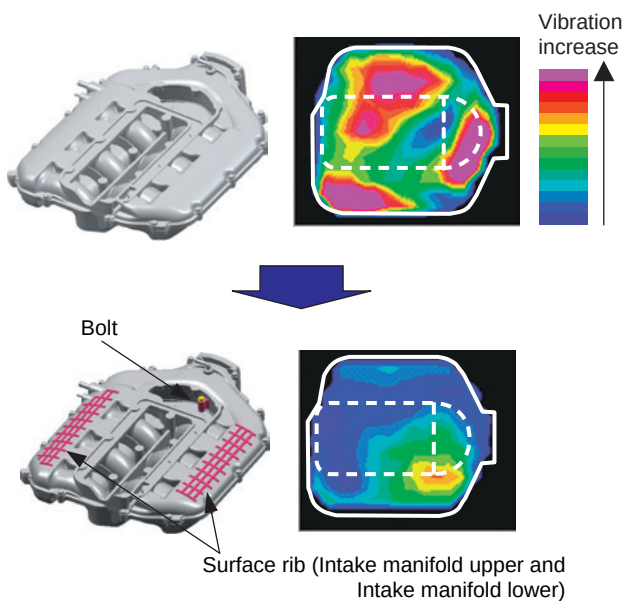


Fig. 9 Effect of surface rib on vibration reduction

4. Magnesium HPDC Casting Technology

In order to create intake manifold dies, casting simulations were carried out, and casting plans established. A high-speed injection type of diecast casting machine was used, and a horizontal layout adopted that enabled a compact casting design.

From the start of the development, a vacuum system was used to enhance molten material flow. The layout of the die heating and cooling circulation lines was established, based on thermal modeling of the inside of the die to obtain a solidification profile.

4.1. Casting Simulations

The solidification time for Mg alloys is shorter than that of aluminum alloy. To reduce the fill time of the castings the magnesium intake manifold upper has a new runner on the side (side gate) as shown in Fig. 10, and a new runner in the space inside the product (bridge). This is different from the runner layout for the Al HPDC intake manifold of the same shape.

The results of the casting simulation showed metal flow that is similar to that of the Al HPDC part. The final area to fill is on the side opposite the sprue, there is no trapped gas, and no areas in the center of the part are the last to solidify.

For the intake manifold lower, as shown in Fig. 11, a bridge was also placed in the center of the part to enhance flow. The results of the casting simulation again showed similar metal flow to the Al HPDC layout. This again confirmed that the final solidification area was on the opposite side from the sprue, there was no trapped gas in the center of the part, and no areas in the center of the part are the final areas to solidify.

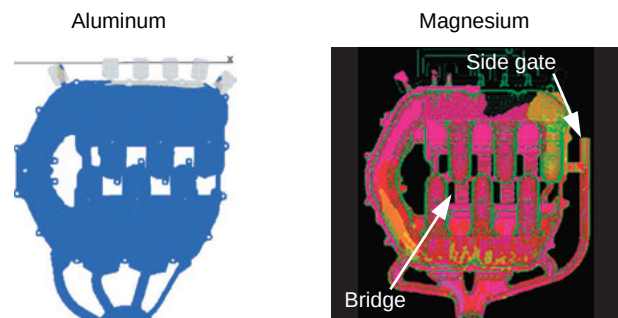


Fig. 10 Simulation of casting of Intake manifold upper

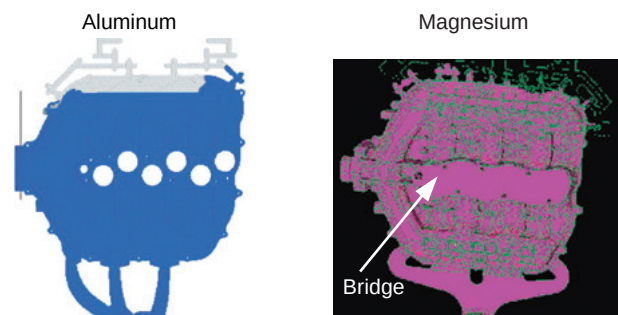


Fig. 11 Simulation of casting of Intake manifold lower

4.2. Setting Casting Conditions

The results of a casting parameter study on prototypes established the final casting conditions.

To prevent the occurrence of shrinkage cavities in the thick areas of the connection boss, Super-Cooling core pin (cooling inside the core pin, combining water cooling and air cooling) were added. The shrinkage cavities, caused by core pin surface overheating, were eliminated from the surface of the thick areas of the connection boss with the Super-Cooling device (circled in Fig. 12).

To ensure clearance between the outer seal surfaces of the intake manifold upper and intake manifold lower, which has a great effect on the outer flange FIPG seal performance, the outer flange flatness as a single part was enhanced by joining the die. Furthermore, to reduce the amount of sealing surface unevenness when removing the sprue after ejection, trim die distortion correction was implemented by clamping the part to the trim die. Increasing the time that the trim die jig holds the outer flange area yields enhanced flatness. As a result, as shown in Fig. 13, enhanced material flatness median values and peak to peak values were achieved.

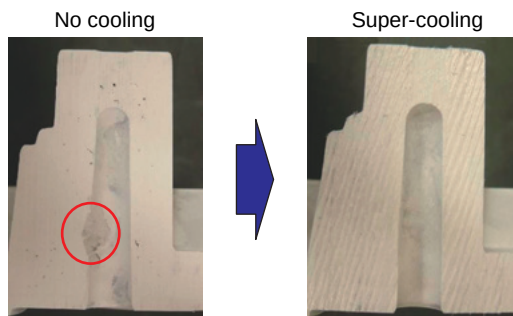


Fig. 12 Effect of Super-cooling

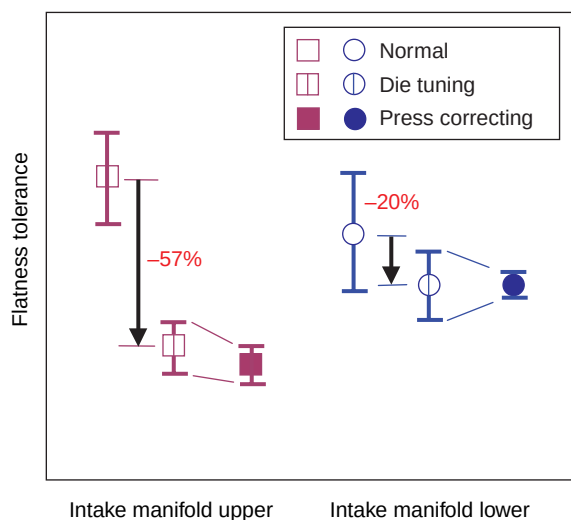


Fig. 13 Flatness of outer flange

5. Weight Reduction Effect

As shown in Fig. 14, the newly developed Mg HPDC intake manifold for the V6 engine is 44% lighter than that of the conventional Al LPDC intake manifold.

Figure 15 shows the appearance of the Mg HPDC intake manifold.

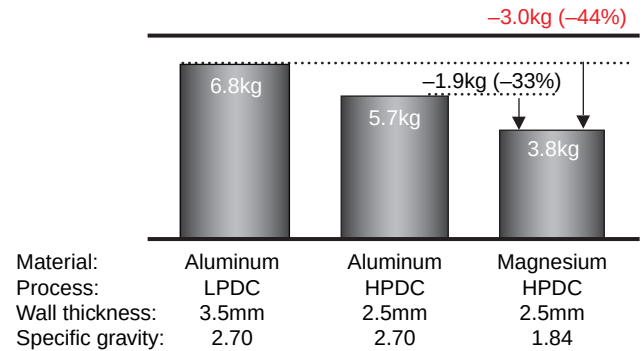


Fig. 14 Comparison of Intake manifold weight

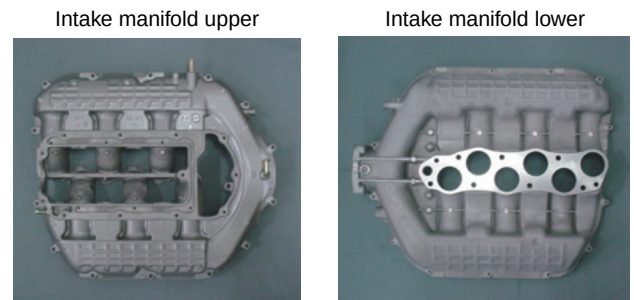


Fig. 15 Appearance of Intake manifold

6. Conclusion

Development of the Mg HPDC intake manifold achieved the following results.

- (1) Three-dimensional seal techniques using Formed-In Place Gaskets, and seal techniques for low rigidity parts using metal gaskets were established.
- (2) Optimizing the shape of low rigidity parts yielded noise and vibration characteristics equivalent to those of mass-produced aluminum LPDC intake manifolds.
- (3) Mg HPDC casting techniques were established for thin walled, large parts, and product quality equivalent to that of Al HPDC intake manifold was achieved.
- (4) Weight reduction of 44% in comparison to the mass-produced Al LPDC intake manifold was achieved.

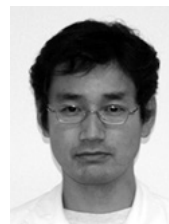
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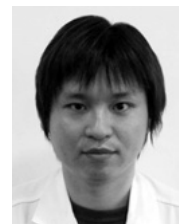
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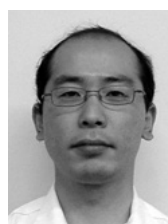
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Application of Non-contact Measurement Technology to Precision Parts

Shigeru TAKAMURA*

ABSTRACT

A non-contact measurement system has been used for the precision analysis of parts in 3D shape evaluation employing CAD data. In recent years, there have been attempts to utilize non-contact measurement systems to respond to demands for 3D shape evaluations of precision parts such as gears. However, there are various optical issues with conventional optical non-contact measurement systems. These issues had to be addressed for application to precision parts, necessitating the development of a new measurement system.

The sensor testing were conducted using a variety of optical sensors to clarify the specific issues arising in the measurement of precision parts with non-contact measurement systems, enabling the optimum optical sensors to be selected. The newly developed system employs two sensors, which enables the achievement of the level of accuracy required for 3D shape evaluation of parts. A multi-axis mechanism which compensates for the weak points of the optical sensors completes the new system.

This paper will demonstrate examples of the use of data measured by the developed measurement system.

1. Introduction

Present-day automobiles are precision machines that demand high performance and high quality. The parts that support these demands also require highly accurate shape measurements. Measurement values of guaranteed accuracy can be obtained with certainty by using a Coordinate Measuring Machine (CMM). The CMM is an effective tool that is equipped with a variety of functions for measuring geometric dimensions. Using it to measure complex surface shapes, however, involves following numerous procedures, from initial setup to the measurement operation itself, so it is a challenge to obtain numerous factors of shape information in a short amount of time using this method.

The advantage of non-contact 3D measurement is that it can obtain large amounts of information in a short period of time in the form of point cloud data. Though this method has been in use for

many years, it must be used correctly in order to obtain highly accurate measurement values by taking advantage of the strengths of this method.

This paper introduces the current state of non-contact measurement technology using optical sensors and the application of that technology.

2. Determining the Measurement Accuracy Required for Precision Parts

Figure 1 classifies measurement systems that are suitable to measurement needs according to their measurement accuracy and measurement range, which are derived from the properties of non-contact sensors and the measurement performance. As indicated by the area enclosed by the dotted line in the figure, however, the

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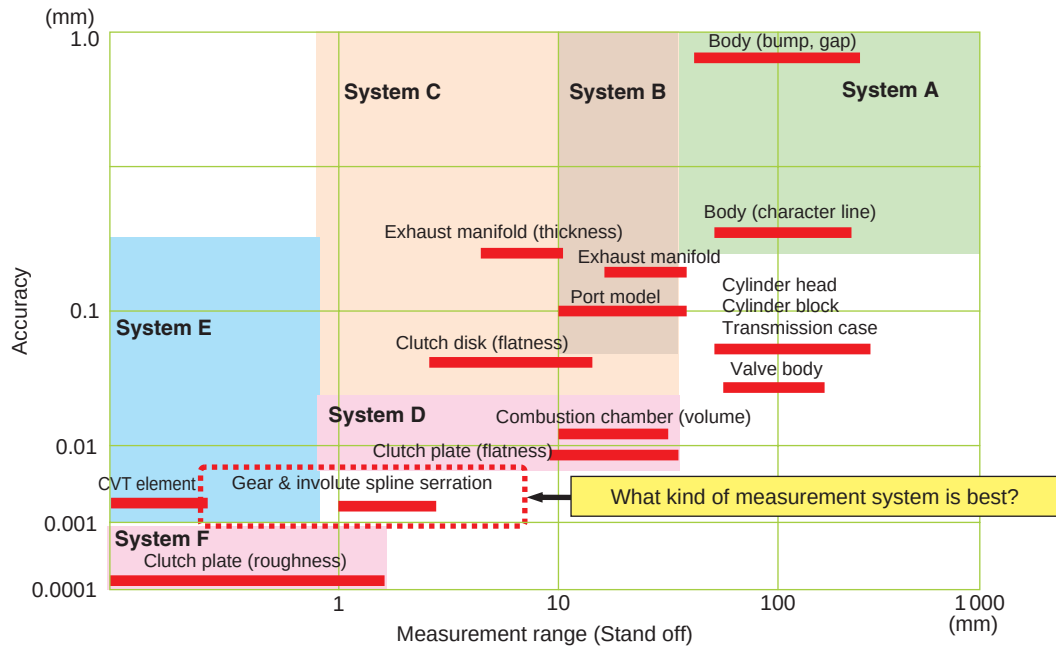


Fig. 1 Classification of non-contact measurement systems

shape measurement of gears and other such precision parts requires a measurement range of 1 – 10 mm and accuracy of about 0.005 mm. There was no appropriate measurement system to meet these requirements.

3. Issues in Conventional Optical Non-contact 3D Measurement

Conventional non-contact measurement systems involve the following kinds of issues:

- (1) Their measurement accuracy is insufficient for precision-machined parts.
- (2) They can only measure objects that have been coated with powder.
 - Accurate measurements of glossy areas cannot be obtained because of halation error and the reflection of environmental light.
 - Accurate measurements of black or dark-colored areas cannot be obtained because the light illuminating the object is absorbed and therefore cannot be detected by the sensor.
- (3) It is a challenge to measure narrow and deep sections where the light does not reach.

Even though attempts to measure an object are made from several directions, detection is a challenge because the light does not reach all the way into deep indentations.

Figure 2 shows parts involving these issues.

Research and development of a non-contact measurement system was initiated in order to address these issues. Gears, which are precision-machined parts, were chosen as the automobile parts to work with.

4. Issues with Gear Parts

Due to their characteristic shape (Fig. 3), gear parts can be expected to present challenging issues. These issues were organized into the following categories:



Fig. 2 Parts with the issues

- (1) Tooth shape: There are shadowed portions because of the irregular shape, and measurements cannot be obtained in areas where light does not reach.
- (2) Part surface: Practically the entire surface is a machined or ground surface, and these kinds of glossy surfaces reflect light.
- (3) Bottom shape: The measuring light causes multiple reflections from the curved surfaces (R sections) of the material, resulting in measurement errors.

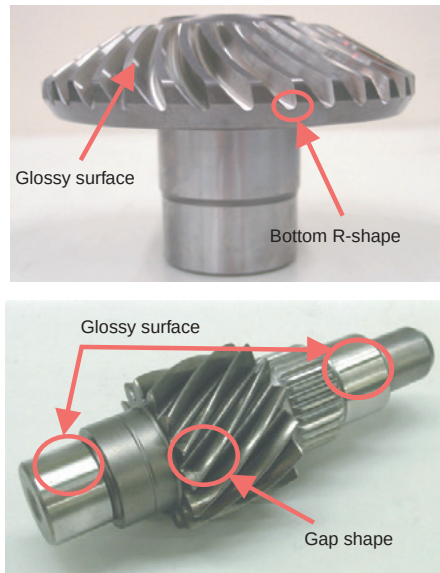


Fig. 3 Shape and surface of gear parts

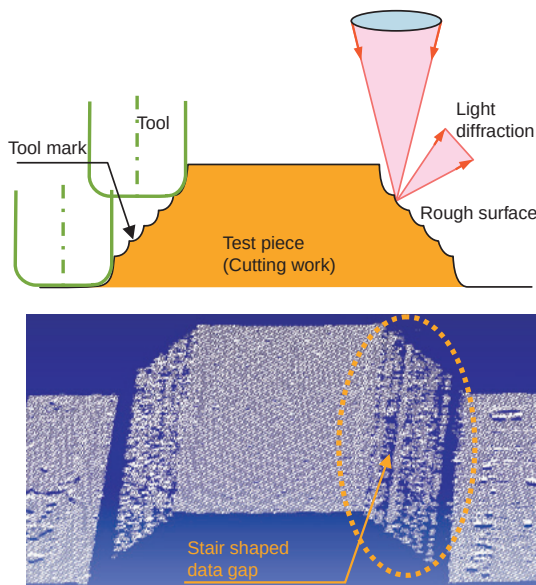


Fig. 4 Data gap caused by optical properties

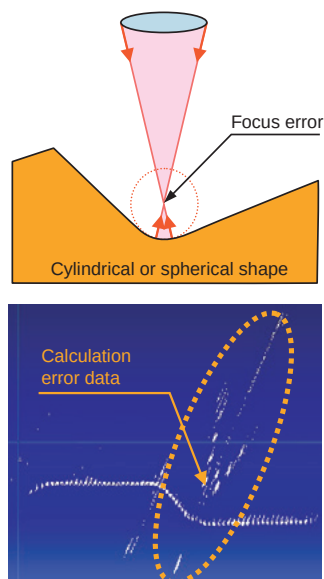


Fig. 5 Data error caused by optical properties

A test piece was fabricated and tests were carried out to verify the measurement values obtained when conducting shape measurements of gear parts.

Figure 4 illustrates how minute irregularities like tool marks in the surface shape cause full reflection from parts of the surface, according to its angle to the measuring light. As a result, stair-shaped gaps in the measurement values thus appear along the tool marks.

Figure 5 shows how cylindrical and spherical surfaces that are glossy result in the measuring light focusing at a distance from the shaped surface. The measurement therefore returns values for a location that are not correct.

In either case, this is a phenomenon that results because of the glossy measurement surface. The conventional method would be to cover the glossy surface with anti-reflective powder. The dispersion in thickness of the powder coating would be added to the precision part, however, so the measurement values would not match the actual object.

When the measured part is later used in testing with actual machines, the residue of powder particles on sliding surfaces and other areas affect the performance. Just like destructive inspection, this makes further use of the part unlikely after measurement.

5. Optical Non-contact Sensors Suited to Use with Precision Parts

Actual gear parts were used to carry out tests with a variety of optical non-contact sensors. The characteristics of the three sensor types that yielded valid results are described below (Fig. 6).

(1) Conoscopic coaxial laser sensor⁽¹⁾

- Strengths: If measurement is carried out in a direction perpendicular to the part surface, measurement values can be obtained with few errors, even from parts with glossy areas.
- Weaknesses: Measurement requires a scanning operation, and shape measurement is time consuming.

(2) Confocal multi-beam sensor⁽²⁾

- Strengths: Measurement in the optical axis direction provides high resolution, and perpendicular to the axis of the light provides highly accurate measurement values (resolution: $0.2 \sigma \mu\text{m}$).
- Weaknesses: The region that can be measured in a single shot is quite limited, so shape measurements are time consuming. (6 mm × 6 mm)

(3) Pattern projection 3D sensor⁽³⁾

- Strengths: The area measured in a single shot is quite extensive, so measurement values can be obtained for a wide area at once. (30 mm × 40 mm)
- Weaknesses: No pattern is projected for areas that are shadowed, so no measurement values are obtained for them.

The light is reflected off of glossy surfaces without projecting a pattern, so no measurement values are obtained.

6. Development of a Non-contact Measurement System

The above test results showed that making good use of sensor strengths would require a mechanism capable of maintaining the proper sensor attitude, relative to the work shape.

Gear parts are evaluated by taking the rotational axis as the standard. Shape data measured by a sensor unit, therefore, does not provide sufficient information to calculate the geometrical basic dimension. Consequently, it is necessary to have a system with a mechanism that can obtain measurement data on the same coordinate system, even for the reference cylinder surface and datum end face used to calculate the geometrical basic dimension. Given the need for use in gear strength analysis, and considering shape reproducibility in the software used for analysis, the tolerance requirement for the measurement values must be about 5 μm . It was decided to develop a new measurement system that would satisfy these requirements.

Sensors (1) and (2) were selected for mounting onto the instrument because of their advantages with regard to accuracy, glossy surfaces, and irregular shapes. Sensor (3) was not selected because multiple reflections occurred in the projected pattern between opposing gear teeth, causing measurement errors.

The measuring device was made multi-axial to enable reproduction of many different sensor attitudes. The unit was equipped with two types of sensors, and the specifications call for a movable 5-axis configuration. This measurement system was developed jointly with Opton Co., Ltd.

In order to obtain shape measurement data with a conventional contact measurement system, first the geometrical basic dimension is measured and the work coordinate system is calibrated. Then trace measurements are carried out.

There are various trace measurement methods. If the starting point and ending point are designated, the measuring system can conduct an auto-scan while computing an estimated shape; another method involves providing the system with a reference shape as nominal data before measurement. Both methods require huge amounts of time and effort, and neither is practical. The advantage of the developed measurement system is the ability to obtain precise shape measurement data simply and conveniently, without

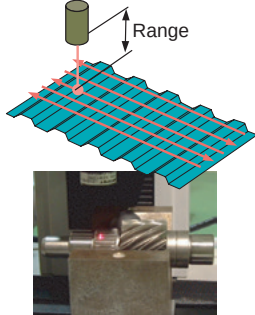
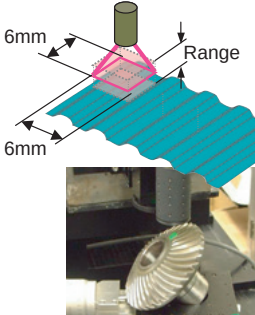
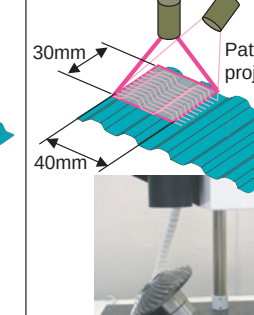
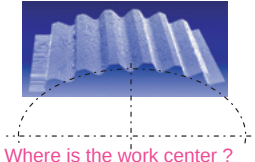
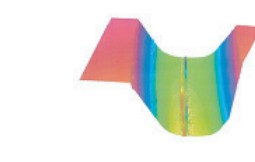
	(1) Conoscopic coaxial laser sensor	(2) Confocal multi-beam sensor	(3) Pattern projection 3D sensor
Type of optical 3D sensor			
Measurement data	 Where is the work center ?	 Where is the work center ?	 Where is the work center ?

Fig. 6 Types of optical measurement sensor

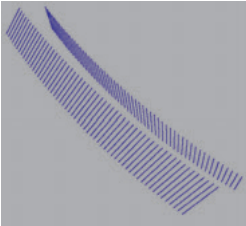
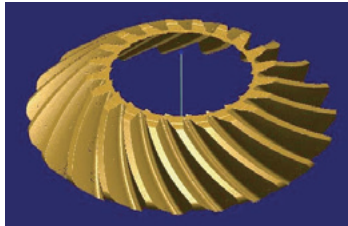
	Conventional system	Newly developed system
Measurement system	 Contact measurement system	 Non-contact measurement system
Measurement data	 Grid point data	 High-density point cloud data

Fig. 7 Conventional and newly developed measurement systems

employing such procedures. Figure 7 shows a comparison of a conventional contact measurement system and the newly developed system.

7. Accuracy of Actual Measurements

The measurement accuracy of the developed measurement system was verified by comparing its measurements with those of a conventional contact measurement system. The gear tooth measurements from a contact gear measurement system and the gear tooth shape measurements obtained by non-contact 3D measurement were superimposed on the same reference shape, and their measurement accuracy was verified in terms of the differences between the data. The procedure was to generate the nominal data line, created for gear tooth measurement purposes, on the 3D model created with point cloud data from the non-contact measurement. The probe of the contact gear measurement system was then traced over this data line, and the measurements were obtained (Fig. 8).

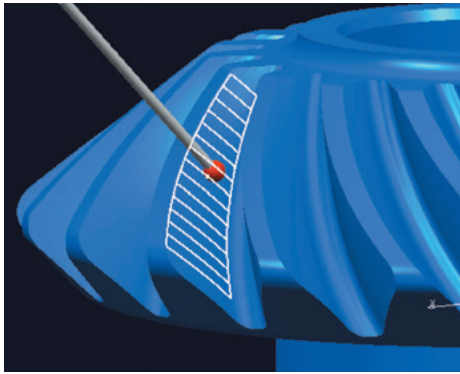


Fig. 8 Nominal data line generated on reverse actual model

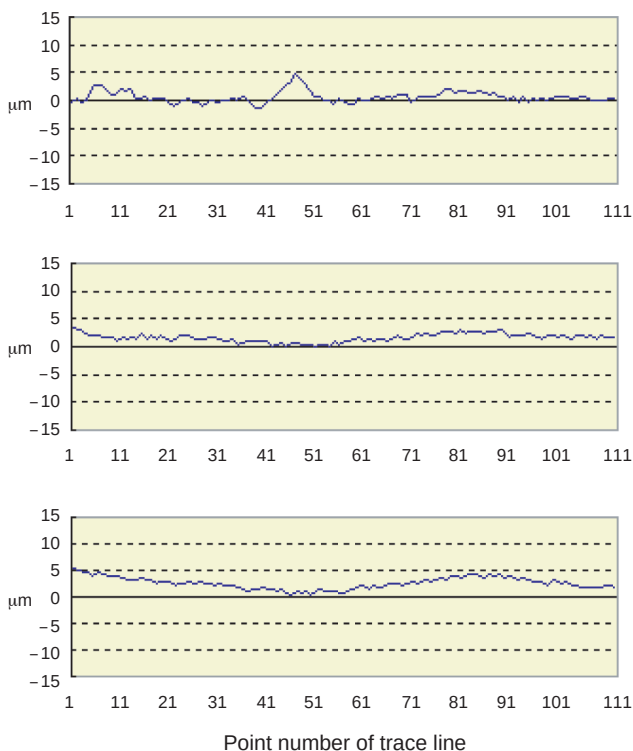


Fig. 9 Difference between measurement data

The differences between these measurement values and the original point cloud in the normal direction of the gear teeth surface were then identified and found to be within 5 μm . The differences were plotted on each tooth height trace line, and the resulting graphs of the tooth shapes are shown in Fig. 9.

8. Application Case Example

The highly accurate, high-density measurement data obtained through non-contact 3D measurement includes an enormous quantity of information about the part features. It is thought that this information will have numerous useful applications.

Here, the point cloud data from the measurement of gear parts were used to generate a 3D model (actual model) that accurately represents the part shape. This was subjected to CAE analysis, and the design specifications were then examined in a trial study. This case is shown in Fig. 10.

The development of automobiles involves the fabrication of large numbers of prototype vehicles of various types, which are subjected to running tests. In the initial stages of development, parts alone are fabricated and subjected to specification studies, without fabricating prototype vehicles. The ability to estimate at this stage whether the performance of a part will pass or fail may help to reduce the number of tests performed on a prototype vehicle. As a result, development time and cost reductions could also be expected.

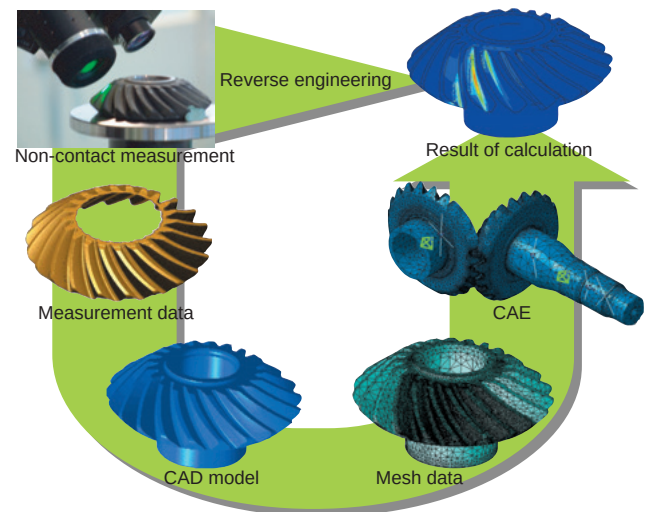


Fig. 10 Reverse engineering

9. Conclusion

The above practical application to precision parts measurement demonstrated the applicability of non-contact 3D measurement. The results achieved and issues for the future are summarized below.

- (1) This method can be used to non-destructively measure objects, leaving them undamaged. Consequently, the method can be used in the quality control and analysis of precision parts, procedures which were previously not possible using contact measurement methods.
- (2) The high density and high volume of measurement values

make it possible to perform shape evaluation that takes the surface as its measurement unit. Thus the method can deal with quality discrepancies that could not be adequately evaluated or analyzed previously by points and lines.

- (3) The developed measurement system makes it possible to measure parts such as gears that have complex shapes presenting a measurement challenge. Also, the measurement system should be applicable to other precision parts apart from gears, as well.
- (4) Recent 3D CAD analysis is able to handle polygon data generated from measurement point cloud data, and measurement results can therefore also be used in CAM/CAE analysis by reverse engineering.
- (5) The accuracy of actual measurement values is still far from adequate for evaluating the contact of gear teeth and other such matters. Therefore, efforts to increase the accuracy of the system will be continued.

Current non-contact measurement technology has been in a period of transition. It appears likely, however, that this technology will be applied in a variety of different fields in the times ahead, and new developments in this technology can be expected to emerge. It is hoped that this paper will be of some help in bringing this about.

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■ Authors ■



Shigeru TAKAMURA

Development of Electronically Controlled Multi-purpose Engine *iGX440*

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Mashu KURATA*

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ABSTRACT

Honda's *iGX440* general purpose engine utilizes electronic control technology to increase ease of use with high fuel economy and low noise performance.

The engine control system employs an electronic control unit (ECU) enabling more precise engine speed control in response to the numerous different operational requirements of a general purpose engine. This has resulted in an engine displaying power characteristics and operational performance meeting the requirements of users in a variety of different applications.

To increase the environmental friendliness of the engine, exhaust emissions have been reduced below regulation levels and fuel economy has been increased. In addition, the noise level of the engine has been reduced to meet Stage 2 requirements of the EU Directive on Noise Emission by Outdoor Equipment. This has been achieved mainly by reducing the noise generated by the mechanical system.

A precoated steel sheet fuel tank, a cold-forged camshaft, and a thermoplastic resin molded ECU have also been employed in the new engine.

1. Introduction

A vertical cylinder and side valve configuration had previously been used for general purpose engines but the GX general purpose engine series developed in the 1980's adopted an inclined cylinder and OHV configuration, providing both a compact structure and high power output in addition to being highly reliable and durable. Therefore, after sales had begun, the GX engines became the standard within the industry particularly for commercial and industrial applications. However, in recent years, various companies have developed inclined cylinder compact configuration engine line ups that can be installed interchangeably with GX series engines and that have equivalent levels of engine performance. From this background, the *iGX440* has been developed, equipped with electronic control, with the concept of "evolution and increased intelligence," in addition to a further evolution in low fuel

consumption and low noise performance. This engine provides ease of use in which work that previously required users to depend on experience and intuition can now be performed "reliably," "quickly," and "simply." This paper introduces the development of the *iGX440*, the first of the "*iGX*" engine series.

2. Development Goals

Performance and functional requirements of general purpose engines include power output sufficient for the work load, ease of starting and use while working, inexpensive use, and low noise performance. In consideration of the above performance requirements, the development goals of the *iGX* engine were set as "enhancement of ease of use" and "reduction of fuel consumption and noise."

Figure 1 shows an external view of the *iGX440*.

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Fig. 1 External view of iGX440

2.1. Enhancement of Ease of Use

To develop a general purpose engine that is easy for users to use, the following concrete targets were established.

- (1) Ensure stable governor characteristics provided in conjunction with mounting on any type of working equipment.
- (2) Achieve operating conditions under which users can operate easily, by automatically changing the engine speed corresponding to working conditions when mounted on work equipment.

2.2. Reduction of Fuel Consumption and Noise

To develop a general purpose engine that is environmentally friendly from the perspective of exhaust and noise, the following items were defined as concrete goals.

- (1) While meeting EPA Phase 2 exhaust requirements, achieve an increase in fuel economy for EPA mode operation.
- (2) Achieve EU noise directive stage 2 standard compliance for power generation equipment levels [LwA97 dB (operating condition: 3 000 min⁻¹, at 3/4 rated power), which is to be implemented January, 2006.

3. Main Specifications

Table 1 shows the main specifications of the iGX440. The engine has a displacement of 438 cm³, uses an OHC driven by an in oil timing belt, and has two valves. In consideration of engine interchangeability in relation to working equipment, the positioning of auxiliary devices such as the air cleaner and fuel tank are the same as those on the GX engine series. In addition, to provide easier use, an adaptive control type electronic governor system⁽¹⁾ is employed as the governor mechanism and the ECU and throttle control mechanism are placed on top of the carburetor. Figure 2 shows a primary sectional drawing.

4. Electronic Control Technology

A centrifugal governor for engine speed control has been used for the previous GX series; however, the iGX engine uses an adaptive control electronic governor that has enabled robust governor characteristics relative to various dynamic characteristics.

Table 1 Engine specifications

Item	iGX440	GX390
Engine type	4-stroke, single-cylinder	4-stroke, single-cylinder
Displacement (cm ³)	438	389
Bore × Stroke (mm)	88.0 × 72.1	88.0 × 64.0
Compression ratio	8.1	8.0
Valve train	OHC 2-valve	OHV 2-valve
Valve inclination angle (deg)	22	0
Lubrication system	Splash	Splash
Ignition system	CDI	Transistorized magneto
Governor system	Electronic	Mechanical
Choke	Automatic/Manual	Manual
Fuel valve	Automatic/Manual	Manual
Starting system	Electric/Recoil	Electric/Recoil
Dimensions: L × W × H (mm)	407 × 505 × 454	380 × 450 × 443

In addition, the ability to change the target engine speed of this electronic governor based on work equipment requirements has enabled enhanced ease of use.

4.1. Enhancement of Ease of Use from Installation of Electronic Governor

Centrifugal governors have a mechanical characteristic of the engine speed being reduced relative to the target engine speed proportional to load increases. By employing an electronic governor, it is possible to achieve operation at a prescribed speed relative to the target engine speed, irrespective of an increase or reduction in load. Output characteristics of a centrifugal governor and of an electronic governor are shown in Fig. 3.

In addition, the ability to change the target engine speed of the electronic governor based on the requirements of work equipment has enabled the realization of various operating modes such as those shown in Fig. 4.

4.1.1. Auto throttle mode

Status confirmation of load on/off by the engine ECU based on the correlation of throttle opening and engine speed of this system

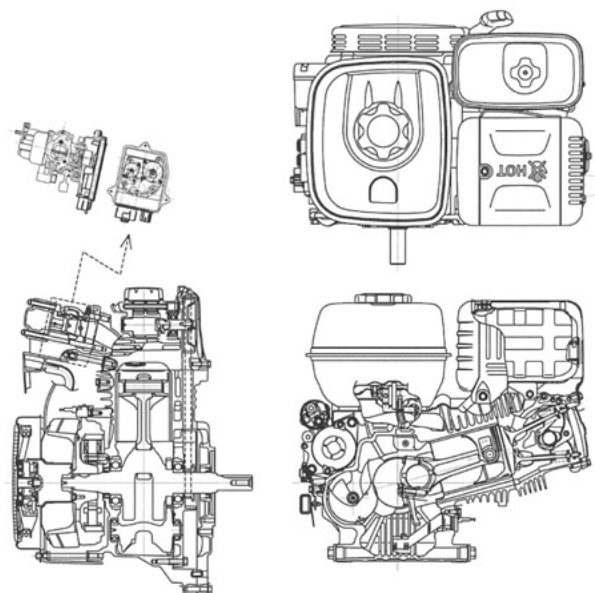


Fig. 2 Engine configuration

enables automatic throttle operation corresponding to the given load on working equipment. For example, when mounted on work equipment where the load is repeatedly turned on and off such as with welders and high pressure washers, by using auto throttle mode, automatic idling can be enabled when the load is turned off, without requiring the addition of a load sensor on the completed equipment.

4.1.2. Communication mode

If this engine is installed on work equipment that uses an electronic control system such as an ECU, it is possible to control engine speed using instructions from the work equipment ECU by making a mutual connection. In this case, as load information is not only received from the engine throttle opening, but also from the work equipment, highly precise load detection becomes feasible, which enables fine tuned engine speed control.

For example, for cases of completed equipment where power requirements for the working portion is provided by the engine and the self-propelled drive system is installed separately, the ECU of the work equipment can receive both load information concerning the working portion from the engine in addition to load information from the drive system, and control engine speed.

In addition, linking control of the working equipment drive section, such as the drive speed using this type of load information, enables construction of a system that provides ease of use for the whole working equipment corresponding to user requirements. The system may include a "working time management operation mode," a "fuel economy prioritizing mode," or an "operation mode prioritizing torque characteristics during transition."

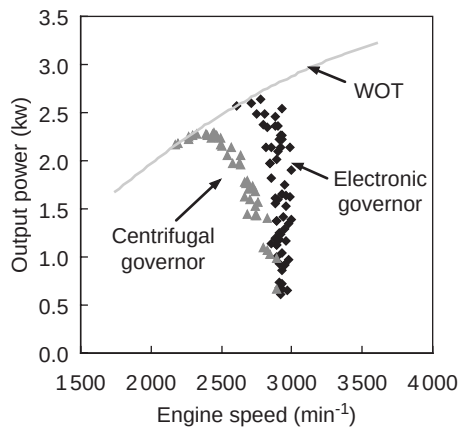


Fig. 3 Governor performance

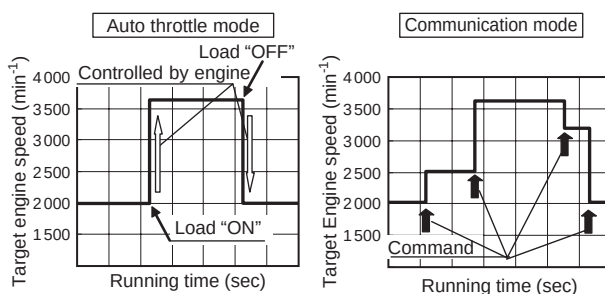


Fig. 4 Operation mode

4.2. Drive System for ECU without a Battery

Many types of work equipment equipped with general purpose engines do not have a battery and even if a battery is used, there are cases whereby the equipment is stored for long periods and the battery loses its charge. In order for the engine to start and for operation to be possible in these cases, a drive system where the ECU does not require a battery is employed.

To make the ECU work without requiring a battery, a power coil that supplies power to the ECU from the rotation of a flywheel with a multi-pole magnet has been established.

When the engine is started and engine speed exceeds 1 000 min⁻¹, power is supplied by the power coil to the ECU which controls the throttle so that the target engine speed is reached. In addition, when stopping, the engine switch is turned off and at the same time, the throttle valve is moved to a wide open position using the power derived until the engine rotation is stopped, preparing for the next time the engine is started. A drive system for an ECU without a battery is shown in Fig. 5.

4.3. Configuration of the ECU and Throttle Drive

As a general purpose engine, it is necessary to place the ECU and wire harness without increasing overall dimensions. Therefore, the throttle drive motor and gears were meshed together, in 3 layers forming a compact configuration and placed on top of the carburetor. In addition, of the elements built on the ECU circuit board, large elements such as capacitors were placed on one side, and the motor was placed in the space opened inside the motor case. This layout has enabled an overall height reduction of 16 mm compared to a box style ECU with uniform thickness. Also, the wire harness from the ECU mounted below the air cleaner to the control panel was placed in front of the fan cover, enabling a configuration where the communication connector discussed later can easily be connected. In addition, by using an airtight construction for the motor case, water resistance and dust resistance of the internal components such as the motor is ensured.

Figure 6 shows a sectional drawing of the ECU.

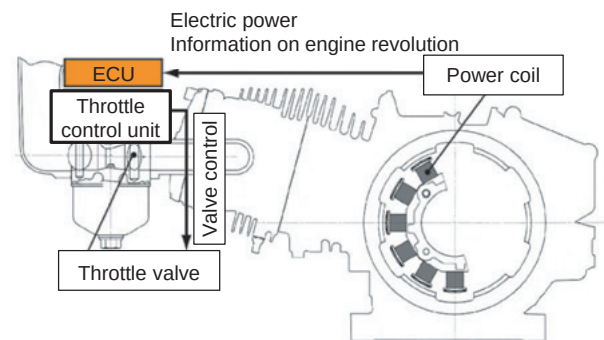


Fig. 5 Engine control system

4.4. ECU Communication System

Engine operation mode and engine speed settings based on the working equipment on which the engine is installed can be written into the ECU and engine operating conditions and operational history can be read from the ECU. Specifically, one configuration allows easy access to the communication connector by removing the electronics panel, and by linking with an exclusive communication unit, writing to and reading from the ECU are enabled. The wiring harness that connects the communication connector and the ECU is shown in Fig. 7.

Assembling a basic mode ECU on the assembly line and having mode parameters written to the ECU at the end of the assembly line eliminated the need for control derived from assembly of ECU.

In addition, when performing inspection of the completed equipment at the plant for shipment, inspection of the completed equipment in operation mode as previously established is enabled by automatic operation communicated from the inspection equipment. At the same time, historical control of operating conditions is conducted enabling enhanced quality control.

In the case of failure diagnosis, utilizing this control information enables prompter issue resolution.

In addition, when dealing with customers who request several types of ECU characteristics, the need to stock multiple engine types or exchange ECU's is eliminated by ability to rewrite ECU.

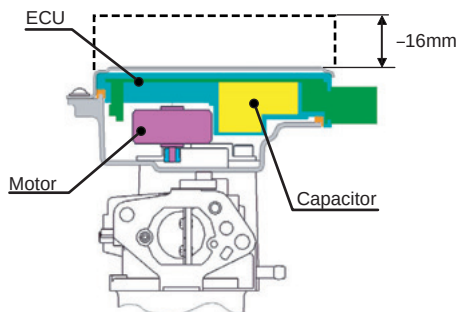


Fig. 6 Throttle motor and ECU

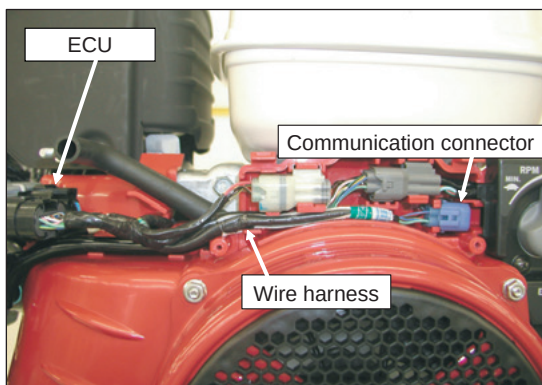


Fig. 7 ECU, wire harness and communication connector

5. Engine Output, Fuel Economy, and Emissions Enhancement Technology

5.1. Engine Output

Figure 8 shows the power output performance of the engine.

Figure 9 shows a comparison of intake valve diameters for this engine and for conventional engines. Based on the current model, by changing the intake valve diameter from 40 mm to 31 mm, the flow speed of the intake gas has been increased. At the same time, the amount of offset of spark plug placement with respect to the bore center was reduced to 9 mm, compared to 23 mm for the GX390, enabling a shortening of the flame propagation distance inside the combustion chamber. These modifications have enhanced knock avoidance performance.

Valve port shapes and valve timing were reviewed to prevent any reduction in fill efficiency caused by the smaller intake valve diameter. As the cross sectional shape of the intake port gets closer to a trapezoid, the cross section area is increased by 10% compared to a circular cross sectional surface. The valve timing overlap period of intake and exhaust valve lift has been shortened, compared to that of the GX390, and the amount of combustion products remaining in the cylinder after the exhaust stroke reduced.

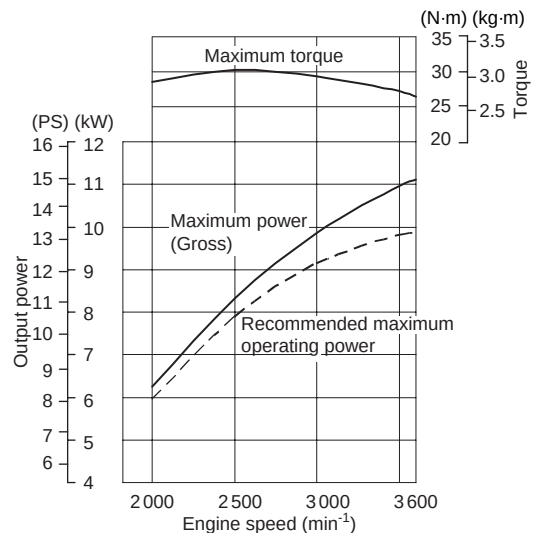


Fig. 8 Engine performance

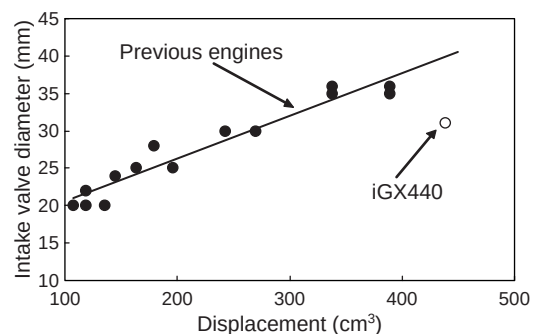


Fig. 9 iGX440 intake valve diameter

5.2. Fuel Economy Enhancement Technology

Figure 10 shows the level of fuel economy and exhaust gas emissions achieved by this engine. By employing the technology shown below, a 15% increase in fuel economy for EPA mode operation has been achieved, without increasing emissions which meet EPA Phase 2 exhaust requirements.

5.2.1 Shape of combustion chamber

The GX390 has a wedge type combustion chamber with a stroke bore ratio of 0.73. In contrast, the iGX440 is equipped with a compact combustion chamber of which its shape is closer to being spherical with stroke bore ratio of 0.82 and a valve angle of 22 degrees. Because of this, the combustion chamber surface/volume ratio (S/V ratio) was enhanced by 6%.

5.2.2. Offset cylinder and piston specifications

In order to reduce piston side pressure friction, an offset cylinder configuration was employed. As shown in Fig. 11, with respect to the crank shaft, the centerline of the cylinder is offset in the direction opposite the direction of piston thrust.

In conjunction with the offset cylinder configuration, piston side pressure on the thrust side is reduced; however, piston side pressure on the anti-thrust side is increased; therefore, while evaluating friction, piston noise and conditions of the piston skirt hitting the side, the amount of offset was set to 7 mm.

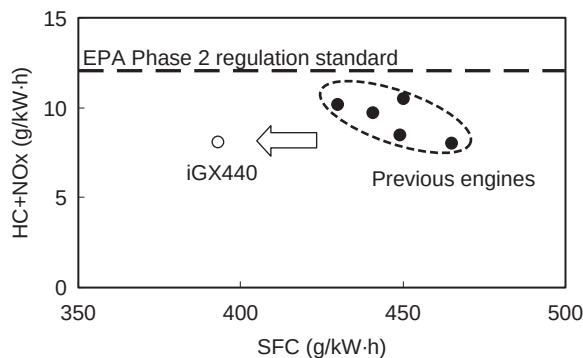


Fig. 10 Results for fuel economy and mass emissions in EPA Phase 2 emission mode

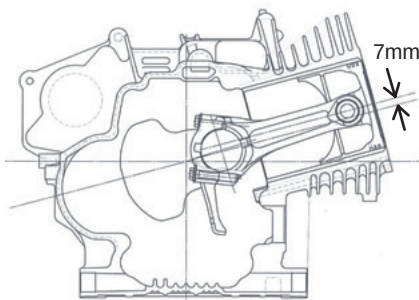


Fig. 11 Offset cylinder

In order to increase the following performance of piston rings with the cylinder, the ring width dimension was reduced and to further reduce friction, the tensile strength of the piston rings was reduced. Compared to the GX390, the top ring and second ring were changed from a width of 2 mm to 1.5 mm and the oil ring was changed from a width of 2.8 mm to 2.5 mm. The total tensile strength of the piston ring was reduced 45% compared to that of the GX390. To address the reduction in heat dissipation from piston rings to the cylinder associated with the piston rings being smaller, the dipper, which is integrated with the connecting rod and flips up oil to cool the piston, was enlarged and the amount of oil flipped onto the back side of the piston increased. With respect to the reduced tension strength of the piston rings, the temperature distribution around the perimeter of the cylinder was made more uniform and the amount of blow-by gas and oil consumption equivalent to that of the GX390 achieved.

5.2.3. Ignition timing control

Ignition timing for this engine is changed based on engine speed and throttle opening. Ignition timing is determined by the ECU using engine speed and throttle opening information, and ignition using CDI is performed.

Figure 12 shows the ignition timing map. These settings help avoid knock and reduce emissions at high loads, reduce fuel consumption at medium loads and reduce piston noise at low loads.

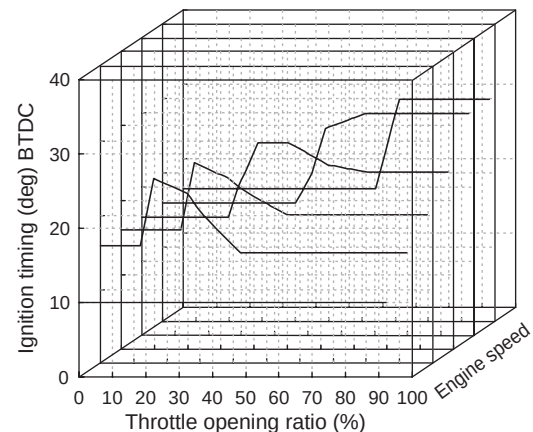


Fig. 12 Ignition timing map

6. Noise and Vibration Performance

6.1. Noise Reduction Technology

In order to maintain cooling performance, fan noise of this engine increased compared to that of the GX390; however, through efforts to reduce mechanical system noise, noise emitted from tank and exhaust noise, a noise level of 96 dB, which is lower than the target LwA value of 97 dB, was achieved. Figure 13 shows analysis results for noise contribution ratios of the iGX440 and GX390.

By using a bearing holder as shown in Fig. 14, the crank shaft receiver distance was reduced by 15 mm with respect to the GX390 and shaft diameter was increased for each part of the crank shaft. As a result, crank pin rigidity relative to in-cylinder combustion pressure was increased by 10%, with respect to the GX390, and crank system noise was reduced. Also, as discussed in section 5.2.2,

by reviewing the piston side pressure reduction effect of utilizing an offset cylinder and of the amount of piston pin offset as well as changing the ignition timing, piston slap noise has been reduced. In addition, the use of in-oil timing belt drive OHC construction has reduced cam gear noise. As a result, the noise energy value of the mechanical system was reduced by 47.5%.

With respect to exhaust noise, the muffler layout is shown in Fig. 15 and placement of a rib on the plating reduced noise emissions. The inside of the muffler is formed by 3 chambers and the employment of a diffuser pipe, from the 3rd chamber to the opening of the exhaust, increased sound transmission loss performance. In addition, a baffle plate was placed at the exhaust opening to reduce flow noise. As a result, the noise energy value of the exhaust system was reduced by 43.8%.

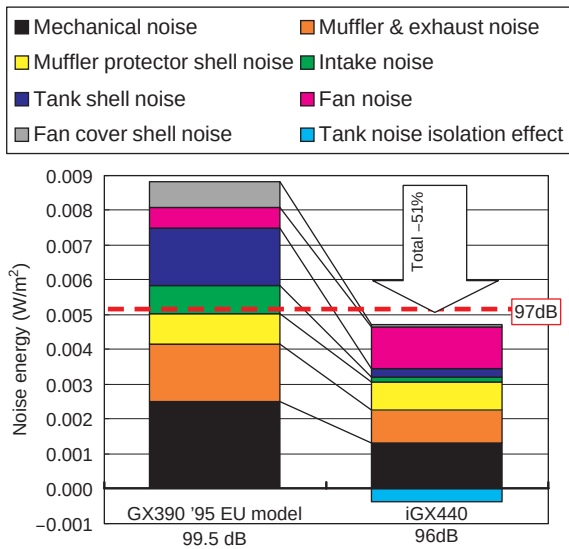


Fig. 13 Effective noise level

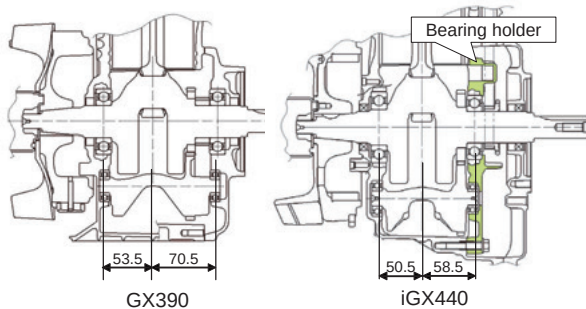


Fig. 14 Crankshaft and bearing layout

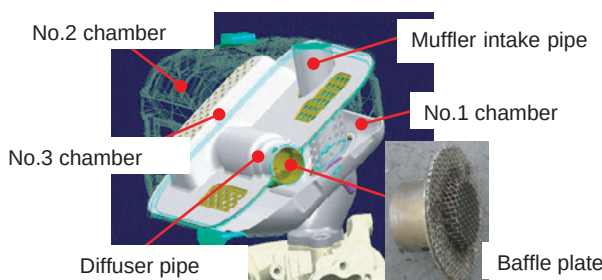


Fig. 15 Muffler structure

By using a rubber mount support construction for the fuel tank, solid borne sound from the engine is shielded resulting in a reduction of noise emissions, reducing the noise energy of the tank by 84.2%. And for cases where the tank is used, the construction exhibits a noise isolation effect for fan and mechanical noise and has enabled a reduction in the overall noise energy by 7%. Through these efforts, with respect to the GX390, overall engine noise energy was reduced by 51%.

6.2. Vibration Reduction Technology

Similar to the GX390, the iGX440 uses a single axis single order rotation balancer. The balance efficiency of the crankshaft counterweight and the rotation balancer was revised so that the 1st and 2nd order vibration of the mounting joint of the base of the engine are minimal. In addition, a nearly uniform distribution of the left to right ratio of the crankshaft counterweight suppressed the couple of force applied on the engine. As a result, as shown in Fig. 16, the vibration acceleration of the engine mount joint was reduced compared to that of the GX390. Figure 16 shows measurement results using rubber mounts for the engine with an engine speed of 3 600 min⁻¹ and at no load.

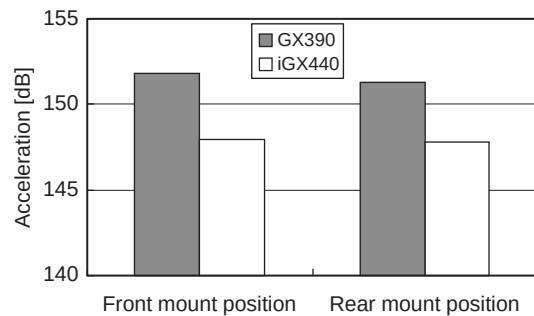


Fig. 16 Vibration performance

7. New Manufacturing Technology

7.1. Pre-Coated Steel Sheet Fuel Tank

With the goal of removing the coating process, pre-coated steel sheets were used for the fuel tank material.

Pre-coated steel sheet is used in general for residential electrical products. To use the material for the fuel tank, additional performance enhancements were required in terms of gasoline contaminants, the appearance after deep draw, the weldability, and rust that develops on the ends of the plate.

Therefore, joint development was performed with a steel plate manufacturer; for gasoline resistance, the hardness of the coating was adjusted to make it difficult for the pigment to permeate the coat layer; for the appearance after deep draw, an appropriate color tone was distributed in two layers of coating to prevent gloss from deteriorating. Also, verification was performed using a press simulation so that plate stretching would be held within 15%. A gloss meter was used to measure gloss at an incident angle of 60° and a gloss greater than 80 was obtained. Clinching was used in place of spot welding. Tank seaming was changed as shown in Fig. 17 and a construction where the end of the plate is not exposed is used to make rust less prevalent.

Figure 18 shows the external view of the fuel tank.

7.2. Cold Forged Camshaft

In order to simplify the manufacturing process using near net forging of the camshaft material, a cold forged camshaft was used.

The point that makes this camshaft feasible is that there is no space in the linkage between the intake and exhaust cam pieces. Therefore, it can be formed from round material and as shown in Fig. 19, separated by the parting line, the punch side produces the intake cam and the die side produces the exhaust cam. This makes it possible to make cam pieces with separate profiles on one object. As the cam profile shape is forged using a multistage parts former, the grinding process was eliminated.

7.3. Thermoplastic Resin Molded ECU

With the goal of enhancing manufacturability through a reduction of the manufacturing cycle time, and reduced size and weight, an ECU with a circuit board protected by a thermoplastic resin molding was used⁽²⁾.

A conventional method for protecting circuit boards includes the casting method of flowing a thermosetting resin into the case for the circuit board. In thermosetting resin casting, injecting resin and the hardening processes take a lot of time. Another characteristic of the casting method is that the depths of cases are made uniform so that they are deep enough for large components such as capacitors, causing a lot of space and materials to be wasted. By selecting a resin that can be molded at lower than normal molding resin temperature and pressure for thermoplastic resin molding, it is possible to perform molding without damaging the electronic circuit board and the following advantages have been obtained.

Forming the mold to the shape of the components enabled the height of the case to be reduced by 20% and weight by 50%, compared to the use of casting. In addition, compared to the 1 to 12 hours that are conventionally required depending on shape, resin hardening requires roughly 10 seconds, enabling efficient production. Figure 20 shows a comparison of the manufacturing processes, and Fig. 21 shows an external view of the ECU.

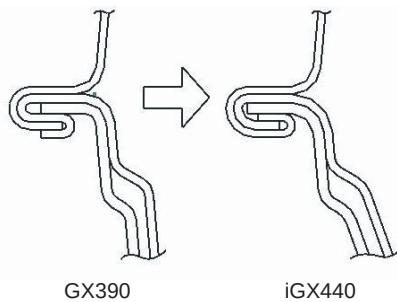


Fig. 17 Cross-section view of fuel tank



Fig. 18 View of fuel tank

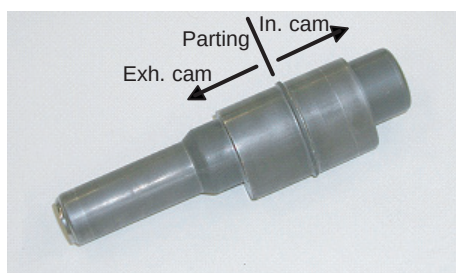


Fig. 19 View of camshaft

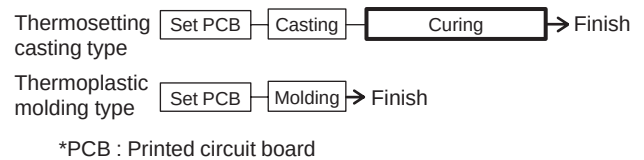


Fig. 20 Comparison of process between casting and molding



Fig. 21 View of ECU

8. Conclusion

A general purpose engine iGX440 was developed using electronic control technology, pursuing further functionality in terms of the ease of use, low fuel consumption, and low noise.

- (1) In order to secure stable rotation for any type of work equipment, an adaptive control type electronic governor was adopted. The operation mode was made changeable according to requirements of the working equipment, and a system that provides ease of use was created.
- (2) An ECU system that does not require a battery was employed.
- (3) The ECU and throttle drive parts were placed in a compact manner, enabling installation without increasing the overall dimensions compared to that of the conventional engine series.
- (4) By connecting an exclusive communication unit, a system where engine output characteristics, operating conditions and

other information can easily be written to and read from the ECU was achieved.

- (5) A 15% increase in fuel economy has been achieved for EPA operating mode without increasing emissions, meeting EPA phase 2 regulations.
- (6) A level of 96 db was achieved for base engine noise, making it possible to meet EU noise directive stage 2 standards. Also, by suppressing the couple of force created by the engine, vibration acceleration of engine mounting joints on the completed equipment has been reduced.
- (7) The following new manufacturing technologies have been implemented in mass production.
 - Pre-Coated steel sheet fuel tank
The coating process was eliminated.
 - Cold forged camshaft
Machining and grinding processes were eliminated.
 - Thermoplastic resin molded ECU
Compared to thermosetting resin casting, weight was reduced by 50% and the height of the case by 20%.
Manufacturing efficiency was increased by shortening the resin hardening process.

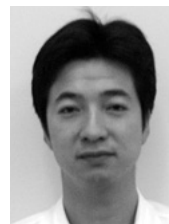
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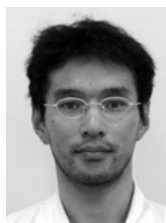
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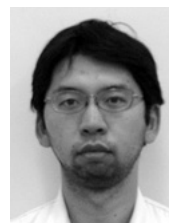
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Application of Thermoplastic Resin Molding to Control Unit of General Purpose Engine

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ABSTRACT

The majority of electrical components utilized in general purpose work equipment are manufactured by potting employing thermosetting resins, taking into account their water, chemical and vibration resistance properties. The research discussed in this paper adopted a different approach, and attempted to mold these components using thermoplastic resins in order to reduce the size and weight of general purpose engines. Thermoplastic resin molding is typically only utilized when manufacturing small electrical components, due to the characteristics of the injection molding process. Molding conditions were optimized to enable the procedure to be employed for a large electronic circuit for engine control, a process which involved the study of resin materials and the layout of the electronic components, in addition to resin flow analysis. Manufacturing components using these optimized conditions enabled a 20% reduction in height and 50% reduction in weight of the control unit, in addition to reduction of the time required for the molding process.

1. Introduction

Electrical parts for general-purpose equipment utilized as work equipment, like general-purpose engines, are required to have high heat resistance characteristics as well as vibration and water resistance, because of their system requirements. Potting solidification using thermosetting resins has been conventionally used for this protection. Two liquid component epoxy resins are frequently used for this application. The electrical components such as Printed Circuit Board (PCB) are put into plastic or metal cases, which are then potted with resin and thermal processing is performed according to prescribed hardening requirements. Thermal processing, which differs according to the type of resin,

requires not only equipment investment and management of the heat curing furnace, but the process also involves a considerable amount of time and effort. For example, the cases need to be placed in the furnace from 3 to 10 hours at 100 °C in a controlled environment.

Products that use epoxy resin potting, for example PCBs, require cases with sufficient depth to cover the tallest installed components and enough resin to fill the cases. Therefore, the amount of resin used naturally increases, making the size and weight of the units larger than necessary and incurring excess costs for the resin.

On the other hand, reductions in size, weight and costs are continual requirements for electrical equipment; therefore, this molding method which focuses on thermoplastic resin

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characteristics, as a method for achieving better environmental performance of electrical equipment, an effective method that would replace thermosetting resin potting was investigated.

This paper describes the knowledge obtained during development and application of a molding method using thermoplastic resin to control units for general-purpose engines.

The thermoplastic resin molding method is one that replaces conventional thermosetting resin potting and has been called “hot melt molding technique” for the last several years. In Japan, the procedure has been used for sealing extremely small circuit boards, sensors, and wire harnesses for residential applications, in order to enhance reliability and reduce costs. This technique, as compared to the temperature and pressure requirements of injection molding using normal thermoplastic resins of 240 – 300 °C and 40 MPa – 130 MPa, requires much lower temperature and pressure conditions, roughly 200 °C and less than 5 MPa, for insert resin molding of circuit boards and sensors.

As injection pressures are low, a simple aluminum mold can be used for the mold, thus greatly reducing the cost of the mold and the injection machine, in addition, the mold tightening device can be greatly simplified. There are also several advantages compared to thermosetting epoxy resin potting, the main procedure used up until now; however, the thermoplastic resin mold method has previously not been used with this large of a circuit board (roughly 100 mm × 70 mm).

2. Basic Structure and Target Requirements

Throttle valve control is one of the features of the new general purpose engine “iGX440” (Fig. 1)⁽¹⁾ that uses a control unit where this method has been applied. The ECU that performs this control is integrated into the top of the carburetor, together with the gear case and the motor that drives the throttle valve. As the ECU is placed on top of the gear case, motor wiring has been simplified. In addition, the endothermic effect of the carburetor has a mitigating effect on the heat of the motor and engine.

The actual layout is shown in Fig. 2. Utilizing this method’s feature of being able to form the external shape of the mold freely, as space can be used effectively, the motor in the gear case and tall parts such as capacitors and transformers in the ECU were mounted so that they mesh together. The gear case is of waterproof construction with a grommet inserted between the aluminum gear case and the ECU, in addition to a metal plate inserted from the outside for sealing and impact protection. This enables the use of

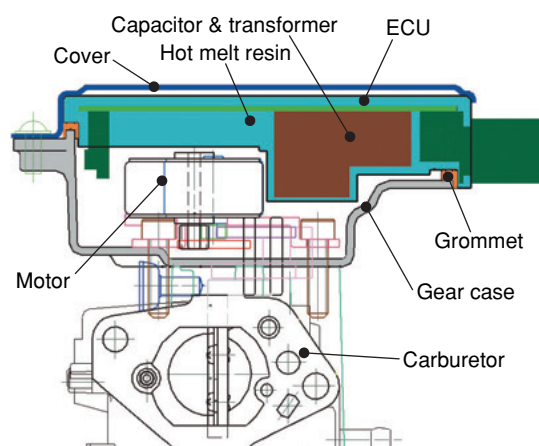


Fig. 2 Layout around ECU

small, non-waterproof parts for the motor and connectors, in addition to adding a dustproof countermeasure to the internal gear.

The following target conditions are required for ECUs that employ this method as described above.

Making the mold thinner (space efficiency).

- (1) Fixation of the circuit board.
- (2) Covering the surface of the circuit board completely (water resistance).
- (3) Ensuring the bonding performance of connectors with surface components (water resistance).
- (4) Maintaining flat seal area surfaces (seal ability).

3. Investigation of Resin Materials

While focusing on the chemical and water resistance properties of the copolymerization polyester resin during development of this method, and in repeating mold shape and environmental performance testing on test circuit boards, requests for changes to resin characteristics were made with a resin manufacturer assisting with joint development, and all of these factors accordingly have contributed to the selection of the resin currently being used. Copolymerized polyester resin has more than an order higher melt viscosity compared to that of the polyamide resin that had been investigated. It was expected that a lot of work would be required for mold and flow control. The fundamental characteristics of the resin used for this method are shown in Table 1⁽²⁾. When the investigation of this method first began, the resin used was different from that of the current resin and was a dymer acid polyamide, a resin that had frequently been used in this method. However, from the results of environmental performance testing required for general purpose electrical equipment, and in evaluating gasoline resistance, chemical resistance, water resistance, and so on, it turned out that an enhancement in performance was required. Investigation was then initiated for a resin that would meet these requirements accordingly.

4. Mass Production Equipment

With respect to the balance between preserving circuit functionality and resin insert molding, control of the resin temperature and injection pressure are important.

First, with respect to resin temperature, focusing on the correlation between resin temperature inside the mold and resin



Fig. 1 External view of iGX440

Table 1 Resin characteristics for hot melt molding

	Properties	Unit	Condition	Spec	Test Method
1	Specific gravity	—	RT	1.05	JIS K 7112
2	Hardness	(Shore A)	RT	70	ASTM D2240
3	Melting point	(°C)	RT	155	DSC
4	Brittle point	(°C)	RT	< -65	JIS K 6261
5	Tensile strength	(MPa)	RT	4.3	JIS K 6251 *1
6	Elongation	(%)	RT	1300	JIS K 6251
7	Flexural strength	(MPa)	RT	0.63	ASTM D790 *2
8	Flexural modulus	(MPa)	RT	9	ASTM D790
9	Shear strength	(MPa)	RT	2.0 <	ASTM D732
10	Coefficient of expansion	(K ⁻¹)		1.9×10^{-4}	TMA
11	Volume resistivity	($\Omega \cdot \text{cm}$)	RT	$1.0 \times 10^{11} <$ (dry condition)	ASTM D257 *3
12	Dielectric strength	(kV/mm)	RT	20 (dry condition)	JIS C 2110

*1 Dumbbell No.3 Tensile rate 500 mm/min

*2 Sample: 100 × 12.5 × 6.5mm Flexural rate 3mm/min

*3 Dry condition: 60°C × 8hours drying under <0.1MPa atmosphere

temperature at the end of the injection gun, the resin temperature at the end of the injection gun is used as a control parameter for mass production. Also, the temperature was established considering solder's melting point and the heat dissipation of the mold.

With respect to the injection pressure, as pressure can have the effect of deforming some of the components, injection at the lowest pressure possible was targeted. With optimization of the injection gate layout and an increase in the injection gate thickness, both methods that have been used for normal injection molding, a reduction in the injection pressure was achieved.

Generally divided, mass production equipment consists of a resin injection portion, a mold tightening portion, a product attachment portion, in addition to a product removal portion, making up a vertical rotary molding machine (Fig.3).

The mold tightening portion holds the circuit board and in order to suppress opening due to resin pressure buildup on opposite sides (upper and lower sides in this case), a specification for generating a tightening force 2 times that of the force generated by the theory⁽³⁾ used for injection molding where:

$$\begin{aligned} &\text{The required mold tightening force (ton)} \\ &= \text{average pressure in resin (kgf/cm}^2\text{)} \times \\ &\quad \text{filling projection area (cm}^2\text{/1000).} \end{aligned} \quad (1)$$



Fig. 3 Hot melt molding system

The product attachment and removal parts have specifications that conform to general injection molding equipment.

5. Optimization of Molding Requirements

5.1. Resin Flow Pre-investigation

The item for which thermoplastic resin molding will be used is a PCB. In molding, the following items must be observed while maintaining its shape.

- (1) The heatproof temperature of any of the components must not be exceeded.
- (2) The resistance pressure of any of the components must not be exceeded.
- (3) Any changes in the shape of components are to be within permissible limits.

The circuit board was larger than the one conventionally used, with tall components (roughly 15 mm) such as a capacitor for power supply smoothing and a transformer for ignition power supply on the board making experiential estimation of the flow difficult and resin flow analysis imperative.

Flow patterns that show flow conditions, molding temperature and pressure are flow analysis results that have been obtained (using Toyobo's CAE flow analysis service). An actual example of a flow pattern is shown in Fig. 4.

In the above analysis example, in the center of the lower area (inside gear case), resin filling is late; therefore, the pressure balance between the upper and lower areas is upset and it is anticipated that deformation of the circuit board will occur. In temperature analysis while performing molding, filling temperature results identified the areas on the circuit board at which temperature increases are recognized, and it has specifically been confirmed that these temperatures are below the solder melting point. Using pressure analysis during molding enables the estimation of required resin pressure settings when filling.

By performing the analysis above, the circuit board layout, mold shape and injection gate shape are determined more efficiently, shortening the time and reducing the costs required to actually make the mold.

5.2. Heat and Pressure Resistance and Molding Confirmation

Although temperatures and pressures are low compared to normal resin injection molding, the affect these factors have on electrical parts populated on the circuit board can not be ignored, when comparing to sealing with conventional thermosetting epoxy resin potting. As an example, temperature and pressure resistance for all of the electrical parts used in the current engine control unit were determined. Characteristics are summarized below.

- (1) Explosion proof construction of large electrolytic capacitors makes them weak under pressure, and the external resin film has poor heatproof performance.
- (2) The heatproof temperature of the film used for the film capacitor is lower than that of the melting point of solder.
- (3) Parts other than this capacitor will basically not have an issue as long as the temperature is held under that of the melting point of solder.

Temperature comparisons and verifications were performed while molding each electrical component on the circuit board with the standards above.

In order to prevent obstruction of the flow path injection gate, an extremely thin, wire diameter thermo-couple was used to measure the temperature of several locations on the circuit board. The temperature measurement locations were based on the results of the resin flow analysis, and with the location nearest the injection gate used as standard, the surface temperatures of vital components were measured. The temperatures at all locations were below each

standard and thus verified that there were no issues (Fig. 5).

On the other hand, stress applied to the circuit board during molding was measured. From this measurement, it can be seen that stress is applied on the circuit board at the time of initial injection pressure application and holding pressure application that comes after the completion of resin filling (Fig.6).

5.3. Bonding Performance of Resins with Circuit Board and Electrical Parts

Judging from the characteristics of the resin being used, a certain level of bonding performance with the circuit board and electrical components was expected; however, with the actual molding, from the temperature of each of the areas during injection molding, the resin temperature was found to be slightly lower than necessary to obtain sufficient resin bonding performance. For the current circuit board, the resin temperature could not exceed the melting point of solder which is 180 °C. Due to the small temperature difference with the melting point (155 °C) of the resin being used, strong bonding performance, similar to that obtained when molding is performed using thermosetting epoxy resin, can not be expected. It was necessary to work with the external shape of the mold, in order to ensure water resistance of the ECU.

5.4. Ensuring Mold Quality

As a step to establish target molding requirements for mass production equipment, of the test circuit boards, dummy PCBs populated with large components (large electrolytic capacitors, transformers, connectors, power transistors, LED, film capacitors) were used to determine limitations on conditions where molding can be performed using resin temperatures and initial injection pressures as parameters.

Concerning high resin temperatures, all components were populated on a similar circuit board that was made operable and whether or not the circuit board could be operated was also tested (Fig.7).

With the current manufacturing method, in order to maintain the quality of molding, it is necessary to prevent the melted viscosity of used resin with a large effect from obstructing the resin flow path. The following items must be considered.

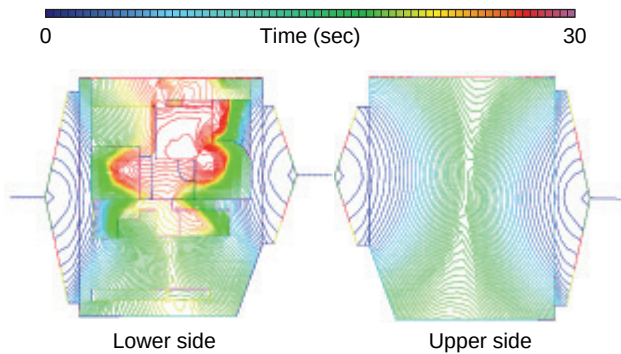


Fig. 4 Flow pattern

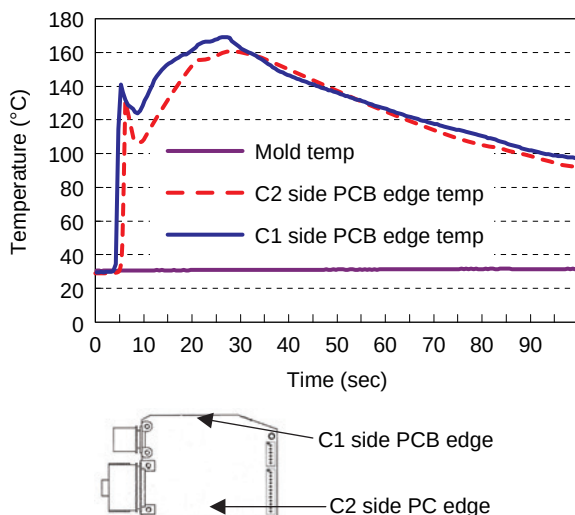


Fig. 5 Thermal transition on PCB surface and resin

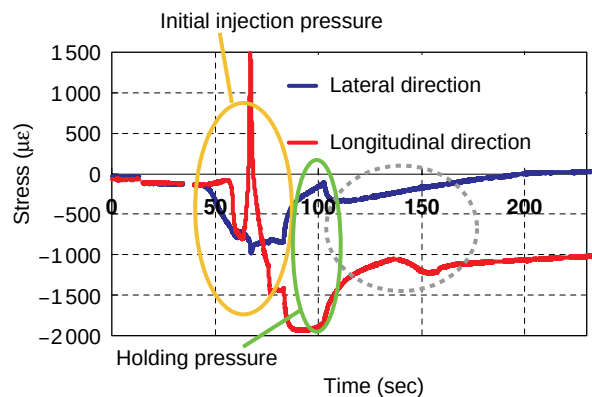


Fig. 6 Stress on PCB with resin injection

- (1) Resin temperature: while keeping electrical part and solder temperature requirements in mind, use as high a temperature setting as possible.
- (2) Use part shapes and layouts designed so that pressure is not concentrated in the same location.
- (3) Design such that resin flow is not concentrated from the resin flow gate.
- (4) In order to maintain somewhat of a balance of resin flow above and below the parting line, do not place components in a manner that would severely obstruct the flow to one side.

Also, for a molding shape, bonding between the resin and parts is not all that strong; therefore, using a configuration where resin covers the parts and not one with resin just being a simple adhesion surface with parts populated on the circuit board is desirable. Figure 8 shows an external view after molding.

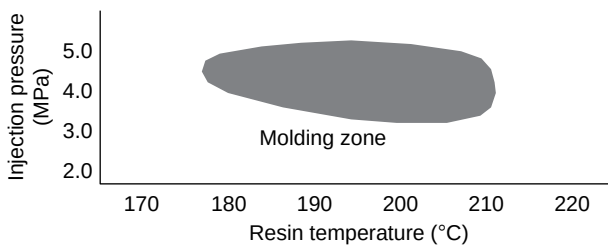


Fig. 7 Practical molding conditions



Fig. 8 Outside view of a hot melt molded ECU

6. Durability and Reliability

This manufacturing method differs from that of the normal thermosetting epoxy resin potting method. In this method, pressure is applied for the molding while resin is flowing. Therefore, there was concern for the effect of high temperature resin flow and that of molding pressure stress on solder areas of the parts that have a lot of electronic elements such as the ECU. In addition, this is the first time that an ECU was installed directly on a general-purpose engine and high reliability and durability requirements of the ECU were needed. In order to provide such features, temperature and vibration-related testing was mainly performed.

With this method, as the effect to the solder junctions was expected to be large, thermal shock testing was performed extensively for verification. A sample was taken at each prescribed cycle of the thermal shock test and solder condition of CPU attached portion was confirmed using x-ray CT equipment. The results are as shown in Table 2. There were no cracks that developed in the solder joints due to the expansion and contraction of the resin relative to thermal stress. In order to understand limitations, the verification of solder conditions was performed for a number of cycles exceeding the test requirement; however, development of cracks large enough to obstruct circuit operation was not seen.

Table 2 Solder conditions of micro-processor and capacitor

Function	Micro-processor	Capacitor
500 Cycles	Crack growth rate 0% 	Crack growth rate 0%
700 Cycles	Crack growth rate 13% 	Crack growth rate 0%
1 000 Cycles	Crack growth rate 23% Minimum pass on soldering junction	Crack growth rate 0%

In order to establish overall reliability in a short amount of time, a temperature humidity cycle and vibration energizing test was conducted, but no issues were found.

Changes in vibration, cooling and heating, and humidity do not affect the populated electrical parts or the circuit board, and it was also verified that this unit has equivalent reliability and durability as a unit manufactured using a conventional thermosetting epoxy resin potting method. This unit fully satisfies requirements of the molded unit that is used for a general-purpose engine.

7. Conclusion

A thermoplastic resin molding method, that replaces the conventional thermosetting epoxy resin potting method used for environmental performance of electrical equipment installed on motors, was developed such that it can be applied to a control unit used for general-purpose engines.

In addition, the following knowledge has been obtained in conjunction with this research.

- (1) If a PCB is molded using this manufacturing method, as the resin temperature can not exceed the heatproof temperature of electrical components or the solder melting point, while on the other hand, the temperature needs to be set as high as possible to provide mold quality, resin temperature and pressure conditions must precisely be controlled.

- (2) Resin flow changes dramatically with shape and placement of components as well as shape and location of the injection gate, and as this easily affects molded product quality, a CAE resin flow analysis performed prior to making test samples is extremely effective.
- (3) Unit height has been reduced by 20%, weight has been reduced by 50% and a 2-order enhancement in molding work processing time has been achieved.

As a sealing method for electrical equipment, starting with control units for general-purpose engines, application of this method will be expanded.

Acknowledgements

In conjunction with this research, especially with development of the resin material and CAE flow analysis, we express our thanks to TOYOBO Co., Ltd. and to all others who have assisted.

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Governor System for General Purpose Engine Using Adaptive Control Theory

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ABSTRACT

General purpose engines are required to keep the engine speed constant, and are therefore equipped with governor mechanisms. However, the engines are applied to various applications, and the dynamic characteristics of the controlled engine changes according to different operating conditions. So it is necessary for a conventional mechanical governor to change the spring rate and tune the idle speed; similarly, gains must be tuned in electronic governors using PID control. To enhance robustness, an adaptive control system that uses a self-tuning regulator has been constructed. Applying elliptical gears to the mechanism for driving the throttle valve, it was possible to realize both high resolution near the closed position and a reduction in operating time of the throttle valve.

1. Introduction

In automobiles and on motorcycles, people estimate conditions using the speedometer and their five senses and then use the throttle (accelerator pedal) for control. In comparison, products with general purpose engines, most notably the small single cylinder engines (in generators, lawn mowers, tillers, snow throwers, water pumps, etc.) have a mechanism that automatically adjust the throttle opening of the carburetor, either because it is not possible for a person to respond fast enough to sudden load changes or for unattended operation. This is a mechanism for keeping the engine speed constant. Many products use mechanical feedback control which applies James Watt's centrifugal force governor mechanism⁽¹⁾ such as the one shown in Fig. 1. This mechanism detects changes in engine speed using the centrifugal force of a counterweight, and a linkage adjusts the throttle opening of the carburetor. The throttle opening becomes steady at where the centrifugal force and governor spring are balanced. However, there is mechanically a difference

in engine speed between idle and when a load is applied.

In recent years, use of inexpensive high performance microprocessors has enabled the introduction of governors that are electronic (hereinafter; electronic governors) as generators, instead of mechanical governor mechanisms. The electronic governors drive the throttle valve shaft using a motor and converge engine speed to a target value by PID control, etc. However, as a governor for general purpose engines where dynamic characteristics of the controlled object change, due to the fact that the engines are used for several different applications, and the environmental use conditions such as temperature and pressure change, it is not sufficiently robust and gain adjustments adapted to changes in dynamic characteristics are required. In the present research, an electronic robust governor using an adaptive control theory has been developed, which has used Self-tuning Regulator control^{(2),(3)} (hereinafter, STR control) for changes to dynamic characteristics of the controlled object in order to control the engine speed of the small-single cylinder engines.

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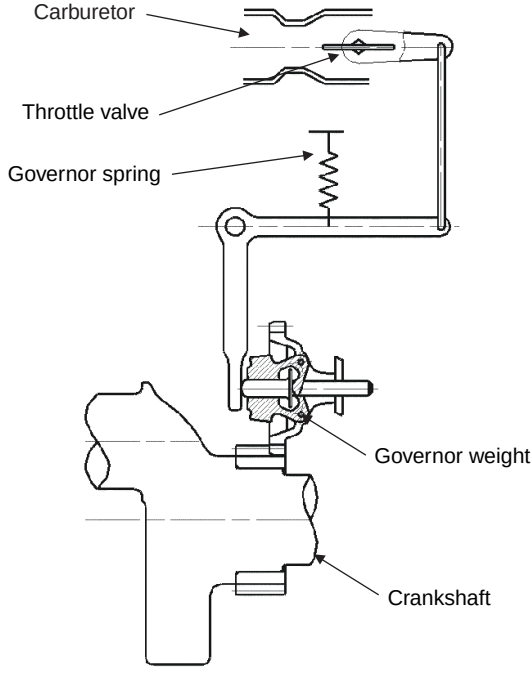


Fig. 1 Centrifugal type governor

2. Implementation of STR Control

2.1. Engine Modeling

Figure 2 shows a simplified engine model where throttle opening is used as an input. Considering the block diagram of a general purpose base engine that uses a carburetor, mass air flow G_a and mass fuel flow G_f are determined by the throttle opening θ_{th} and engine speed NE when the air fuel ratio is fixed. A mean effective pressure P_{mi} generated by combustion of the intake mixture is converted to engine speed by the crank linkage system. The engine load characteristics, connected to this crank linkage system of general purpose engines, differ with the engine application (for generators, lawn mowers, tillers, snow throwers, water pumps, etc.). Even when the engine is used on the same application, the load characteristics differ with the moment of inertia of the attachment. In order to control an electronic governor, the throttle opening angle θ_{th} is selected as the input and engine speed NE selected as the output, with the area surrounded by dashed lines is used as a single block model for the controlled object.

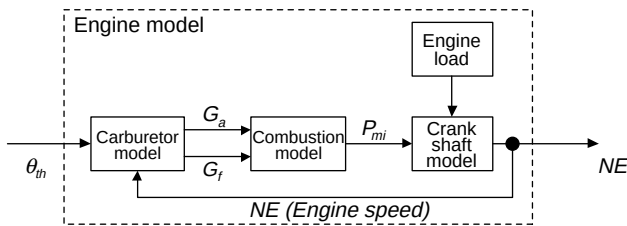


Fig. 2 Block diagram of engine model

A step response of the engine speed relative to step input of the throttle valve is shown in Fig. 3. With the premise of performing adaptation control using the ARX (Auto-Regressive Exogenous) model⁽⁴⁾, a minimum square model (Equation 1), with throttle opening θ_{th} as the input u and engine speed NE as the output y , modeling was performed while focusing on the order and lag-type.

$$A(q^{-1})y(k) = q^{-1}B(q^{-1})u(k) + w(k) \quad (1)$$

$$A(q^{-1}) = 1 + a_1q^{-1} + \dots + a_nq^{-n} \quad (2)$$

$$B(q^{-1}) = b_0 + b_1q^{-1} + \dots + b_mq^{-m} \quad (3)$$

$a_1 \dots a_n, b_0 \dots b_m$: Model parameters

$w(k)$: White noise

In Eq. (1), with the target engine speed defined as y_m the output error is calculated using the following equation.

$$e_1(k) = y_m(k) - y(k) \quad (4)$$

The control object is automatically adjusting the throttle opening u , the input, allowing the engine speed y , the output of the controlled object, to become equivalent to the target engine speed y_m . Here, the load fluctuation is a primary unknown parameter. In addition, the whole parameters of the engine combustion model, including the load, are estimated as the unknown parameter by sequential calculation.

2.2. Controller Design

With the introduction of the Diophantine equation⁽⁵⁾ and configuring Eq. (1) in the following manner, variance of the output error can be minimized, enabling achievement of the control object.

$$u(k) = \frac{1}{b_0} \{y_m(k+1) - \bar{\theta}^T \bar{\zeta}(k)\} \quad (5)$$

Here,

$$\bar{\theta}^T = [b_1 \dots b_m, h_0 \dots h_{n-1}] \quad (6)$$

$$\bar{\zeta}(k) = [u(k-1) \dots u(k-m), y(k) \dots y(k-n+1)] \quad (7)$$

However, in the actual system, as the parameter is unknown, sequential estimation is performed using parameter estimation law below.

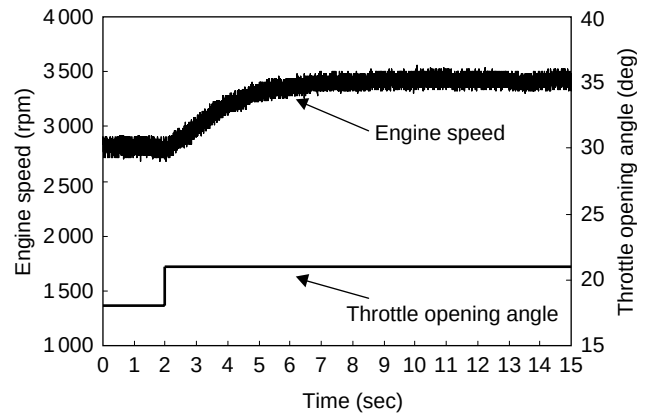


Fig. 3 Step response

$$\hat{\theta}(k) = \hat{\theta}(k-1) - \Pi(k-1) \zeta(k-1) \varepsilon_1(k) \quad (8)$$

$$\Pi(k) = \frac{1}{\lambda_1(k)} \Pi(k-1) - \frac{1}{\lambda_1(k)} \left[\frac{\lambda_2(k) \Pi(k-1) \zeta(k-1) \zeta^T(k-1) \Pi(k-1)}{\lambda_1(k) + \lambda_2(k) \zeta^T(k-1) \Pi(k-1) \zeta(k-1)} \right] \quad (9)$$

Here,

$$\theta^T = [b_0, \bar{\theta}^T] \quad (10)$$

$$\zeta^T(k) = [u(k), \bar{\zeta}^T(k)] \quad (11)$$

Here $0 < \lambda_1(k) \leq 1$, $0 \leq \lambda_2(k) < \lambda_1(k)$, $\Pi(0) = \Pi^T(0) > 0$

If the unknown parameter converges to the true value, the system becomes the minimum variance control using the configuration of Eq. (5). Therefore, a robust controller for a general purpose engine speed governor with respect to dynamic characteristic fluctuations can be attained.

3. Mechanism Characteristics

An outline form of the electronic governor mechanism installed on iGX440⁽⁶⁾ is shown in Fig. 4.

The following items were considered in tandem with the application of an electronic governor to a general purpose engine that is used in several applications and various environments.

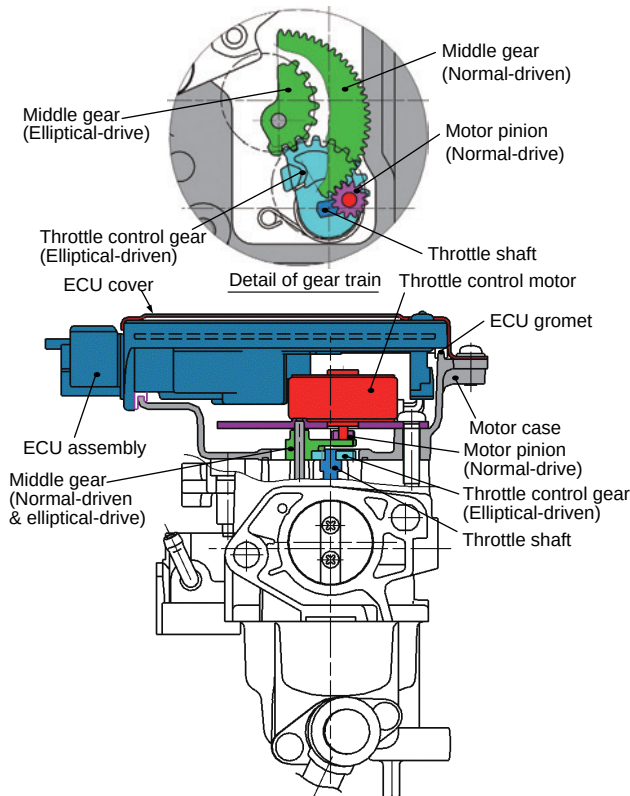


Fig. 4 Carburetor assembly

Table 1 Characteristics of actuator

	Stepping motor	DC motor	Solenoid
Control method	Open loop	Feedback	Feedback
Position control	Number of pulse	Feedback by position sensor	Feedback by position sensor
Speed control	Pulse rate	PWM control	Current control
Power supply	Constant voltage	PWM control	Current control
Life	Bearing life (long life)	Brush life (short life)	Bearing life (long life)

- Feasibility of operating without a battery.
- Reducing sensors to a minimum to help achieve reliable durability.
- Use of a simple drive circuit enabling use of a compact ECU.

Stepper motors, DC motors, and solenoids were considered as the actuator for the throttle drive. Table 1 shows characteristics for each of these actuator types. Accordingly, in order to have stop position control without requiring sensors, a simple drive circuit, and long life, a stepper motor was selected as the actuator for driving the general purpose engine throttle.

However, due to all the various characteristics required of a general purpose engine, just applying a stepper motor is insufficient and the following technical methods have additionally been applied.

- Non-linear reduction to satisfy required characteristics of the throttle
- Compact layout for integral construction ECU and carburetor
- Cost and size minimization of electronics in order to maintain weather and dust resistance

Details concerning these three points are described below.

3.1. Non-linear Reduction Mechanism

The relationship between throttle opening θ_{th} and mass air flow G_a controlled by a butterfly throttle valve is shown in Fig. 5. Independent of the engine speed, within the limits of $\theta_{th} < 30$ degrees, the amount of fluctuation of G_a with respect to the throttle opening is sensitive; therefore, in order to stabilize engine speeds when in idle and slight throttle opening states, control resolution must be fine.

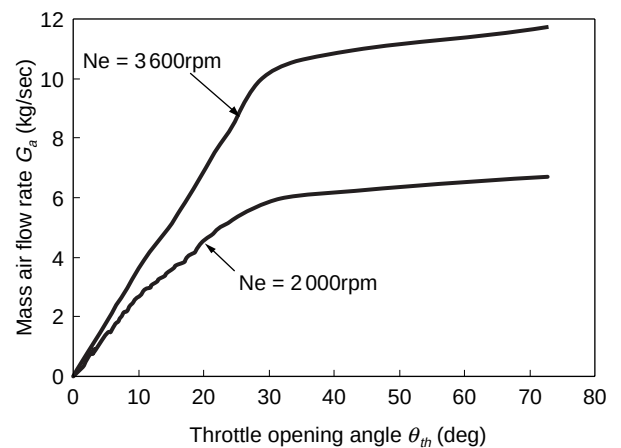


Fig. 5 Mass air flow rate

A Permanent Magnet type stepper motor (hereinafter; PM type stepper motor) has been selected for use in this mechanism as the actuator to control the throttle valve. The angle control capability of a PM type stepper motor is determined by the pitch of the rotor magnet and pole stator and there are physical limits with the enhancing resolution of the motor itself. Therefore, enhancement of resolution is generally provided by gear reduction. If the reduction gear train is formed using normal gears and resolution is set fine to meet slight opening requirement characteristics, throttle opening and closing speed is reduced due to the stepper motor characteristics involved.

General purpose engines frequently have 0% to 100% and 100% to 0% sudden load fluctuations. In order to enhance the ability to minimize engine speed fluctuations during load fluctuations, it is necessary to increase the throttle opening and closing speed.

Increasing the speed of input pulses is a method for increasing motor speed; however, as shown in Fig. 6, torque generated by the motor is reduced, leading to a reduction in surplus torque relative to that required to drive the throttle, which is itself a cause for loss of synchronization. On the other hand, if resolution is set loosely, matching the requirements of a large throttle opening, resolution at slight opening is insufficient and can cause hunting.

In order to allow for these conflicting requirements, elliptical gears⁽⁷⁾ have been utilized. Figure 7 shows a linkage that uses elliptical gears.

With θ_m , θ_{th} as the centers of the ellipses, and respective focus F_m , F'_m , F_{th} and F'_{th} of gears of the same shape with long diameter $2a$, and short diameter $2b$ and with F_m and F_{th} as rotational axes, $|A_m F_m| = \rho_m$, $|A_{th} F_{th}| = \rho_{th}$ and θ_m , θ_{th} as the rotation angles.

$$\rho_m = \frac{r}{1 + \varepsilon \cos \theta_m} \quad (12)$$

$$\rho_{th} = \frac{r}{1 - \varepsilon \cos \theta_{th}} \quad (13)$$

$$\rho_m + \rho_{th} = 2a \quad (14)$$

$$\text{However, } r = \frac{b^2}{a}, \varepsilon = \frac{\sqrt{a^2 - b^2}}{a}$$

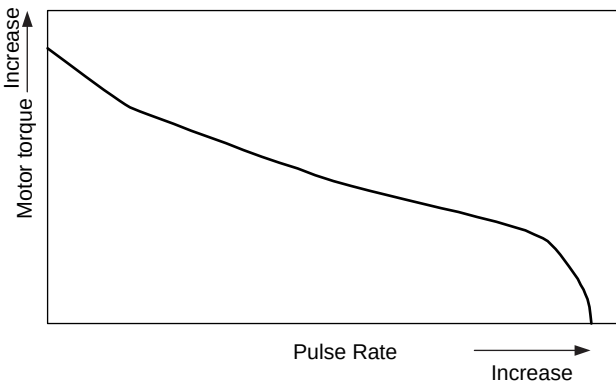


Fig. 6 Characteristics of motor torque

Therefore,

$$\theta_{th} = \cos^{-1} \left\{ \frac{(1 + \varepsilon \cos \theta_m) \cdot (2a - r) - r}{[2a \cdot (1 + \varepsilon \cos \theta_m) - r] \cdot \varepsilon} \right\} \quad (15)$$

The relationship of θ_m and θ_{th} with $\varepsilon = 0.25, 0.425$, and 0.65 is shown in Fig. 8.

When the elliptical gear θ_m rotates through a differential angle $d\theta_m$, θ_{th} rotates through an angle $d\theta_{th}$. This leads to the following relationship becoming equal.

$$\rho_m \cdot d\theta_m = \rho_{th} \cdot d\theta_{th} \quad (16)$$

However, as $\frac{d\theta_m}{dt} = \omega_m$ and $\frac{d\theta_{th}}{dt} = \omega_{th}$ rewriting of Eq. (16) becomes

$$\omega_{th} = \frac{\rho_m}{\rho_{th}} \omega_m \quad (17)$$

With ω_m operating at uniform speed it can be seen that when $\rho_m < \rho_{th}$, $\omega_m > \omega_{th}$ and if $\rho_m > \rho_{th}$, $\omega_m < \omega_{th}$. In other words with O_m as the drive gear and O_{th} as the driven gear (throttle drive), with the gears set in full closed position as shown in Fig. 7, the amount of change of θ_{th} with respect to θ_m near where the throttle is fully closed is insensitive; however, the amount of change of θ_{th} with respect to θ_m near wide open throttle is more sensitive. This characteristic is determined by the eccentricity ε of the ellipse. The

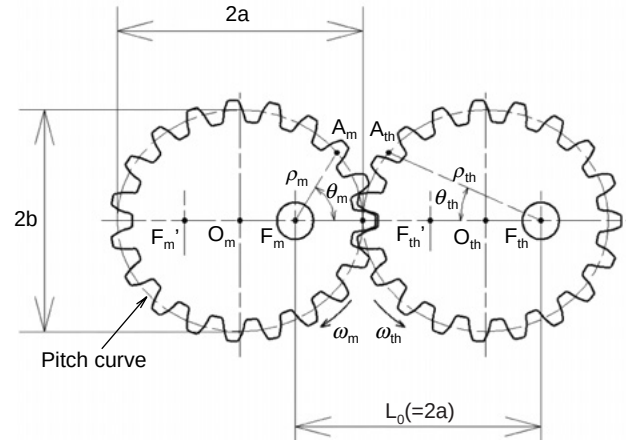


Fig. 7 Elliptical gear train

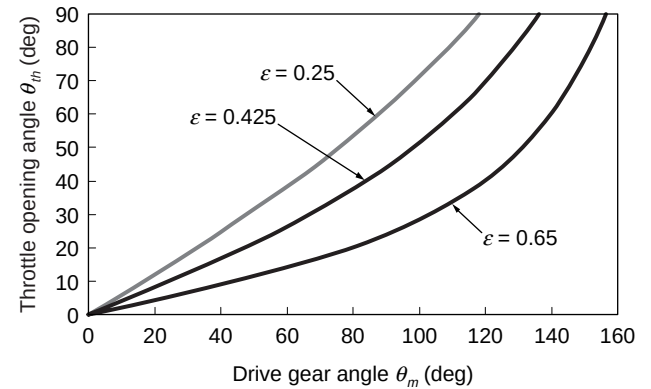


Fig. 8 Characteristics of angle with difference in elliptical gear eccentricity

relationship of $d\theta_{th}/d\theta_m$ with $\epsilon = 0.25, 0.425$, and 0.65 is shown in Fig. 9. It can be seen that increasing the value of ϵ leads to an enhancement in resolution for low throttle valve openings.

The calculation of resolution with respect to the throttle opening with ϵ at maximum permissible with a given layout is shown in Fig. 10. By using a non-linear resolution mechanism such as this, a linear relationship between motor step pulse and mass air flow G_a such as shown in Fig. 11 is achieved. If the non-linear relationship is corrected mechanically, linearization processing using the control system is no longer needed.

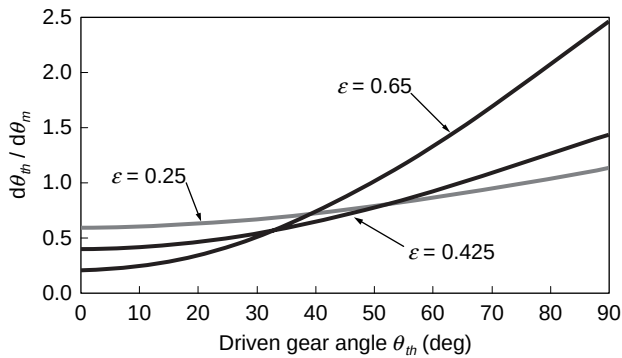


Fig. 9 Characteristics of angular velocity with difference in elliptical gear eccentricity

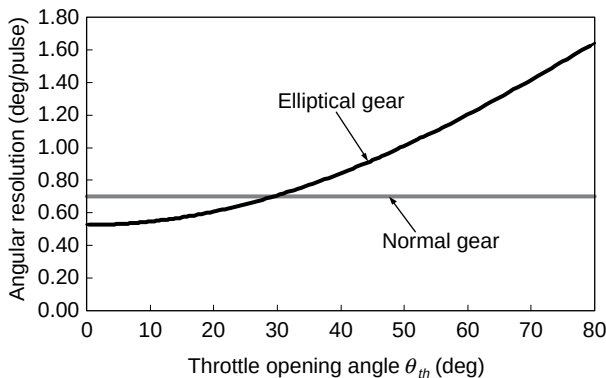


Fig. 10 Characteristics of angular resolution

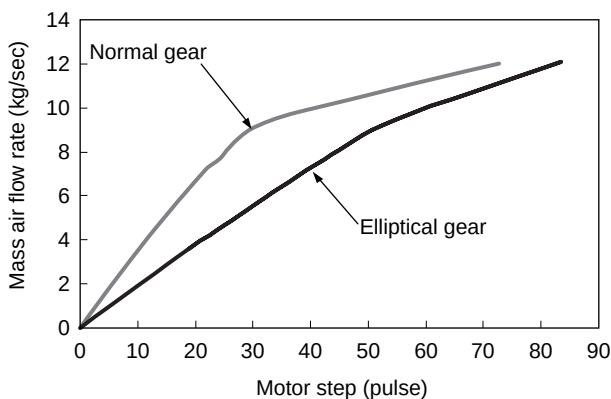


Fig. 11 Relationship between motor step and mass air flow rate (Engine speed = 3600rpm)

A comparison of a normal gear reduction with minimum resolution equivalent to the resolution curve shown in Fig. 10 and the number of steps needed from where the throttle is fully closed to it becomes wide open is shown in Fig. 12. The number of steps used for an elliptical gear mechanism is 35% less than the number used for a normal gear.

In conjunction with use of an elliptical gear mechanism, the relationship between throttle drive torque requirements and actual throttle drive torque is shown in Fig. 13. Because the throttle drive torque is inversely proportional to resolution, elliptical gear mechanism throttle drive torque is large when the throttle is closed and becomes smaller when the throttle is wide open. Throttle torque was designed to be at least 1.5 times that of maximum throttle drive torque requirements. During sudden accelerations or decelerations, where the throttle suddenly moves from being wide open to one where it becomes all the way closed, a larger than normal negative pressure is applied to the throttle valve causing resistance to the movement of the throttle shaft. The fact that maximum torque of this mechanism is generated where the throttle valve is closed contributes to increased toughness with respect to the resistance of the throttle shaft movement.

3.2. Compact Layout

As a general purpose engine, it is necessary to position the ECU and electronics for the throttle control mechanism, etc. so as not to increase overall dimensions. The motor and gear case combination is placed on top of the carburetor and the motor and reduction gears are placed inside the silhouette of the carburetor. Through centralizing the placement of large components on the circuit board in the ECU on the motor case, compared to the conventional method

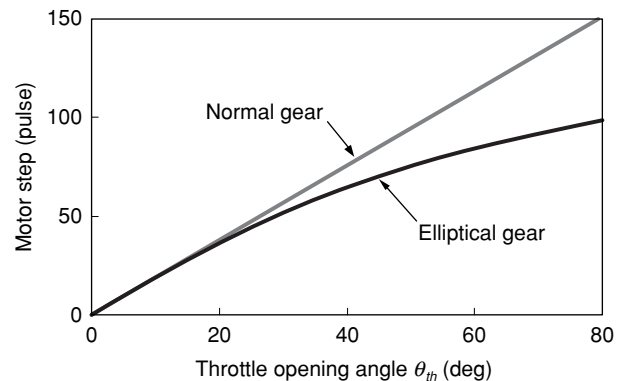


Fig. 12 Relationship between throttle opening angle and motor step

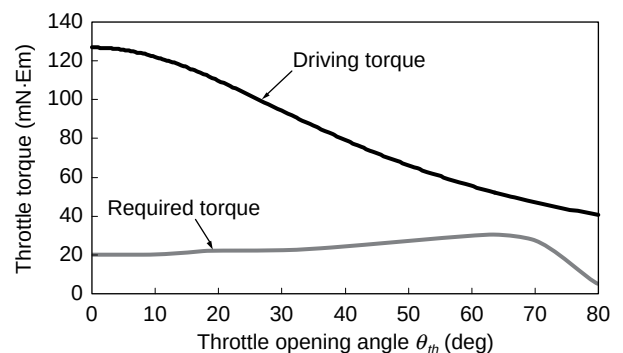


Fig. 13 Characteristics of throttle torque

of molding an ECU in a uniform depth case, the overall height has been reduced by 16 mm.

Furthermore, as the connector that connects the ECU and motor is placed in the same case, this contributes to a reduced connector size, enabling a simplification of the wiring, and contributing to a compact layout.

3.3. Water and Dust Resistance

As shown in Fig. 4, sealing is provided by an O-ring that is placed between the aluminum motor case and the carburetor and by a grommet between the ECU and motor case, making the motor case and ECU air tight. As electronics are placed inside of these units, water resistance and dust resistance thus become unneeded for individual electronic parts, enabling a reduction in size and cost.

4. Installation in Equipment Results

With a test engine GXV160 (Table 2) installed in a lawn mower, Fig. 14 shows results comparing governor characteristics between its use in a conventional mechanical governor and in an adaptive control type electronic governor. The conventional mechanical governor has steady state error, where the engine speed fluctuates depending on the load. With respect to this point, when using an adaptive control type electronic governor, it can be seen that engine speed is uniform, independent of the load.

In addition, a test engine GX160 (Table 2) installed on a cylindrical power generator was equipped with an elliptical gear as shown in Fig. 10, and with a normal circular gear, and their idle stability were compared. For this comparison, the same STR control forming an electronic governor was used, the electric load was set to no load and the target engine speed set to 1 400 rpm. Results for the use of the elliptical gear mechanism are shown in Fig. 15 and results where the normal circular gear was used are shown in Fig. 16. Through the use of the elliptical gear mechanism, as compared to the use of a normal circular gear mechanism, idle stability has been enhanced.

An iGX440 engine that uses an adaptive control type electronic governor system is shown in Fig. 17. With this engine installed on a cylindrical power generator, output power characteristics relative to different engine speeds are shown in Fig. 18. The convergence capability of engine speed fluctuations during rated load intermittent

Table 2 Test engine specifications

Model	GX160	GXV160
Type	Air-cooled, 4-stroke, single-cylinder gasoline (OHV)	Air-cooled, 4-stroke, single-cylinder gasoline (OHV)
Displacement (cm ³)	163	163
Dimensions L × W × H (mm)	304 × 362 × 335	415 × 359 × 354
Dry weight (kg)	15	14.6
Bore × Stroke (mm)	68 × 45	68 × 45
Max. power (kW / rpm)	4.1 / 3600	4.1 / 3600

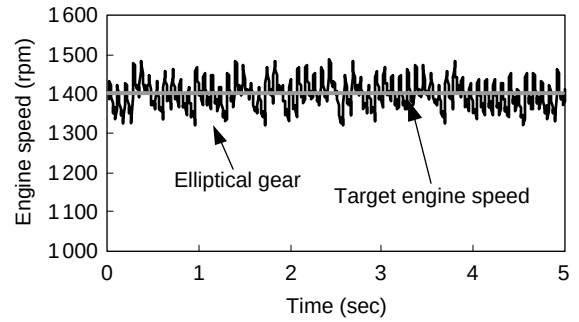


Fig. 15 Idle characteristic with elliptical gear

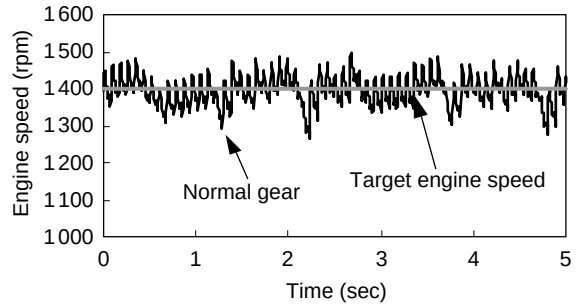


Fig. 16 Idle characteristic with normal gear

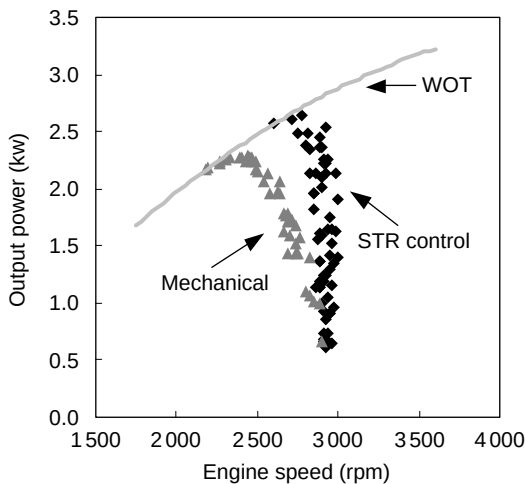


Fig. 14 Dynamic characteristic of governor



Fig. 17 External view of iGX440

operation (from 100% to 0% and 0% to 100%) is shown in Fig. 19.

As shown in Fig. 18, engine speed fluctuation from idle to that of a wide open throttle for an arbitrary engine speed setting was held to within 2%, demonstrating that the strength of an electronic governor, high engine speed stability, was obtained. In addition, as shown in Fig. 19, engine speed fluctuations relative to the set speed during rated load intermittent operation show a favorable convergence capability of 8%, which is within the original goal of 10%. Both stability of engine speed for no loads and convergence of engine speed during load fluctuations have been achieved.

Furthermore, an adaptive control type electronic governor was installed in commercialized products (lawn mowers, snow throwers, pressure washers, etc.) to perform practical durability testing for confirmation. Tuning was not required for the individual products where favorable operation characteristics were obtained from the beginning through to the end of the durability testing verifying the validity of this system.

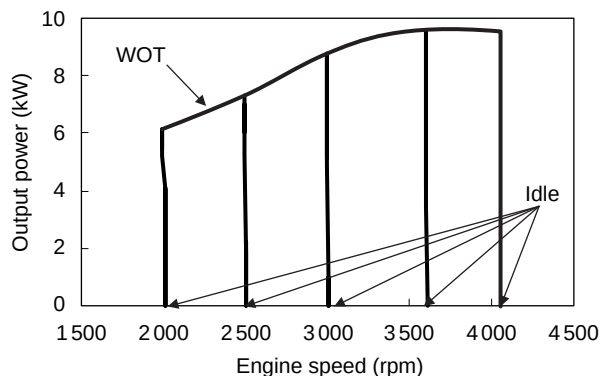


Fig. 18 Static performance of governor

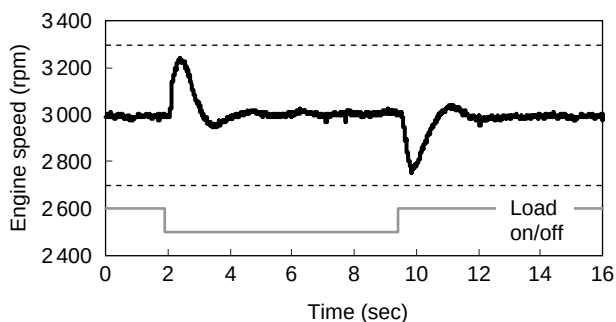


Fig. 19 Dynamic performance of governor
(Target engine speed = 3000rpm)

5. Conclusion

Through applying adaptive control and a non-linear gear mechanism to the governor of a general purpose engine, the following features have been achieved.

- (1) A robust governor relative to fluctuations of dynamic characteristic of the engine, the controlled object, has been developed, allowing use in a general purpose engine, for which applications are unlimited.

- (2) The realization of both a high resolution near the closed position and a reduction in operating time of the throttle valve made it possible to control the engine from idle condition to sudden load change condition.
- (3) Through linearization of the relationship between motor step and mass air flow, linearization of the input signal in control calculations becomes unnecessary, leading to a simplified control system.

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Development of Low Emission Diesel Engine with DPF System

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ABSTRACT

Honda has developed a turbo-charged common rail diesel engine for the European version of its CR-V Sports Utility Vehicle (SUV). During this development measures have been implemented to address the issue of the SUV body having high running resistance. The promotion of homogeneity in the air-fuel mixture through the employment of a shallow dish combustion chamber and the reduction of the diameter of the fuel injection nozzle holes has enhanced combustion and reduced engine-out Particulate Matter (PM). In addition, the use of a highly efficient EGR cooler has increased the volume of EGR. Combined with the achievement of optimal EGR gas temperature control by means of an EGR cooler bypass system, these factors together have enabled the achievement of a reduction in NOx emissions.

The enhancement in exhaust emission performance resulting from these innovations has made the CR-V the first SUV which is compliant with the 2005 European diesel exhaust emissions standards (EURO-IV).

A diesel particulate filter (DPF) is also employed in the CR-V. Optimization of the multi-fuel injection system, which is able to inject up to four times per cycle, control of the DPF temperature via the use of a variable geometry turbocharger (VGT), and intake control by means of the throttle valve have all resulted in reliable DPF regeneration performance.

These various emission reduction technologies have resulted in the achievement of PM emissions less than one-tenth of those stipulated by EURO-IV.

1. Introduction

Concern for environmental issues is growing worldwide. Year by year the demand for diesel engines increases, as they provide both a means of reducing CO2 emissions and high dynamic performance. On the other hand, emission regulations for diesel engines are becoming more stringent, and in some regions, there is

movement beyond the current regulations towards favorably treating vehicles fitted with diesel particulate filters (DPF) that can remove much particulate matter (PM). The demand for vehicles with DPF is rising steeply.

In 2003, Honda developed and introduced its first proprietary diesel engine for the Accord^{(1),(2)}, which has been well received in the market. Based on the diesel engine for the Accord, a new engine has been developed for the CR-V Sports Utility Vehicle (SUV)⁽³⁾.

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The CR-V is the first SUV that complies with the 2005 European diesel exhaust emissions standards (EURO IV) and at the same time achieves both high output and low fuel consumption. Combining this engine with the newly developed DPF system has made it possible to greatly reduce PM emissions. This paper reports on the engine technologies that comply with EURO-IV and on the DPF system.

2. Development Goals

In the development of this SUV engine, the following targets were set.

- (1) To be the first SUV to comply with EURO-IV.
- (2) To ensure top-level fuel economy and dynamic performance for an SUV diesel vehicle.
- (3) To create a DPF system (To reduce PM emissions).

3. System Outline

3.1. System Introduction

Figure 1 shows the i-CTDi engine system with DPF, as employed in the CR-V.

With the adoption of the DPF, throttle valves, exhaust temperature sensors, and pressure difference sensors were newly added, to enhance DPF regeneration performance.

Table 1 shows the main engine specifications. The fuel injection system is the BOSCH 2nd generation common rail system, which was also used in the Accord. In order to increase EGR gas, cooling performance of the EGR cooler was enhanced. In addition, an EGR cooler by-pass system was provided to reduce HC and CO gas emissions due to excessive cooling of the EGR gas at low water temperatures.

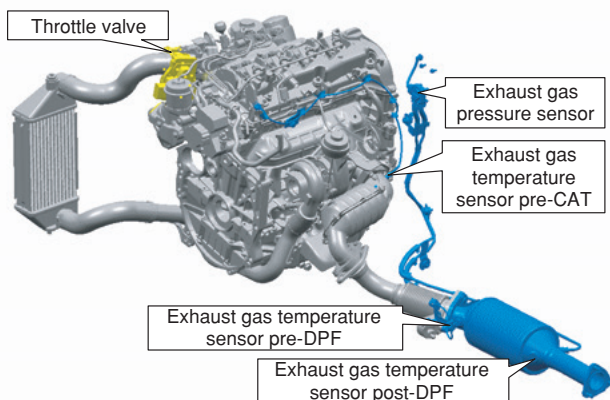


Fig. 1 Schematic diagram of i-CTDi engine with DPF system

Table 1 Main specifications of engine

Engine type	DOHC-4VLV
Displacement (cm ³)	2 204
Bore × Stroke (mm)	85 × 97.1
Compression ratio	16.7
Piston (Bowl diameter) (mm)	52.8
Bowl shape type	Shallow dish
Injector (mm)	φ 0.134 × 6holes
Fuel injection system	Common rail system
Maximum Power (kW/rpm)	103 / 4 000
Maximum Torque (Nm/rpm)	340 / 4 000
After treatment	DOC + DPF
EGR system	EGR cooler with bypass system

4. Engine Unit

4.1. Enhanced Combustion

With an SUV body having large running resistance, the amount of fuel that is injected increases with the increase in engine load. This expands the range where the local air-fuel ratio in the combustion chamber is small, and combustion worsens, leading to an increase in PM emissions. At the same time, the combustion temperature rises, leading to an increase in NO_x emissions. Accordingly, in working towards EURO-IV compliance, it is necessary to simultaneously reduce PM emissions through combustion enhancement, and to reduce NO_x emissions through advances in the EGR system^{(4), (5)}.

4.1.1. Reduction in PM emissions

PM emissions are thought to be caused by poor distribution of the mixture having a low local air-fuel ratio, and its expansion in the combustion chamber. Therefore, the following modifications were made, to homogenize the fuel spray and to distribute the spray near the bowl wall surface.

- (1) The diameter of the injector nozzle holes was reduced (micro-sac type 6-hole nozzle) to promote finer fuel spray particles.
- (2) The piston bowl diameter was expanded, and modified to a shallow dish, to promote wider distribution of fuel spray.

Figure 2 shows the results of a CFD simulation air-fuel mixture distribution analysis with the above injector specifications and combustion chamber shape changes. It is clear that reducing the diameter of the holes in injector nozzles promotes finer fuel spray particles, enabling homogenization of the mixture within the bowl.

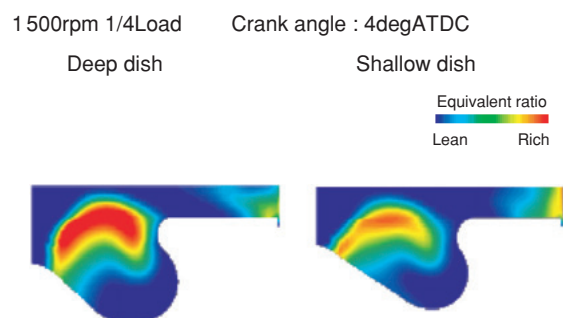


Fig. 2 Effect of combustion system modifications for mixture distribution

Moreover, expanding the bowl diameter promotes wider fuel spray distribution near the bowl wall surface, mitigating poor distribution.

The combustion chamber shape selected as a result is shown in Fig. 3.

These measures promoted air-fuel mixture homogenization, and combined with swirl control that responds to engine speed and load using a linear variable swirl control valve (SCV), PM under a fixed level of NO_x are dramatically reduced. Figure 4 shows a comparison of emission performance.

4.1.2. Reduction in NO_x emissions

NO_x emissions tend to increase accompanying increases in engine required load. In order to reduce NO_x emissions and minimize PM emissions, this engine increased EGR gas amounts by enhancing the cooling performance of the EGR cooler. However, there is the issue of increasing HC and CO emissions when the combustion temperature drops excessively due to the drop in the EGR gas temperature when cooling the engine. To address the matter, a new mechanism bypassing the EGR cooler was adopted. The EGR system is shown in Fig. 5.

In order to minimize the increase in HC and CO emissions due to a drop in combustion temperature when the water temperature is low, an EGR cooler bypass valve can be switched to the bypass side, and a drop in the EGR gas temperature is avoided. Then, by switching the valve to the cooler side when the water temperature rises, it is possible to reduce NO_x emissions without increasing HC and CO emissions.

As shown in Fig. 6, compared to the situation without an EGR bypass, it is clear that HC emissions are dramatically reduced in the low water temperature - low load range where the combustion temperature is also low. As the water temperature rises, the effect

of the EGR cooler bypass gradually decreases, and when the water temperature reaches a predetermined value, the valve switches to the EGR cooler side, to increase EGR.

4.2. Output Performance

Figure 7 shows the engine output performance for this engine. Among engines with the same displacement, this engine provides

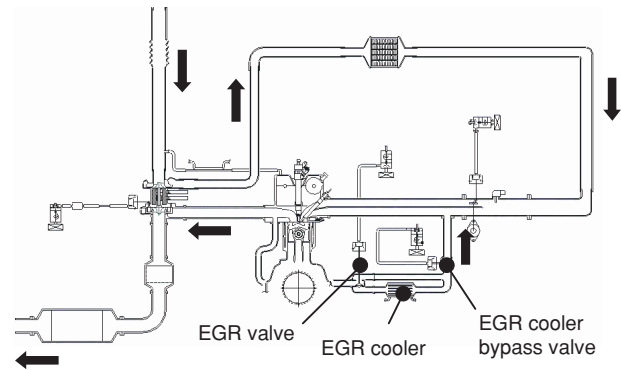


Fig. 5 Schematic diagram of EGR control system

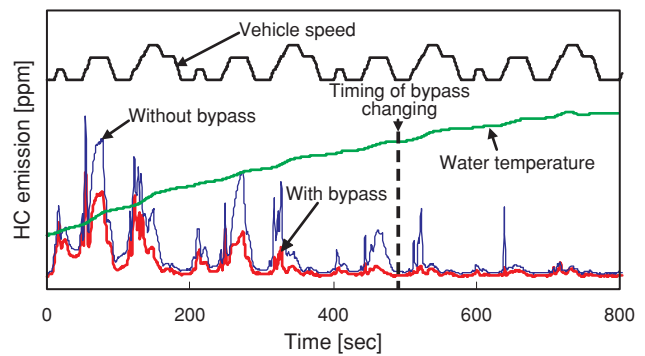


Fig. 6 Effect of EGR bypass system

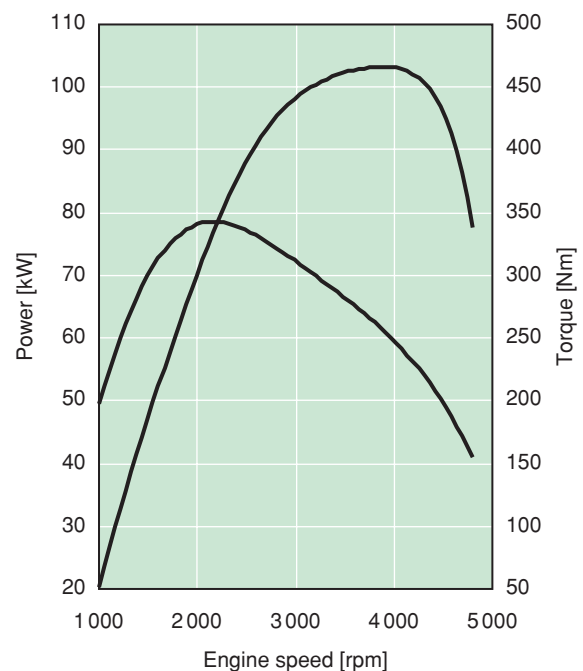


Fig. 7 Power curve of i-CTDi engine

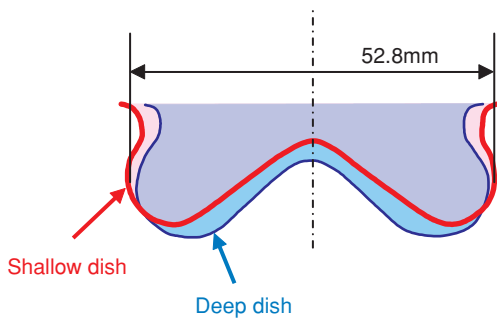


Fig. 3 Shape of combustion chamber

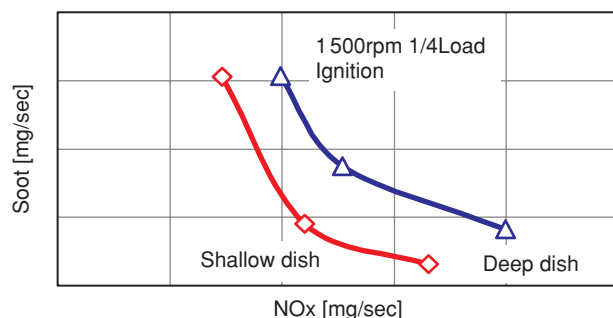


Fig. 4 Effect of combustion system refinement

the highest level of power performance in its class. The turbocharger system adopted a VGT, and optimum setting of the compressor enables optimum turbocharging in the full range. Boost pressure is controlled so it is at the target pressure, by opening and closing vanes, according to the boost pressure detected in the intake manifold. Negative pressure, electronically controlled by an Electric Vacuum Regulating Valve (EVRV), is used to open and close the vanes. The center housing is water cooled, and protects bearings and engine oil exposed to high heat. In addition, an air cooled intake cooler is provided, achieving a maximum turbo pressure of 130 kPa.

5. DPF System

5.1. System Outline

Figure 8 shows the basic configuration of the DPF. PM is collected, as exhaust gas passes through the DPF wall (pores of 9-13 μm).

Figure 9 shows an outline of PM collection and DPF regeneration in the DPF system. As the figure shows, the PM emitted from the engine is collected by the DPF, and when a specified amount is accumulated, the PM is burnt away due to the rise in exhaust temperature, regenerating the DPF.

Table 2 gives the basic general specifications of the DPF. The DPF pore diameter is determined based on PM collection capability. Porosity is determined based on regeneration performance. Diameter and capacity of DPF substrate are determined based on the vehicle life distance.

In order to create a system that demonstrates optimal DPF performance, without losing fuel economy or emissions performance, the following issues should be addressed.

- To enhance DPF regeneration performance during low load driving.
- To guarantee reliability when fuel dilutes engine oil.

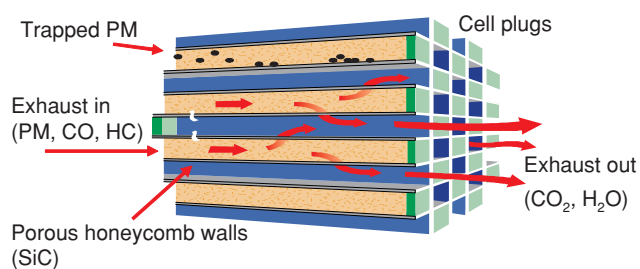


Fig. 8 Structure of DPF

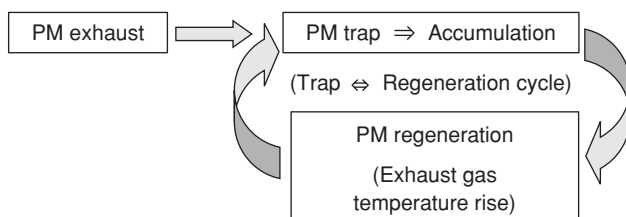


Fig. 9 Outline of PM regeneration

Table 2 DPF Specifications

DPF Size (mm)	$\phi 132 \times 228.6$
DPF Volume (liter)	3.1
Cell number (cells/in ²)	300
Pore diameter (μm)	11
Porosity (%)	42

5.2. DPF Regeneration Control

By forcibly increasing the DPF temperature to 550 °C or higher, using DPF regeneration control, the combustion speed of the accumulated PM is dramatically increased, and the PM burns in a shorter time.

Figure 10 shows the DPF regeneration control configuration. The PM emission amount calculated from engine speed, fuel injection quantity, intake air mass flow, and the PM combustion amount calculated from the DPF temperature are used to calculate the amount of PM accumulation. If the PM accumulation reaches the specified amount, DPF regeneration starts. During DPF regeneration, the DPF temperature rises, and when the PM combustion temperature is reached, the amount of PM accumulation is reduced, and when it reaches zero, DPF regeneration ends.

To make DPF regeneration possible under a wide range of driving conditions, it is necessary to increase the DPF temperature to 550 °C or higher. The methods are different for each driving range and control is performed according to different driving conditions. Figure 11 shows regeneration methods in each driving range.

In the full load range, the temperature making DPF regeneration possible is reached without regeneration control, and natural regeneration is possible.

At high loads, controlling the VGT vane towards the open side and reducing the intake air volume increases the exhaust temperature so that it reaches the DPF regeneration temperature.

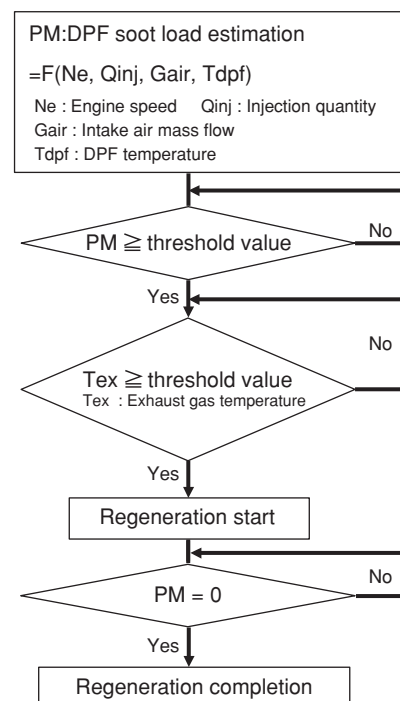


Fig. 10 DPF control software architecture

At medium loads, exhaust temperature rises by reducing the intake air volume through further throttling, and unburned fuel is supplied through post-injection in the expansion process. Utilizing the heat generated from the oxidizing catalytic converter located at the front of the DPF, the exhaust temperature reaches the DPF regeneration temperature.

At light loads, because the exhaust temperature does not reach the reaction temperature of the oxidation catalytic converter, the exhaust temperature is further raised by fuel injection carried out immediately after the main injection (after-injection). This enables DPF regeneration as the oxidation catalytic converter reaction temperature is reached.

Figure 12 shows the temperature shifts during DPF regeneration under actual driving conditions in European City mode. In traffic congestion, low exhaust temperatures are detected, and DPF regeneration stops. In suburban driving, the exhaust temperature rises and DPF regeneration starts. Once the DPF temperature reaches the PM combustion temperature, it is possible to maintain the DPF temperature even during congested driving.

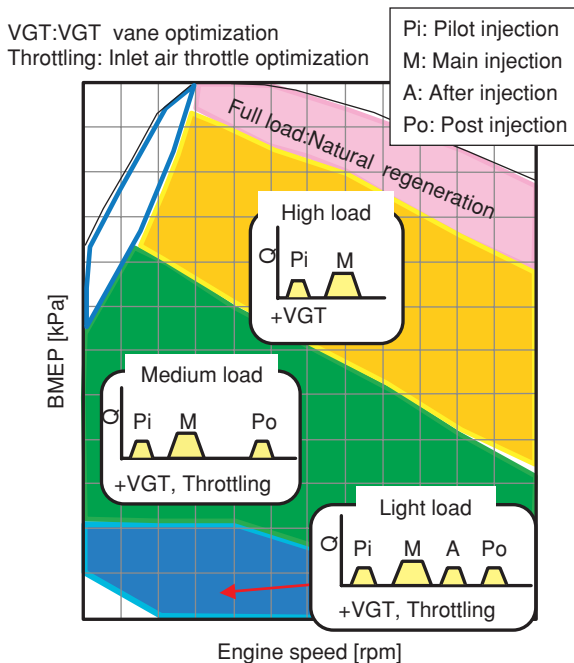


Fig. 11 Engine management strategy for DPF regeneration

In constant congested driving, DPF regeneration may not be possible and the driver is prompted by an indicator light to carry out DPF regeneration. The driver can complete DPF regeneration by driving for 10 minutes at 60 km/h or more.

5.3. Effect on Reliability (fuel dilution through post-injection)

When using post-injection during DPF regeneration, the unburned fuel that sticks to the sleeve wall becomes mixed in with the engine oil, which promotes deterioration such as a drop in viscosity. Therefore, as shown in Fig. 13, post-injection limits are established.

If a specific amount of PM has accumulated and DPF regeneration is carried out using post-injection, when the total post-injection amount reaches a specific limit, post-injection during DPF regeneration is terminated.

Then, after a specific distance or greater has been traveled, DPF regeneration post-injection resumes. Once DPF regeneration is complete, the post-injection limit is reset.

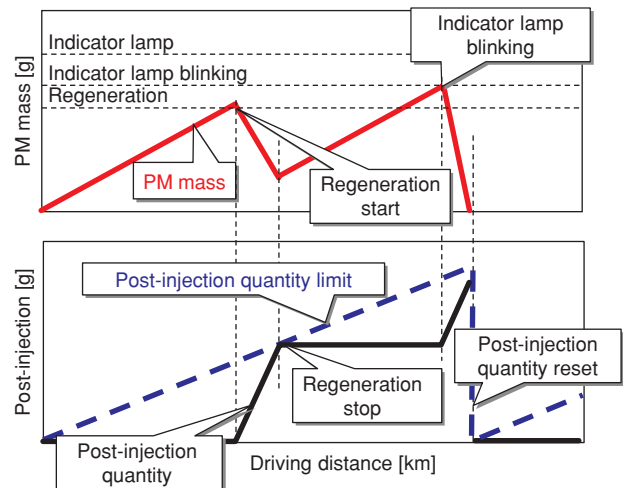


Fig. 13 Outline of post-injection limit control

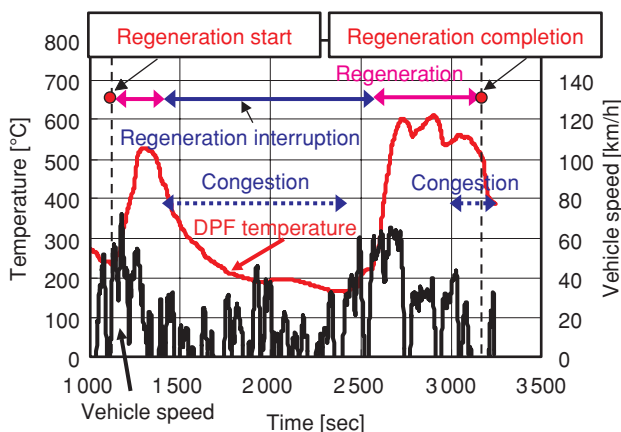


Fig. 12 DPF temperature in European city mode

6. Vehicle Performance

6.1. Emissions Performance

Figure 14 shows the emissions achieved. Through the aforementioned enhancements, PM and NO_x are dramatically reduced, and the vehicle is the first SUV to comply with EURO-IV standards. In addition, with the adoption of the DPF, PM was reduced to less than one-tenth of the EURO-IV regulation value. The resulting PM emissions value is in compliance with that of the Austrian Incentive standard, the most stringent worldwide at the present time.

6.2. Dynamic Performance

Output characteristics emphasize acceleration at low to medium speeds. From city driving through highway driving, the engine

exhibits high performance under a wide variety of driving conditions.

In combination with an optimized gear ratio, NEDC mode fuel consumption reaches 6.7 L/100 km. This fuel economy is best-in-class for an SUV diesel engine.

Figure 15 shows this fuel economy and power performance achieved.

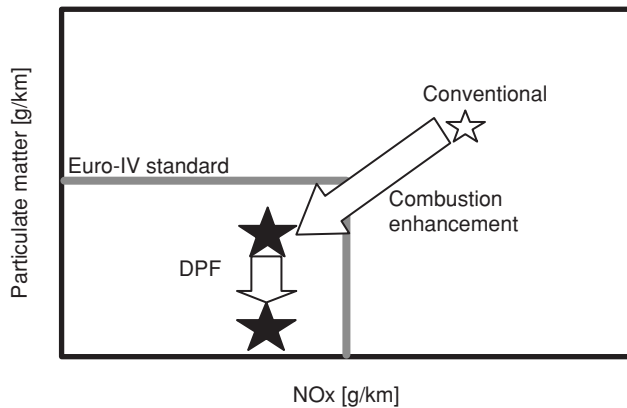


Fig. 14 Effect of combustion enhancement and DPF for exhaust gas emission

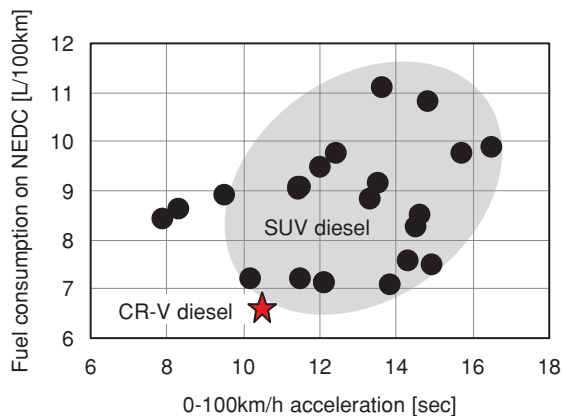


Fig. 15 Fuel consumption and acceleration performance of CR-V diesel

7. Conclusion

A diesel engine that has best-in-class SUV total performance was developed. The following results have been achieved through the development of this engine:

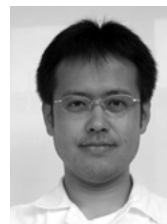
- (1) Modifications of the shallow dish combustion chamber shape and fuel injection nozzle as well as the use of a highly efficient EGR system made the engine the first SUV to comply with EURO-IV standards.
- (2) In terms of dynamic performance, quietness, and fuel economy, SUV best-in-class performances were also attained.
- (3) With the adoption of the DPF, PM emissions were reduced to less than one-tenth those of the EURO-IV regulation value.

- (4) By addressing the DPF system issues of enhancing DPF regeneration performance during low load driving and of fuel diluting the engine oil, Honda's first proprietary DPF system has achieved high performance and high reliability not seen previously.

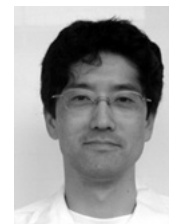
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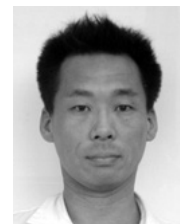
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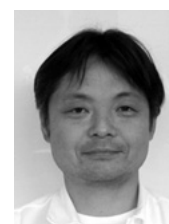
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Research on ISG Belt Drive System for Idling Stop System

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ABSTRACT

To enable a belt-drive Integrated Starter Generator (ISG) to be used in an idle stop system, the belt system must be capable of both driving the accessories and starting the engine. Accomplishing this with a conventional belt system required an increase in the tension of the belt, which also increased friction losses and is thus incompatible with the intention of increasing fuel economy.

Testing quantified the effect of the increased belt tension on increasing friction losses. With this knowledge the research team focused on the auto-tensioner mechanism and the design of a superior belt system without the requirement for an increase in tension. Belt tension was reduced through the employment of an auto-tensioner in which contraction can be controlled by exploiting the incompressibility of fluid and by modifying the belt layout to enable optimal positioning of the new auto-tensioner. The use of reverse torque was tested and proven to be an effective method of further reducing the tension of the belt. The combination of these innovations has resulted in the development of a superior belt system well matched to the requirements of an idle stop system.

1. Introduction

The Kyoto Protocol of the United Nations Framework Convention on Climate Change became effective in February 2005, thus mandating the signatory nations to reduce their carbon dioxide emissions. As shown in Fig. 1, the emission of carbon dioxide in Japan in 2004 exceeded the targeted value for 2008 by 14%. Demands are being placed on the automobiles, which accounts for a significant percentage of emissions, to further increase fuel economy. An idling stop system can promptly achieve enhanced fuel economy.

Utilization of an idling stop system that uses a belt-driven Integrated Starter Generator (ISG) requires a belt system that is able to drive the engine accessories including ISG, and also start the engine. In the conventional auto-tensioner belt system which

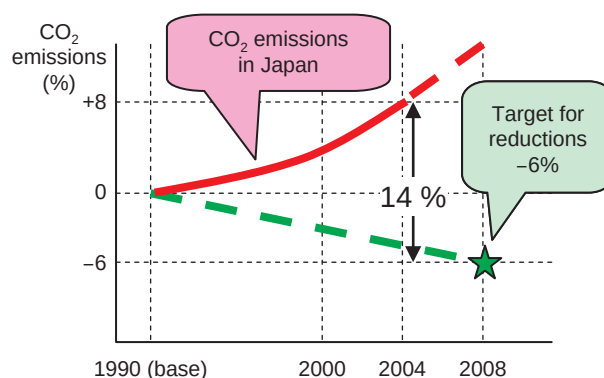


Fig. 1 Status of CO₂ emissions and target for reductions in Japan

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uses a single belt to drive several accessories, the auto-tensioner would be pushed back when the engine starts, causing the belt to slacken. This requires the tension to be set higher during starting conditions. Research was conducted on a new auto-tensioner mechanism and belt layout that enabled the mechanism to be mounted in the optimum location, and also to determine an ISG control method for engine starting. As a result of this study, the above-mentioned issues have been overcome. This paper describes the configuration and operation of the system, in addition to the results of engine starting tests.

2. ISG Drive and Auto-tensioner Issues

2.1. Auto-tensioner Layout Issue

The optimum mounting position of an auto-tensioner, for driving accessories only, was found to be on the belt segment just behind the crank pulley, along the running direction of the belt at which the set tension can be minimized. When the engine was being started by the ISG, a significantly larger tension than that generated during normal accessory drive was generated between the crank pulley and the ISG, along the running direction of the belt. This caused the belt to elongate, while on the opposite side the reverse phenomenon had occurred. It was possible to increase the tension on the overall belt by shortening the portion of the belt over which high tension was applied, and by minimizing the difference in length resulting from the belt elongation and contraction. The optimal position of the ISG was found to be just behind the crank pulley as an accessory along the running direction of the belt⁽¹⁾. Figure 2 shows the combined optimum positions of the auto-tensioner and

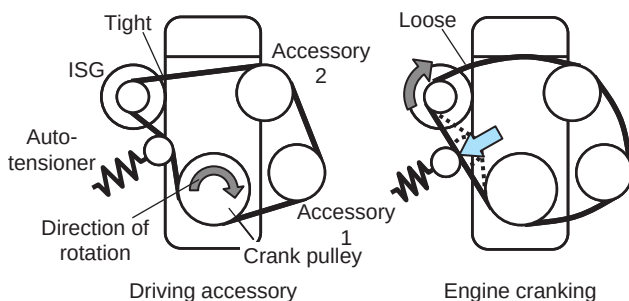


Fig. 2 Issue of auto-tensioner

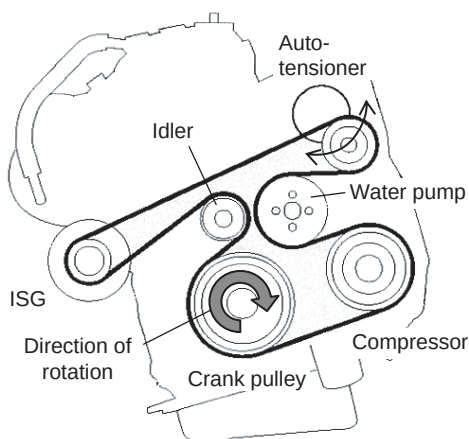


Fig. 3 Example of previous ISG system configuration

the ISG as a single layout.

These positions were found to be optimum for driving the accessories, but when the engine was being started by the ISG, the tension generated by the ISG caused the auto-tensioner to be pushed back, reducing the overall tension on the belt. As a result, it was not possible to obtain adequate friction, so there was the possibility of slippage occurring^{(2), (3)}. In order to prevent this phenomenon, it was necessary to raise the tension setting of the auto-tensioner to a value higher than that of the generated tension. However, this increased belt tension during normal accessory driving, so there remained the issue of the durability of the belt itself, and of the accessory bearings.

2.2. Conventional Solution and Tension Issue

Figure 3 shows a practical example of the layout of the ISG and auto-tensioner which was intended as one solution for this issue^{(4), (5)}. Normally, when the engine is being started, the belt segment, just behind the ISG in the direction of the belt motion, loses tension. The auto-tensioner pulls up the segment, maintaining the belt tension on the slack side, which is important in transmitting friction. This position was found to be optimum for driving the engine⁽²⁾.

On the other hand, because the auto-tensioner had been moved away from its optimum position for driving the accessories, it was necessary to set tension higher. In other words, it was necessary to superimpose the maximum effective tension required to drive the ISG^{(2), (3)}, while it generates electricity, on the set tension of the auto-tensioner, which results in the tension being greatly increased. Thus with this solution as well, high tension remained an issue.

3. Friction Loss and Tension Accompanying the Accessory Belt Drive

3.1. Friction Loss Testing Equipment

The belt tension should be set to the minimum value necessary to enable the belt to transmit drive force, and it was expected that any tension greater than this level will increase power loss of the bearings to which the tension is applied. However, concrete knowledge and information concerning the ratio of the loss related to the belt drive, to the whole power loss of the engine friction, and the relationship between the increase in belt tension and friction loss could not be obtained from past publications.

Because the reduction of belt tension was an issue, it was necessary to obtain an accurate understanding of the tension and its affects. A test was completed to measure the belt friction using the test apparatus shown in Fig. 4.

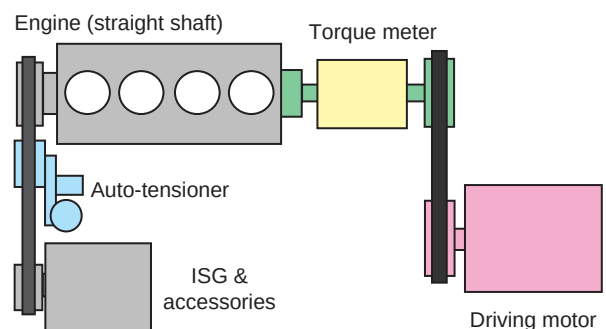


Fig. 4 Equipment for belt friction measurement

The accessories and the belt layout of the test engine are shown in Fig. 8 of the next section. The original crankshaft was replaced with a round, straight bar and the engine crankcase consisted of the crank journal alone, with the connecting rods and pistons removed. The chain that drives the camshaft was removed so that the crankshaft drove only the belt mounted accessories via the crank pulley. A torque meter was used to measure the friction loss torque generated by the belt drive, including that of the crank journals. One end of the torque meter was attached to the flywheel side of the crankshaft, and the input end was driven by a driving motor via the belt. The friction loss power was calculated from the product of this torque and the rotational speed.

3.2. Friction Loss Test Conditions and Results*

In order to make the temperature conditions of the engine and each bearing the same as those of a warm engine, the engine oil temperature was raised by the heat of motoring.

In order to understand the effect of tension on friction loss torque, the tension was set to two values: the necessary amount for conventional accessory drive, and to the original amount plus a 40% increase. The measurements were performed over the entire engine speed range. Also, in order to measure the friction loss of the crank, the test was repeated with the accessory belt removed. The results are shown in Fig. 5.

Figure 6 shows the results of calculating the ratio of increase of the friction torque due to increased tension to the whole engine friction which was measured in advance.

According to the measured results, the friction loss torque related

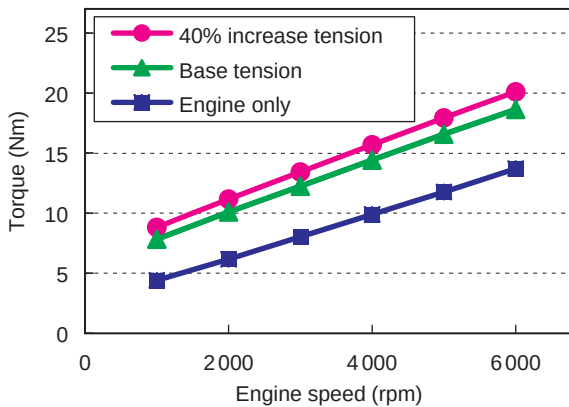


Fig. 5 Increase in friction torque due to increased tension

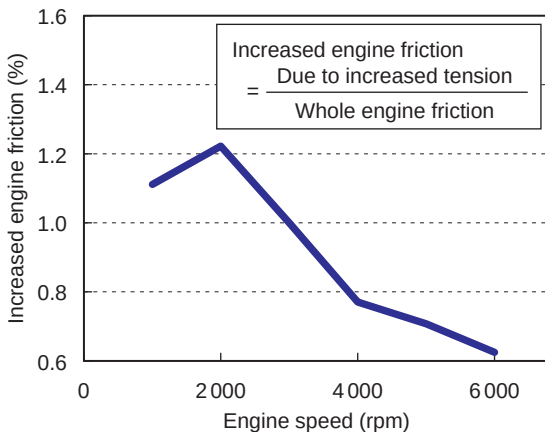


Fig. 6 Influence of increased tension

to the belt drive on the high tension side at 1 000 rpm accounted for about 50% of the entire friction loss. As the rotational speed increased, the belt drive friction loss increased. However the increase in the friction loss of the crank journal also became larger, and at 6000 rpm accounted for about 30% of the entire loss.

The increase in the friction loss torque of the belt drive due to the increase in belt tension was a maximum of 12.5% at low rotational speeds, and 8% at high rotational speeds.

The ratio of the friction loss (due to the increase in belt tension) to the friction loss of the whole engine is in the range of 1.2% to 0.6%. At low engine speeds, which are in the normal engine speed region, the ratio exceeds 1%.

While the engine was running, friction loss due to the increase in belt tension occurred unavoidably, regardless of the running conditions, resulting in a corresponding increase in fuel consumption.

Also, because the crankshaft bore has journal bearings, which are generally force-lubricated by an oil pump, the starting torque from standstill, in which the oil pressure is zero, is directly affected by the belt tension, causing it to increase. From the engine starting aspect as well, the increase in tension is a detrimental factor.

Because any increase in the belt tension has an adverse affect both on the fuel consumption when the engine is running, and on engine startability, the necessity of reducing the belt tension in an idling stop system can be recognized significantly.

4. Belt-driven System that Uses an HLT (Hydraulic Locking Tensioner)

4.1. HLT Mechanism and Layout of Accessories

A solution to the issue of the layout of Fig. 2 that permits the lowest tension to be applied while driving accessories, which is almost the entire driving period, is sought from a new viewpoint: an automatic-tensioner. An auto-tensioner must be capable of elongating and contracting freely, to maintain the belt tension, and it must incorporate a means of suppressing contraction due to the increase in belt tension when the engine is being started. In order to meet these conflicting requirements, we focused on a new mechanism that could suppress contraction. In the mechanism, a damping force due to the flow of fluid accompanying the mutual elongation and contraction of the cylinder and the piston is generated, and by closing the flow path, it was possible to utilize the incompressibility of the fluid shut in the cylinder. Specifically, in the construction employed, a solenoid valve for opening and closing the fluid path was installed on an auto-tensioner that provided a damping force generated by shearing of the oil.

Figure 7 shows the construction of an HLT (Hydraulic Locking Tensioner), and Fig. 8 shows the layout of the accessories including the HLT. In the HLT, the cylinder is installed on the tensioner pulley side, and the piston on the engine side, and a coil spring generates a repulsive force against contraction. When the HLT elongates, oil is drawn into the cylinder through a check valve at the bottom of the cylinder. When it contracts, the oil flows out through an orifice in the piston, thus generating a damping force. When the solenoid is energized, the valve mounted on the piston closes, shutting the oil in the cylinder and thus enabling the contraction of the piston to be suppressed.

Even in this condition, oil can flow into the cylinder enabling the piston to elongate, due to the repulsive force of the spring. As a result, the function that applies tension to the belt is maintained, permitting optimization of the belt tension for driving accessories

and for engine starting, which were issues with earlier conventional systems.

4.2. Verification of the Effectiveness of the HLT

4.2.1. Belt behavior testing equipment

To enable the engine to be started by the belt of an ISG system, it was necessary to understand the various factors that are related to the transmission of torque by friction. The method used to measure the belt pulley tension involved converting the strain at the support shaft into tension. The tension on the taut side when the engine is being started was measured using an idler, and the tension on the slack side was measured by installing an additional idler, between the crank pulley and the compressor.

In order to study the frictional transmission between the belt and the pulley, it was necessary to determine the running speeds of the ISG pulley, the crank pulley, and the belt, respectively. Accordingly, the surface speeds of the back of the belt, and the points on the side of the ISG and crank pulleys where the surface speed must be equal to that of the back of the belt, were measured using a laser Doppler vibration meter which permits non-contact, highly accurate speed measurements. The elongation and contraction of the HLT were measured by installing a stroke sensor between the fixed end and the free end, on the arm side. Also, a microphone was installed in the vicinity of the belt, and the slip of the belt was checked from the noise emitted.

All of these signals were recorded by a single recording device

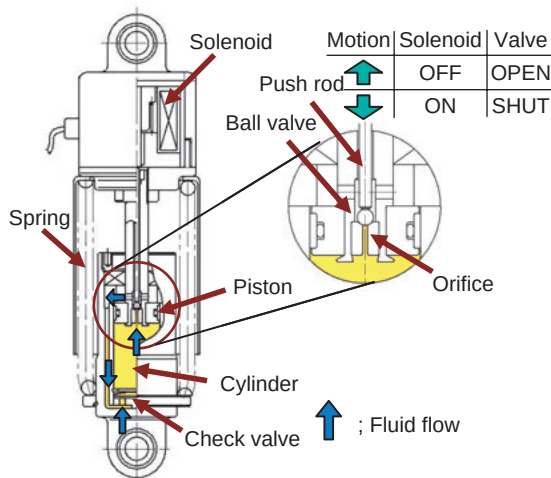


Fig. 7 Mechanism of HLT

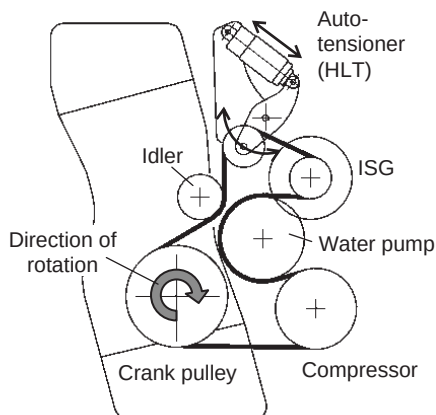


Fig. 8 Accessory drive layout with HLT

on the same time line, enabling the time correlation to be understood. Figure 9 shows the configuration of the measuring instruments. Figures 10 and 11 show the actual installation of the laser Doppler vibration meter and the microphone.

4.2.2. Performance of the HLT and engine starting results°

Figure 12 shows the results of actually using an HLT to start the engine. In order to clarify the performance of the HLT, the engine was also deliberately started without suppressing the contraction of the HLT, for comparison. The results are shown in Fig. 13. The engine was cranked by making the ISG generate the maximum torque from the commencement of starting. In the case

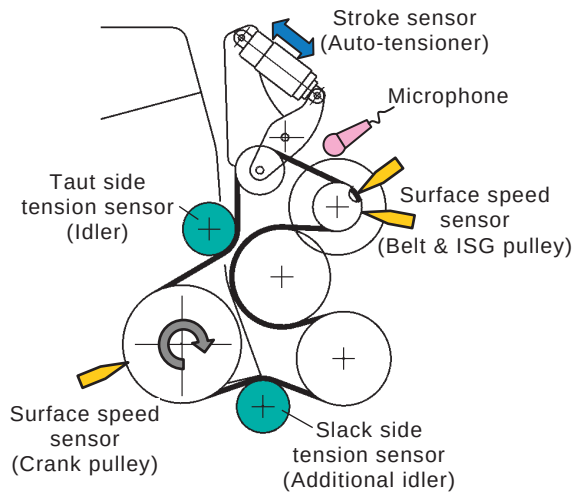


Fig. 9 Measuring instrument layout

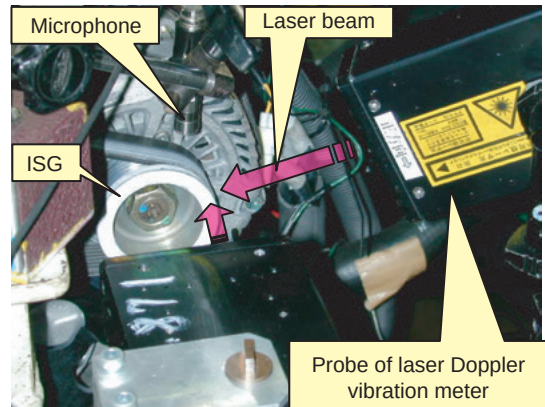


Fig. 10 Surface speed sensor for ISG

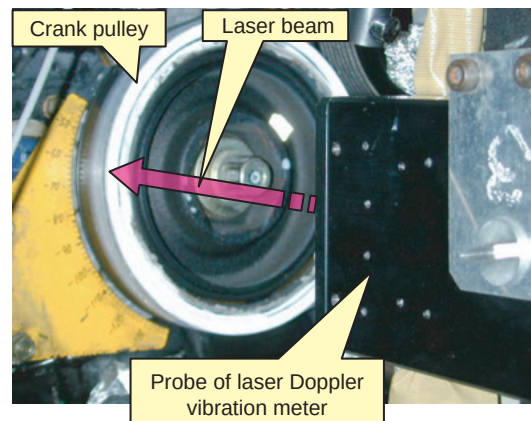


Fig. 11 Surface speed sensor for crank pulley

where contraction was not suppressed, the HLT contracted due to the belt tension on the taut side, the tension on the taut side fell rapidly, and the tension on the slack side became roughly zero. Because of this tension reduction, the frictional transmitting force between the pulley and the belt decreased, resulting in a difference in the speeds of the ISG pulley surface, the crank pulley, and the belt.

In other words, belt slippage occurred. The slip noise increased when the surface speed difference became large, thus verifying this phenomenon. In order to prevent excessively high rotational speeds, the ISG limits the drive torque at high speeds, which causes the surface speed of the ISG to exhibit sinusoidal wave behavior. It was presumed that if the drive torque had not been limited, the speed would have risen excessively.

Because of this slippage, almost no power was transmitted from the ISG to the crankshaft, and the engine failed to start.

On the other hand, when contraction control was performed, a small amount of contraction occurred in the HLT. However, the tension on the taut side and the slack side was symmetrical with respect to the initial tension level, thus ensuring that adequate effective tension for starting the engine was maintained. As a result, there was almost no discrepancy between the surface speed of the ISG and that of the crankshaft, and the engine was able to be started.

From these two results, it was possible to identify the various factors related to the starting of the engine. In addition, the function and performance of the HLT has clearly been confirmed.

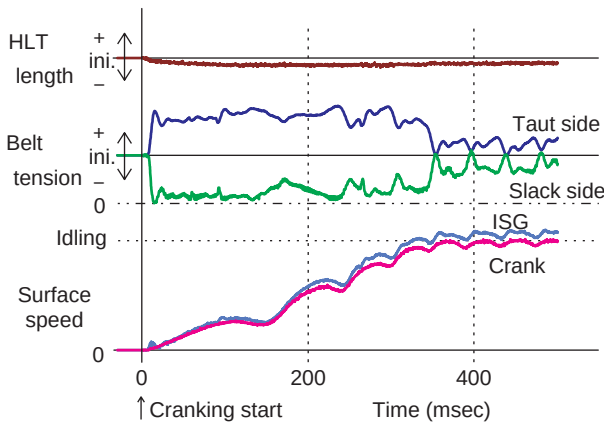


Fig. 12 Engine start with HLT locking

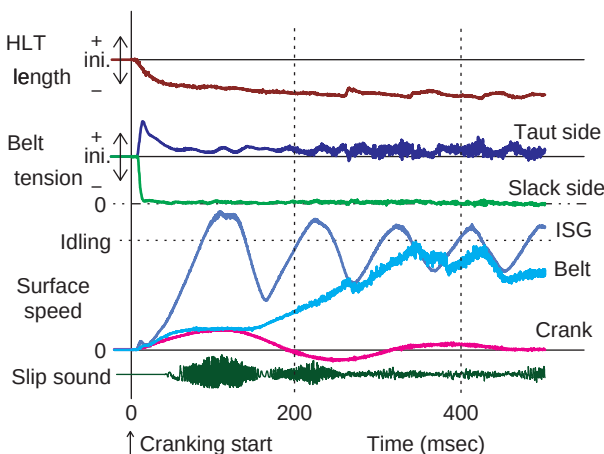


Fig. 13 Engine start without HLT locking

5. ISG Control at Engine Starting

5.1. Aim of ISG Reverse Torque Control

Engine starting necessitates a belt drive force that exceeds the maximum load to drive all of the accessories. It was confirmed empirically in the layout of the test engine accessories, including the HLT, that the tension on the taut side and the slack side would increase and decrease, respectively, by one half of the effective tension generated by the ISG. In order to maintain the frictional transmission on the belt, it was necessary to maintain the tension on the slack side, so it was necessary that the initial tension be at least one half of the effective tension as that of the ISG. This value was higher than the minimum necessary value for driving the accessories. Considering the load on the belt as well as the load on the bearings of each accessory and of the engine, it was desirable to set the initial tension to a lower value.

While searching for a method to set the initial tension to as low a value as possible, and to raise the tension immediately before starting the engine, a solution was found in the characteristics of the HLT and drive control of the ISG.

The HLT suppresses contraction as a result of the energization of the solenoid, however elongation can occur freely. When slackness occurs in the segment, it is always possible to maintain the tension. In this condition, if the ISG generates a torque that acts in the reverse direction to that of the torque generated when the engine is started, the segment on the HLT side will slacken, and the segment on the opposite side will become tauter. When reverse torque is generated for a minimum amount of time, the HLT elongates within the time, while the reverse rotation of the engine remains extremely small. When the reverse torque is released, the tension on the belt is increased by an amount corresponding to the elongation of the HLT. It is considered that if the ISG is caused to generate a forward torque immediately after this, the belt will be assisted by the increased tension, and will adequately transmit friction force, thus ensuring that the engine starts reliably.

5.2. Confirmation of the Performance of Reverse Torque Control

The following is a description of the results of starting the engine by combining the HLT with ISG control. In order to confirm the increase in tension due to reverse torque control, a test in which zero torque time was inserted into the interval during which the torque changes, from the negative direction to that of the forward torque, was also performed. Figures 14 and 16 show the torque command to ISG, and Figs. 15 and 17 show the results of engine starting. When the ISG generates reverse torque during time T_r indicated in Fig. 14, after the HLT performs contraction control, the HLT elongates in order to maintain the tension on the slack segment. The mean value of the tension on the taut and slack sides, which indicates the overall level of the tension, increases by about

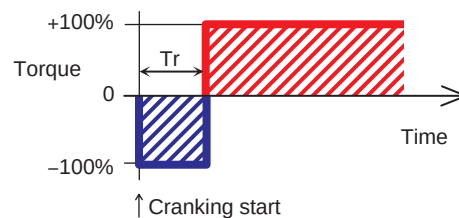


Fig. 14 Torque command for ISG

one half of the effective tension with respect to that of the initial value. Next, the engine commences to start as a result of the forward torque. Though the HLT contracts by a small amount, the tension was maintained during this process. Because of the high tension, an adequate frictional transmission force was obtained, and the engine was started under a condition in which the difference in the surface speed of the ISG and the crankshaft was smaller than that for the case where HLT contraction suppression alone was used.

In the results (Fig. 17) of the starting test carried out using the torque commands indicated in Fig. 16, a reverse torque was applied for a period of T_r , then the torque was made zero for a period of T_i , after which a forward torque was applied.

After the convergence of the tension fluctuation due to the removal of the reverse torque, it was possible to confirm an increase in tension with respect to the initial value, which was of the same order as that obtained in the previous test on the taut and slack sides, thus verifying the performance of this control method.

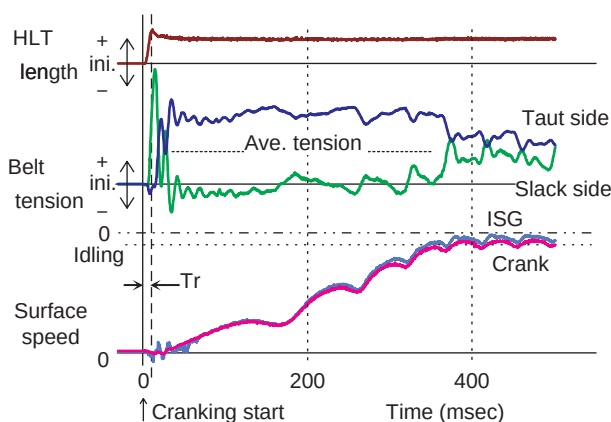


Fig. 15 Engine start with reverse torque

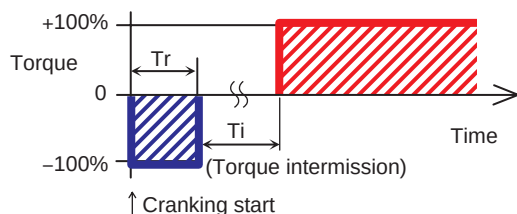


Fig. 16 Torque command with intermission

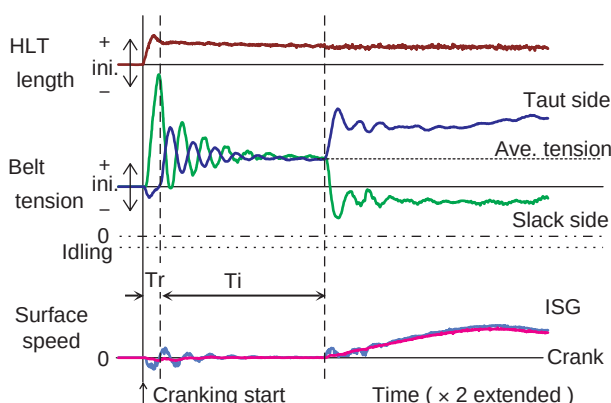


Fig. 17 Engine start with reverse torque and intermission

As described above, by applying reverse torque control from the ISG, it was possible to increase and maintain the tension, including that on the slack side, thus confirming that this is an effective method of reducing the initial tension setting.

6. Conclusion

- (1) Research was carried out on a belt drive system including an HLT, and as a result, the different requirements of driving the accessories and starting the engine were understood.
- (2) It was found possible to greatly reduce the initial tension setting by combining the characteristics of the HLT with ISG control.
- (3) A detailed measurement of the behavior of the various factors related to engine starting was performed, and the results obtained verified the effectiveness of HLT and ISG control.
- (4) The friction loss due to the belt tension was clarified empirically, and thus the necessity of reducing the tension was also clarified.

The results of the research performed by the authors enabled the issues of tension in the ISG belt drive system to be resolved. The development of this ISG and HLT belt system can considerably reduce torque loss by stopping the engine instead of idling it, and also by maintaining a minimum amount of belt tension. Both of these enhancements to the belt system will enhance fuel economy.

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Clarification of Mechanism of Exhaust Orifice Noise via Exhaust Pulsating Flow Simulation

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Yuuya KASAJIMA**

ABSTRACT

Regulations relating to environmental noise generated by automobiles are becoming increasingly stringent, making it essential to develop simulation technologies to contribute to the reduction of exhaust noise. The authors attempted to make the pulsing exhaust flow produced by an internal combustion engine visible using Schlieren photography and to visualize the source of exhaust orifice noise using noise holography. The knowledge obtained from these experiments was used to develop an exhaust pulsating flow simulation technology for exhaust orifice noise that enables quantitative prediction of levels. The major components of exhaust orifice noise are compression waves and ring vortices; the new simulation technology enabled further clarification of the mechanism of production of this noise.

1. Introduction

Exhaust noise is classified into two types, radiation noise and orifice noise. The mechanism of radiation noise already revealed that pressure fluctuation in the pulsing exhaust flow excites the exhaust system structures to be a source of radiation noise⁽¹⁾. This research presents an analysis of the mechanisms for generating orifice noise. Although orifice noise largely affects environmental noise, currently, R&D has lagged behind in understanding the generation mechanisms and in simulation technology. In order to clarify the generation mechanisms of orifice noise, it is necessary to gain a better understanding of the phenomena arising at the end of the exhaust pipe. The authors carried out a Schlieren visualization of exhaust orifice flow and attempted another visualization of the orifice noise source using acoustic holography. After understanding the phenomena, Computational Fluid Dynamics satisfying the conditions for analyzing vortex rings in detail was selected, and numerical analysis was performed to find the orifice flow noise source. Calculating the sound power of the sources of orifice flow

noise also pointed the way to a practical and precise procedure for finding propagation into the sound field. This paper reports that the visualization experiments and exhaust pulsating flow simulation technology were used in an attempt to clarify the mechanisms involved in generating exhaust orifice noise.

2. Generation Mechanisms of Orifice Noise

First, the mechanisms by which exhaust pulsating flow is propagated into the surrounding field as noise is considered. Part of the internal energy due to the pressure and velocity of the exhaust pulsing flow appears as structural excitation forces that apply to the exhaust pipe and the silencer, and from there, propagates to the surrounding field as noise.

The authors' conclusions from their previous experiments and calculations led them to the assumption that orifice noise originates from the noise source at the orifice end, and propagates away as an acoustic wave (Fig. 1).

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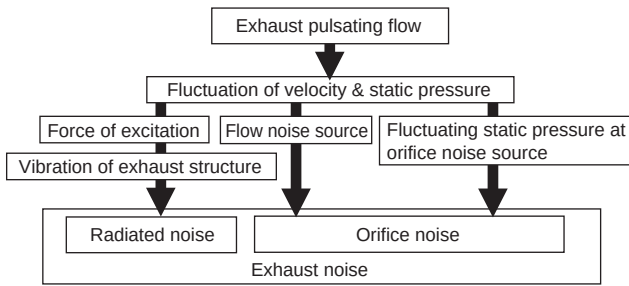


Fig. 1 Mechanism of generation of exhaust noise

Two main sources of orifice noise have been identified, the noise source at the orifice of the exhaust pipe, and that of the flow noise. A flow visualization experiment was therefore conducted in order to confirm the above assumption and gain a better understanding of the phenomena. Next, a Computational Fluid Dynamics (CFD) simulation was performed to find the orifice noise source; the sound power of the exhaust orifice flow was also calculated to compare the predicted noise level in the sound field quantitatively with measured sound pressure level.

3. Technology for Experimental Analysis

The exhaust pulsating flow in the exhaust pipe arrives at the orifice and is ejected as orifice flow. Schlieren photography was used to visualize the flow, in order to view the noise sources for the orifice noise. Then, non-steady acoustic holography, which can be used with time history data, was employed to visualize a wide area around the source of the exhaust orifice noise.

3.1. Experimental Method for Schlieren Photography Visualization

The beam from a xenon source was reflected from a concave mirror to create a parallel beam which passed through the region to be measured (target), collected by another concave mirror and focused onto a knife-edge. The portions of the target from which the beam was diffracted toward the knife-edge were darkened, and the portions from which the beam was diffracted away from the knife-edge were brightened. This method enabled visualization by contrasting brightness in the density distribution of the target. The photographs were taken by synchronizing the light source triggering

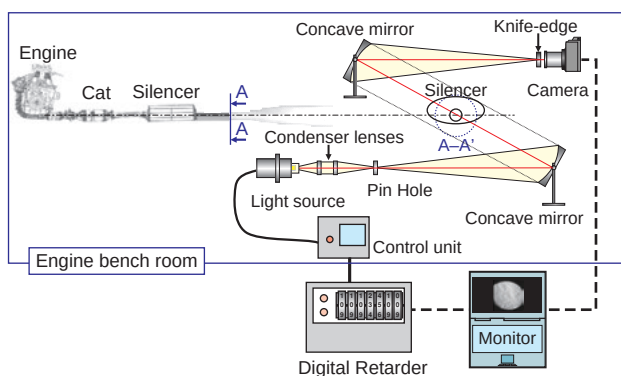


Fig. 2 Schematic of experimental apparatus

with the engine speed and setting the trigger delay time with a digital retarding circuit to obtain frames of the desired conditions. Figure 2 shows a schematic view of the experimental apparatus.

3.2. Results of Schlieren Photography Experiment

Figure 3 shows Schlieren photographs of the exhaust orifice flow obtained at 3000 rpm when firing (with combustion) and motoring (without combustion).

The following items were observed in this visualization:

- (1) A large number of vortices were confirmed in the exhaust orifice flow. The density of vortices was found to be higher at higher exhaust temperatures (Firing mode). The density of vortices was lower during motoring mode.
- (2) A large occurrence of vortex rings was confirmed. These were clearly visible under the lower density fluctuations in the orifice flow during motoring (blue lines in Fig. 3). They were less clearly defined during firing, due to a large number of vortices.
- (3) No shock waves⁽²⁾ were observed in the exhaust pipe near the orifice.

3.3. Method for Non-steady Acoustic Holography Visualization Experiment

The acoustic intensity was measured at intervals of $240 \mu\text{s}$ with a microphone array (6×11) whose center was placed 100 mm from the end face of the silencer (Fig. 4).

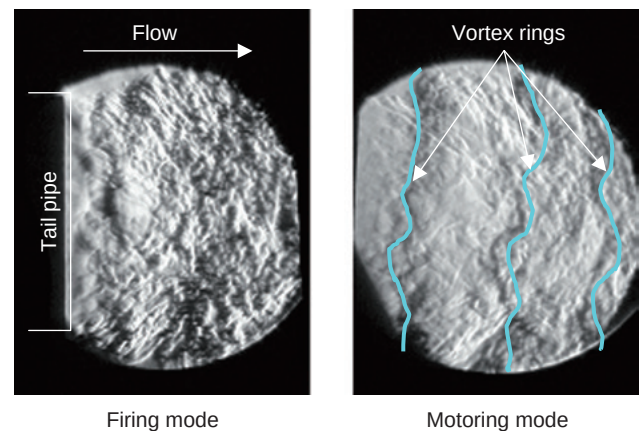


Fig. 3 Schlieren photograph of exhaust orifice flow (3 000 rpm, Th15%)

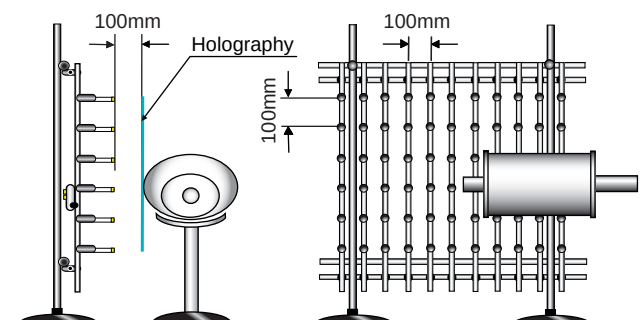


Fig. 4 Schematic of nonsteady acoustical holography

3.4. Non-steady Acoustic Holography Experimental Results

The non-steady acoustic holography visualization of the orifice noise source indicated the following two phenomena:

- (1) Each source of noise released from the orifice propagates at sonic speed and diffuses in a spherical pattern.
- (2) The centers of some orifice noise sources were observed to migrate away from the orifice (white spots in Fig. 5).

This migration matches exactly with the flow speed during the pulsations of the orifice flow. It was assumed that the sources migrate due to the flow.

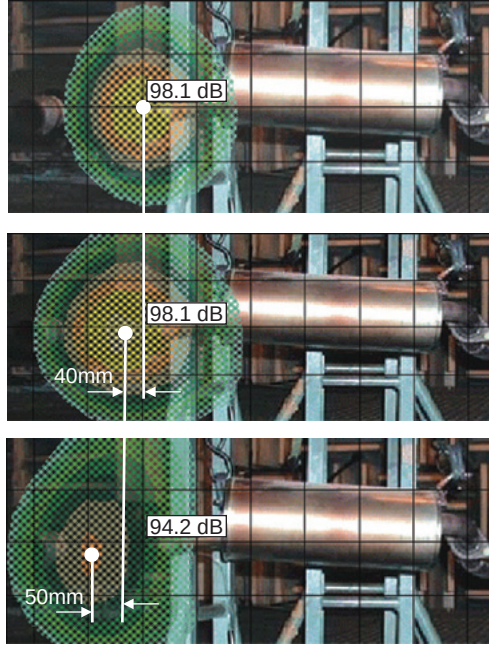


Fig. 5 Nonsteady acoustical holography of exhaust orifice noise (5 000 rpm WOT:Wide Open Throttle)

4. Simulation Technology

In real internal combustion engines, the exhaust pulsation is produced by the exhaust process, after cylinder combustion process. Inconsistencies in the combustion and other factors affect the exhaust orifice noise. In this research, a physical model of a high-speed exhaust pulsation generator that does not have combustion inconsistencies or other such factors was employed in order to compare and verify the simulated orifice noises.

4.1. Verification Experiment Apparatus

A comparison test was conducted using computational numerical analyses and actual results obtained in a high-speed exhaust pulsation generator⁽³⁾ which modeled an internal combustion engine, without any actual combustion process. This flow generator used the rotary valve diagrammed in Fig. 6. Rotation of the internal element induced a pulsating flow in the right portion of the test pipe. The pulsation flows propagated down the pipe to the outlet, where they entered an anechoic room. The orifice flow was visualized with Schlieren photography. The test pipe was 1 m long. Semiconductor pressure sensors were installed in the pipe 100 mm downstream from the valve (Point A) and 100 mm upstream from

the orifice (Point B) to observe changes in the static pressure due to pressure waves traveling through the pipe⁽³⁾. The absolute pressure in the plenum tank (left side of the pipe) was 1.5 times that of the ambient atmospheric pressure (Pressure ratio = 1.5). The pressure of the orifice noise was measured at the sound field with a microphone 1 m from the right exit of the pipe and at a 45° angle off the pipe centerline.

4.2. Computational Numerical Analysis

The testing revealed that all of the following conditions for numerical methods need to be satisfied simultaneously, in order to conduct a successful computational numerical analysis of the exhaust pulsating flow in the pipe and of the orifice flow:

- (1) Compressible fluid analysis for the exhaust pipe inner flow including compression-expansion waves and shock waves
- (2) Non-steady analysis of fine structures (turbulence, vortex rings, etc.) in flow fields
- (3) Aeroacoustic analysis for flow field and sound field pressure fluctuations

In this research, a commercially available CFD solver⁽⁴⁾, satisfying the above conditions by applying the governing Navier-Stokes equation, the law of mass conservation, and the law of energy conservation was used for conditions (1) and (2). For condition (3), the sound power level was practically calculated to predict the acoustic pressure level, accounting for attenuation with distance, at the observation point.

4.3. Compressible Flow Numerical Model

The numerical model was the 2-dimensional axially symmetric model shown as Model A in Fig. 7. The node pitch was 0.25 mm. The initial inner pressure of the high-pressure tank was set at 150 kPa. The exhaust pulsations were generated at the valve, which was described using an Arbitrary Lagrangian-Eulerian (ALE)⁽⁵⁾ mesh (Fig. 8). It is necessary to solve for shock waves and other compressible fluid phenomena in the exhaust pipe, as mentioned above in 3.1, so the simulation model with compressible fluid analysis was used, assuming conditions of an isentropic heat exchange between the inside and outside of the pipe.

4.4. LES Calculation Model

In order to solve the fine flow structure at the orifice, a Large Eddy Simulation (LES)⁽⁶⁾ was used as the model for turbulence in a pseudo-compressible approach (Model B, Fig. 7). The calculation results at 100 mm upstream from the orifice in Model A (Point B) were used as the boundary conditions for the entrance to Model B

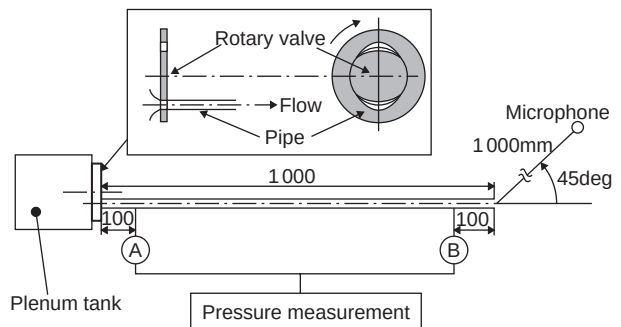


Fig. 6 Schematic of experimental apparatus

(Point C). The boundary of the orifice flow model was made Silent boundary.

A quad-shell mesh with equal pitch of 0.25 mm was used in both Models A and B. The total number of elements in both was about 500 000.

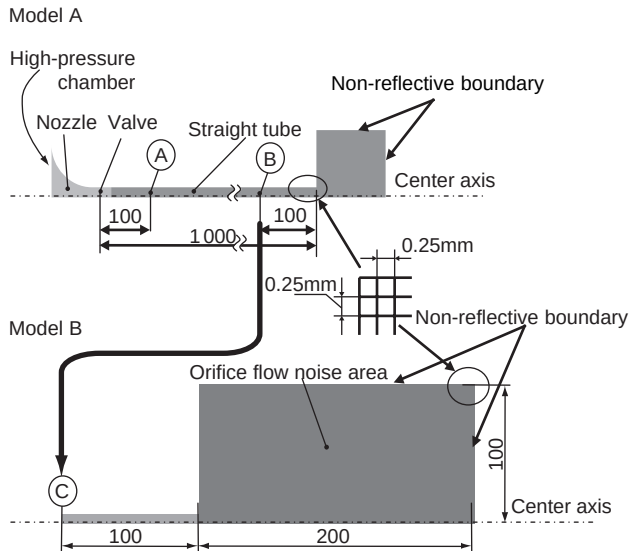


Fig. 7 Calculation model

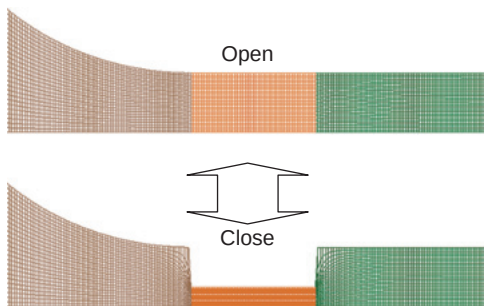


Fig. 8 Valve ALE model

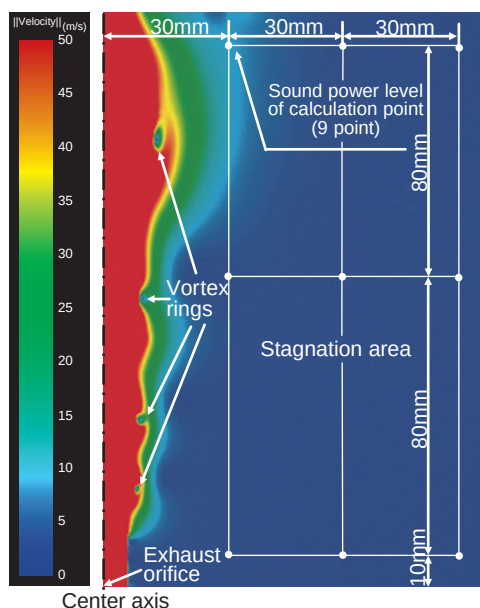


Fig. 9 Calculation model of sound power level

4.5. Sound Power Calculation Model

The near sound field, affected by the orifice flow, comprises pseudo-sound waves⁽⁷⁾. “Pseudo-acoustic waves” mean the field of pressure fluctuations near the flow field which is affected by orifice flow. An acoustic numerical method employing the acoustic source based on Lighthill’s analogy can be used to find acoustic pressure levels in a sound field which is unaffected by pseudo-acoustic waves^{(8),(9)}. In this research, however, the sound power of the orifice flow was calculated, and the sound pressure level remaining at the observation point, after attenuation with distance, was estimated. This method requires the model for calculating sound power to be in a stagnation region unaffected by the flow field. Nine locations in the stagnation area of the flow unaffected by the orifice flow were selected for measuring sound power (Fig. 9), using the results of the numerical fluid model.

4.6. Results of Numerical Calculations for Pulsating Pipe Flow

Figure 10 shows the numerical results for the pressure distribution in the pipe. Figures 11 and 12 show the time history of the pressure in the pipe 100 mm downstream from the valve (Point A) and 100 mm upstream from the orifice (Point B), respectively.

The numerical results reproduced the compression and rarefaction wave distribution in the pipe. Notably, the compression wave region included a complicated distribution, due to the combined influences of reflected waves from both the orifice and the valve (Fig. 10).

Figures 11 and 12 indicate a generally good match between the experimentally observed results and the calculated results. The compression portion of the waveform was steep at the pipe end, verifying the process of growth toward a shock wave.

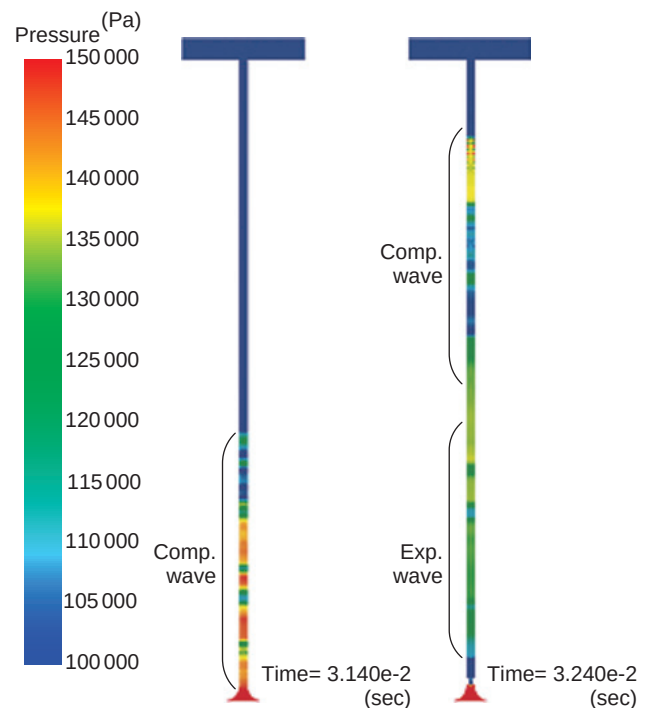


Fig. 10 Pressure distribution (contour map)

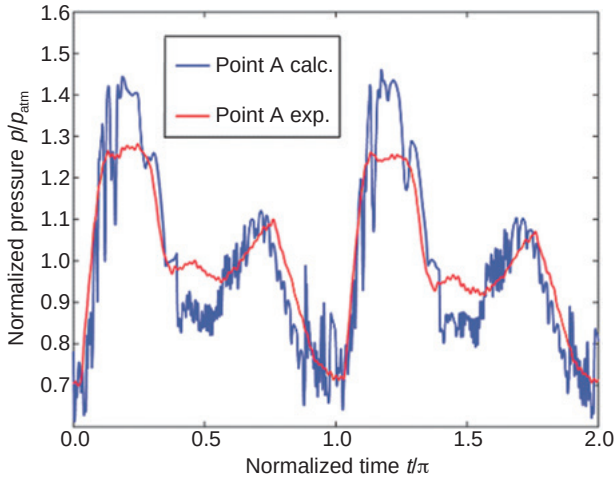


Fig. 11 Pressure history at Point A

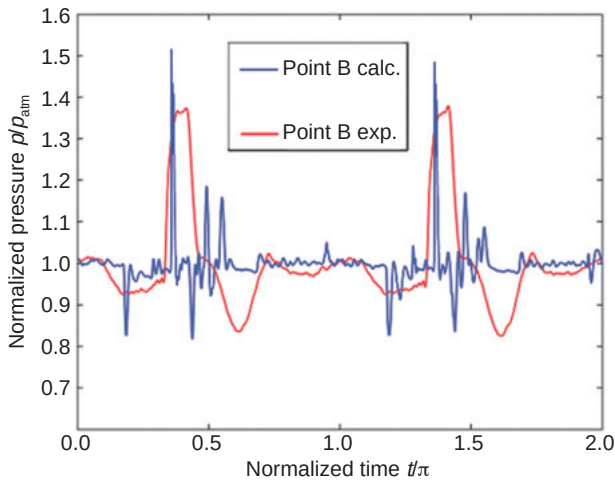


Fig. 12 Pressure history at Point B

4.7. Results of Numerical Calculations for Flow near Pipe Orifice and Orifice Flow

Figure 13 shows the calculated results in the vicinity of the orifice. Compression waves arriving at the orifice are shown to spread out in a spherical pattern centered on the outlet, developing into orifice flow including multiple vortex rings. This phenomenon was also observed in the visualization by Schlieren photography (Fig. 14).

It is common to use multiple-stage expansion structures in silencers to attenuate compression waves⁽¹⁰⁾.

Figure 15 presents the calculated orifice flow in the exhaust space. There is a procession of pressure fluctuation distributions centering on the vortex rings. The fluctuations in pressure increase as the vortex rings develop. This indicates that vortex rings are one of the main orifice noise sources⁽¹¹⁾.

4.8. Results of Numerical Calculations for Orifice Noise

The mean acoustic pressures in the frequency spectrum was found using multiple CFD nodes that comprised the measurement planes in the axially symmetric sound power model. The sound power levels were calculated and, in turn, the orifice noise in the sound field was obtained.

The general relationship between the acoustic energy density E [w·sec/m³] and acoustic source output W [w] is given by Eq. (1):

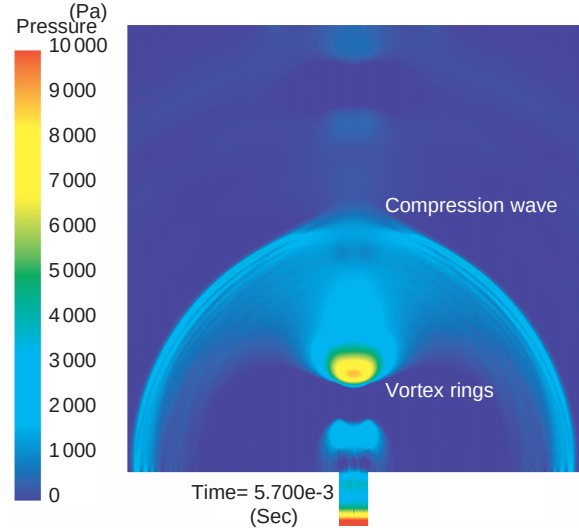


Fig. 13 Pressure distribution at exhaust orifice

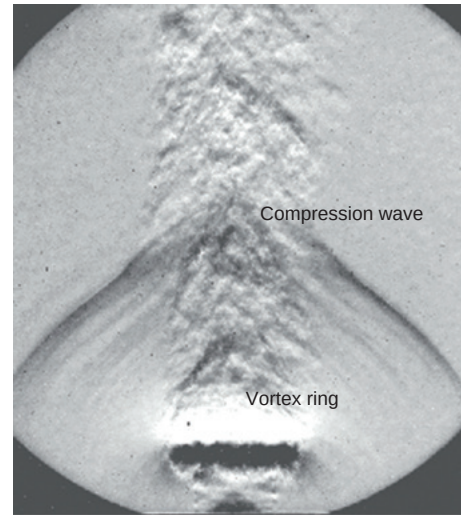


Fig. 14 Schlieren photograph at pressure ratio 4.0

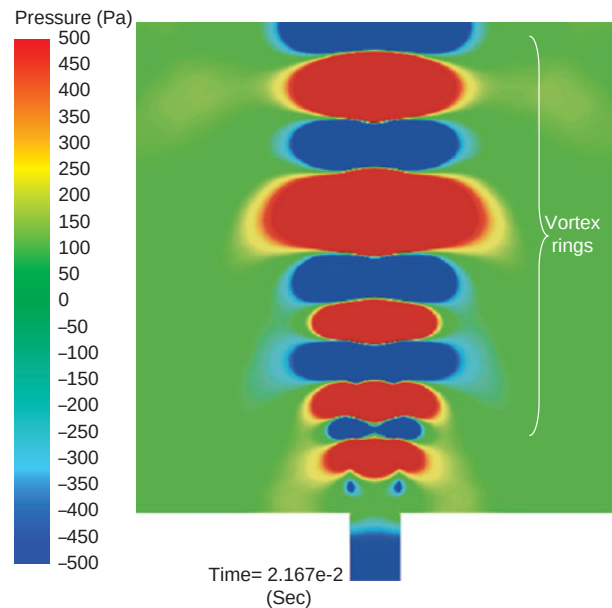


Fig. 15 Pressure distribution of vortex rings

$$E = \frac{W}{4\pi r^2 c} \quad (1)$$

r : Distance from acoustic source [m]

c : Sound speed [m/sec]

When the acoustic pressure at the observation point is L_{mic} [dB], the sound power level is L_w [dB] and $10E-12$ [w] = 0 dB, the following relation is found:

$$L_{mic} = L_w - 11 - 20\log_{10}r \quad (2)$$

The relationship between sound power level L_w and the mean pressure at the observation point for sound power L_p [dB] was

$$L_w = L_p + 10\log_{10}(S) \quad (3)$$

S : Area over which sound power is measured [m²]

This provided the orifice noise pressure level L_{mic} [dB] at the observation point.

Figure 16 shows a comparison of the calculated frequency spectrum and measured orifice sound pressure levels at the sound field 1 m from the orifice. The peak error was generally within 10 dB up to about 4 000 Hz, a comparatively good match. Above 4 000 Hz, the measured acoustic pressure levels tended to exceed the calculations. This is because of the short sound power levels in the vicinity of vortex rings at high frequencies used for these calculations. It is likely that the settings for the sound power observation point affects the precision of measurements of the air flow noise.

4.9. Mechanisms of Orifice Noise Generation according to Numerical Analysis Results

The numerical analysis clarified the following mechanisms of orifice noise generation:

- (1) The pressure distribution of compression-expansion waves was seen in the pipe and the compression waves showed a sharp pressure rise with distance along the exhaust pipe. Upon reaching the orifice, the pressure spread in a spherical pattern centered on the compressible wave arriving at the orifice. These phenomena were confirmed using high-speed exhaust pulsation generator and orifice flow visualization experiments.

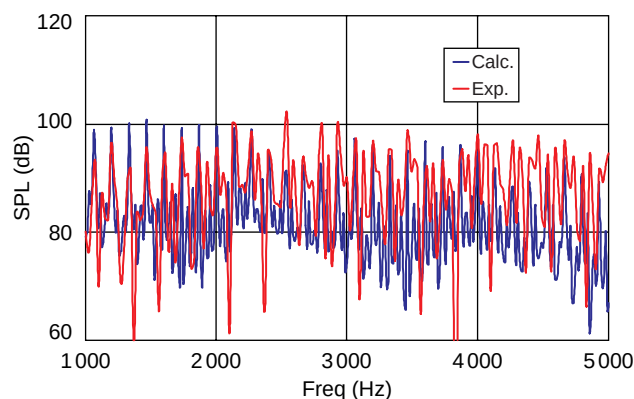


Fig. 16 Comparison of orifice noise at microphone (1 m)

- (2) Vortex rings were formed in the orifice flow region and a pressure distribution was observed to spread in the vicinity of this region with movement of the vortex rings. These phenomena, including the vortex rings, were confirmed using Schlieren photography to visualize the orifice flow.
- (3) The acoustic pressure spectrum of the sound field (1 m from the orifice), calculated using the sound power level in the orifice flow field, exhibits the pulsating noise that contains sharp peaks due to its multiplicity of compressive waves synchronized with the exhaust flow. The spectrum also has a broadband pressure distribution in the high-frequency spectrum, the main component of air flow noise, yielded by calculating the sound power of noise sources near the vortex rings. The calculated spectrum was confirmed to show nearly the same trends as those of the spectrum of the noise measured at the observation point.

5. Conclusion

On the basis of experimental and computational numerical analysis, exhaust orifice noise, which is the main component of exhaust noise, has been shown to occur due to the following mechanisms:

- (1) The main factors in the generation of exhaust orifice noise are pressure fluctuations occurring when compression waves and orifice flows in the exhaust pipe are ejected from the orifice. Orifice noise sources have been broadly classified into two types, propagation of acoustic waves due to compression waves centering on the orifice, and pressure fluctuations centering on vortex rings in the orifice flow.
- (2) Pressure fluctuations due to compression waves propagate into the sound field in a spherical pattern and become pulsating noise. They are synchronized with exhaust flow pulsations.

Multiple vortex ring acoustic sources move with the orifice flow. They grow, diffuse and become air flow noise, having a broadband pressure distribution in the high-frequency spectrum.

A numerical solution procedure created in this paper is confirmed to be effective in simulating exhaust pulsating flow and in clarifying these phenomena. PC calculation resources now allow flow simulations, using flow calculations limited to 2-dimensional axially-symmetric flow, and sound power calculation. A practical tool for analyzing the mechanisms of phenomena in exhaust orifice noise has been created.

Acknowledgements

The authors extend their appreciation to the following for their help with the experiments in visualization of exhaust orifice noise: Prof. Junjiro Iwamoto of the Mechanical Engineering Dept. at Tokyo Denki University for the Schlieren photographs visualizing high-speed pulsing flow, and Mr. Ono of Sankei Giken Kogyo Technology Center for the visualization photographs by non-steady acoustic holography.

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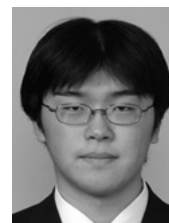
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Research of Low Heat-mass 4 in 1 Exhaust Manifold Noise Reduction Technology

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ABSTRACT

A noise generation mechanism of intermittent high-frequency noise during deceleration and high-frequency noise under load in a dual-thin-wall 4-in-1 exhaust manifold (hereinafter referred to as 4-1 manifold) have been investigated by using Computational Fluid Dynamics (CFD) and actual measurement of exhaust gas pressure fluctuation in the exhaust manifold. Modified gas flow in the exhaust manifold by CFD enabled a reduction in actual exhaust gas pressure fluctuation, which is the excitation force, and actual radiated noise has been reduced.

1. Introduction

As requirements for automotive emissions performance⁽¹⁾ become more stringent, increasing use is being made of the type of 4-in-1 (4-1) exhaust manifolds and manifolds with low heat-mass structures⁽⁴⁾ in order to enhance the temperature rise performance of the catalyzer^{(2), (3)}.

However, 4-1 exhaust manifolds and low heat-mass (thin-wall) structures generate more high-frequency noise than conventional type of 4-in-2-in-1 (4-2-1) exhaust manifolds, causing a decline of the quietness in cabin. Radiated noise levels were reduced in existing exhaust manifolds in one of two ways: either structural response is reduced by increasing the thickness of the walls, or separators are added into the collector area of the manifold to reduce excitation (semi 4-2-1). Both of these methods, however, directly increase heat mass, and are therefore incompatible with the enhancement of emissions performance. The purpose of the research discussed in this paper is to outline the noise generation mechanism in order to develop a technology capable of reducing noise by reducing excitation force, thus enabling radiated noise of a low heat-mass manifold equivalent to that of a 4-2-1 manifold.

The noise reduction technology developed in this project has been applied in the next mass production model, and has significantly reduced the high-frequency noise level radiated from the manifold under high load operating condition.

2. High-frequency Intermittent Noise at Deceleration

2.1. Characteristics of High-frequency Intermittent Noise at Deceleration

Figure 1 shows time history data for high-frequency intermittent noise, pressure measured at the exits of the manifold ports, the noise level close to the manifold (9k-12k [Hz] B.P.F.) and TDC pulse (2 [Pls./Rev.]) when the engine is decelerating.

These data indicate that high-frequency intermittent noise is generated 4 times for every 2 revolutions of the engine, and that negative pressure waves are generated in exhaust manifold ports 1 to 4 with the same timing. The timing of the negative pressure wave generation corresponds to the ignition timing in the test engine.

Figure 3 shows pressures measured at various parts of the engine shown in Fig. 2 in operation, plotted with crank angle.

During acceleration, as shown in Fig. 3(a), at exhaust valve open timing, combustion chamber pressure is higher than port pressure, and there are no negative pressure waves shown in Fig. 1 at the exit of the exhaust manifold ports. During deceleration, however, as shown in Fig. 3(b), fuel cut control suspends combustion and therefore at exhaust valve open timing, the combustion chamber pressure is lower than the pressure at the exits of the exhaust manifold ports. This induces a reverse flow, causing the rapid pressure drop at the exits of the ports, thus generating a

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negative pressure wave of the type shown in Fig. 1.

CFD was used to analyze the reverse flow of the gas, based on the assumption that this pressure fluctuation is the excitation force of high-frequency intermittent noise at deceleration.

2.2. Presumption of Noise Generation Mechanism by CFD

Steady flow CFD analyses were conducted using the model shown in Fig. 4 to analyze the reverse gas flow from the collector to the ports.

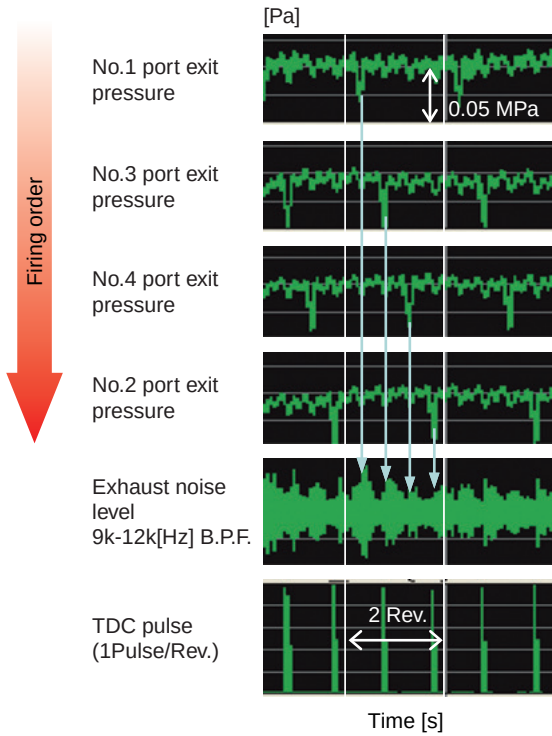


Fig. 1 Generating timing of high-frequency intermittent noises at deceleration

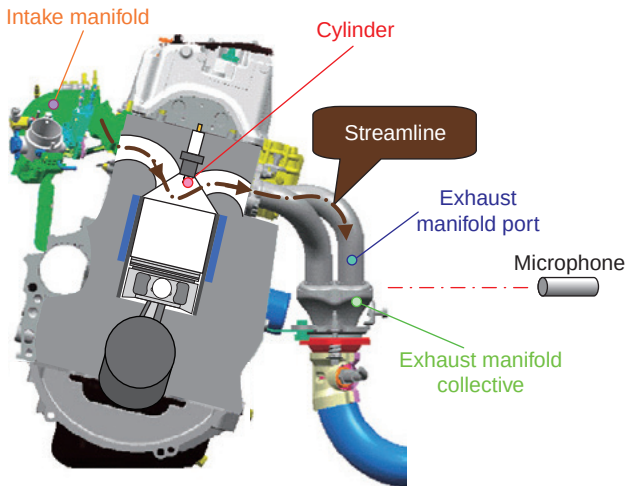
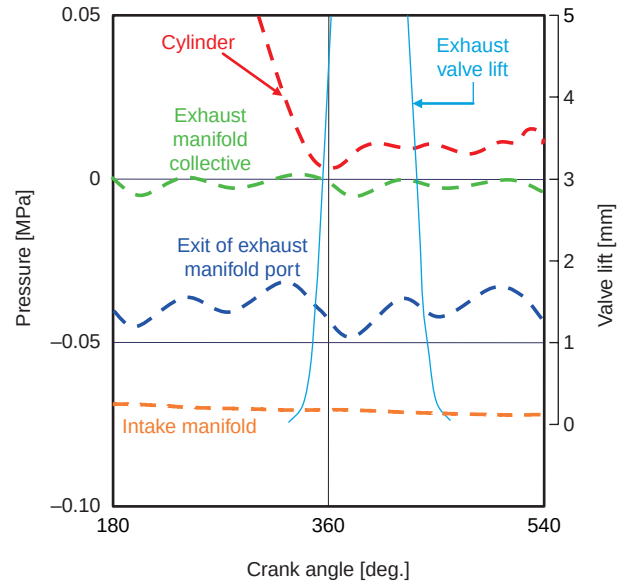
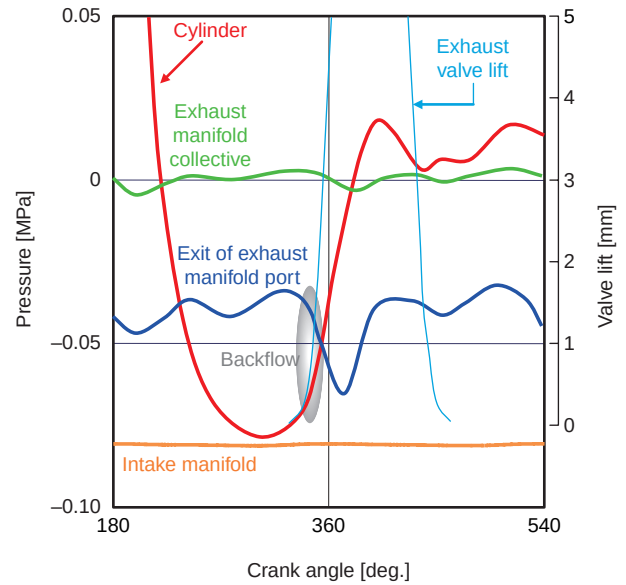


Fig. 2 Measurement position of engine and exhaust manifold pressure



(a) Acceleration



(b) Deceleration

Fig. 3 Pressure relationships of engine and exhaust manifold

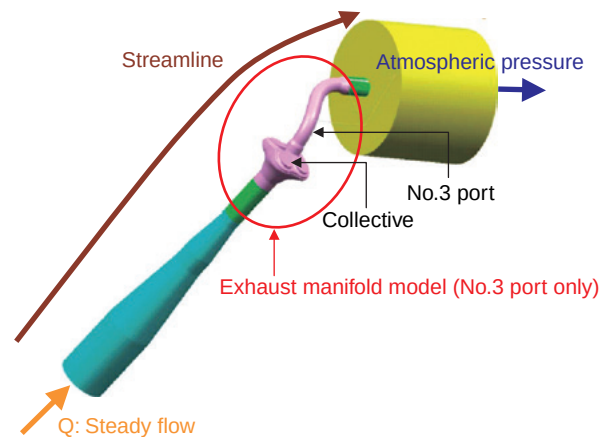


Fig. 4 CFD model for reversed flow analysis

Figure 5 shows the streamlines around the port collector by CFD analysis. As Fig. 6 shows, in a conventional dual-thin-wall configuration, the bulges near the edges of the ports enable the structure to slide, producing gaps at the port edges. When exhaust gas flows in reverse it is blocked by these gaps, and this is believed to produce an excitation force that leads the increase in the level of negative pressure waves. Calculations made with the gaps closed showed a reduction in reverse flow in the collector.

2.3. Effect of Port Edge Shape Modification

Figure 7 shows the negative pressure waves in exhaust manifold ports with and without the gap at the edge of the port exits, measured

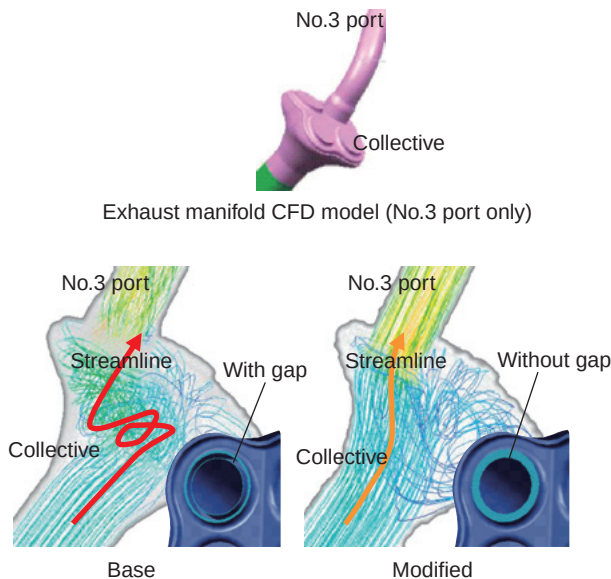


Fig. 5 Result of reversed flow streamline analysis by CFD

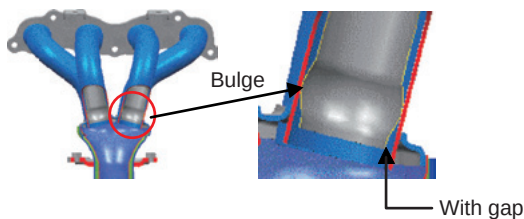


Fig. 6 Edge shape of exhaust manifold port

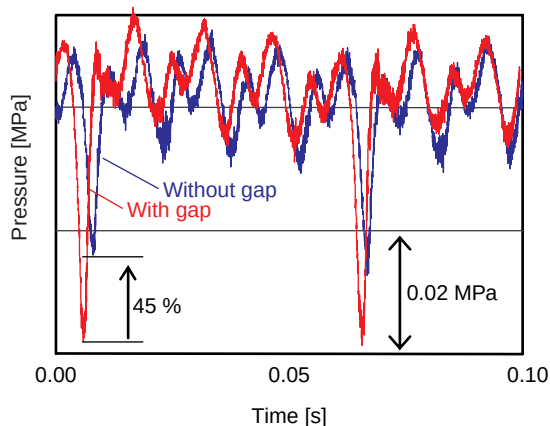


Fig. 7 Port exit pressure of effective edge shape

on an actual engine on the basis of the results of the CFD analysis described in section 2.2. The port without the gap produces negative pressure waves approximately 45% lower than that of the port with the gap.

Figure 8 demonstrates the effect of eliminating the gap at the edge of the port exits on high-frequency intermittent noise at deceleration. Intermittent high-frequency noise in the 9-12 kHz range is reduced by a maximum of 8 dB.

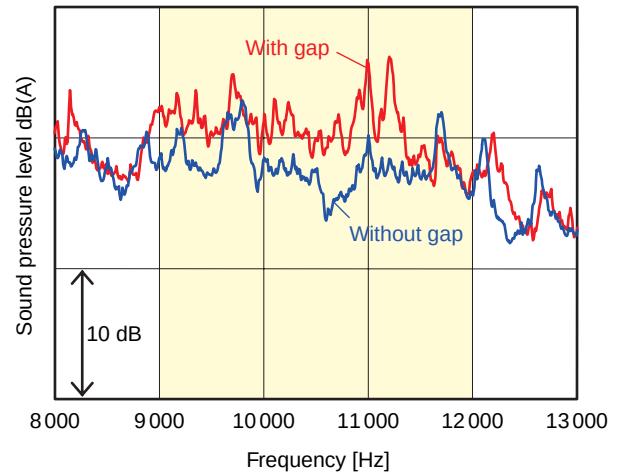


Fig. 8 Level of high-frequency intermittent radiation noises of exhaust manifold

3. High-frequency Noise when Engine is Under Load

3.1. Characteristics of High-frequency Noise when Engine is under Load

Figure 9 shows the high-frequency noise level measured close to the exhaust manifold when the engine is under load. The sound pressure level of the 4-1 low heat-mass manifold at frequencies of above 4 kHz is approximately 8 dB higher than that of the conventional 4-2-1 manifold.

In addition, as Fig. 10 shows, the exhaust gas pressure level in the exhaust manifold collector is approximately 7 dB higher at above 4 kHz in the 4-1 low heat-mass manifold than in the 4-2-1 manifold.

3.2. Presumption of Exhaust Gas High Frequency Fluctuation Generation Mechanism

CFD was used to calculate fluid velocity vectors and pressure distribution in order to determine the cause of the pressure fluctuation difference in the port collector of the manifolds.

Figure 11 shows an enlarged view of the edge of a port exit in a 4-1 low heat-mass exhaust manifold. Where the bulge causes the cross-sectional area to change, there is a change in the gas flow speed, causing pressure to increase.

Figure 12 shows results for (a) pressure distribution on the inner wall of the exhaust manifold and (b) fluid velocity vectors obtained by steady flow CFD.

Pressure on the wall of the port collector was higher in a 4-1 manifold than in a 4-2-1 manifold, as shown in Fig. 12(a). In

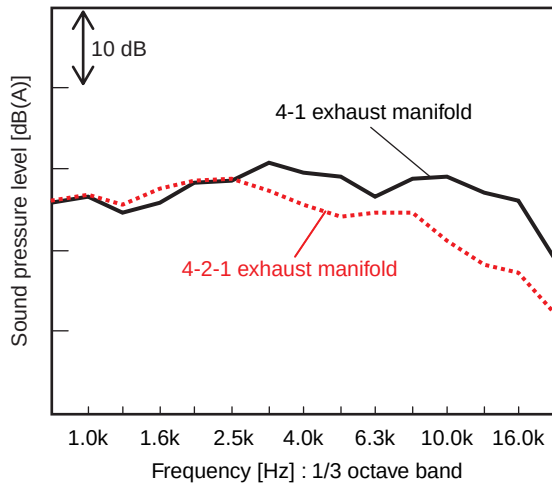


Fig. 9 Sound pressure level close to exhaust manifold

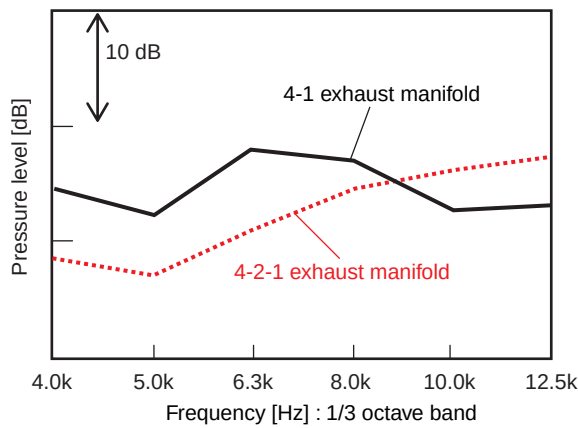


Fig. 10 Pressure at collective of exhaust manifold

addition, the fluid velocity vectors shown in Fig. 12(b) demonstrate that in the 4-1 manifold the flow path of gas flowing from the port to the collector is constricted, and the gas flow impinges on the collector wall.

These results led to the presumption that the cause of the high levels of high-frequency pressure fluctuation in 4-1 low heat-mass exhaust manifolds is the increase in the flow speed of gas flowing into the collector caused by the change in the cross-sectional area at the edge of the port, and by the impact of the gas with the collector wall.

3.3. Countermeasure Study of High-frequency Noise when Engine is under Load

Studies were conducted to determine a shape for the exhaust manifold that would enhance gas flow and reduce high-frequency exhaust pressure fluctuation.

(1) Modifications to change flow speed at edge of port exit

As Fig. 13 shows, the bulge was eliminated to make the port exit straight.

In addition, SUS mesh support was adopted as the method of support for the outer and inner pipes.

(2) Modifications to reduce gas impingement on collector wall

The incident angle of the ports was changed to 20° and the shape of the collector has also been changed according to this angle change

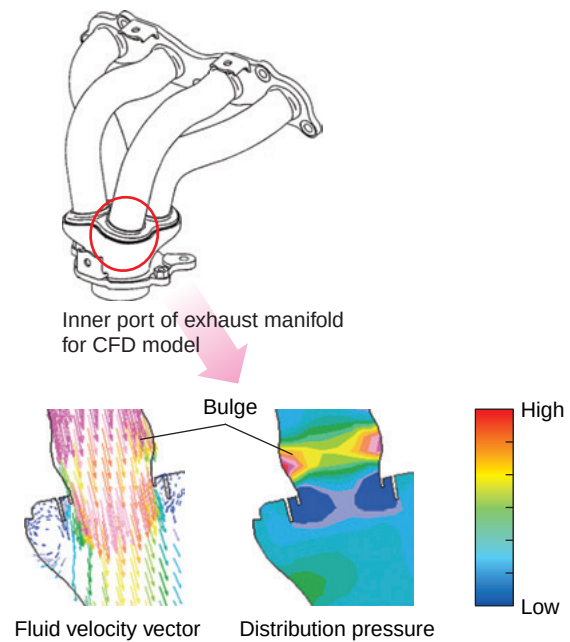
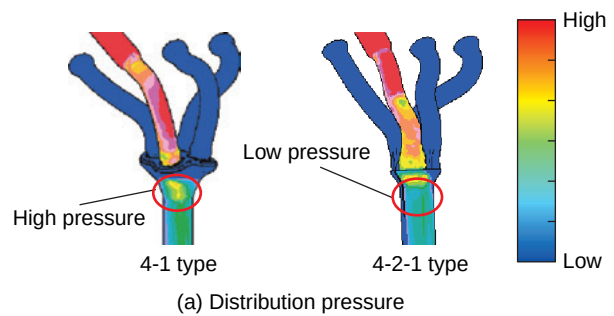
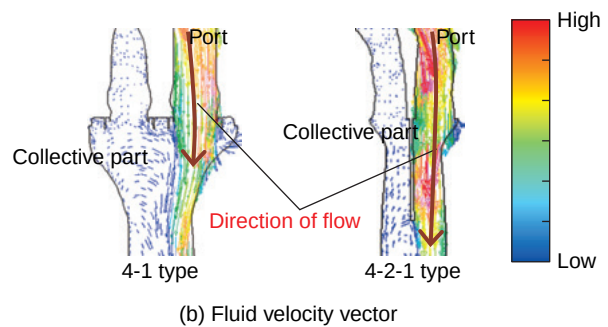


Fig. 11 CFD result for bulge effects



(a) Distribution pressure



(b) Fluid velocity vector

Fig. 12 CFD result for steady flow

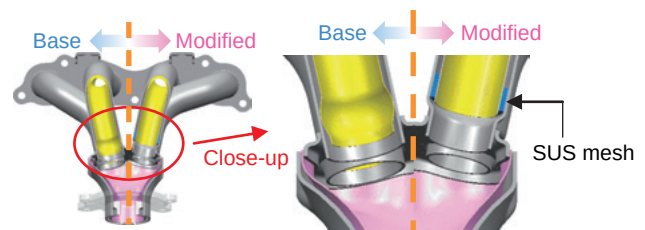


Fig. 13 Enhancement of edge shape of exhaust manifold port

as shown in Fig. 14.

Figure 15 shows CFD results for the modifications shown in Figs. 13 and 14.

The elimination of the bulge, in other words, straightening of the edge of the port, has reduced pressure at the port exits, and the incident angle change of the ports and the shape change of the collector has reduced the pressure increase caused by gas impingement on the collector wall.

3.4. Verification of Effectiveness of Modified Manifold in Actual Engine

Figures 16 and 17 respectively show comparisons of pressure at the edge of a port and in the collector, before and after modifications.

These results show that the change of gas flow has reduced the high-frequency exhaust pressure fluctuation that acts as the excitation force for high-frequency noise when the engine is under

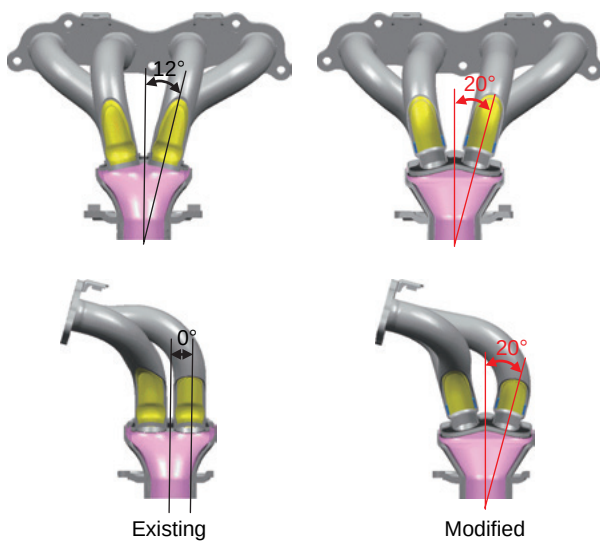


Fig. 14 Enhancement of angle of incidence of exhaust manifold port

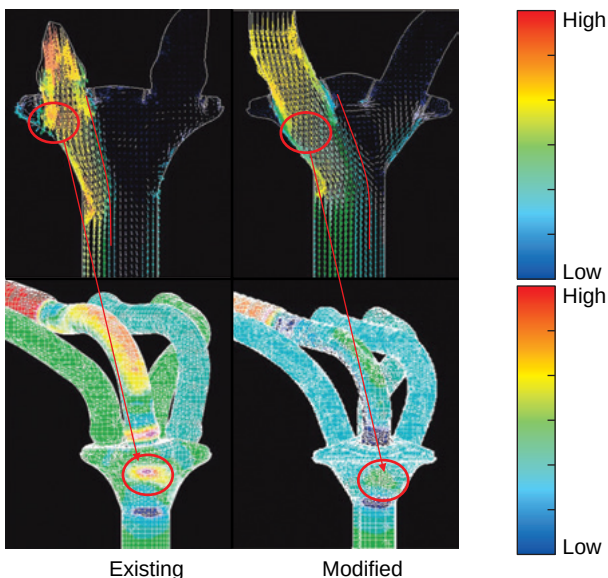


Fig. 15 CFD result for fluid velocity vector and pressure

load by approximately 6 dB.

3.5. Effect of Change in Inner Port Support Structure

Tests were conducted to verify the effectiveness of using SUS mesh as the support structure for the inner ports of the modified exhaust manifold shown in Fig. 13. Figure 18 shows the results of dividing the outer port vibration level by the inner port exhaust pressure level (excitation force), to compare the vibration level of the outer port per unit excitation force. Sensitivity in the high-frequency range has been reduced by 6 dB in the manifold employing SUS mesh support.

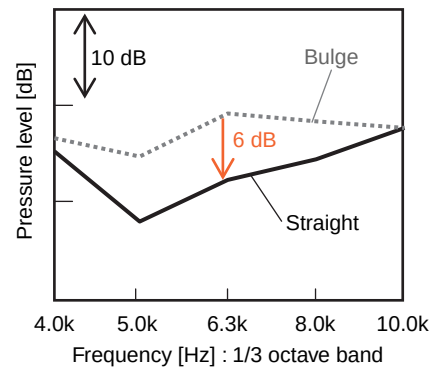


Fig. 16 Effect of edge shape on pressure level

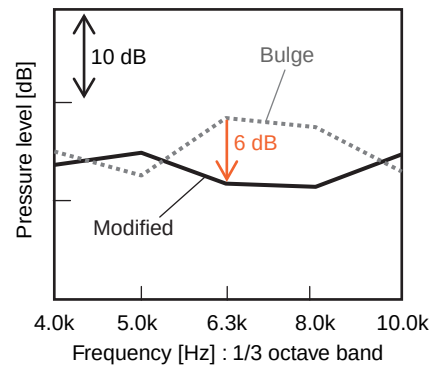


Fig. 17 Effect of enhancement of angle of port

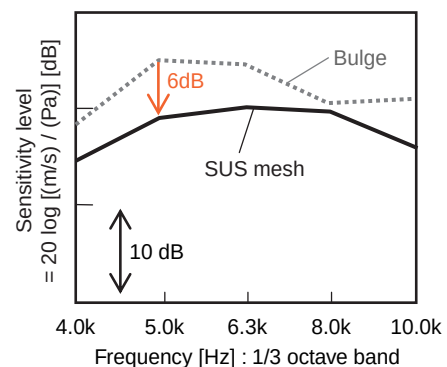


Fig. 18 Estimate of sensitivity to vibration/pressure of exhaust manifold

3.6. Effectiveness in Radiation Noise Reduction in an Actual Manifold when Engine is Under Load

Figure 19 shows measurements for noise from conventional and modified low heat-mass manifolds compared with target measurements from a 4-2-1 manifold.

The measurements show that modification of the flow in the modified exhaust manifold has enabled achievement of the initial target: the same level of high-frequency noise as produced by a 4-2-1 manifold when the engine is under load.

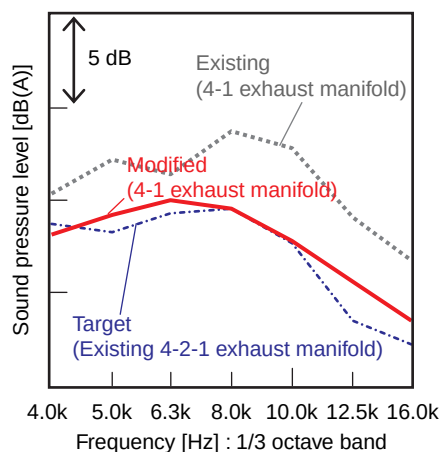


Fig. 19 Radiation exhaust manifold noise

4. Conclusion

This project outlined the noise generation mechanism of high-frequency intermittent noise during deceleration, and of high-frequency noise when the engine is under load, in a dual thin-wall 4-1 exhaust manifold. The development of a noise reduction method to reduce excitation force enabled the achievement of a noise level equivalent to that produced by a 4-2-1 exhaust manifold while retaining a low heat-mass structure.

- (1) This research demonstrated that high-frequency intermittent noise during deceleration is caused by reverse flow in the exhaust manifold. Elimination of the gap at the edge of the ports of the dual-wall manifold reduced the level of negative pressure waves, thus reducing the generated noise level.
- (2) This research demonstrated a correlation between the impact of the gas flow against the exhaust manifold collector wall and the generation of high-frequency noise when the engine is under load. Modification of the port exit shapes and the collector (angle of incidence of the ports) modified the gas flow, reducing the high-frequency component of exhaust pressure fluctuation.
- (3) The employment of SUS mesh support for flotation between the inner and outer pipes in the dual-wall structure was extremely effective in reducing surface vibration in the exhaust manifold.

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Reduction of Noise and Vibration in Compact Car Engine Mount System

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ABSTRACT

A new engine mount system has been developed to enhance noise and vibration performance and driving comfort, and thus increase competitiveness, in the next generation compact car. The basic system configuration is a principal axis of inertia type mounting system with torque rods that display excellent vibration isolation performance. In addition, the following were employed as guidelines for system characteristics in order to maximize performance. (1) Rod transmission characteristic that increases vibration isolation, to reduce booming noise and engine noise. (2) Rod mounting direction that reduces body shaking when the accelerator pedal is employed. (3) Control of engine rigid body vibration modes to increase driving comfort. (4) Dynamic stiffness of mounting points to reduce engine noise. The achievement of these characteristics has enabled the development of an engine mount system for a highly competitive compact car platform.

1. Introduction

A recent trend in automotive development is that vehicles in the same rank category have begun sharing platforms and developing derivative models such as sedans, station wagons and SUVs by interchanging the upper body and other parts. Sharing a common platform makes it possible to shorten the development period and reduce new costs, so development of platforms with high basic performance is demanded. On the other hand, booming noise while driving and engine noise are issues associated with principal axis of inertia type engine mount system commonly used in vehicles equipped with conventional in-line 4-cylinder engines. In addition, the noise and vibration produced by the large driving force and torque fluctuation vibration applied to the front and rear stoppers when starting are also issues for vehicles equipped with automatic transmissions and continuously variable transmissions (CVT).

This study aimed to construct a new engine mount system with

greatly enhanced noise and vibration performance and ride comfort as part of the development of a new public car platform with increased dynamic performance.

2. Overview

2.1. System Overview

Figure 1 shows the system configuration. The power plant is supported by the two mounting points located on the principal axis of inertia from the right side of the cylinder block towards the upper left part of the transmission, a torque rod that links the upper part of the cylinder block side mount with the top of the damper base, and a torque rod that links the bottom of the oil pan with the subframe. The structure is such that the power plant weight is borne by the two mounting points, and the driving reaction force is received by the two torque rods.

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2.2. Advantages of the Torque Rod System

The conventional principal axis of inertia type engine mount system has a structure which locates the stoppers that receive the driving reaction force in the vertical direction and supported by the front and rear beams of the subframe. The newly developed engine mount system with torque rods features a structure that receives the driving reaction force with the torque rods in the longitudinal direction as shown in Fig. 2. This structure has the following two advantages with respect to noise and vibration.

- (1) The torque rods do not simply receive the driving reaction force and control engine roll. The torque rods are elastically supported on both ends by rubber, and are comprised of rods made from steel, aluminum, plastics and other materials, so the vibration characteristics of the torque rods themselves provide significant vibration isolation effect.
- (2) The noise transfer function of the subframe, which is the input point for the vibration transmitted by the torque rods to the chassis, is generally lower in the longitudinal direction than the vertical direction. This means that an input format which is advantageous for idling vibration as well as noise and vibration when starting or otherwise operating at low engine speeds can be adopted by receiving the torque fluctuation vibration which acts in the direction of the crank axial rotation in the longitudinal direction⁽¹⁾.

The newly developed system makes maximum use of the above basic characteristics and applies the following technical elements to ensure high competitiveness. The torque rods are positioned horizontally for the reason in (2) above, but the mounting direction has been optimized to control the power plant behavior caused by the driving reaction force. Ride comfort is enhanced by employing

hydraulic engine side mounts and controlling the engine vibration in accordance with the spring characteristics of the transmission mounts. Dynamic stiffness targets that achieve the noise transfer function targets for the engine vibration transfer were set for each mounting point, and structural investigations were performed.

3. Basic Characteristics of the Torque Rod System

3.1. Reduction of Engine Booming Noise by Double Vibration Isolation Effects

A double vibration isolation structure establishes resonance systems in series within the vibration transfer path and provides significant vibration isolation effects after those resonance frequencies. Rubber floating of the subframe reduces the road and engine noise based on this principle. In the torque rod system, the rod weight and the spring setting of the rubber bushes on both ends make it possible to obtain these effects from frequencies of 100 Hz or less. Therefore, isolation of vibration can be expected to reduce booming noise and engine noise during driving. In vehicles with a conventional principal axis of inertia type mount system, booming noise caused by the 2nd order inertial force of a 4-cylinder engine is significantly affected by the side and rear mount vertical input as shown in Fig. 3. In contrast to this, the torque rod system provides significant isolation effects by attenuating the engine vibration at engine speeds higher than the rod resonance frequency as shown in Fig. 4. However, the vibration isolation effects decrease at the rod resonance frequency, so booming noise while driving is vastly

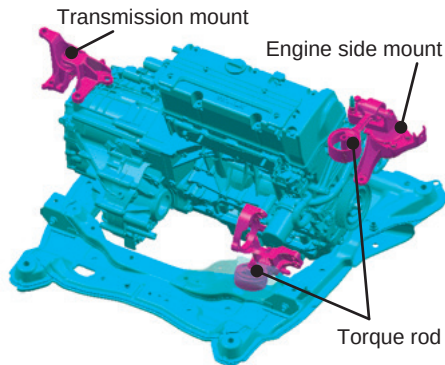


Fig. 1 Torque rod mount system configuration

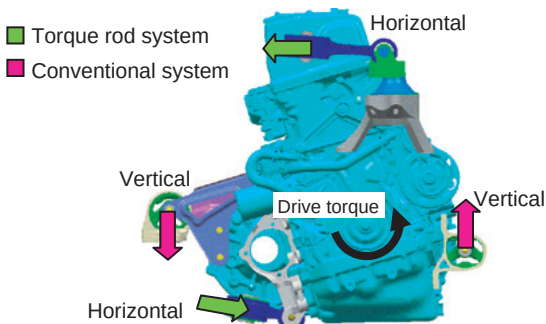


Fig. 2 Direction of principal input force

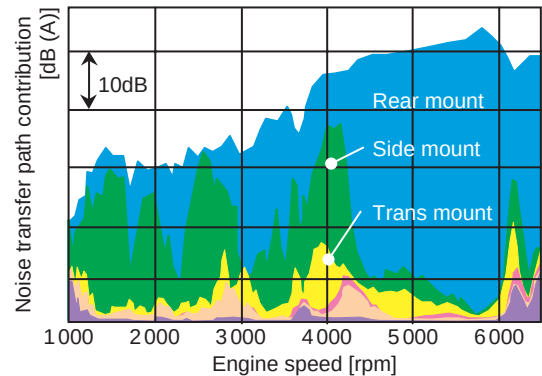


Fig. 3 2nd order booming noise transfer path contribution in conventional system

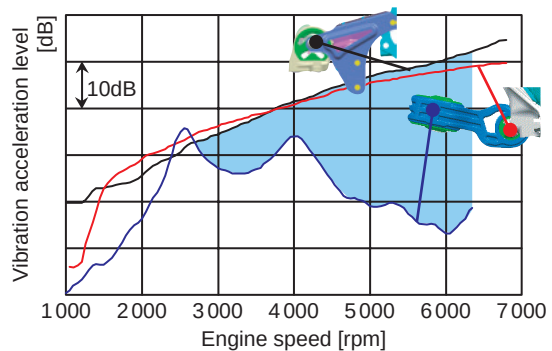


Fig. 4 Torque rod double floating effect in 2nd order vibration

reduced by setting the resonance frequency in a low sensitivity frequency band in view of the chassis sensitivity and other factors. Figure 5 shows the effects confirmed using a test vehicle.

3.2. Isolation of Torque Fluctuation Input

A large driving force is produced when a vehicle equipped with automatic transmission or CVT starts, and booming noise and vibration may occur due to an increase in torque fluctuation vibration and decrease of the mount transfer characteristics. In the conventional principal axis of inertia mount system, the vertical input component from the rear mount is dominant among the effects of the input from the engine mounts with respect to this phenomenon as shown in Fig. 6(a).

This is because the noise transfer function of the subframe is dependent on the following factors which are more severe in the vertical direction, as shown in Fig. 9.

- (1) The body shear bending mode shape, shown in Fig. 7 has a large effect on the sound pressure, and is easily excited by the subframe vertical input.
- (2) The subframe section shape is generally thin due to layout restrictions as shown in Fig. 8, which is disadvantageous in terms of vertical stiffness.

Therefore, in the principal axis of inertia type engine mount system with torque rods, engine vibration caused by driving reaction force and torque fluctuation can be input in the longitudinal direction, which is extremely advantageous with respect to these noise and vibration phenomena. Figure 6(b) shows the distribution of the sound pressure contribution for the principal axis of inertia

type engine mount system with torque rods, which shows that the sound pressure due to the main contributing factor of the torque rod longitudinal input is reduced, and that the overall sound pressure is also greatly reduced. Figure 10 shows the 2nd order booming noise reduction effects in a test vehicle.

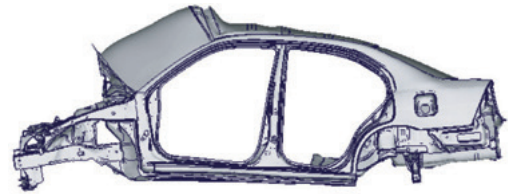


Fig. 7 Body bending mode



Fig. 8 Subframe section

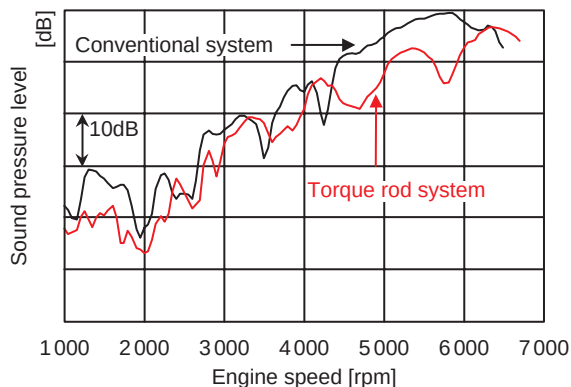


Fig. 5 Enhancement of 2nd order booming noise level

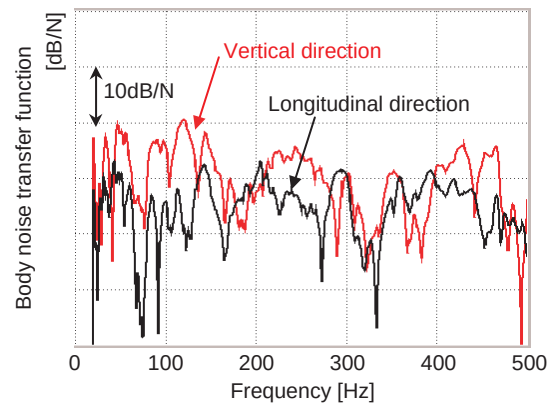


Fig. 9 Body noise transfer function of subframe

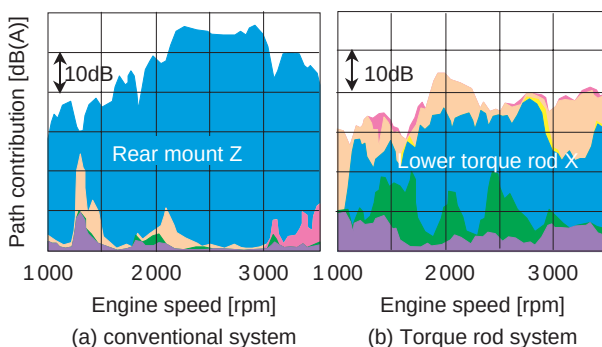


Fig. 6 2nd order booming noise transfer path contribution

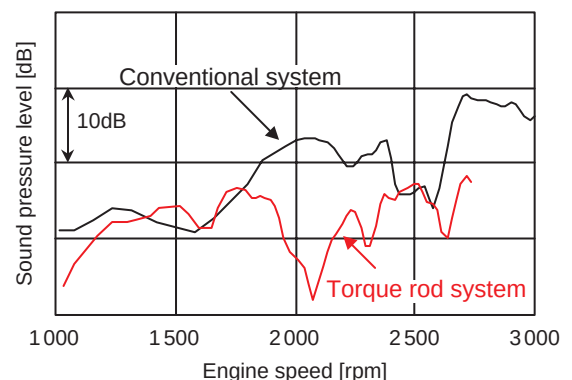


Fig.10 Enhancement of 2nd order booming noise

4. Torque Rod High Frequency Vibration Transfer Characteristics

Like booming noise while driving, engine noise performance can also be reduced up to a frequency of 1 kHz or more which is affected by the mount transfer characteristics by controlling the vibration transfer characteristics of the torque rods, based on the chassis sensitivity and other factors. This study defined the double vibration isolation effects of the torque rods by the transmissibility of the discrete torque rods as shown in Fig. 11.

$$\text{Transmissibility} = F_1/F_0 = X_1 \times K_1/F_0 = \int \int A_1 \times K_1/F_0$$

The torque rod transmissibility was quantitatively predicted using a discrete CAE model that takes into account the rod and the bushes on both ends, for which the rod shape and bush spring rates were determined. Figure 12 shows the calculation results for the transfer function. The vibration isolation effects generally tend to increase as the rod weight increases and the resonance frequency is set at low frequencies. However, vibration isolation performance can be satisfied by setting the peak of the torque rod transfer characteristics in the frequency band with low chassis sensitivity in the same manner as that of the booming noise region, described in the previous section.

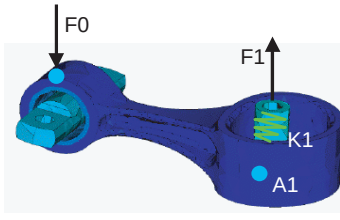


Fig. 11 Transmissibility simulation model of torque rod

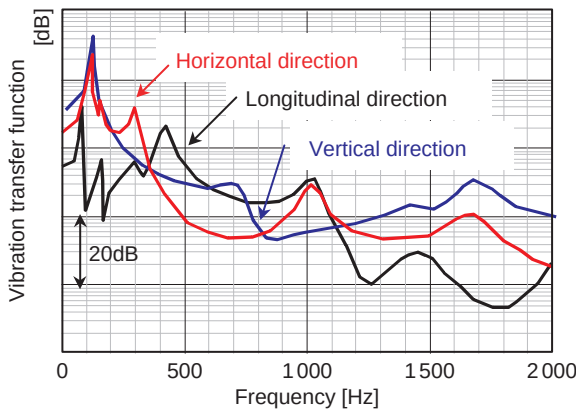


Fig. 12 Vibration transfer function of torque rod

5. Torque Rod Mounting Direction

The required rotational stiffness of the engine mount system is set to ensure that there are no part interference issues within the engine room and that the shock and body shake phenomena of manual transmission vehicles are at an acceptable level. In the torque rod system, the mounting direction is also important in addition to the bush spring rate. The torque rods have a small spring rate in the bush rotational direction, and rod displacement can only be restrained in the axial direction. Therefore, roll stiffness can be

increased by reducing the offset angle between the displacement direction of the rod parts that are connected to the power plant and the rod axial direction. On the other hand, the translational motion of the power plant may have an adverse effect on body shaking, but this can be enhanced by shortening the distance between the power plant center of rotation and center of gravity⁽²⁾. The roll center of the power plant when accelerating or decelerating can be controlled according to the torque rod mounting direction as shown in Fig. 13. Therefore, this study determined the torque rod mounting direction as shown in Fig. 14 to minimize the translational motion of the center of gravity. This reduced the body shaking by approximately 30%.

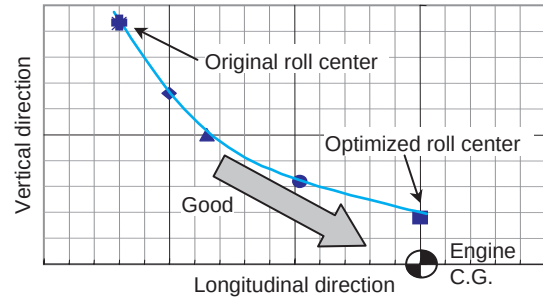


Fig.13 Optimization of engine roll center axis

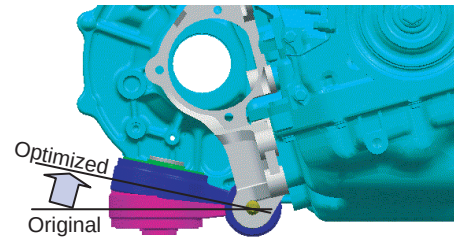


Fig.14 Optimization of torque rod angle

6. Enhancement of Ride Comfort

It is important to control the power plant rigid body vibration modes in order to reduce low frequency (5 to 20 Hz) chassis vibration due to the road surface input⁽³⁾. The newly developed system focused on the bounce mode and pitch mode (longitudinal axial rotation) of the power plant vibration, and performed mode control. The horizontal positions of the power plant center of gravity and the elastic center of the mount can be shifted as shown in Fig. 15, and bounce and pitch vibration can be excited with respect to the tire input by increasing the vertical spring ratio of the engine side mount and transmission mount. This produces a phase difference between the vertical vibration on the right and left sides of the power plant as shown in Fig. 16, which reduces chassis vibration by causing mutual interference between the inputs to the chassis. The results of bench vibration tests are shown in Fig. 17, and confirmed that vibration decreased by approximately 3 dB compared to the conventional model.

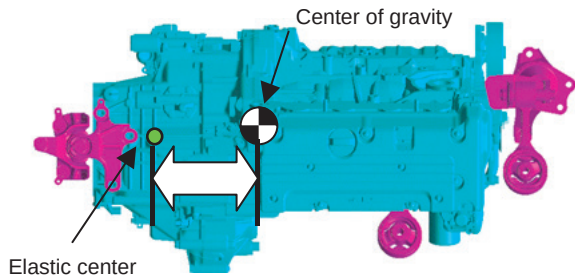


Fig. 15 Horizontal distance between engine center of gravity and elastic center of mount system

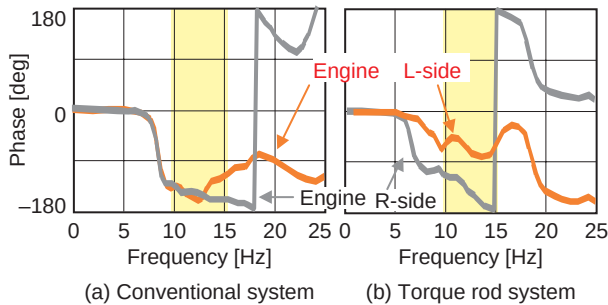


Fig. 16 Comparison of body transmissibility phase

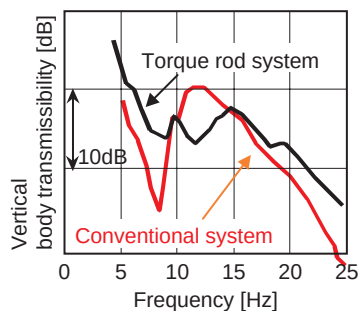


Fig. 17 Comparison of body transmissibility

7. Engine Mount Mounting Point Stiffness

In order to achieve the high frequency noise transfer function target for the engine mounting points, the correlation between the noise transfer function of the mounting points and dynamic stiffness values was first obtained through experiments. The noise transfer function and the dynamic stiffness values are roughly proportional within the same chassis, so the system target was replaced with the dynamic stiffness values and structural investigations were performed using a finite element model. The mounts fastened to the side frame employ mounting bolts with a collar structure as shown in Fig. 18(a), (b) and (c), stays were attached between the mounts and the wheel housing or the upper member as shown in Fig. 18(d) and (e) to increase lateral stiffness. In addition, the target stiffness was ensured by using aluminum die cast and steel as appropriate for the mounts and stays. Figure 19 shows the noise transfer function level achieved at the side engine mount. This shows that the acoustic sensitivity in the 100 to 300 Hz range can be greatly reduced by increasing the dynamic stiffness to the target value.

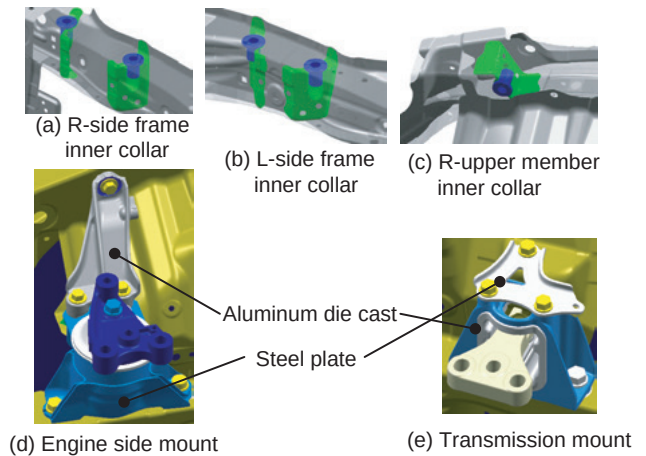


Fig. 18 Body construction in mount area

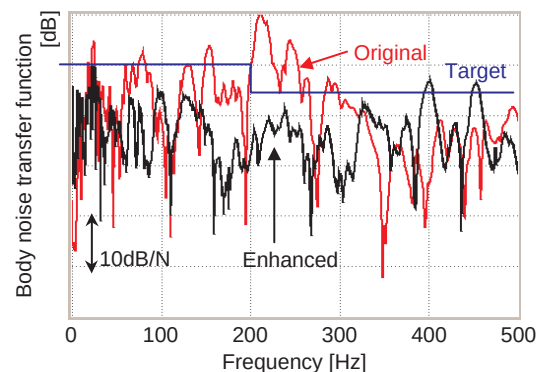


Fig. 19 Enhancement of body noise transfer function at engine side mount

8. Conclusion

The following technologies for reducing noise and vibration were applied to a principal axis of inertia type engine mount system with torque rods developed for a full model change for the next generation compact car.

- (1) Torque rod high frequency vibration isolation performance
Vibration isolation performance and weight reduction were simultaneously achieved by defining the double vibration isolation effects with the transfer characteristics of the discrete torque rods and performing optimization using a finite element model.
- (2) Torque rod mounting direction
Body shaking was reduced by ensuring rotational stiffness and by controlling the engine center of rotation.
- (3) Enhanced ride comfort
Chassis vibration was reduced by determining the vertical spring of the mounts on both sides so that the engine bounce vibration and pitch vibration are coupled.
- (4) Enhanced dynamic stiffness of mounting points
Engine noise was reduced by quantitatively setting dynamic stiffness targets that have high correlation with the noise transfer function, and by determining the structure based on these results.

Applying these technical elements in addition to the reliable basic noise and vibration characteristics of the torque rods made it possible to develop a highly competitive principal axis of inertia type engine mount system with torque rods for a public car platform.

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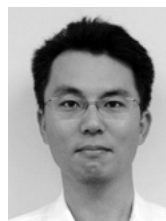
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Development of Simulation Technique for Element Stress Prediction of Metal Pushing V-belt

Toshihiro SAITO*

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ABSTRACT

A simulation method enabling simultaneous prediction of dynamic behavior and stress distribution on element has been developed for durability evaluation of dynamic strength of metal pushing V-belts. The finite-element-method that enables contact analyses in time history with large-scale model was adopted to reproduce the dynamic behavior of the V-belt in high rotational speed range of CVT. This paper focuses on the element strength in actual operation of CVT, and also discusses modifications made to the previously reported simulation method to enable the prediction of detailed stress. A new technique named inertia-relief is introduced needless of applying the constraint conditions when calculating the detailed stress on element in respectively. This results in allowing the stress distribution on any element to be found at any position on the trajectory of the V-belt. Using this technique, it has been found that the stress on the necks of elements at the entrance and exit of the pulleys affects the dynamic strength of the V-belt. Furthermore, a method of evaluating this stress was also determined. In addition, consideration was given to the effect of the elements tail side concave shape on the dynamic strength of the V-belt.

1. Introduction

Against a background of rising demands for responses to environmental issues, automotive manufacturers are directing attention towards CVTs as a technology enabling the achievement of a balance between driving performance and fuel economy. In terms of driving performance, CVTs offer quietness and smooth acceleration without shift shock. From the perspective of fuel economy, cooperative control between the CVT and the engine enables engine speed to maintain the high combustion efficiency, resulting in a higher level of fuel economy than is available from an AT. Therefore, the needs for increases in transmitting capacity of CVTs will become greater^{(1), (2)}. The important factor in attempting to increase transmitting capacity is ensuring the dynamic

strength of the metal pushing V-belt. Dynamic strength cannot be determined simply from the maximum input torque, but must be determined experimentally in relation to the shift ratio, pulley piston pressure as well as the maximum rated speed. This is because it is generally difficult to measure element stress, a factor which is indispensable in determining the strength of the belt in actual CVT operation^{(3), (4)}. It is therefore both technically and procedurally difficult to optimize the shape of the belt for higher-speed operation, making it necessary to develop a theoretical method of determining the effect of the shape of the belt, and in particular the shape of the elements, on dynamic strength⁽⁵⁾. A previous study⁽³⁾ by one of the present authors focused on dynamic stress on the multi-layered rings in actual operation of the belt, and developed a technique for simulating belt behavior. The effect of the deflection rigidity of

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the pulley on ring stress has also been studied⁽⁶⁾. This paper discusses the development of a method of calculating the detailed stress on individual elements using element contact pressure obtained from the belt behavior simulation mentioned above. This has made it possible for the first time to determine the stress on elements at any point on the circumference of the belt in the high rotational speed range. This paper will clarify the effect of element neck stress on the dynamic strength of the belt when the metal pushing V-belt is close to the entrances and exits of the pulleys.

2. Development Goals

Figure 1 shows a cutaway view of the main section of a CVT, and Fig. 2 shows a metal pushing V-belt manufactured originally. Drive torque is transmitted from the drive pulley to the driven pulley, by means of friction between the elements and the pulleys.

This friction is generated by there being pulley piston pressure constricting the belt. The stress on the elements and rings during vehicle operation is determined by the following factors: shift ratio, pulley piston pressure, engine speed and level of torque transmitted⁽⁷⁾. Tests have shown that the fatigue load of the pulleys and elements is lower than that of the rings⁽⁸⁾, and therefore reduction of impurities in the ring materials, to increase fatigue strength, has been adopted as a method of increasing the overall dynamic strength of the belt⁽⁹⁾.

Given the increasing requirements for expanding the ratio range of CVTs and employing them in combination with high-speed engines, determining the fatigue strength of the elements acted on

by centrifugal force is becoming more important. Modifications of materials have resulted in an increase in the fatigue strength of the rings⁽¹⁰⁾, where now the elements rather than the rings have become the points of failure in high-speed durability tests⁽¹¹⁾ (Fig. 3). It is essential to understand the behavior of the elements and the stresses produced on them when the belt is in operation to address this issue. Therefore, the research discussed in this paper attempts to develop a technique to enable prediction of the element stress, in order to clarify the mechanism of failure of the elements in high-speed operation.

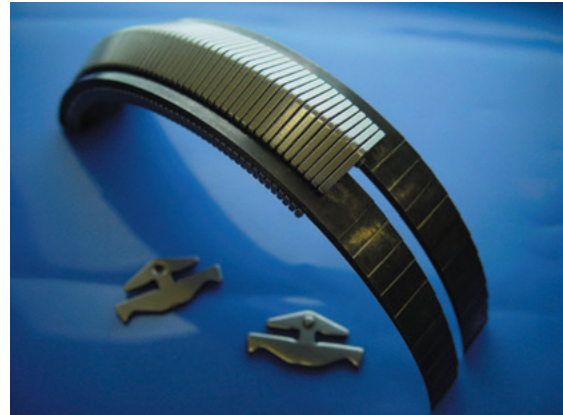


Fig. 2 Metal pushing V-belt

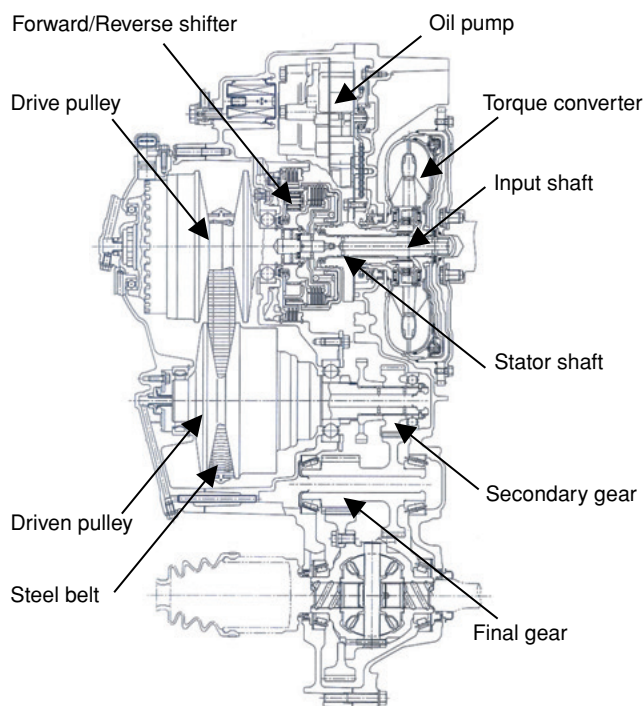


Fig. 1 CVT main section



Fig. 3 Damaged element

3. Technique for Element Stress Prediction

The simulation method used in this project is the standard method for predicting element stress, but too much calculation time was required when focusing on the dynamic stress on the multilayered rings⁽⁶⁾. However, employing parallel computation, using the latest domain decomposition, enabled calculations to be conducted in approximately the same time as that required for durability testing.

3.1. FEM Model of Metal Pushing V-belt

Figure 4 shows an FEM model of our original metal pushing V-belt. Two deformable solid elements are employed in the direction of thickness for individual elements to reproduce bending deformation. Three deformable solid elements are used to divide the R section of the element neck, where stress is thought to concentrate. Reproduction of detailed form of the nose and hole sections has been prioritized, and therefore modeled with rigid

elements. Four deformable shell elements have been used in the direction of width to model each layer of rings to enable reproduction of the crowning of the rings. In addition, the ring-element, element-element, ring-ring and element-pulley contact surfaces have been defined, and appropriate friction characteristics have been assigned in each case.

3.2. Pulley Rigid Surface Model

Pulley rigidity has a tremendous influence on the dynamic behavior of the belt, and connector elements were therefore used to reproduce deflection rigidity. Different rigidity values were assigned to the pulley that moves in the axial direction, and to the fixed shaft pulley. The rigidity values were reverse-calculated from figures for flexion obtained from applying a static belt load and pulley piston pressure to a 3D FEM pulley model. However, the pulley surfaces are defined as rigid surfaces. If they were modeled with deformable solid elements, it would be unlikely to reproduce the smooth surfaces of the pulleys.

3.3. Belt Misalignment

As it wraps around the drive and driven pulleys, the center of the belt does not move within a single plane, but displays slight deviations outside the plane. This deviation of the belt from the plane is termed misalignment. Durability tests have shown that a large misalignment reduces the lifespan of the belt. However, even if the pulleys were defined as rigid bodies, misalignment would not be eliminated; the size of the misalignment is determined by the distance between the axes of the pulleys and the shift ratio. A full 3D model rather than a plane symmetry model was therefore used in the simulation.

3.4. Analytic Procedure

Figure 5 shows the steps used in analyses employing the belt behavioral simulation. Finite element codes, which enable large-scale contact analyses over a time history were employed in the simulation. With the belt in its initial, no stress, and perfectly circular state, the pulleys are moved a fixed distance to produce a transmission ratio. Next, misalignment is imparted to one pulley

and pulley piston pressure is applied, as the drive pulley is gradually forced to rotate until the target speed is reached. During this procedure, counter-torque, representing the engine input torque multiplied by the shift ratio, is gradually applied to the driven pulley. This continues until rotation speed, pulley piston pressure and torque are judged to be steady.

3.5. Accuracy of Simulated System

In actual operation, feedback control of the pulley piston pressure of the pulley around which the largest diameter of the belt is wrapped is used to maintain the target ratio⁽¹²⁾. Because the pulley piston pressure is constant in the simulation, the shift ratio and the rotational speed of the driven pulley respond as a system. A time history of the response indicates the accuracy of the simulated system. As Fig. 6 shows, the target ratio is maintained when load conditions become stable.

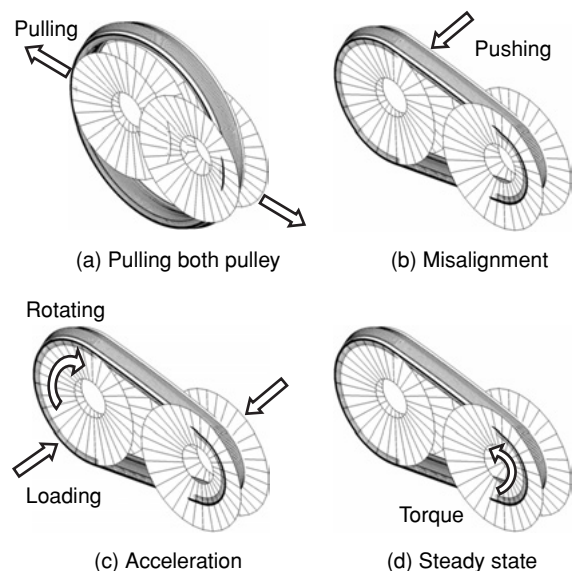


Fig. 5 Analysis procedure

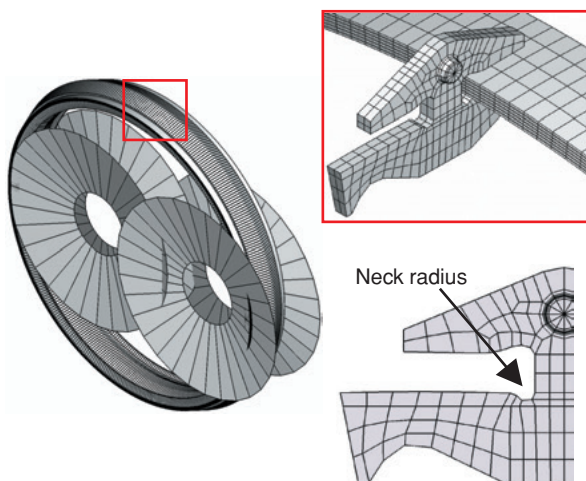


Fig. 4 Belt model defined as deformable bodies

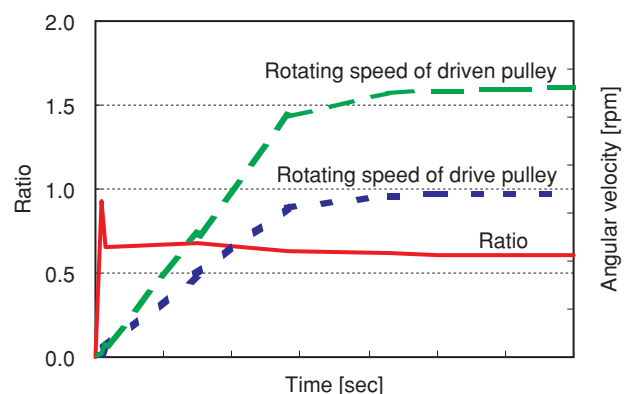


Fig. 6 Time history of ratio

3.6. Element Stress during Actual Belt Operation

Figure 7 shows a contour diagram of the stress on the elements in this simulation when the belt is rotating at a constant speed. The compression stress near the locking edges, the main cause of which is believed to be the pushing force between the elements, increases from the drive pulley entrance, reaches its peak close to the drive pulley exit, then disappears at the driven pulley exit. The stress on the element necks, the assumed point of failure of the elements, is highest as the belt approaches the entrances and exits of the pulleys. The stress generated by the pushing force is at its highest at the exit of the drive pulley. This is believed to be because of the reaction force generated on the element V surfaces at the instant the belt enters and exits the pulleys, as shown in Fig. 8⁽¹³⁾.

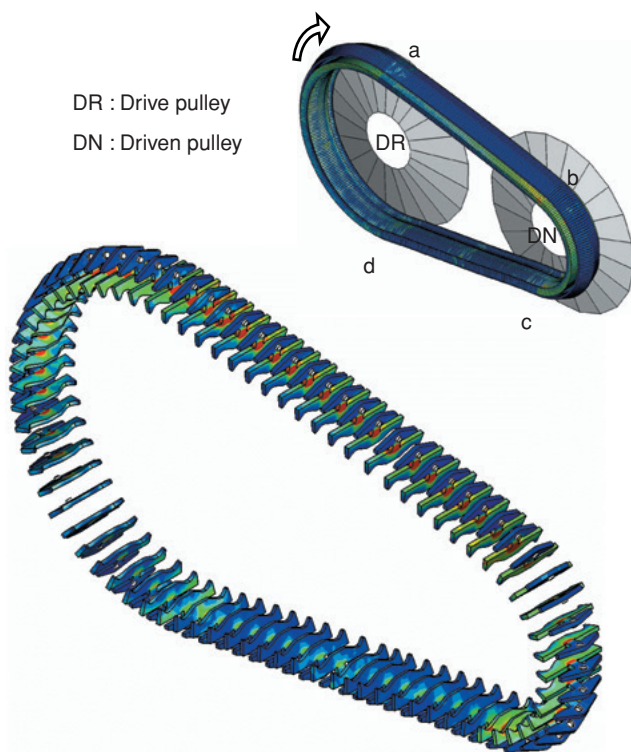


Fig. 7 Element stress distribution

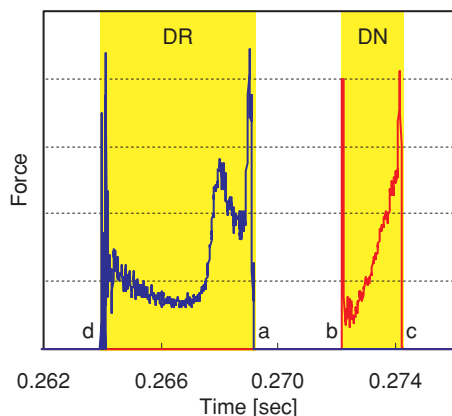


Fig. 8 Reaction force on V-surface

4. Development of Technique for Prediction of Detailed Stress on Elements

As indicated at the beginning of the previous chapter, the calculation time required makes it impractical to use a stress simulation in which all the elements are defined as deformable bodies in the development process for metal pushing V-belts. It was therefore decided to use results from the previously mentioned belt behavior simulation, in which the elements are defined as rigid bodies, to simulate element stress.

4.1. Inertia Relief Method

The detailed stress on the elements was found from this belt simulation model, and the distribution of the contact pressure acting on the elements was determined. This contact pressure was applied to a finely subdivided deformable element model of an individual CVT element. Normally, it would be unlikely to conduct a static stress analysis without applying constraint conditions at several nodes of the element model and stopping operation as a rigid body. However, when constraint conditions are applied, stress concentrates around the points of constraint, causing the originally deformed section to become fixed and preventing the achievement of an accurate stress distribution. A method of static stress analysis known as inertia relief was therefore employed. In this method, the inertia of residual forces is automatically applied to the center of gravity, in order to balance the element to which surface pressure is applied in space. This enables static stress analysis without the requirement for constraint conditions.

4.2. Distribution of Detailed Stress on Elements

Figure 9 shows the distribution of detailed stress on elements obtained using this method. This example shows elements approaching the entrance and exit of a pulley on the trajectory of the V-belt (Fig. 7). The results of stress analysis using the belt

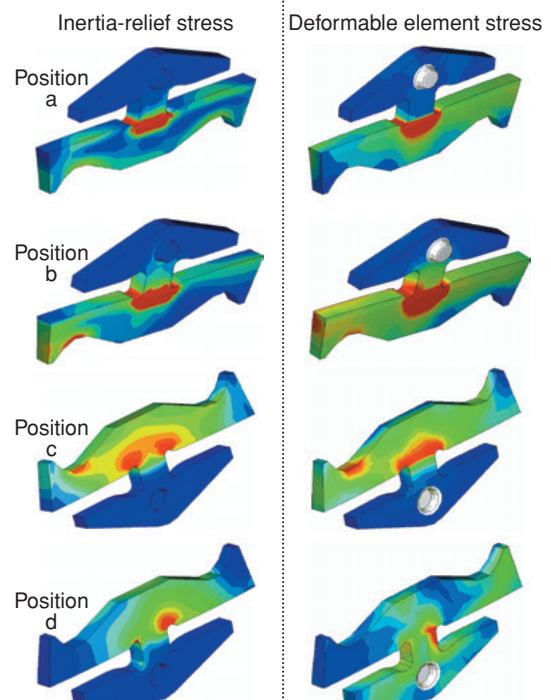


Fig. 9 Stress comparison at each position

behavioral model in which the elements are defined as deformable bodies are also shown. There are slight differences between the two sets of results, but they reproduce a basically similar stress distribution.

5. Element Strength in the High Rotational Speed Range

5.1. Engine Speed and Element Stress

Figure 10 shows the detailed stress on an element in the high rotational speed range. Element stress increases dramatically as

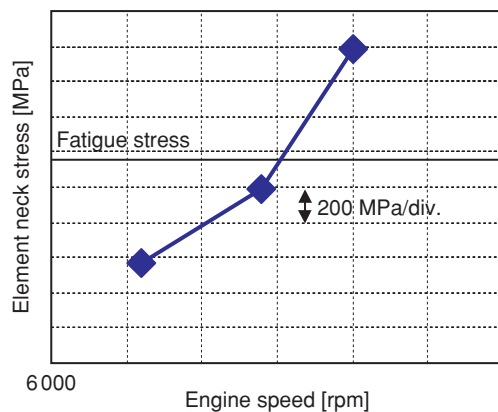


Fig. 10 Element neck stress influenced by engine speed

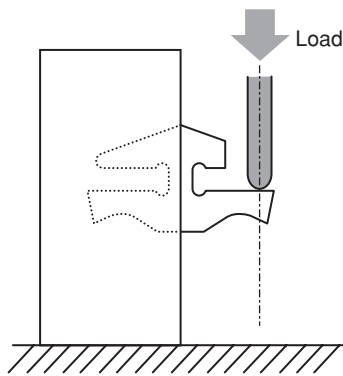


Fig. 11 Schematic of element fatigue test

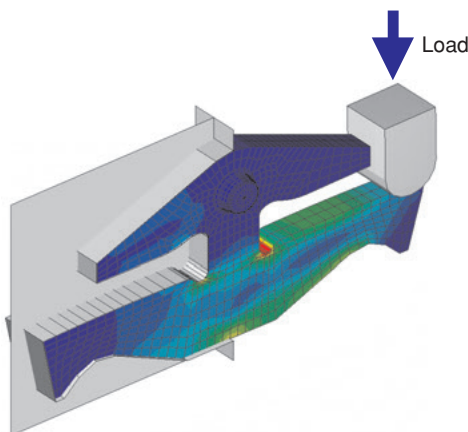


Fig. 12 Element fatigue stress

engine speed increases. This plot is for the maximum stress, generated at the entrances and exits of the pulley. Focus was placed on the base of the neck, R, of the element. In durability tests conducted under the same operating conditions, the belt comes closer to the limit of its dynamic strength as the speed of rotation increased. Simulation results for element stress showed a good correlation with test results.

5.2. Fatigue Strength of Individual Elements

Next, fatigue tests were conducted on individual elements to determine the accuracy of the stress on the elements in operation. Figure 11 shows the element fatigue test apparatus. In the tests, half of the element is inserted into a mold to fix it in position, and a sine load is applied to the tip of the element edge. This test enables the fatigue limit load to be ascertained⁽¹¹⁾.

Fatigue limit stress was determined using the analysis model shown in Fig. 12. The same finite element model was employed in this analysis as was that used in the analysis of detailed stress with inertia relief. The fatigue limit stress values determined from the element test correlated well with the results of durability tests, indicating its usefulness as a benchmark for quantitative judgment of the element stress when the belt is in operation.

6. Application to Metal Pushing V-belt Development

Element strength can be increased in a number of ways. Just as ring strength can be increased by attention to materials, heat treatment and shot peening can be applied to the elements to increase their surface hardness. However, the issue under consideration in this paper is the influence of the shape of the elements on the dynamic strength of the belt.

As Fig. 8 shows, the load on the element V-surface becomes a reaction force at the entrance and exit of the pulleys. This reaction force is closely related to the deflection rigidity of the pulleys. A previous study has demonstrated that reducing deflection rigidity reduces the reaction force on the elements, resulting in increased dynamic strength. The project therefore considered an element shape that would reduce the reaction force. As Fig. 13 shows, a concavity was added to the tail of the elements to reduce the area of contact between them.

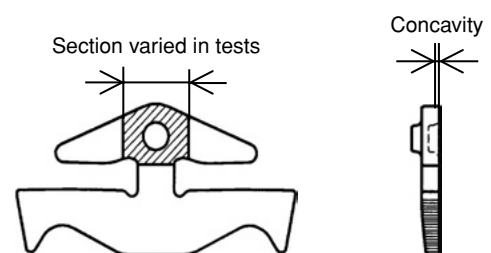


Fig. 13 Concave shape for control of contact area between elements

Elements with concave areas of differing widths were modeled and labeled X, Y and Z in descending order of width. Figure 14 shows results obtained from conducting the belt behavior simulation using these elements. The graphs show that the V-surface reaction force varies with the width of the concavity of the element. Focusing on the load peak at the abovementioned exit of the drive pulley shows that reaction force does not vary uniformly with changes in the width of the concavity, but that a minimum value exists. In durability tests of belts with elements of varying concavities, belt life increased as the V-surface reaction force peaks decreased. Case Y, which produced the lowest load peaks in the simulation, ensured the dynamic strength of the belt in high-speed operation, verifying the effectiveness of the method discussed in this paper. This phenomenon cannot be explained solely in terms of the rigidity of the elements; it can be explained only after element behavior and load have been identified.

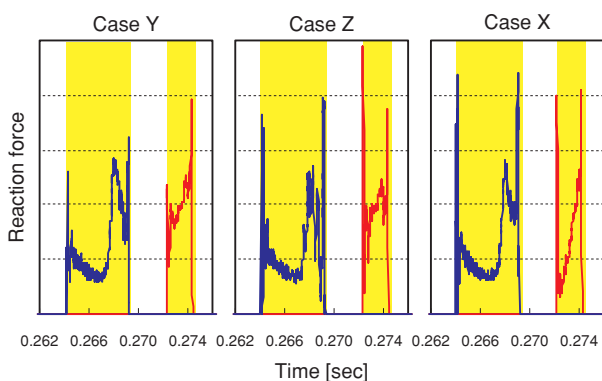


Fig. 14 Reaction forces on V-surface for each case

7. Conclusion

The following techniques were developed to study the dynamic strength of metal pushing V-belts in operation:

- (1) A static analysis technique that combined the inertia relief method and contact pressure drawn from a belt behavior simulation using rigid elements was developed.
- (2) This allowed the detailed stress on any element to be determined at any position on the trajectory of the V-belt.
- (3) Stress distribution determined using this method showed good correlation with the results of a belt behavior simulation using deformable elements.

The following results were obtained using the new technique:

- (1) The stress on the neck of the element, which is presumed to be the point of failure, is highest close to the entrances and the exits of the pulleys. This is because of the reaction force generated at the instant the belt enters and exits the pulleys.
- (2) At high rotational speeds, element stress rapidly increases, bringing the elements closer to their strength limits. Data on fatigue limit stress obtained from individual element tests provides a benchmark for judgment of element strength in actual operation.
- (3) Optimization of a concavity at the tail of the elements increased the dynamic strength of the belt in the high rotational speed range.

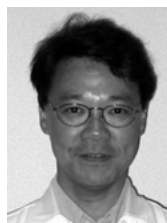
8. Afterword

Increasing the transmitting capacity of metal pushing V-belts has rapidly become a significant requirement in the automotive industry. The complexity of the subject makes a theoretical approach to be a challenge, and the authors hope that they can enhance analytic techniques to contribute to this field.

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Correlation of Subjective and Objective on Centre Steering Feel

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ABSTRACT

Previous studies have described a new analysis technique for objective measurements of passenger car steering systems carried out using the ISO standards. This analysis technique is based on deriving linearised stiffness and damping, which are objective metrics, from the non-linear results typically obtained for vehicles.

In this paper, the previous work is extended by carrying out a more comprehensive study of the correlation between these objective metrics and driver subjective assessments using a wide range of vehicles.

It is concluded that subjective assessments of on centre steering effort level, response level, friction and delay correlate well in many cases with the proposed objective metrics.

1. Introduction

Due to the common use of multi-lane, relatively straight, high speed roads (Autobahns, Motorways, etc.) the response of passenger cars to small angular inputs made by the driver is important. These small angular inputs (up to ± 20 deg at the hand wheel) are known as on centre steering feel.

In the past, on centre steering feel has been judged subjectively using expert drivers and some attempts have been made to objectively quantify the vehicle's response for a given condition of steering input speed and amplitude^{(3), (4)}. However in real situations these conditions vary considerably along with the dynamic response of the vehicle, especially steering torque which is strongly non-linear with changing steering input speed (frequency) and amplitude.

Recently a measurement technique was developed^{(1), (2)} which quantified on centre steering feel objectively and in a concise manner, as steering input frequency and amplitude is varied. These papers^{(1), (2)} detail the use of simple comparison studies with subjective evaluations to show that the objective measurement technique gave a good representation of driver subjective judgement.

This paper details a continuation of the research, where a more comprehensive study of the validity of the technique was undertaken using a wide range of passenger cars. To achieve this, a number of different vehicles were assessed subjectively by an expert driver on public roads and have been measured using the objective technique on a test track. Finally a detailed correlation between the objective and subjective results to ascertain the accuracy of the objective measurement technique in characterising the on centre steering feel of passenger cars is given.

2. Application of the Objective Technique

The objective measurement technique needed to be implemented in an efficient and portable manner to enable rapid testing of multiple vehicles. This involved the development of suitable instrumentation and software algorithms to not only acquire the data, but also to post process and display the results. All the testing and data processing was done to ISO136704-1:2003(E)⁽⁴⁾ standard to help achieve good repeatability of results.

The vehicle was equipped with: an instrumented clamp-on steering wheel to measure angle and torque; a gyro to measure

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yaw velocity at the centre of gravity and a GPS speed sensor to give an accurate indication of forward velocity. Data acquisition equipment was used in conjunction with a laptop to convert the analogue signals into digital data which was scaled, filtered, displayed and saved to the hard disc as ASCII files. Whilst trying to replicate each sine wave at each required frequency and amplitude, the driver was given audio and visual cues from the data acquisition software to produce accurate steering input.

The post processing stage was split into two halves, the creation of the steering hysteresis loops and then the parameter derivation. To attain the hysteresis loops, the continuous time history data was manually searched for the set of waveforms which matched the required waveform the closest. To allow an average to be made from the results, at least three continuous waveforms were found and then used to give the steering hysteresis loops, see Fig. 1.

The second stage involves using software algorithms which apply the “analytical method of equivalent linear system using the restoring force model of power function type”^{(1), (2)} to derive a set of parameters from these steering hysteresis loops. The exact generation of the parameters is given in detail in⁽²⁾, but can be briefly explained as calculating the equivalent spring stiffness K and damping coefficients C for an equivalent non-linear system. The derived K and C objective metrics are linearised metrics obtained from the non-linear steering hysteresis loop data and hence, they are functions of steering amplitude and steering speed.

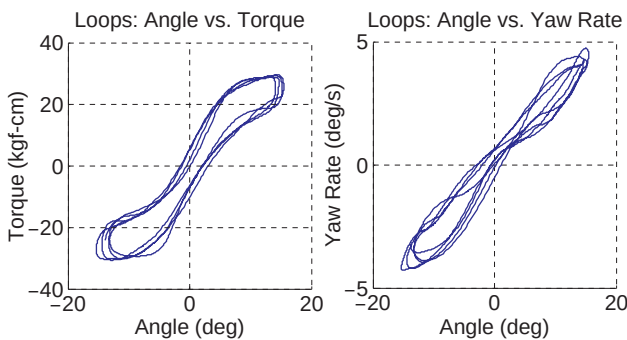


Fig. 1 Data processing stage 1: Steering hysteresis loops

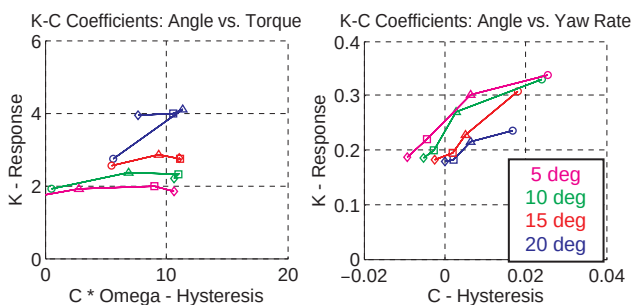


Fig. 2 Data processing stage 2: Steering characteristic parameters at various steering frequencies and amplitudes for a single vehicle

This is done for each waveform in the time history, which are then averaged together to generate four parameters for each steering frequency and amplitude (K_p, C_i – from steer torque versus angle data and K_y, C_y – from yaw rate versus angle data). Figure 2 shows how, for a single vehicle, these parameters vary with steering frequency and amplitude. Each coloured line is at constant steering amplitude and indicates a speed change from 0.25 to 1 Hz.

3. Subjective Evaluation

The vehicles were evaluated for steering feel at motorway speeds whilst driving on public roads, using small steering displacements. This evaluation was also complemented by further assessment on the test track at the test speed (140 km/h) to confirm any changes of behaviour with increasing speed. All the assessment was undertaken by an expert driver with six years experience and his results were also checked against the opinion of other industry experts.

The vehicles tested were split into four categories:

- Compact passenger (7 vehicles).
- Larger passenger (6 vehicles).
- SUV (3 vehicles).
- Sports (1 vehicle).

Although different perceived performances were expected from each of the categories of vehicle, an attempt was made to judge all the vehicles on the same scale.

The main subjective assessment criteria were:

Effort Level: Amount of steering effort required for a given steering input. Vehicles with higher steering efforts were given a higher score.

Response Level: Amount the vehicle reacts to a given steering input. Vehicles with a larger response to a given input were awarded a higher score.

Friction Feel: This was evaluated by the difference in torque between applying and removing lock, and the linearity of torque build-up for small inputs.

Delay: The time lag between steering input and torque build-up, and time lag between input and vehicle response.

4. Objective Tests

Objective testing was performed for all vehicles according to the test method proposed by Tokunaga et al.⁽¹⁾, which describes a series of sinusoidal steer tests at 140km/h, with steer amplitudes ranging from $\pm 5^\circ$ to $\pm 20^\circ$ and steer frequencies from 0.25 Hz to 1 Hz.

Analysis of the relationships between steering angle and steering torque, and between steering angle and yaw velocity is performed using the method described above⁽²⁾. For all steering tests K_p, C_i (from steer torque versus angle data) and K_y, C_y (from yaw rate versus angle data) was found. For the Torque versus Angle curves, the damping coefficient C_i is multiplied by the effective rotational velocity ω (referred to as omega), in order to evaluate the actual damping force.

4.1. Correlation between Objective Data and Subjective Assessment

For each of the subjective assessment criteria, an objective metric was selected from the range of tests performed on all the vehicles.

Many of the metrics were selected from the 0.25 Hz tests, as this frequency is the most typical of normal driver inputs⁽³⁾. The $\pm 20^\circ$ tests were chosen for all the metrics because of their lower sensitivity to road input and other outside disturbances.

Correlation studies could be shown between all test frequencies and amplitudes, but these single values are used here not only for the reasons given above but for conciseness.

Figures 3 to 8 show scatter plots between the relationships of the subjective data and objective metrics. In each case a linear least squares fit was made and the residual values calculated $R^{(2)}$. The R^2 value gives an indication of the accuracy of the linear least squares fit and thus the linearity of the results and therefore correlation between the subjective and objective data. The R^2 value should be above 0.8 (80%) to show a good linear correlation.

4.2. Effort Level

This was correlated with the gain of the Torque vs. Angle K_t relationship at 0.25 Hz. – chosen because the torque gradient can change significantly due to delay characteristics at the higher frequencies.

Figure 3 shows a high level of correlation between the subjective assessment and the objective criteria.

4.3. Response Level

This was correlated with the gain of the Yaw velocity vs. Angle K_y relationship at 0.25 Hz. Again, this frequency was chosen because the response gradient changes due to delay characteristics at the higher frequencies.

Figure 4 shows there is a relationship between the data, although there is one vehicle that falls far from the linear fit (car A). This vehicle was the sole ‘Sports’ vehicle with subjectively a low yaw gain compared to its transient lateral acceleration build-up. This may have affected the subjective evaluation of yaw response. Otherwise, a good correlation is seen for non ‘Sports’ vehicles.

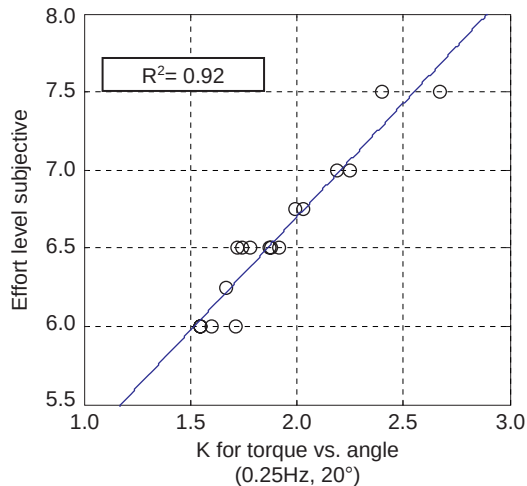


Fig. 3 Effort Level correlation

4.4. Friction Feel

This was correlated by the $C_t * \omega$ value of the Torque versus Angle relationship at 0.25 Hz. The damping coefficient changes significantly with frequency due to steering delay and steering system damping effects.

In Fig. 5 there is a clear relationship between the subjective and measured data, although there is more scatter when the car is rated highly. It is expected that other effects (damping, delay) become more significant when friction levels are lower.

4.5. Delay

Delay was correlated with two metrics. Firstly, the delay in vehicle response, which is measured by the C_y value of the Yaw Velocity versus Angle relationship at 1Hz⁽³⁾. This frequency is shown to give the highest value of delay for most vehicles.

Figure 6 shows no strong correlation between the data, although it does show a relationship between the subjective results and the measured data. Further study did not produce a better correlation.

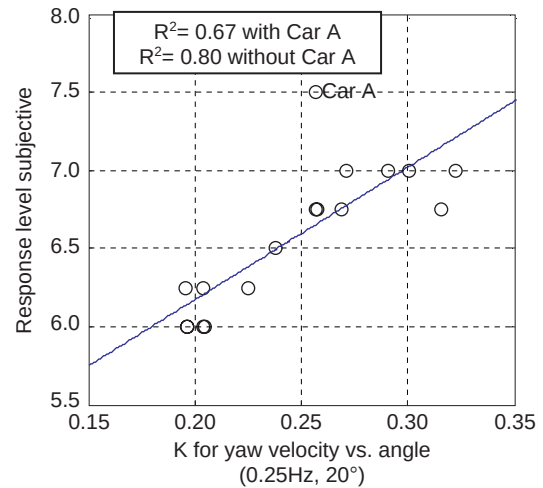


Fig. 4 Response Level correlation

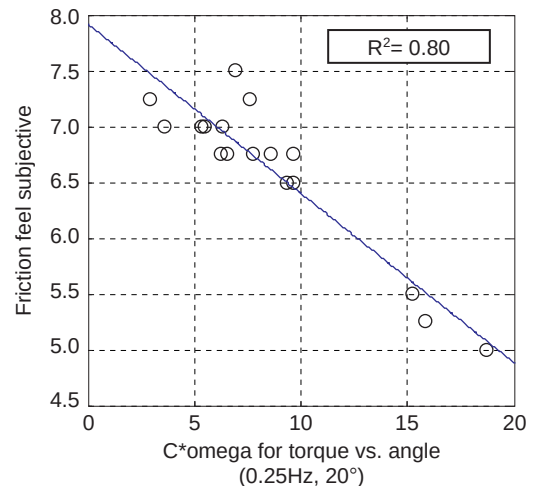


Fig. 5 Friction Feel correlation

The second metric for delay was steering torque build-up delay, which is measured by the reduction in $C_y^* \omega$ values between 0.25 Hz and 1 Hz. A large reduction in damping values corresponds to a narrowing (or even a reversal) of the hysteresis band around the zero steering angle region due to increasing delay in steering

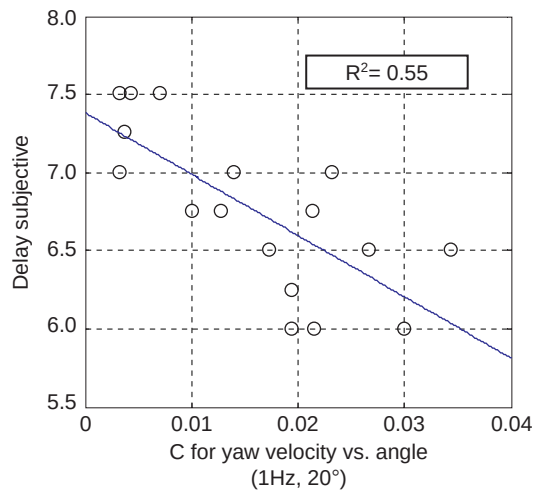


Fig. 6 Response Delay correlation

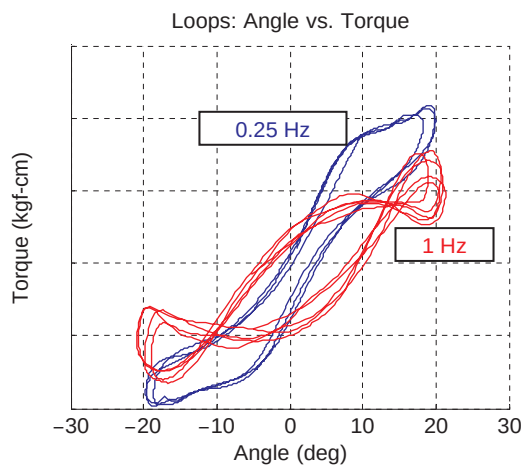


Fig. 7 Effect of increasing frequency on steering torque hysteresis

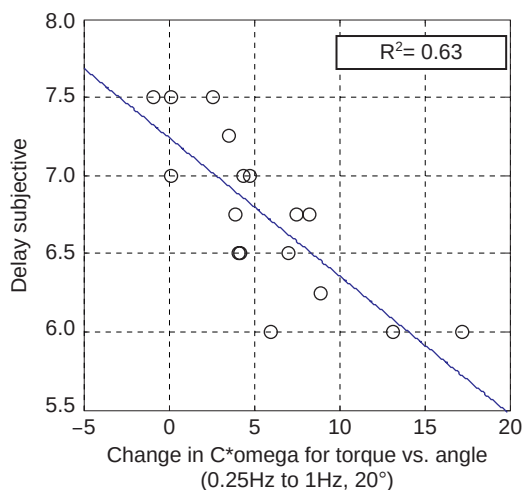


Fig. 8 Torque Delay correlation

torque build-up, see Fig. 7.

Finally, Fig. 8 shows no strong correlation between the data, although it does show a relationship between the subjective results and the measured data.

5. Conclusion

The paper has given details of an extensive subjective and objective study of vehicle steering feel.

These results have then been used to show that the objective metrics proposed by Tokunaga et al.⁽¹⁾ give significant correlation in several important areas of steering feel, namely Steering Effort and Response Level, Friction Feel and Delay, for a wide range of passenger vehicles.

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Assistance Method of Obstacle Avoidance Maneuver by Steering Operation

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ABSTRACT

Assistance types and control methods were studied, in regard to the system to assist driver maneuvers in avoiding obstacles through steering. From the principle that the driver is the focal point of driving operation, a “driver reactive” type of assistance was selected, wherein assistance begins after the driver initiates an avoidance maneuver. Actualization of a four-wheel brake control that equivalently reduces vehicle yaw inertia moment and derivative steering characteristics has enabled the development of a steering reaction torque control system, which reduces driver avoidance operation delay, as a new avoidance assistance control method. Vehicle test results have confirmed that yaw rate gain during an avoidance maneuver rises a maximum of 1.5 times, and that steering delay can be reduced by a maximum of 700 ms; these have led to enhanced driver avoidance maneuvers.

1. Introduction

As a countermeasure to traffic accidents, progress in crashworthiness technology that raises safety in accidents has been remarkable, contributing greatly to the reduction in fatal accidents, though no reduction in the actual number of accidents occurring can be seen. It is believed that systems designed to help prevent accidents would be effective, and accordingly, many automobile manufacturers are developing various active safety and obstacle avoidance systems^{(1) - (3)}.

To actualize such systems, it is necessary to have functions that recognize the surrounding circumstances, using outfield sensors such as radar or cameras, and that assist the driver's driving operation in situations where an accident might occur. However, due to current limitations in outfield sensor recognition capabilities,

the systems implemented so far have been passive safety technologies such as warning systems. On the other hand, by limiting the accident scenarios, there are accident damage mitigating technologies known as pre-crash safety technologies, but accident avoidance assistance technologies using outfield sensors⁽⁴⁾ are still in the research stage.

With this background, the authors have developed a system that assists the steering avoidance maneuver, applying this function with the pre-crash safety system known as collision mitigation braking (CMS)⁽⁵⁾. This research project studied assistance methods, based on the principle that avoidance maneuvers are dependent upon the driver actions. Then, in order to reduce any driver avoidance maneuver delays, studies were made of brake force control reducing the equivalent yaw inertia moment and of steering reaction control system that actualizes derivative handle⁽⁶⁾ with the use of electric power steering (EPS).

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For the avoidance scenario, a double lane change on a low friction coefficient road was established, and enhancements of driver's avoidance operation were verified through vehicle testing. The results show that it is possible to reduce the driver's avoidance operation by raising yaw rate gain during avoidance steering, and the control effect which increases steering reaction torque when the steering wheel is returned to the normal position after undergoing avoidance steering.

2. Background to Development

Figure 1 shows avoidance actions taken by the driver when the driver recognizes the danger of an obstacle ahead⁽⁷⁾. It is clear that at least 90% of drivers take avoidance action by braking or steering, or a combination of the two. In addition, in circumstances where the driver does not recognize the dangerous situation and is informed of this with a warning, it has been reported that roughly 70% of drivers will respond after 1 second⁽⁸⁾. That is to say, 70% of drivers, even if they have not recognized a dangerous situation due to inattention, etc., can take avoidance maneuvers after 1 second with a warning.

In contrast, when the road surface friction coefficient is low, such as when there is compacted snow, even if the driver's own avoidance action delay is reduced with a warning as described above, avoidance maneuvers become difficult due to the large delay in the response of the vehicle itself. Moreover, when the avoidance maneuver is carried out through steering, if the maneuvers are inappropriate, the vehicle's path may become unstable (slide or spin, for example).

Vehicle stabilizing systems, such as the Vehicle Stability Assist System (VSA), operate in these kinds of situations, and have been reported as reducing accidents⁽⁹⁾. In addition, a system was made practical, providing assistance before dangerous situations described as above are reached, by proactively stabilizing vehicle motion from the normal use range⁽¹⁰⁾.

However, with this type of system, in an effort to further enhance avoidance performance, there has been no thought given to correcting driver avoidance maneuvers in relation. The reason for this is that in correcting avoidance maneuvers, it would be helpful to have outfield sensor information that fully understands the given avoidance circumstances. Accordingly, to actualize a function that assists driver avoidance operation while maintaining vehicle motion stability, vehicle motion controls that utilize outfield sensor information are necessary.

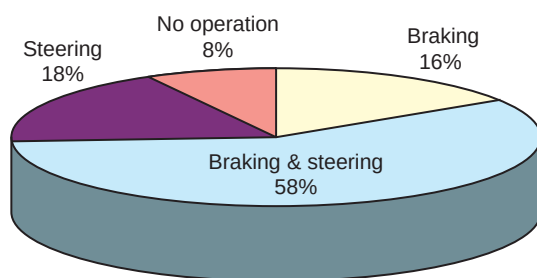


Fig. 1 Driver operation in an emergency avoidance (V=80km/h)

3. Principles of Accident Avoidance Assist Technologies

There are two ways of thinking about how to compensate for the driver avoidance operation delay. The first is a method where, in cases when the avoidance operation delay exceeds the permitted time, the system intervenes and operates before the driver carries out an operation. In this case, the operation delay can be fully compensated for, but the system's environmental recognition capability is required to be as good as that of the driver. In addition, the stronger the system's operation intervention, the greater the operational interference with the driver. This means that there is a limit to the amount of intervention. CMS utilizes automated braking in response to driver avoidance operation delay, but the scope of system operation is limited to scenarios where collisions cannot be avoided. Accordingly, expanding the system's scope of operation to a possible avoidance range will not be possible, as long as the above points cannot be addressed technologically.

Another idea to consider is to detect the driver's avoidance operation, and provide assistance in response to that operation (a driver reactive system). This is an assistance method where the main element of avoidance operation is completely the driver, and there is no means of reliably compensating for driver operational delay. However, required external sensing performance is limited, so compared to the other idea, it is more easily actualized. There are examples of actualization based on this idea⁽¹¹⁾.

In this system, based on the principle that the avoidance maneuvers are dependent upon driver actions, a system was built using the latter idea of a driver-reactive system. The goal is a reduction of collisions with obstacles by taking measures to address operational delays caused by the driver, or by a drop in vehicle dynamic response due to a low friction coefficient road.

4. Outline of the Accident Avoidance Assist System

4.1. System Configuration

Figure 2 shows the system structure diagram. The system configuration was built using existing CMS, EPS and VSA systems. The ECUs of these systems were connected through a Control Area Network (CAN), and data was transmitted.

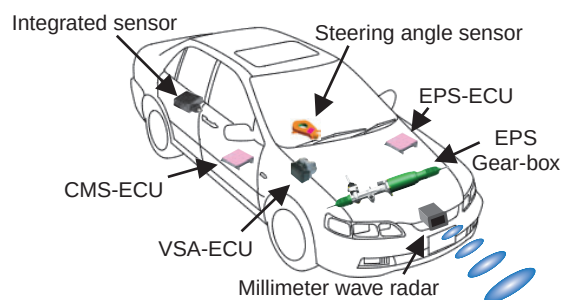


Fig. 2 System structure of obstacle avoidance operation assist system

4.2. System Function

As shown in Fig. 3, the system functions are composed of avoidance target detection, avoidance operation judgment, and avoidance operation assist control blocks.

The avoidance operation judgment block identifies an avoidance operation according to driver steering, in cases where the possibility of collision is high. The criteria value used the linear sum of the squares of steering angle and steering speed, as shown in the equation below.

$$O_t = m \cdot \theta_H^2 + n \cdot \dot{\theta}_H^2 \quad (1)$$

Here, m and n are tuning parameters, and θ_H is the steering angle.

Next, when O_t becomes higher than the specified number, it is judged that an avoidance operation has been carried out, and the avoidance operation assist control block calculates the avoidance yaw rate necessary for avoidance. At this time, if it is judged that

the avoidance operation is complete, the avoidance yaw rate is held at zero, and the avoidance operation assist ends.

4.3. Cooperation with CMS

The scope of this system overlaps with the CMS scope of operation, so it is necessary to ensure the systems do not interfere with each other. Figure 4 shows examples of function cooperation. To give priority to CMS operation, this new system does not operate when the driver takes avoidance action by braking only. However, in situations where the CMS warning braking (pre-braking) operates, when steering avoidance operation is judged, the vehicle shifts from CMS to this system, and accident avoidance operation assist control begins.

On the other hand, in situations where CMS collision mitigation braking operates, because it is not possible to avoid collision even

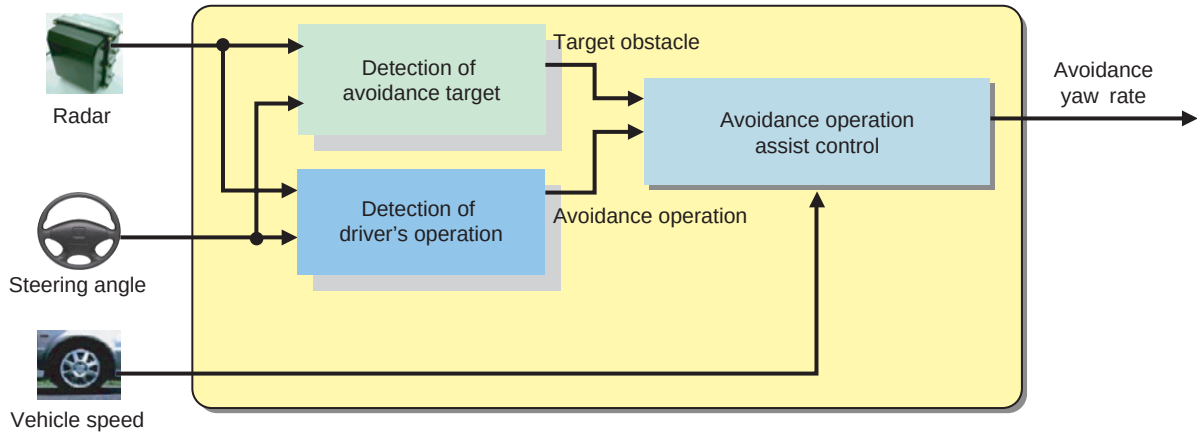


Fig. 3 Structure of obstacle avoidance operation assist function

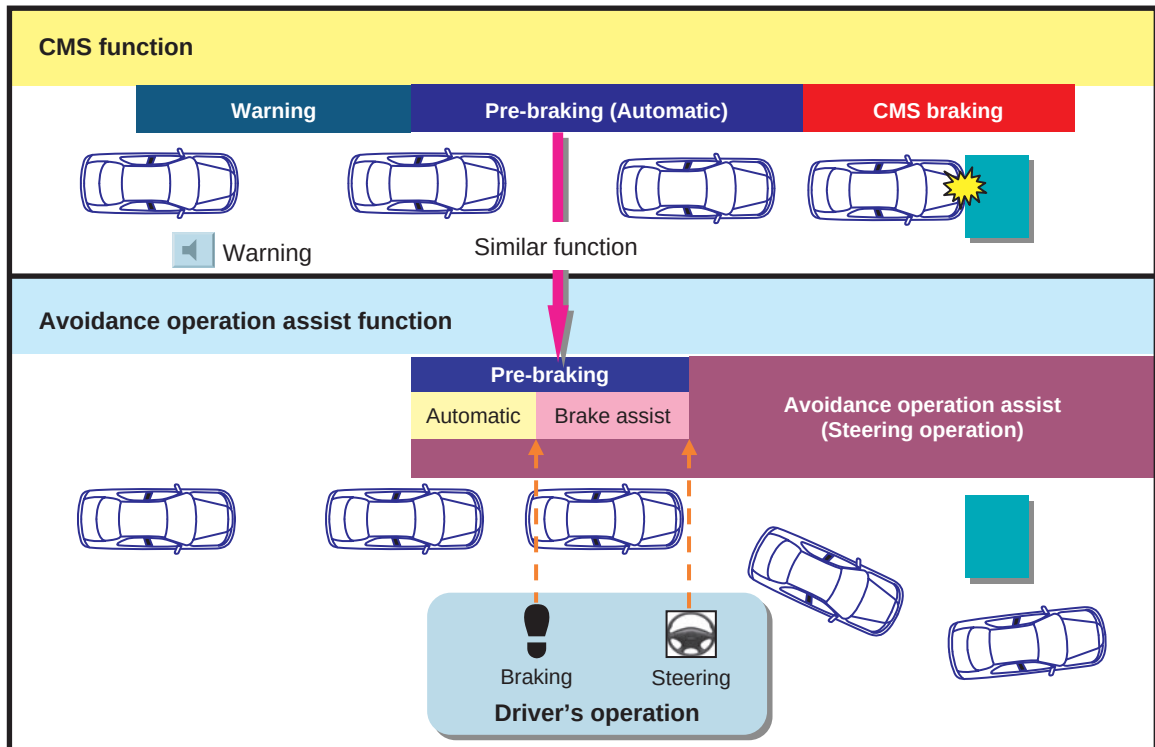


Fig. 4 Assist function comparison with CMS

with steering avoidance, even if avoidance is carried out by the driver under these circumstances, this system does not operate.

5. Design of the Avoidance Operation Assist Control System

A yaw moment control according to steering speed has been reported as a method of reducing avoidance operation delay⁽¹²⁾. In this research, studies were made of yaw moment control through braking force distribution, as a means of actualizing this method.

In contrast, focusing on steering input, steering angle, which is the amount of driver operation, and steering speed, which is the amount of system operation, are combined as inputs to the vehicle. As a result, steering input is phase lead steering (known as derivative handle). Here, studies were carried out concerning steering reaction torque control that actualizes this derivative handle.

5.1. Equivalent Yaw Inertia Moment Reduction Method

This section shows that it is possible to reduce yaw inertia equally, through braking force distribution control.

First, the relationship from yaw moment T_m to yaw angular acceleration $\gamma \cdot s$ is as follows.

$$J_V \gamma \cdot s = T_m \quad (2)$$

Here, J_V is yaw inertia moment, and s is the Laplace operator. Next, control $J_V K \theta_H \cdot s$, which generates supplementary yaw moment from steering speed, is added to Eq. (2). Doing so gives the following.

$$\gamma \cdot s = \frac{1}{J_V} (T_m + J_V K \theta_H \cdot s) = \frac{1}{J_V} T_m + K \theta_H \cdot s \quad (3)$$

In addition, the transfer characteristics from steering angle to yaw rate are defined as follows.

$$\gamma = F(s) \theta_H \quad (4)$$

Here, the following equation is determined from Eq. (3) and Eq. (4).

$$\gamma = \frac{F(s)}{J_V (F(s) - K)s} T_m \quad (5)$$

This means Eq. (5) changes yaw inertia moment as in the following equation.

$$\tilde{J}_V = J_V \frac{F(s) - K}{F(s)} \quad (6)$$

Accordingly, by setting steering speed gain K appropriately, it is possible to make the yaw inertial moment smaller.

5.2. Pseudo Derivative Handle Effect of Steering Reaction Torque

With an ordinary vehicle, it is not possible to change the front tire steering angle, independently of the driver's steering angle. Therefore, it has not been possible to actualize derivative handle unless there is an actuator system such as steer-by-wire.

Therefore, by applying steering reaction torque, derivative handle effect has been actualized by advancing the phase of the driver's steering itself.

First, the following equation is defined from the driver's steering angle.

$$S_{rto} = \alpha \cdot \theta_H + \beta \cdot \dot{\theta}_H \quad (7)$$

This is called the derivative handle ratio, and the range of the provisional value is defined as shown below.

$$-1 < S_{rto} < 1 \quad (8)$$

Here, parameters α and β are setting parameters.

Next, considering the size $d_{\max}$ of the obstacle to be avoided, the amount of lateral movement in the avoidance target is determined as shown below.

$$dist_{ref} = d_{\max} \times S_{rto} \quad (9)$$

Furthermore, if the shortest time to collision TTC that permits the driver's steering avoidance is $TTC_{\min}$, the target amount of lateral movement shown in Eq. (9) can be expressed as the lateral acceleration necessary for avoidance, as below.

$$G_a = \frac{dist_{ref}}{TTC_{\min}^2} \quad (10)$$

In contrast, the yaw rate and lateral acceleration are known to have the following relationship⁽¹³⁾.

$$G_l = V (\gamma + \dot{\beta}) \quad (11)$$

Here, G_l is lateral acceleration, V is vehicle velocity, γ is yaw rate, and β is side slip angle. At this time, assuming that the side slip angle is sufficiently small, the yaw rate required to achieve the lateral acceleration necessary for avoidance is as shown below, according to Eq. (11).

$$\gamma = \frac{1}{V} G_l \quad (12)$$

Accordingly, expressing the lateral acceleration necessary for avoidance G_a as yaw rate becomes the following.

$$\gamma_a = \frac{1}{V} G_a \quad (13)$$

By using the deviation between avoidance yaw rate γ_a obtained in this way, and the actual yaw rate, the system generates steering reaction torque so that the driver's steering amount becomes close to derivative handle characteristics.

5.3. Vehicle Stability Compensator in Avoidance Operation

During avoidance operation on low friction coefficient road surfaces, vehicle side slips tend to occur, and the relationship between the yaw rate and lateral acceleration no longer reaches the approximation of Eq. (12).

According to this formula, when control is carried out that generates a yaw rate, a large side slip angle is generated, and vehicle stability is lost.

Therefore, avoidance assist control is stopped, according to vehicle driving conditions and radar information, and a stability compensator is added to dampen the yaw rate. From the results of preliminary tests using a vehicle, the following conditions are set for the application of the stability compensator.

- When side slip is likely to occur

$$|\gamma| - |G_l| < 0 \quad (14)$$

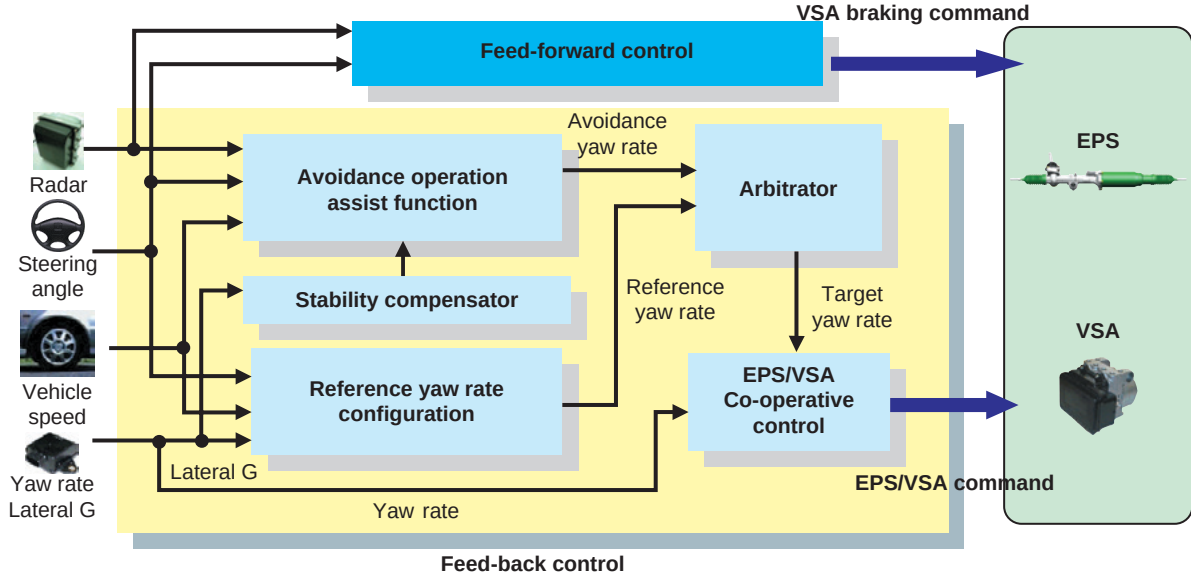


Fig. 5 Control system structure of obstacle avoidance operation assist system

- When obstacle avoidance is certain

$$|d_{ofs}| > d_c \quad (15)$$

- When it is no longer possible to detect an obstacle

$$|d_{ofs}| > L_x \tan^{-1} \theta \quad (16)$$

d_{ofs} is the offset distance detected by radar, d_c is the avoidance possibility judgment value, L_x is the distance to the obstacle, and θ is the radar detection angle. Figure 5 shows the avoidance assist control system structure, based on the principles above.

6. Actual Unit Test Results and Considerations

6.1. Test Conditions

In order to confirm system function and performance, a double lane change course was set up, as shown in Fig. 6. In addition, an avoidance scenario was established wherein the vehicle drives straight forward towards an obstacle in front, recognizes danger due to a collision warning, takes an avoidance action by braking, and then attempts avoidance by steering.

The vehicle used for testing was a 3.0L class passenger vehicle, fitted with studless tires.

Vehicle driving conditions were a road surface friction coefficient of 0.3, straight forward driving at a fixed speed of 60 km/h, and a warning occurring when the time to collision reached 3 seconds, to warn drivers of the pending collision danger. As an avoidance action, the driver took avoidance action with the brakes after the warning, and in this state, avoidance through steering was carried out 22 m (TTC=1.32 sec.) before the obstacle. The test drivers were five men between the ages of 20s and 40s, having ordinary driving skills.

6.2. Test Results

In Fig. 7, the upper row shows yaw rate response and the lower row shows steering angle during avoidance operation. These graphs

are data from the five test drivers, with and without the avoidance operation assist control. Yaw rate responses with the control were enhanced compared to without it, for all test subjects. As the arrow shows in the upper row of Fig. 7, the yaw rate gain peak value was, on average, 1.5 times larger. In addition, concerning steering delay, as the lower row of the figure shows, there were better results for all test subjects, and the time from avoidance action until the steering wheel crossed back to the neutral position was, on average, 700 ms shorter.

In Fig. 8, the upper row shows lateral acceleration response and the lower row shows steering reaction torque. As the arrow shows, there is an average increase of 0.5 m/s² at the lateral acceleration peak. In addition, a characteristic point in steering reaction torque is that, as the part surrounded by the dotted line shows, reaction is large where the steering returns from avoidance steering. This is expressed as an effect suppressing excessive steering by the driver, and an effect assisting return steering. When there is no control,

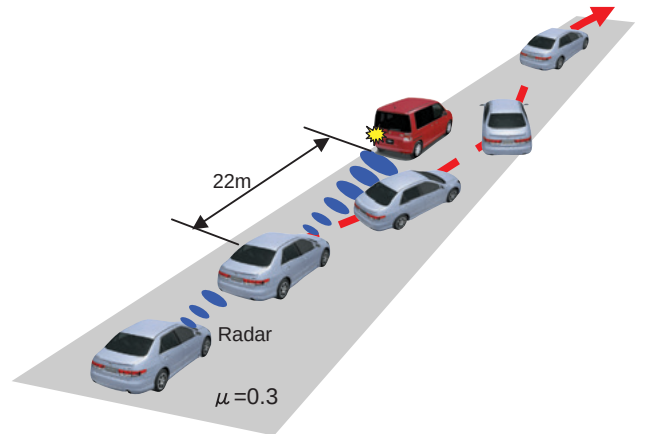


Fig. 6 Control system structure of obstacle avoidance assist system

the steering reaction torque information is not provided, and there is no change to the steering force.

Figures 9 and 10 show the test results for test driver A. These graphs plot avoidance data with and without control, five times each. Test subject A has the largest avoidance operation delay without the control, and almost all attempts at avoidance failed. However, with the control, there were great enhancements in yaw rate responsiveness and in steering delay, and moreover, as shown

by the intersection in Fig. 10, steering reaction torque was large when the steering wheel was turned back, and the reaction information effect was confirmed.

Thus, subject A was able to carry out avoidance operations equivalent to that of other subject drivers. There were no failures in avoidance in these tests.

Figures 11 and 12 show the test results for test driver B. In this case, the driver skillfully avoided the obstacles even without the

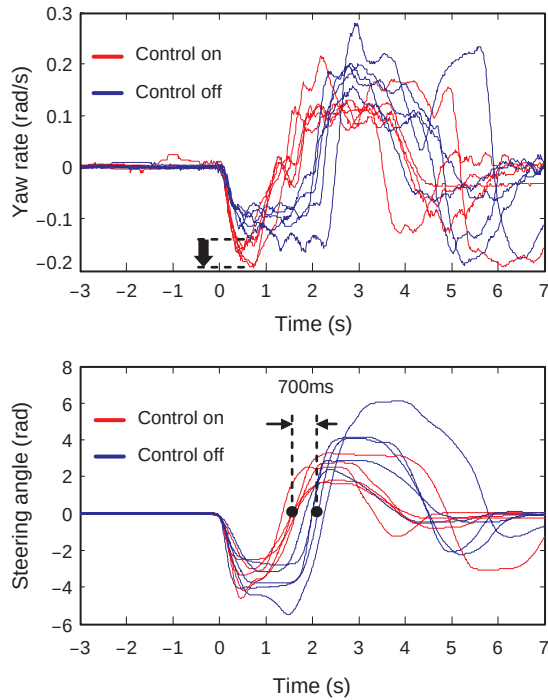


Fig. 7 Test results of total subjects

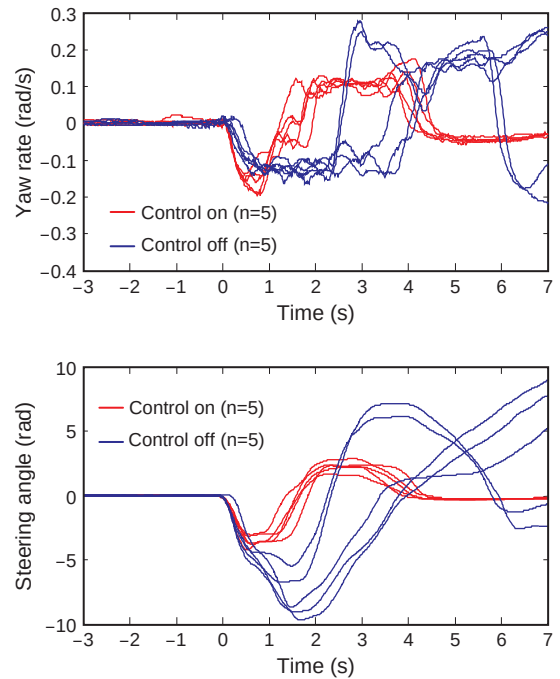


Fig. 9 Test results of subject A

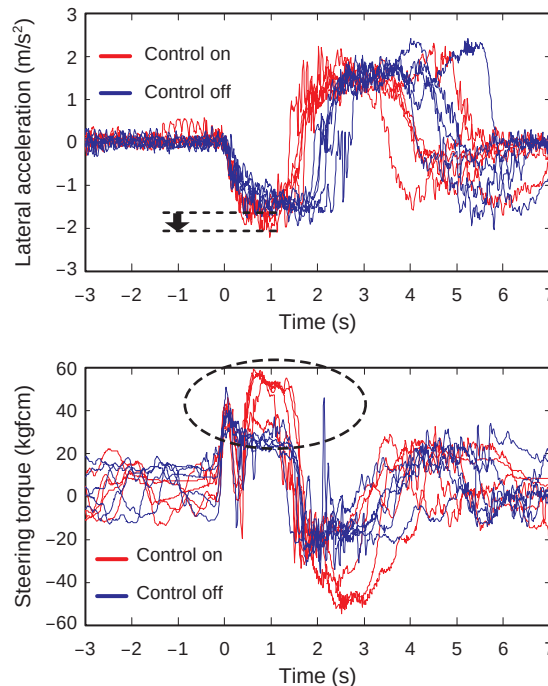


Fig. 8 Test results of total subjects

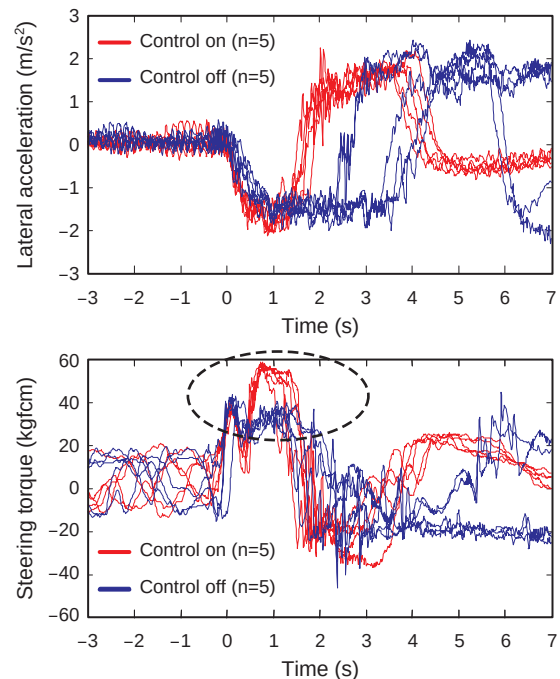


Fig. 10 Test results of subject A

control, and as a result, there was little difference in avoidance when compared to having the control. However, when questioned after the tests, the driver made the evaluation that having the control gave a clear benefit in terms of the ease of operation during avoidance.

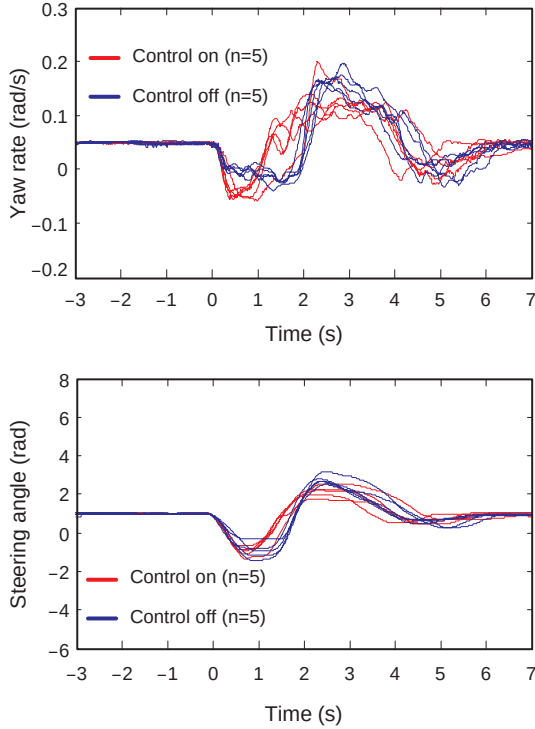


Fig. 11 Test results of subject B

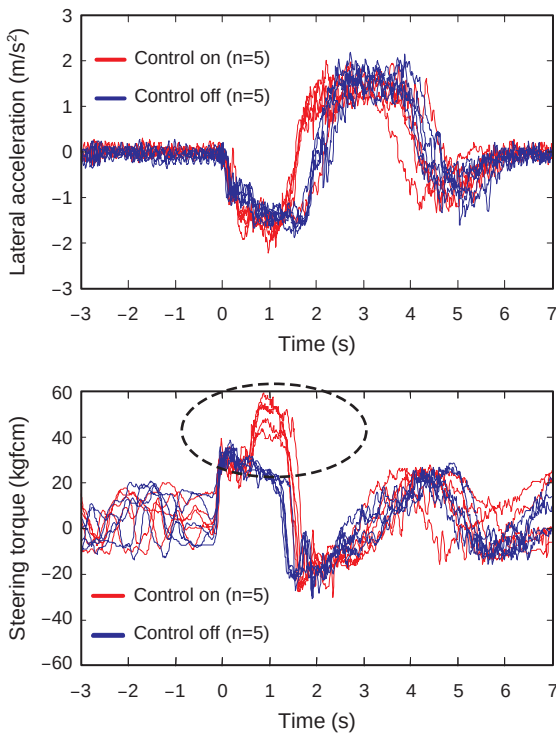


Fig. 12 Test results of subject B

6.3. Considerations

The test results confirmed the following effects.

- (1) Higher yaw rate gain
- (2) A reduction in avoidance steering delay
- (3) Steering reaction torque information against driver operation

Concerning effect (1), in comparison to when there is no control, the yaw rate gain becomes large, even if the steering angle becomes small. Furthermore, during questioning after the tests, all test drivers gave the evaluation that it was easier to steer because the feeling of under-steering was eliminated. Because there is the same effect when a vehicle's yaw inertia moment becomes small, it was verified that the equivalent yaw inertia moment reduction method was appropriate.

In contrast, concerning items (2) and (3), there were beneficial effects on driver steering delay. Here, even test drivers who failed to avoid obstacles without the control, succeeded in avoiding them with the control. This system gave more benefit to those drivers with long steering delays (those with only ordinary skills), than to those with higher skills.

From the above results, it is clear that this system has sufficient benefits as assistance in avoidance operations in the driver-vehicle system. However, there is little difference in the collision avoidance success rate, with or without the control. In order to enhance collision avoidance performance, even more progress is needed in external sensing and in vehicle control devices (for example, steer-by-wire, etc.).

7. Conclusion

This paper has studied systems assisting operations avoiding forward obstacles, and confirms the benefits of the avoidance operation assist function, through vehicle tests. The results of this project are summarized below.

- (1) Generating yaw moment through brake force lateral distribution control, in response to steering speed, enables vehicle yaw inertia moment to be reduced equally, thereby increasing vehicle yaw rate responsiveness during obstacle avoidance.
- (2) An assist benefit was confirmed in reducing driver operation delay by setting the avoidance target yaw rate on the assumption of derivative handle input and using EPS reaction torque control.
- (3) The results of an avoidance test using a double lane change task confirmed that the greater the driver's steering delay (of drivers with only ordinary skills), the greater the system benefit.

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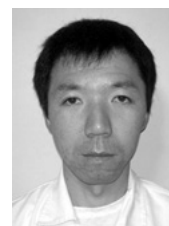
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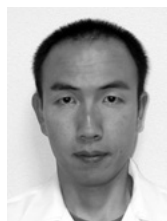
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Development of Light Weight Connecting Rod

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ABSTRACT

The conventional middle carbon non-heat-treated steel used in connecting rods has been used as a base in the development of a new material. High fatigue strength has been achieved by increasing the toughness of the ferrite, which tends to act as a crack origin, while optimal control of the reduction of toughness achieved through addition of Ti has been effective in enabling fracture splitting of the material.

Connecting rods produced using the new steel demonstrate 30% higher fatigue strength than connecting rods produced from conventional non-heat treated steel, and possess stable fracture splitting characteristics. In addition, the new connecting rods show identical strength to connecting rods made of the conventional material, but are 13% lighter.

1. Introduction

It is becoming increasingly necessary to reduce the weight of automotive parts to achieve both increased power and fuel economy. Because of their rotary motion, the mass of connecting rods (hereinafter referred to as connrods) affects engine layout and performance, and the achievement of size and weight reductions in these parts is therefore a particular requirement. In addition, reducing the weight of connrods can be expected to lead to reductions in the weight of other parts.

Titanium and aluminum alloys have been employed to reduce the weight of the connrods in certain sports cars. However, the cost of these lightweight materials is prohibitive⁽¹⁾, and non-heat-treated steel is generally used in the production of connrods for mass-market vehicles; this steel is strengthened to enable weight savings to be achieved⁽²⁾.

Fracture splitting of the connrod big end has recently been increasingly used as a manufacturing method in the United States and Europe. Figure 1 shows a conventional connrod and a fracture splitting connrod, and Fig. 2 shows the respective production processes. The use of fracture splitting enables the number of steps in the production process to be reduced, and does away with the need for pins to align the rod and cap. It also enables the bolt tightening pitch to be reduced, thus reducing the size and weight

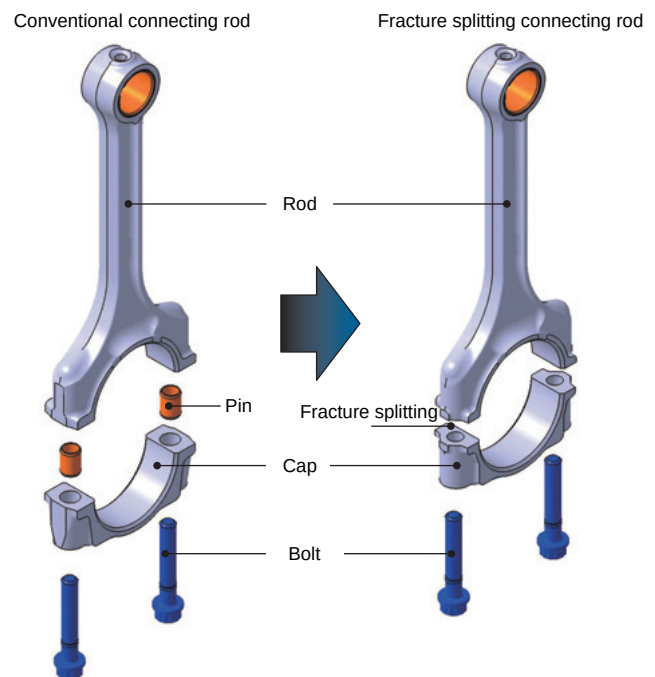


Fig. 1 Comparison of conventional connecting rods and fracture splitting connecting rods

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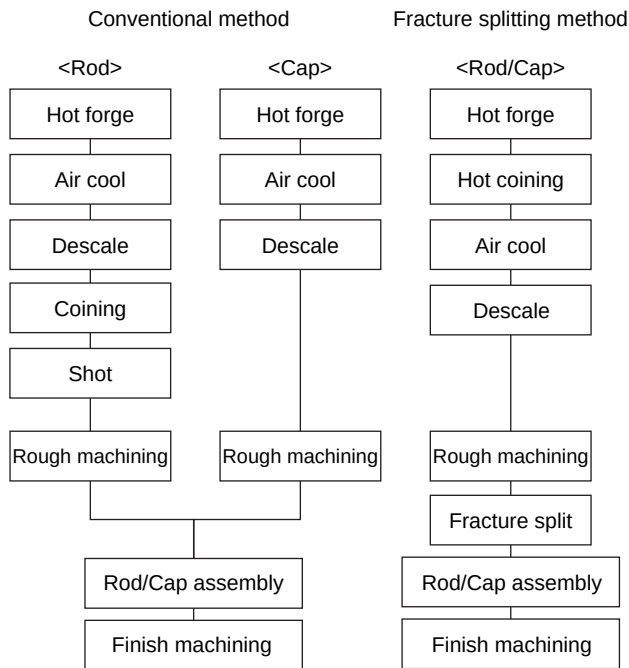


Fig. 2 Production process of connecting rod

of the connrod. However, the fatigue strength of the C70S6 steel (DIN standard) used in the fracture splitting process is identical to that of conventional middle carbon non-heat-treated steel, making it difficult to achieve further weight savings. The project discussed in this paper therefore developed a stronger steel for use in the fracture splitting process, to enable the production of a smaller, lighter fracture splitting connrod.

2. Development Goals

2.1. Fracture Splitting

The following are essential factors in the achievement of stable quality in the fracture splitting process:

- (1) Stable fracture split surface characteristics
- (2) Fracture splitting with a stable cracking force

One of the goals of the development project was to enhance these parameters against those of the C70S6 steel conventionally used in fracture splitting.

2.2. High Strength

Reducing the weight of the connrods by increasing material strength could be expected to contribute to increasing the seizure limit of the crank bearings, increasing the strength margin by reducing the load on the crank, and reducing engine vibration and friction. Past results suggested that a weight reduction of 10% or more would be necessary to ensure the achievement of these benefits⁽²⁾.

A weight reduction of 10% against the current non-heat-treated steel was therefore established as a development target.

Strength can be increased both by increasing fatigue strength and by increasing yield strength. Given the type of engines in which the connrods would be employed, increasing fatigue strength was judged to be most effective in achieving weight savings, and this was therefore the focus of the project.

Table 1 Chemical composition for impact test

(mass %)							
C	Si	Mn	P	S	Cr	Ti	V
0.30	0.70	0.80	0.004 ~ 0.072	0.055	0.20	—	0.200
0.30	0.70	0.80	0.060	0.060	0.20	0 0.190	—

3. Composition of Material

3.1. Concept of Steel under Development

Using conventional middle carbon non-heat-treated steel as a base, the composition of the new steel was designed on the basis of the following concepts, with consideration of increased hardness from the perspective of machinability:

- (1) Reduced toughness
 - Addition of elements to reduce toughness of ferrite, because ferrite has high toughness in ferrite-pearlite structures
- (2) Increased fatigue strength
 - Solid solution strengthening and precipitation strengthening of ferrite, which tends to act as a point of origin for cracks

3.2. Composition Design for Reduced Toughness

Increasing the strength and hardness of the ferrite in ferrite-pearlite non-heat-treated steel is an effective means of reducing the toughness of the ferrite. V is already widely used as an additive to increase the strength of the ferrite⁽³⁾, and the addition of Ti has been reported as a means of strengthening ferrite and achieving optimum toughness⁽⁴⁾. P is also known to act as an impurity and make the grain boundary brittle, reducing the toughness of the ferrite. Tests were therefore conducted using steel with added V, to which P was added in quantities of between 0.004 and 0.072%. Ti was also added in quantities of between 0.10 and 0.19%. The changes in toughness with different quantities of P and Ti were then measured. Table 1 shows the compositions of the steels used in tests. After being heated to 1 250 °C and left to cool to reproduce the conditions employed in hot forging, JIS Z 2202 V notch impact test pieces were formed from the materials. These impact tests were conducted at room temperature. Figure 3 shows the effect of the addition of P, and Fig. 4 shows the effect of the addition of Ti.

Impact value declines as the amount of P and Ti is increased, and Ti is particularly effective in reducing toughness. This is believed to be because of the different precipitation morphology of the V carbonitride and Ti carbonitride precipitating in the ferrite. Figure 5 shows transmission electron microscope (TEM) images of the ferrite thin film in V-added steel and Ti-added steel⁽⁴⁾. In the V-added steel, tiny V carbonitrides can be observed precipitating from the ferrite. By contrast, in the Ti-added steel, there are fewer carbonitrides and they are larger. This causes dislocation loops to form around the Ti carbonitrides, which results in dislocations tending to build up when the material is deformed. Studies by K. Watari have indicated that this results in reduced toughness^{(4), (5)}.

In terms of fracture morphology, fractures occur in both steels within the grain; fractures at the grain boundary were not observed in the P-added steel. Therefore, the reduction in toughness occurring with the addition of P is believed to be the result of solid solution strengthening.

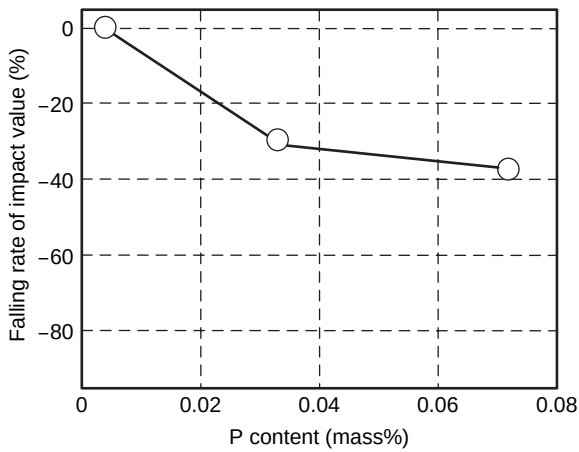


Fig. 3 Relationship between P content and impact value

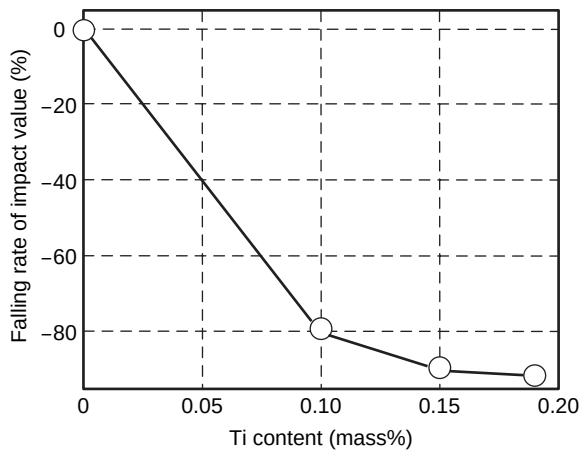


Fig. 4 Relationship between Ti content and impact value

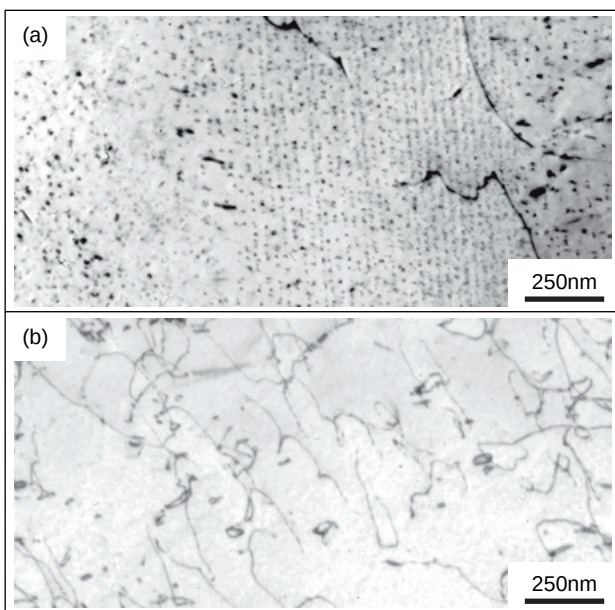


Fig. 5 TEM micrographs of (a) V added steel and (b) Ti added steel

To ensure that the toughness of the new steel was sufficiently low for stable fracture splitting, a design involving addition of both P and Ti in quantities of more than 0.04% and more than 0.15% respectively was decided upon.

3.3. Composition Design for High Fatigue Strength

To ensure adequate machinability, it is desirable that hardness does not increase despite an increase in fatigue strength. It was therefore essential to design the material to increase the endurance ratio, which expresses the ratio between tensile strength and fatigue strength.

Figure 6 shows the effect on the endurance ratio of varying the C content of a carbon structural steel from 0.1-0.6%⁽⁶⁾. This steel has a ferrite-pearlite structure; tensile strength is determined by the ratio between the volume of ferrite and the volume of pearlite, and fatigue strength is determined by the strength of the ferrite, the point of origin of cracks. Therefore, reducing the C content and increasing the strength of the ferrite can be considered to be an effective means of increasing the endurance ratio. In Fig. 6, good correlation was observed with plots of the endurance ratio of the current non-heat-treated steel and C70S6 steel, and therefore a C content lower than that of the currently used non-heat-treated steel was decided upon.

Ferrite can be strengthened by both precipitation strengthening and solid solution strengthening. Figure 7 shows the correlation between increases in hardness and increases in fatigue strength when the composition of different alloys and elements is varied in

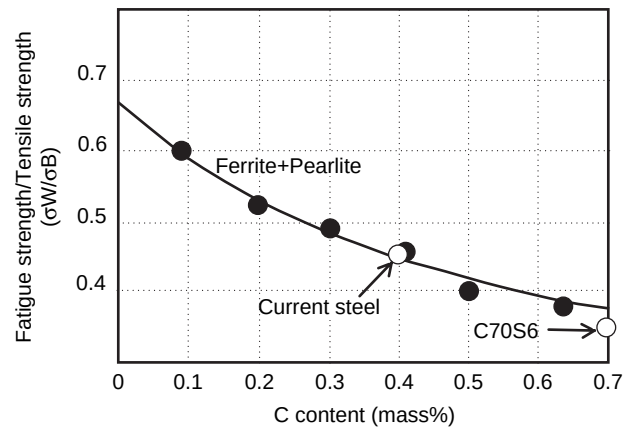


Fig. 6 Relationship between C content and $\bar{E}-W/\bar{E}-B$

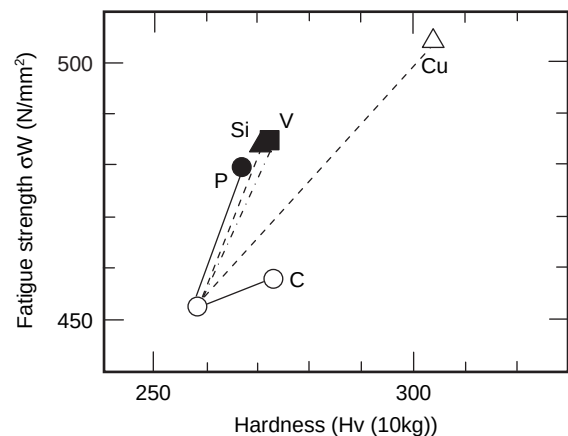


Fig. 7 Relationship between hardness and fatigue strength

middle carbon non-heat-treated steel⁽⁷⁾. These results indicate that solid solution strengthening by the addition of P and Si and precipitation strengthening by the addition of V would be effective in increasing fatigue strength, without increasing hardness. Ti is also known to be a precipitation-strengthening element⁽⁸⁾.

The amounts of P and Ti to be added to the material under development had already been established, and therefore the effect of the addition of Si and V in increasing fatigue strength was studied. The steels shown in Table 2 were subjected to an Ono-style rotating bending fatigue test. JIS Z 2274 1 test pieces were manufactured after the steel had been heated to 1 250 °C and cooled to simulate hot forging conditions.

Figure 8 shows the relationship between Si and V content and fatigue strength. Increasing the V content is highly effective in increasing fatigue strength, while increasing the Si content has little effect on this parameter. Ferrite hardness was studied to determine the reason for this. Figure 9 shows the relationship between Si and V content and ferrite hardness. Increased V content has a noticeable effect in increasing ferrite strength, while increasing Si content does not affect ferrite strength. This is believed to be because it is difficult for the Si to exert a solid solution strengthening effect on the ferrite, because it has already undergone precipitation strengthening by Ti and V carbonitrides. The precipitation strengthening effect of adding V is believed to depend on the respective precipitation strengthening effects of the different

Table 2 Chemical composition for fatigue strength

(mass %)								
No.	C	Si	Mn	P	S	Cr	Ti	V
1	0.28	0.53	0.72	0.045	0.100	0.14	0.154	0.159
2	0.28	0.56	0.73	0.054	0.090	0.17	0.173	0.168
3	0.31	0.68	0.74	0.058	0.066	0.15	0.184	0.199
4	0.32	0.19	0.60	0.035	0.063	0.15	0.179	0.199
Current steel	0.39	0.45	1.05	0.010	0.110	0.09	—	0.110
C70S6	0.70	0.20	0.53	0.008	0.060	0.12	—	0.029

precipitation morphologies of Ti carbonitrides and V carbonitrides, as discussed above. The above results indicated that achieving precipitation strengthening by adding V is effective in increasing ferrite strength in Ti-added steel.

Figure 10 shows the relationship between hardness and fatigue strength in the test steel, the current non-heat-treated steel and C70S6 steel. In the test steel, with its ferrite strengthened, fatigue strength is higher than hardness.

In terms of composition design, a Si content equivalent to that of the former non-heat-treated steel was decided on, and V content was established to achieve a balance between hardness and strength.

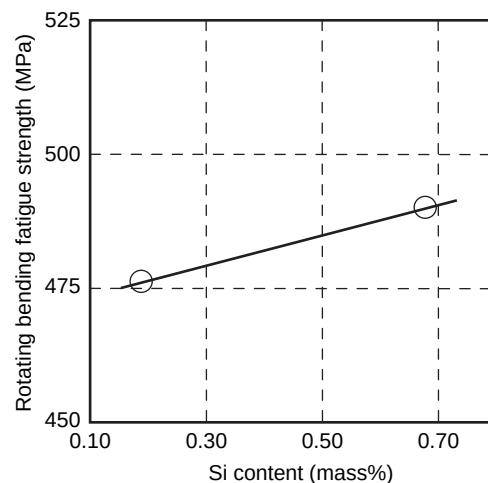
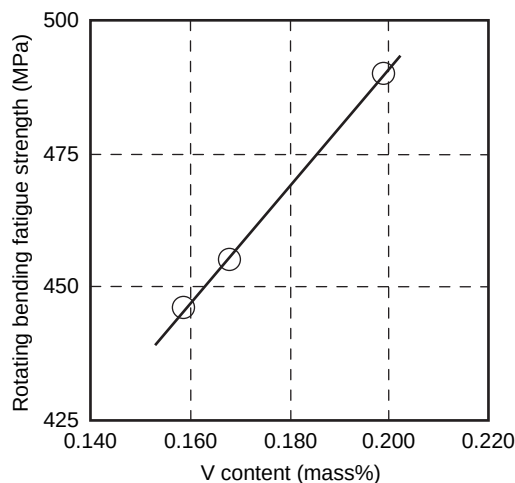


Fig. 8 Relationship between V/Si content and fatigue strength

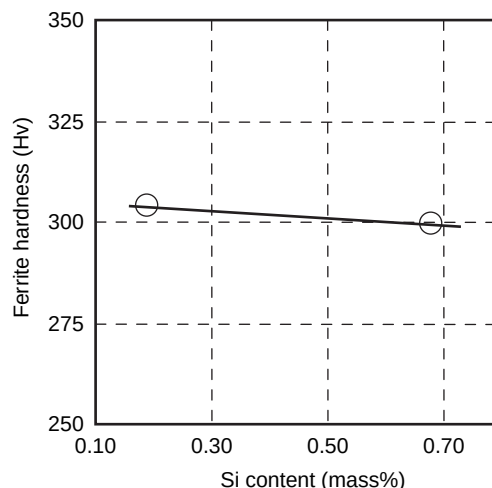
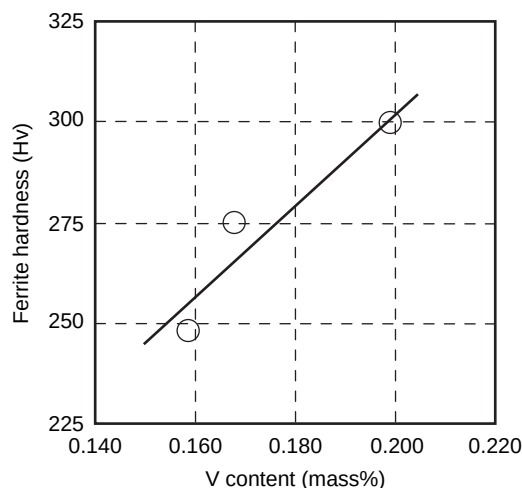


Fig. 9 Relationship between V/Si content and ferrite hardness

3.4. Composition Design

The studies discussed above led to the establishment of a composition design that would ensure ease of fracture splitting while achieving increased fatigue strength by strengthening the ferrite, and with a level of hardness ensuring machinability. Table 3 shows the composition. Figure 11 shows a comparison of the structures of the newly developed steel and C70S6 steel. The new steel has a ferrite-pearlite structure.

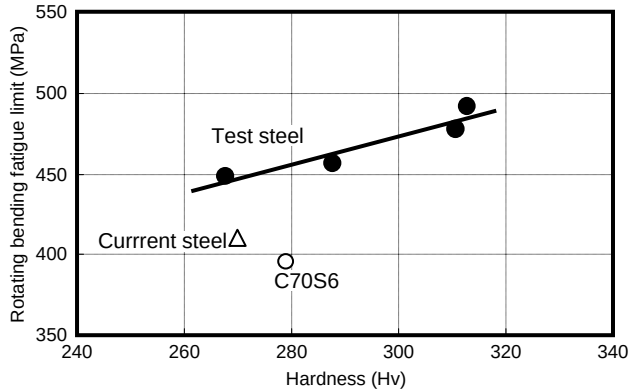


Fig. 10 Relationship between hardness and fatigue strength

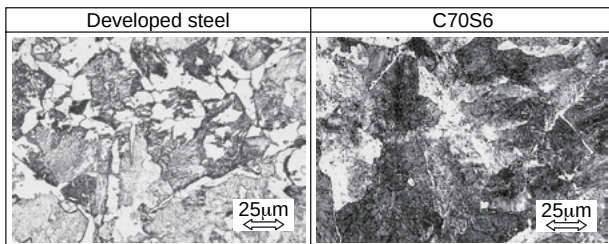


Fig. 11 Micro structure of actual connecting rod

4. Fracture Splitting Characteristics of Newly Developed Connrod

The features and characteristics required when fracture splitting a connrod big end are as follows:

- (1) Required features
 - a: Matching fracture surfaces
 - b: Stable bolt tension
 - c: Close fit between cap and rod when assembled
- (2) Required characteristics
 - a: Brittle fracture of the fracture surface
 - b: Stable level of roughness of the fracture surface
 - c: Alignment of the fracture surfaces within tolerances between parts

With regard to a, when ductile fractures occur on the fracture surface, the sections undergoing plastic deformation prevent the fracture surfaces from fitting together.

With regard to b, when the level of roughness of the fracture surface is high, the surface area increases and the bolt tension is absorbed in achieving a fit between the fracture surfaces; when the level of roughness is low, the bolt tension is low. There is therefore concern that the stability of the bolt tension will decline with

Table 3 Chemical composition of developed steel

(mass %)							
C	Si	Mn	P	S	Cr	Ti	V
0.30	0.50	0.70	0.050	0.100	0.10	0.180	0.180

variations in the level of roughness.

With regard to c, if the rod and cap are not aligned, it will not be possible to correctly fit the connrod to the crankshaft, and its function will be impaired.

4.1. Achieving Stable Roughness of the Fracture Surface

The newly developed steel was used to hot forge a connrod (rod and cap), and it was machined to the desired shape and dimensions. Following this, a notch was machined around the inner circumference of the big end for fracture splitting, and the connrod was placed in the cracking device shown in Fig. 12 for testing.

In this device, after the connrod is fixed in place, a wedge is drawn out, and a mandrel inserted into the big end is extended to produce a fracture split.

To enable quantification of the roughness of the fracture surfaces, a measurement device was used to take 3D measurements of roughness; the measured roughness was defined as surface area ratio (AR) in the following equation. Figure 13 shows the measuring area.

$$AR = \frac{SA}{PA}$$

AR : Surface area ratio

SA : Measurement area

PA : Projection area

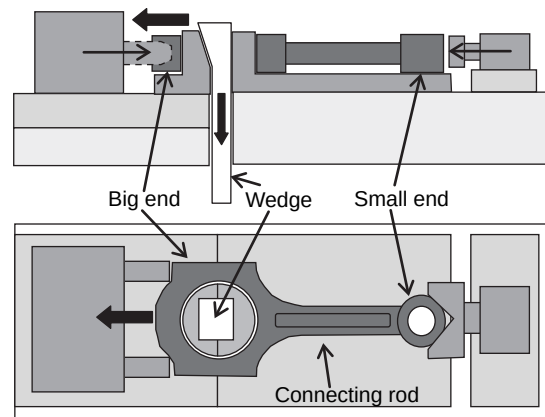


Fig. 12 Fracture splitting device

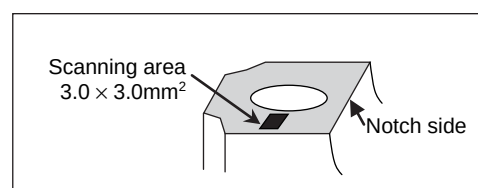


Fig. 13 Measuring area

The effect of the depth of the notch cut around the inner circumference of the big end was studied to enable the roughness of the fracture surface to be controlled. Tests were conducted using conrods with machine-cut and laser-cut notches. A conrod produced using C70S6 steel with a machine-cut notch was also tested for comparison.

As Fig. 14 shows, surface area ratio declines as the depth of the notch increases; a stable surface area ratio is achieved with a notch depth of 1 mm and higher. Results are identical for both machine-cut and laser-cut notches.

The reason for this appears to be that when the notch is shallow, the level of concentration of stress at the edge of the notch is low, and there is therefore a high level of variation in the positions at which cracks occur and in the range of direction of cracks. Because of this, the relative size of the surface area ratio increases, and the level of variation also increases. By contrast, when the notch is deep, there is a high level of concentration of stress, and the position

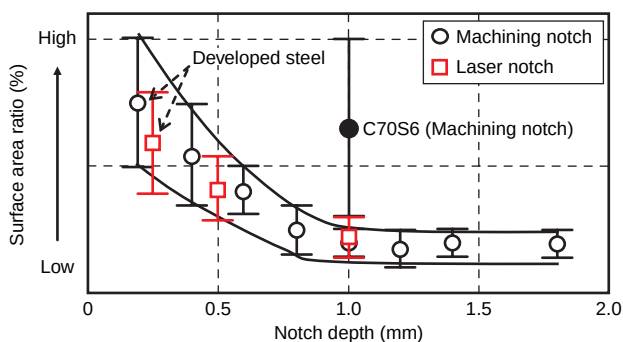


Fig. 14 Relationship between notch depth and surface area ratio

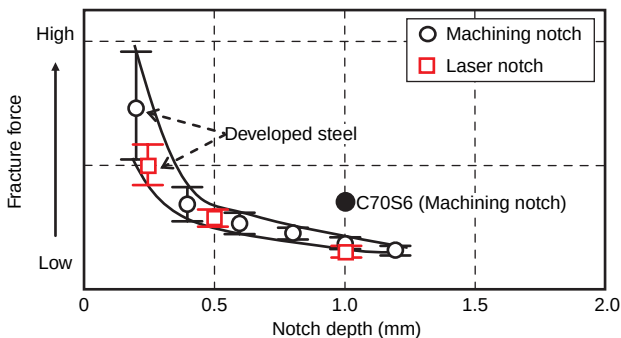


Fig. 15 Relationship between notch depth and fracture force

of occurrence of cracks and the range of directions of cracks are confined within a narrow area; the surface area ratio is relatively small, and therefore the range of variation is lower. In the newly developed steel, a stable fracture surface is obtained with a notch depth of 1 mm and higher.

Figure 15 shows the relationship between notch depth and cracking force. Cracking force is determined by monitoring the change in hydraulic pressure in the hydraulic cylinder used to extract the wedge in the cracking machine.

As was the case in the relationship between notch depth and surface area ratio, as notch depth increases, the cracking force is reduced and the level of variation becomes lower. The cracking force becomes stable with a notch depth of 1 mm or higher. The cracking force is the same whether the notches are machine-cut or laser-cut.

The newly developed steel displays lower variation in fracture surface roughness than C70S6 steel, and can be fracture split with a lower cracking force. These results indicate that the newly developed steel has superior fracture splitting characteristics to those of C70S6 steel. In addition, both machine-cut and laser-cut notches can be employed when fracture splitting the newly developed steel.

Figure 16 shows Scanning Electron Microscope (SEM) images of a fracture split surface. The surface shows a cleavage fracture from the bottom of the notch where the fracture originates to the end of the fracture.

4.2. Self-fitting of Fracture Surface

When a fracture split conrod is assembled, the cap and the rod are required to fit together tightly in alignment.

The concavities and convexities of the fracture surfaces act as a guide in self-fitting, and the control of surface roughness is therefore also important from this perspective.

5. Fatigue Strength of Developed Conrod

Because conrods are used as forged, the tension-compression fatigue strength of the I-beam cross-section of actual conrods was tested. Figure 17 shows the results of these tests. Results for a conrod manufactured from C70S6 steel using the same production process are also shown for comparison. While displaying the same level of hardness, the newly developed steel has a 30% higher fatigue strength than that of the current non-heat-treated steel and C70S6 steel.

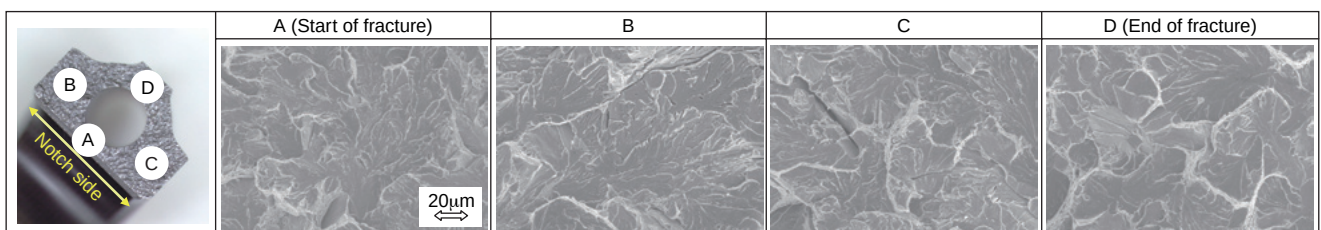


Fig. 16 SEM micrographs of fracture surface

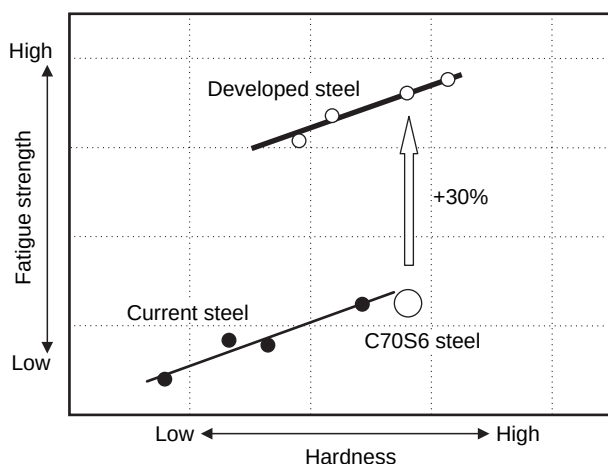


Fig. 17 Comparison of tension and compression fatigue strength between developed connecting rods and C70S6 connecting rods

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- (3) Sato, T., Kurita, M.: Saikin no Hichousitsukou ni tuite, Tanzou Gihou, Vol. 83, p. 16-25 (2000) (in Japanese)
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6. Weight Savings

The use of the newly developed steel has enabled the area of the I-beam cross-section to be reduced, while the employment of fracture splitting has made it possible to eliminate the pins for alignment of the rod and cap and reduce size, resulting in a 13% reduction in weight. This weight saving, together with a consequent reduction in the weight of the crankshaft, has reduced the weight of the engine used in the 2005 full model change CIVIC by approximately 450 g.

7. Conclusion

The project discussed in this paper succeeded in reducing the weight of an automotive connrod through the development of a new steel combining increased fatigue strength with fracture splittability of the big end. The project can be summarized as follows:

- (1) Strengthening the ferrite in the material was effective in enhancing fracture splitting performance. Carbonitride precipitation strengthening of the ferrite by the addition of Ti in particular enabled the achievement of superior fracture splitting characteristics to those displayed by conventional steels.
- (2) Reduction of the C content of the newly developed steel to less than that of the current non-heat-treated steel and the employment of carbonitride precipitation strengthening by the addition of V and Ti were effective in increasing the endurance ratio.
- (3) Use of the newly developed steel has increased connrod fatigue strength by 30% while reducing weight by 13%.

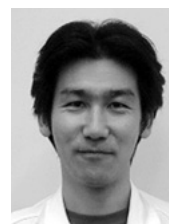
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Chikashi SATO

Development of High-strength Piston Material with Die Casting

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Kenji SATO*

Kazunari FURUKAWA*

ABSTRACT

The pistons are an essential element in the achievement of greater engine efficiency. A method of increasing the strength of the material employed in pistons to respond to increased thermal and mechanical loads and reducing weight has been developed to contribute to the quest for enhancements in power and fuel economy.

This method is used in the die cast production process, which has a rapid rate of cooling. The limit for addition of Ni to the aluminum alloy employed has been increased to 5.5 mass%. At the same time, less than 0.4 mass% of Fe is added and the Mn/Fe ratio is controlled to less than 0.6 in order to limit the coarsening of the crystals. This technique has increased the high-temperature fatigue strength of the material by more than 1.5 times against the material conventionally employed. In addition, the uses of die vacuum and additional pressure have enabled quality exceeding that of conventional GDC pistons.

Optimization of the piston shape to match the characteristics of the new material has resulted in decreasing engine friction by 4.4% and increasing fuel efficiency by 2.2% in engine bench testing.

1. Introduction

In response to increasing demand for more environmentally responsible automobiles, Honda has worked to develop engines with greater fuel efficiency and lower emissions, such as the i-Series of next-generation power plants⁽¹⁾⁻⁽⁴⁾. The demand is growing for pistons that are lighter and capable of withstanding greater thermal and mechanical load, which in turn requires piston materials with better high-temperature strength.

A method of adding Cu, Ni, Fe, Mn, or some other elements that enhance heat resistance, to the AC8A aluminum alloy base has been used to increase the high-temperature strength of piston material. However, in the case of Gravity Die Casting (GDC), the

method used for the majority of pistons, there are limits to the amount of such elements that can be added to the materials due to the low cooling rate of casting, which causes intermetallic phases to form large crystals. To speed cooling, some manufacturers use a powder forging process for two-cycle engine pistons, which are examples of increasing strength by greater quantities of additive elements⁽⁵⁾. However, this method has yet to be adopted for automobile pistons because of several issues related to reliability and cost, such as oxide films formed by the powders, impurities, strength anisotropy (which results from metal flow during forging), and the complexity of fabrication procedures such as quality control.

On the other hand, automobile manufacturers in Europe and Japan^{(6),(7)} are increasingly using die casting for chassis components

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and other aluminum parts; some are even using vacuum die casting for subframes to enable welding. Consequently, die casting technology and the quality of die cast components have advanced considerably in recent years. Since die casting's cooling rate is higher than that of GDC, greater amounts of additive elements can be added due to the rapid cooling effect. Nevertheless, the application of die casting to pistons is limited to general-purpose engines, in which case the technique is used to increase productivity⁽⁹⁾; there are no reports of applications in which the goal was greater piston strength.

Recently, a high-strength piston material with die casting that offers enhanced high-temperature strength without significant cost increases has been developed and, moreover, is now used in the mass-production of automotive pistons. This paper discusses microstructure control used for the new material, its enhanced strength, the high-quality pistons made with the material, and the benefits of using the pistons in engines.

2. Development Goals and Strategy for Achieving Goals

2.1. Development Goals

The primary functions of a piston are to form part of the combustion chamber and to transmit the pressure and heat of combustion. The important material properties are therefore high-temperature strength, abrasion resistance, fracture toughness, Young's modulus, thermal conductivity, and the coefficient of thermal expansion. The goal in developing the new material was to achieve a fatigue strength of at least 1.5 times greater than that of Honda's high-strength material made by GDC in the high-temperature range of 523 K and above (which is the operating environment for the piston materials). Another goal was to achieve casting quality comparable to or better than that of GDC pistons in order to assure the durability that is required of pistons.

2.2. Strategy for Achieving Goals

The piston material exhibits a complex microstructure comprising an aluminum matrix phase with elements such as Cu or Mg in solid solution (the α phase), Si particles and intermetallic phases consisting of crystallized intermetallic products of Ni, Fe, Mn, among others. High-temperature environments of 523 K or greater cause considerable softening of the α phase, from which cracking tends to originate due to stress loading. Increasing high-temperature strength therefore necessitates protecting the α phase by increasing the quantity and micro-dispersion of the crystallized material. Figure 1 shows a schematic of the target microstructure sought for this material.

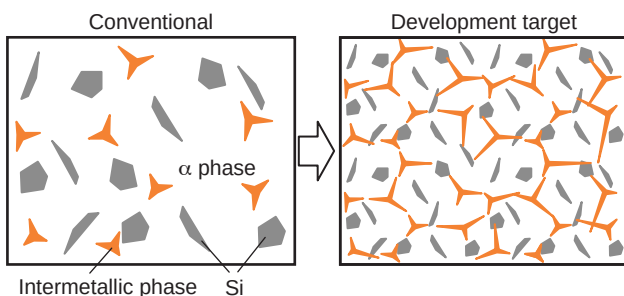


Fig. 1 Schematic microstructures of conventional material and developmental target

Nickel was thought to be a candidate additive material (i.e., to add in greater quantities) because of its large solubility⁽⁹⁾ during quench, low diffusion coefficient in aluminum⁽¹⁰⁾ (an indicator of heat resistance), and low electrical resistivity (an indicator of thermal conductivity)⁽¹¹⁾. The increase in strength and microstructure resulting from greater amounts of Ni for GDC and other techniques are shown in Figs. 2 and 3, respectively. As the figures show, greater amounts of Ni in the case of GDC result in larger crystals and only a slight increase in strength, whereas with die casting, the greater micro-dispersion of the crystals increases strength significantly. It was therefore decided that the strategy would be to increase the quantity of Ni and use die casting in order to increase the high-temperature strength of the piston material. From this strategy, the detailed specifications were determined.

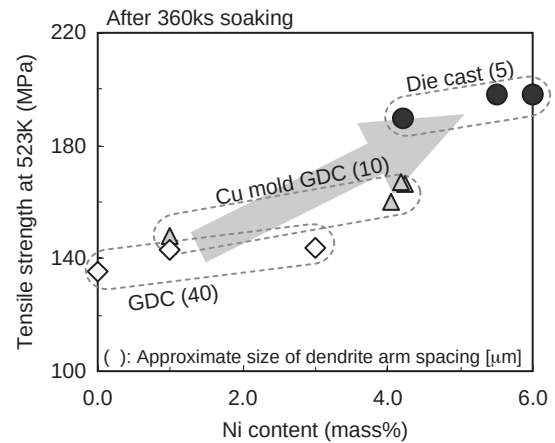


Fig. 2 Effect of Ni on tensile strength in various manufacturing processes

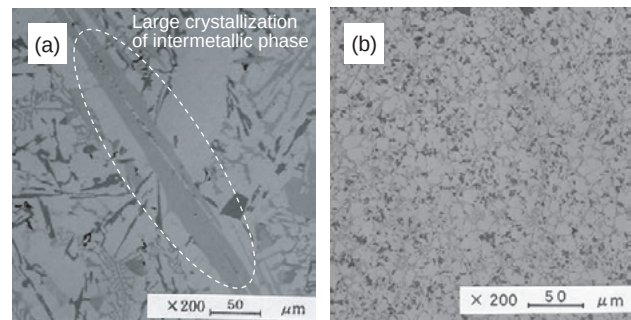


Fig. 3 Microstructures of (a)GDC-Ni 4mass% added and (b)Die cast-Ni 6mass% added

3. Method of Experimentation

Because die casting requires considerable time for determining the casting plan, die fabrication, and determining casting conditions, the following procedure was followed to determine the detailed specifications.

- (1) A wedge-shaped specimen is fabricated by GDC to determine the correlation between microstructure at the time of quenching and Ni amount added. This determines the basic components.
- (2) A simplified piston-shaped specimen (simplified piston) is die cast to determine material properties.
- (3) An actual piston is die cast to determine quality.

3.1. Determining the Basic Components of the Wedge-shaped Specimen

Alloy samples consisting of Al-12%Si-4%Cu-1%Mg as the base alloy and Ni ranging between 4.5% and 6.0% (all mass%) were prepared for determining the maximum amount of Ni that can be added during quenching. Fe, which is believed to have good affinity with Ni crystals, was ranged between 0.2–0.5 mass% to determine the element's effect on crystallization. In addition, since Mn is known to modify the shape of Al-(SiFe) plate crystals⁽¹²⁾, samples were prepared with an Mn/Fe ratio of 0.2 to 1.0 to investigate this effect.

The wedge-shaped specimens were fabricated with a molten-metal temperature of 993 K and a die temperature of 343 K. The microstructure was observed with a metal microscope at the thickness which corresponds to the cooling rate of an actual die-cast piston. Vickers hardness testing was also performed to determine the microstructure's effect on strength. Figure 4 shows a schematic diagram of the wedge-shaped specimen.

3.2. Determining the Material Properties of the Simplified Piston

Table 1 lists the composition of the experimental alloy and a conventional alloy used for comparison. A simplified piston made of the experimental alloy was fabricated with a 350-ton die cast machine at a molten metal temperature of 973 K, a casting pressure of 80 MPa, and a gate speed of 80 m/s. After T5 heat treatment, a 10-mm-thick sample was taken from the crown of the simplified piston. The conventional alloy was cast into a JIS4 boat shape with a molten metal temperature of 1 023 K and a die temperature of

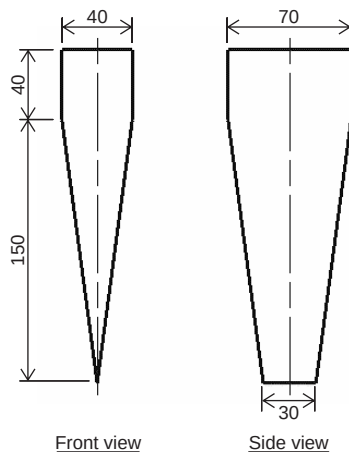


Fig. 4 Schematic diagram of wedge specimen

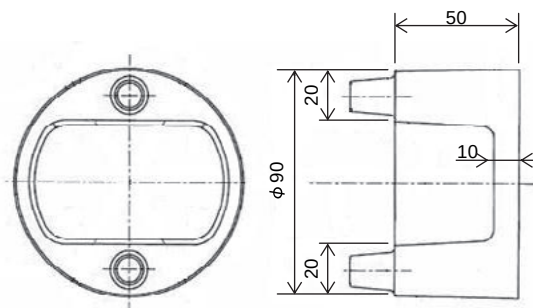


Fig. 5 Schematic diagram of temporary piston

Table 1 Chemical composition of experimental alloy for die cast and conventional alloy for GDC

	(mass%)					
	Si	Cu	Ni	Mg	Fe	Mn
Experimental alloy	12.4	3.9	5.4	1.0	0.4	0.2
Conventional alloy	12.6	4.8	1.0	1.0	0.1	–

Table 2 Test methods and temperature conditions

	Method	Temperature (K)					Soaking time
		R.T.	473	523	548	573	
Tensile strength	JIS Z2241		○	○		○	360ks
Fatigue strength	Push-Pull servo pulser		○	○		○	
Abrasiveness	Abrasion tester			○	○	○	360ks
Young's modulus	JIS Z2280	○					
Thermal conductivity	JIS R1611			○			
Thermal expansion	TMA			○			
Notch fatigue strength	Push-Pull servo pulser			○			

593 K. After T7 heat treatment, a sample was taken from the bottom at the 10-mm point. Measurements were taken at multiple temperatures, and averages were calculated for tensile strength, fatigue strength, hammer abrasiveness, Young's modulus, thermal conductivity, coefficient of thermal expansion, and notch fatigue strength. The simplified piston shape is shown in Fig. 5, the test methods and temperature conditions in Table 2.

3.3. Determining Quality of the Actual Piston

Taking the die cast method into account, thickness at various points on the actual piston was altered based on the piston for a 1.3-L engine; the resultant shape is shown in Fig. 6. The experimental alloy had the composition listed in Table 1 and was formed into the piston using a 250-ton die cast machine with a molten metal temperature of 1 003 K, a casting pressure of 95 MPa, and a gate speed of 80 m/s. To assure high quality, a vacuum to the die cavity and additional pressure were applied.

Liquid penetrant inspection and porosity rate were measured as indicators of internal quality. Porosity rate χ (%), indicator for the space ratio inside the piston, was calculated from theoretical density ρ_r (g/cm³) and actual density ρ_a (g/cm³) using Equation (1):

$$\chi = \frac{(\rho_r - \rho_a)}{\rho_r} \times 100 \quad (1)$$

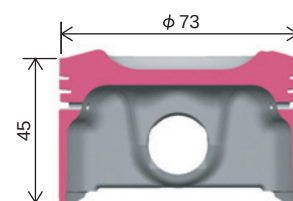


Fig. 6 Schematic diagram of piston

Instead of theoretical density, which is difficult to calculate, this study used the density of a specimen sample obtained by the Ransley method, which is considered to have a low probability of resulting in blowholes.

4. Results and Analysis

4.1. Microstructure

4.1.1. The effect on microstructure by the quantity of added Ni and Fe

Figure 7 shows the correlation between amount of added Ni and Fe and the occurrence of large ($>100\ \mu\text{m}$) crystals in the microstructure. At 4.5 mass% Ni, the desired microstructure, with no large crystals, is observed up to 0.5 mass% Fe. When the amount of Ni increases, the relative amount of Fe that forms large-crystals decreases. At 6.0 mass% Ni, some large crystallization is seen even at 0.2 mass% Fe.

The reason that Fe effects large crystallization is believed to be that Fe is substituted in the Al-Ni crystals, altering the Al-Ni to Al-(NiFe) crystals, thereby promoting large crystallization. A binary phase diagram for Al-Ni indicates that its eutectic point is roughly 5.7 mass%. In view of the danger of primary crystal formation and crystallization due to Fe substitution, Ni and Fe content should not exceed 5.5 mass% and 0.4 mass%, respectively.

4.1.2. The effect on microstructure by the quantity of added Mn

The microstructure and hardness of specimens with varying Mn/Fe ratios were examined in anticipation that the effectiveness of Mn in modifying the shape of Al-(SiFe) plate crystals, as stated in Section 3.1, would manifest itself in Al-(NiFe) crystals. Figure 8 shows the results. Contrary to expectations, a higher Mn/Fe ratio resulted in larger crystals and lower Vickers hardness.

To determine Mn distribution, the component elements of the large crystals were analyzed with an Electro-probe X-ray microanalyzer. Figure 9 shows the results. It was revealed Mn substitution is seen in Al-(SiFe), Al-Fe, and Al-(NiFe) crystals,

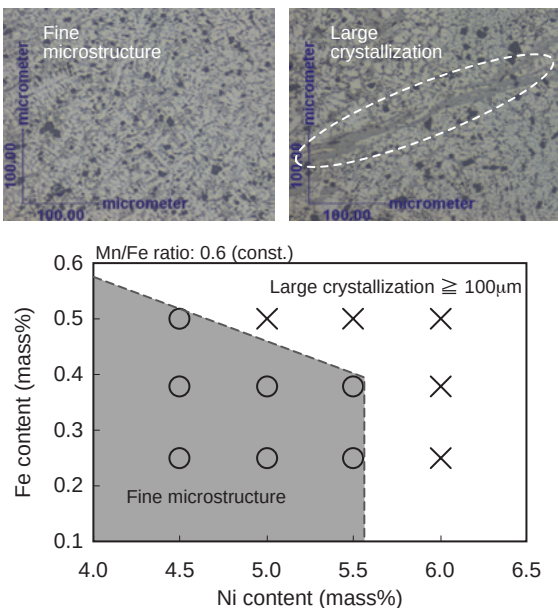


Fig. 7 Effect of Ni and Fe content on microstructure

Table 3 Chemical composition of developed alloy

	Si	Cu	Ni	Mg	Fe	Mn
Developed alloy	13	4	5	0.8	0.4	≤ 0.2

and some Al-(SiFe) integrated with Al-Fe lumps due to the Mn substitution, but it was found that Mn substitution caused Al-(NiFe) to form large plate crystals.

The preceding suggests that the decrease in hardness due to the addition of Mn in a high-Ni alloy containing Fe is because Mn promotes the formation of large Al-(NiFe) plate crystals, reducing the amount of micro-dispersed crystals. Although the mechanism by which Mn affects the various crystals is a subject for further research, the results of this study suggest that the Mn/Fe ratio should not exceed 0.6.

4.2. Basic Composition Determined in This Research

In view of the results detailed in Section 4.1, the following composition was decided for the new alloy: 5.0 mass% Ni, 0.4 mass% Fe, and 0.2 mass% or less Mn. Table 3 shows chemical composition.

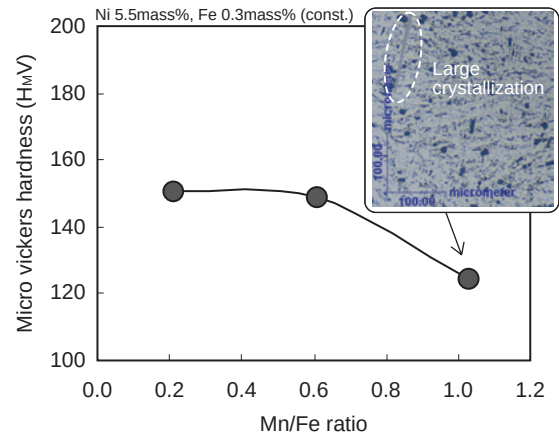


Fig. 8 Effect of various Mn/Fe ratios on microstructure and hardness

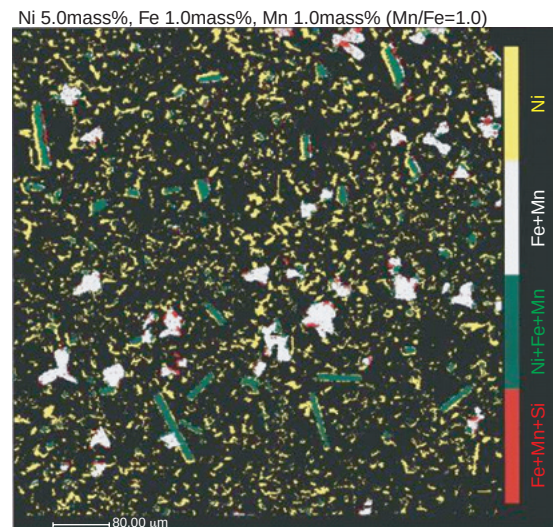


Fig. 9 Photograph of EPMA element mapping

4.3. Material Properties

4.3.1. High-temperature strength

Figure 10 shows testing results for tensile strength and fatigue strength. The new material achieved the goal for fatigue strength by exhibiting 1.5 and 1.8 times the fatigue strength of the conventional material at 523 K and 573 K, respectively.

4.3.2. Ring groove abrasiveness

Figure 11 shows the results of hammer abrasion testing. The new material exhibited one-sixth of the abrasion of the conventional material at 523 K. This is possibly because the lower surface area of the aluminum substrate, due to the addition of Ni, delays the onset of microwelding and lessens the softening of the material at high temperatures. Figure 12 compares the microwelding onset time and softening of the new material and of the conventional material.

4.3.3. Fracture toughness and physical properties

Figure 13 gives the test results for notch fatigue strength, Young's modulus, coefficient of thermal expansion, and thermal conductivity. Although the new material's notch fatigue strength increased by a factor of only 1.2 over the conventional material, which is less than the increase exhibited by the unnotched test piece, it did demonstrate sufficient fracture toughness. Young's modulus and the coefficient of thermal expansion were 16% higher and 6% lower, respectively. Nickel has been known to greatly enhance these properties^{(13), (14)} and apparently performed the same function in this material. Although thermal conductivity was 11% lower compared to the conventional material, Computer Aided Engineering (CAE) predictions of temperature increase indicate that this would entail an increase of only 5 K over the conventional material. This, in addition to the absence of any observed decrease in knocking properties in an actual engine, suggests that the material possesses the properties required for pistons.

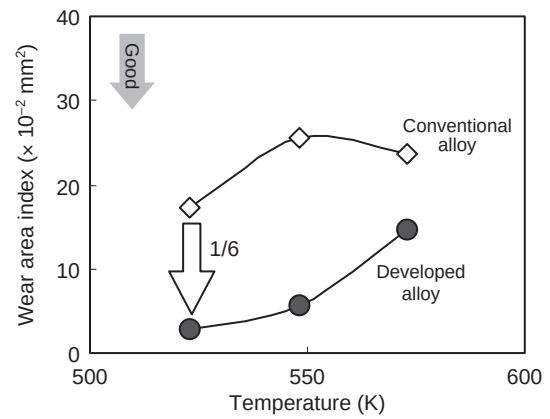


Fig. 11 Abrasiveness

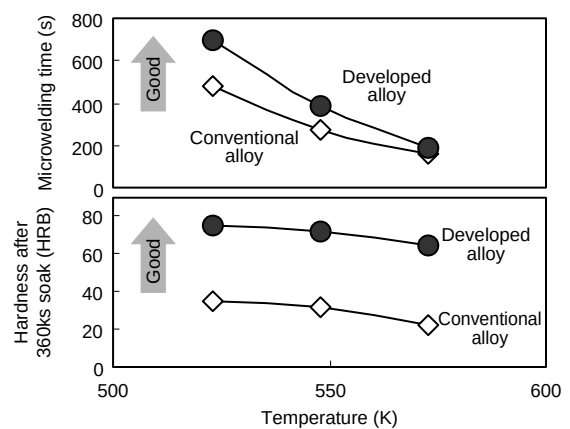


Fig. 12 Comparison of microwelding time and hardness

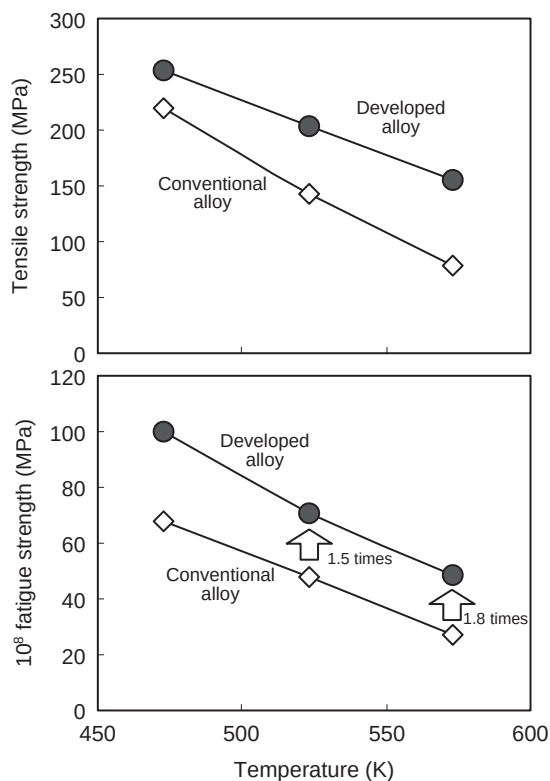


Fig. 10 Tensile and fatigue strength

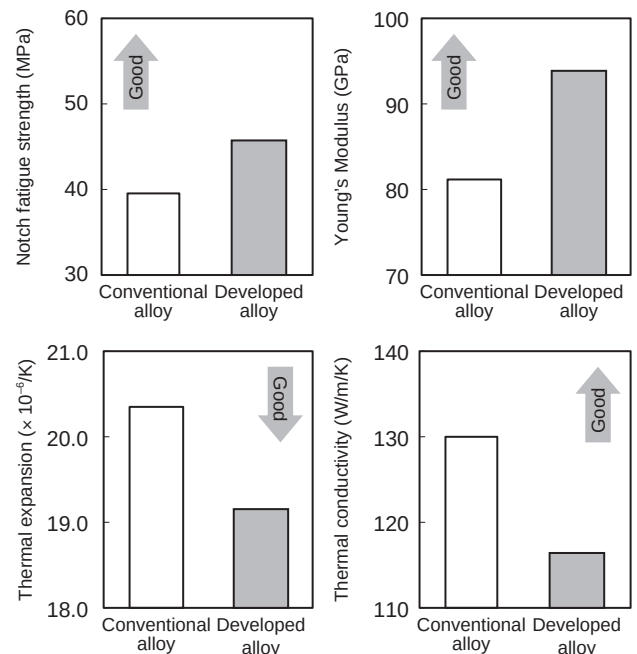


Fig. 13 Notch fatigue strength and physical properties

4.4. Casting Quality

Figure 14 shows the increase in quality made possible by the use of vacuum and additional pressure. With die casting, internal blowholes caused by the infusion of air are often an issue, but the application of vacuum and additional pressure have made it possible to achieve porosity rates lower than those of GDC pistons. Additional pressure has a greater porosity-lowering effect than vacuum, indicating the importance to compensate solidification shrinkage in die casting.

With regard to the sludge, oxide film, and scattered chill structure that occur during melting or injection of the molten metal, high quality was achieved by optimizing the casting plan, molten metal treatment, the type of plunger sleeve material, the amounts of mold release agent and sleeve lubricant, among others.

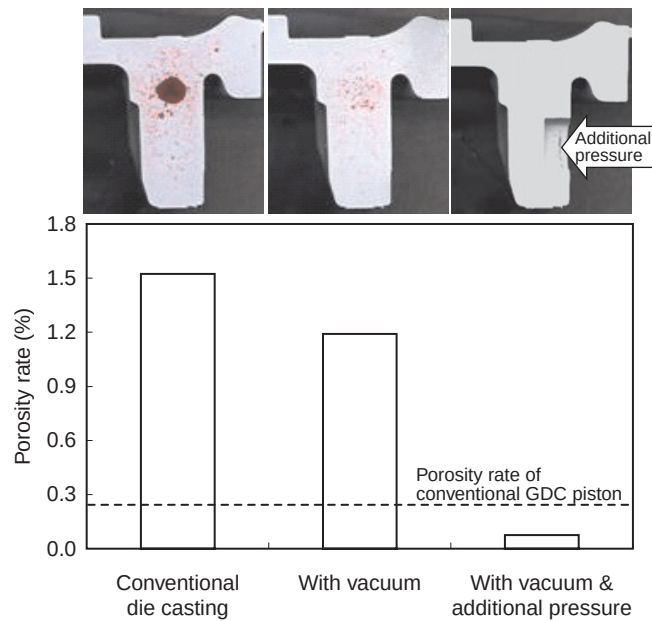


Fig. 14 Effect of vacuum and additional pressure on casting quality

5. Effectiveness in Actual Piston Application

5.1. Basic Piston Specifications

Table 4 lists the basic specifications of the piston made with the new material. The engine selected was one designed for optimum fuel efficiency. Using CAE, the shape of each part of the piston was optimized for minimum friction without sacrificing durability. An example of surface pressure distribution on the piston skirt, optimized with three-dimensional piston dynamics calculations, is shown in Fig. 15.

5.2. Application Effectiveness

Friction and fuel-efficiency measurements of the piston described in Section 5.1 are given in Fig. 16. In engine bench testing, friction was 4.4% lower and fuel efficiency was 2.2% higher.

Table 4 Specifications of developed piston

	Conventional piston	Developed piston
Diameter (mm)	73	73
Compression height (mm)	26.2	26.2
Crown thickness (mm)	5.2	4.5
Land height [top/2nd/3rd] (mm)	5.55/3.0/2.5	5.55/3.0/2.5
Skirt width x height (mm)	34 x 29.5	38 x 29.5
Skirt thickness [skirt/side wall] (mm)	2.0/3.5	2.5/3.45
Piston weight (g)	207	205
Piston shape		
Specific power of applied engine (kW/L)	47.1	52.3
Top ring material/Outer surface treatment	Carbon steel/Hard chromium plating	Carbon steel/Ion plating

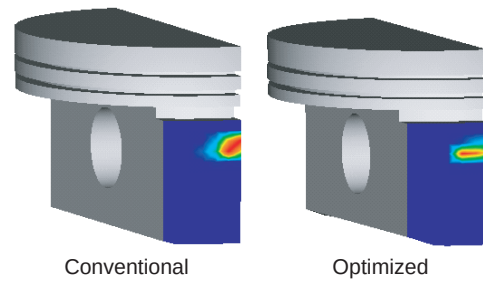


Fig. 15 Three-dimensional piston dynamics calculation

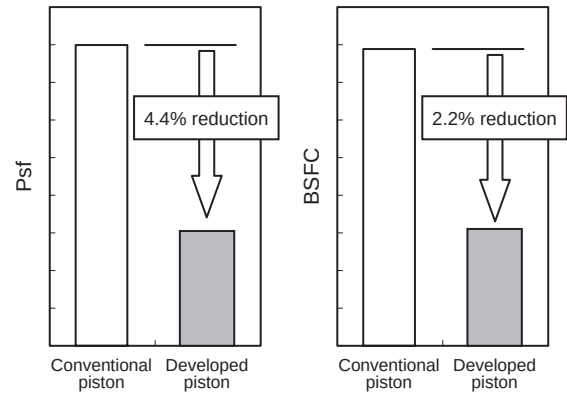


Fig. 16 Effect of developed piston on brake specific fuel consumption (BSFC) and friction loss (Psf)

6. Conclusion

The technology for a new, high-strength piston material was developed by using die casting to raise the limit of Ni that can be added to aluminum alloys. The conclusions obtained in this development are as follows.

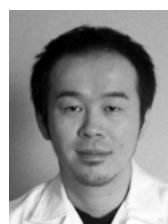
- (1) Using the die casting method allowed the amount of Ni added to be increased to 5.5 mass%, raising the fatigue strength of the new material at temperatures of 523 K or higher by a factor of 1.5 over a conventional material made by GDC.

- (2) The amounts of Fe and Mn added must be controlled to prevent large crystallization: Fe added must be kept within 0.4 mass%, and the Mn/Fe ratio to within 0.6.
- (3) Application of vacuum to the die cavity and using additional pressure enabled quality exceeding that of conventional GDC pistons.
- (4) Pistons made with the new material decreased engine friction by 4.4% and increased fuel efficiency by 2.2% in engine bench testing.

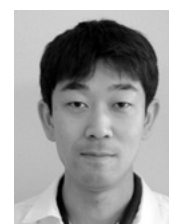
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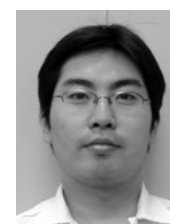
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Research for High Concentration of Precious Metal and Thin Layer of Catalyst Top Layer

Syouji INOSE*

Tatsuya OKAYAMA*

ABSTRACT

Reduction of the precious metal loading of a catalyst generally causes a decline in light-off performance.

Study of the low temperature reaction mechanism of catalysts indicated the importance of the top layer in the reaction, thus increasing reactivity in the top layer was established as an objective. The concentration of precious metals in the top layer was increased, and the thickness of the layers was reduced. At the same time, the precious metal loading of the bottom layer, which contributes little to light-off performance, was reduced by 80%. However, increasing the concentration of precious metals in the top layer caused a reduction in performance due to a sintering effect between the metals. Research was therefore conducted on the employment of a high-thermostability support material (alumina) to control the sintering.

This research has enabled the development of a three-way catalyst with significantly reduced precious metal loading while retaining conversion performance equivalent to that of conventional catalysts.

1. Introduction

The tightening of automotive emissions regulations has recently been increasing globally, as seen in the Tier-2 and LEV2 regulations in the US, EURO4 regulations in the EU, and 2005 regulations in Japan.

Catalyst conversion performance plays an important role in reducing emissions, enabling manufacturers to respond to the tightening regulations. Conversion performance is strongly dependent on the precious metal loading of the catalyst, and therefore the amount of precious metals used per vehicle has tended to increase with tightening regulations.

The majority of automotive emissions occur immediately after engine starts, before the catalyst has warmed up. Achieving enhanced light-off performance is therefore important in terms of

reducing overall emissions. However, light-off performance generally declines when precious metal loading is reduced.

The research project discussed in this paper considered the relationship between precious metal loading and light-off performance theoretically from the perspective of the catalytic reaction mechanism. Based on these considerations, research was focused on minimizing precious metal loading while maintaining conversion performance equivalent to that of conventional catalysts.

2. Development Goals

2.1. Emissions Characteristics

Figure 1 shows the results of measurements of hydrocarbon (HC) emissions from the 2003 model Accord (in-line 4-cylinder

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2.4L engine) under testing in FTP LA-4 mode⁽¹⁾. The 2003 Accord cleared the PZEV regulations, the most stringent in the US.

Of the various emission components, carbon monoxide (CO), Hydrocarbon (HC) and nitrogen oxides (NOx), it is HC that presents the greatest challenge in clearing regulation targets. In the test cycle shown in Fig. 1, Bag-1 represents the cycle which follows engine start, while Bag-2 and Bag-3 are hot cycles, i.e. they occur after the engine has warmed up. Approximately 80% of total HC emission occurs in Bag-1. This is due to advances in air-fuel ratio control systems⁽²⁾ and enhancements in catalyst characteristics which have enabled contemporary emissions conversion systems to significantly reduce emissions in Bag-2, 3 where the catalyst is activated after the engine has warmed up.

To achieve further emissions reductions in the future, it will therefore be necessary to reduce emissions in Bag-1 at engine start, and early activation of the catalyst will be essential in enabling this.

2.2. Emissions Reduction Technologies

Both rapid heating of the catalyst by means of engine management⁽³⁾ and close coupled positioning of the catalyst are currently employed to achieve early activation of conventional catalysts. In addition, research has been conducted on the following catalyst types to enhance light-off performance and reduce HC emissions:

- (1) Zone coat catalysts, in which the precious metal loading is increased at the exhaust gas entrance⁽⁴⁾
- (2) Catalysts employing complex oxide catalysts (perovskite structures, etc.) to supplement precious metals^{(5), (6)}
- (3) Transient adsorption of the pre-activation HC emittants⁽⁷⁾

However, to clear increasingly stringent future emissions regulations and reduce cost, research on methods of making maximally effective use of the precious metals in the catalyst is required.

2.3. Issues of Conventional Catalysts

For activation, a catalyst must be heated to their respective functional temperatures (activation temperature) or above. The catalyst is heated by heat exchange with exhaust gases. When the engine is stopped, the temperature of the catalyst equilibrates to ambient temperature; after engine start, the catalyst is heated to activation temperature by the temperature of exhaust gases. However, until the activation temperature is reached, exhaust gases are emitted without catalytic conversion. Low temperature light-off performance is therefore essential in a catalyst.

Figure 2 shows the relationship between precious metal loading and light-off performance in a conventional catalyst. Conventional catalysts present the issue that reducing precious metal loading causes light-off performance to decline.

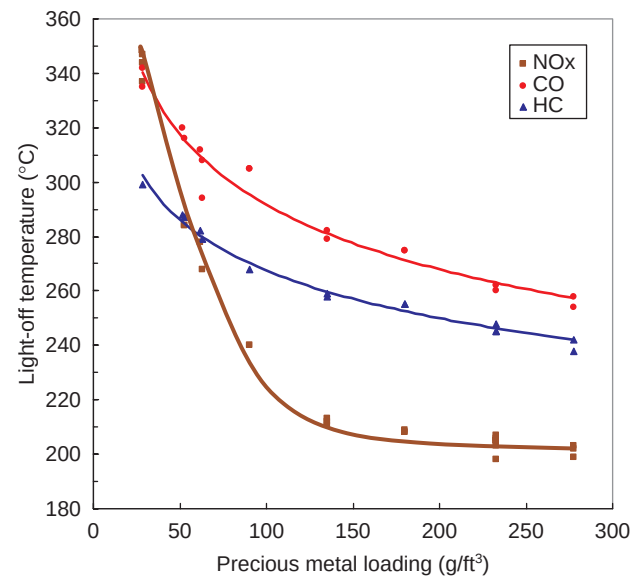


Fig. 2 Relationship between loading of precious metal and light-off temperature

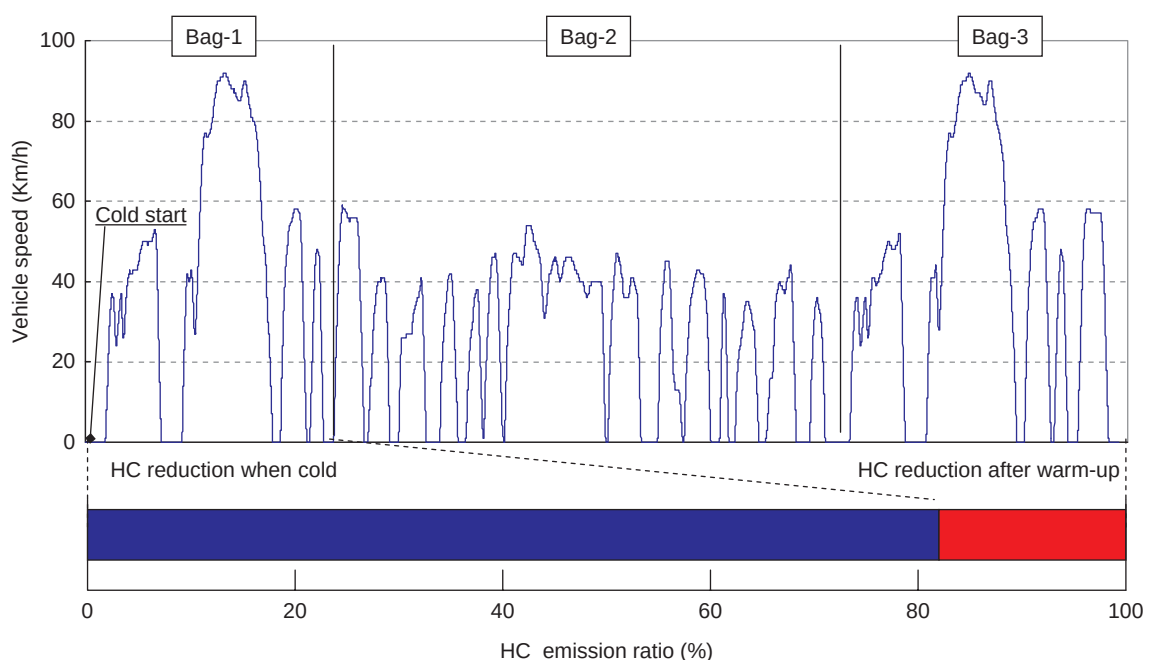


Fig. 1 HC emission in the cold condition and after warm-up (US LA-4 test cycle)

Theoretical consideration of the relationship between precious metal loading and light-off performance, from the perspective of the catalyst reaction mechanism, led to the establishment of the target of the project discussed here as the development of a catalyst that displays identical light-off performance to that of a conventional catalyst with half the precious metal loading.

3. Theory

3.1. Theory of Catalyst Reaction

Here, the authors conduct a simple theoretical examination of catalytic reactions on solid surfaces to investigate methods of effectively utilizing the precious metals in the top layer of the catalyst, to enhance light-off performance.

The following two factors are important in the chemical reaction:

- (1) Activation energy, which forms an energy barrier for reaction
- (2) Molecular energy, which crosses the energy barrier for reaction, to cause a reaction

The development process of chemical industrial catalysts focuses on (1), and generally changes the active species to reduce activation energy. However, from the perspective of durability and conversion performance, no active species have been discovered that can supplant the precious metals platinum (Pt), palladium (Pd) or rhodium (Rh) which are used in current automotive three-way catalysts. Therefore, this project focused on (2). With the aim of achieving enhanced light-off performance, a scenario in which high-temperature exhaust gas was introduced to a room temperature catalyst was assumed, and activity at the atomic and molecular levels was studied using classical molecular collision theory without considering quantum effects.

3.2. Molecular Collision Theory on a Solid Surface

Molecular collision theory is effective in dealing with reactions in the gas phase. This project applied this classical theory to a system including solid catalyst surfaces to determine a method of effectively utilizing the energy in the system.

In molecular collision theory, the essential condition for a reaction to occur is that the translational energy generated by molecular collisions exceeds activation energy. We will use this theory to consider reactions on solid catalyst surfaces. When the molecules making up exhaust gas are absorbed by a solid catalyst surface, translational energy is transformed into surface free energy. The essential condition for a reaction to occur is for the sum of the transformed energy and the energy in the solid surface system to be greater than that of the activation energy (Ea) of the catalyst. Based on these considerations, we will apply molecular collision theory to solid catalyst surfaces to study methods of achieving enhanced light-off performance.

3.3. Modification of Catalyst to Reduce Light-off Temperature

3.3.1. Application of molecular collision theory to solid surface

Assuming that the temperature of the catalyst is sufficiently low in relation to the temperature of the exhaust gas, the energy required for a reaction to proceed is mainly imparted by the translational energy of the gas phase molecules. The reaction is therefore largely determined by the magnitude of the translational kinetic energy of

the molecules in the gas phase, and can be theorized as follows: The gas molecules impart surface free energy to the catalyst through collisions, but the essential condition for a reaction to proceed is that the translational energy is greater than Ea' , the sum of the activation energy of the parts of the catalyst involved in the reaction (Ea), and the energy imparted to the surface free energy of the parts not involved in the reaction.

The distribution of molecular velocity in the gas phase is dependent on temperature, and can be expressed as the following Maxwell-Boltzmann distribution:

$$\frac{dN}{N} = 4\pi \left(\frac{m}{2\pi kT} \right)^{\frac{3}{2}} \exp \left(-\frac{mv^2}{2kT} \right) v^2 dv \quad (1)$$

N : Total number of molecules

v : Velocity of molecules

m : Mass of molecules

k : Boltzmann constant

T : Temperature

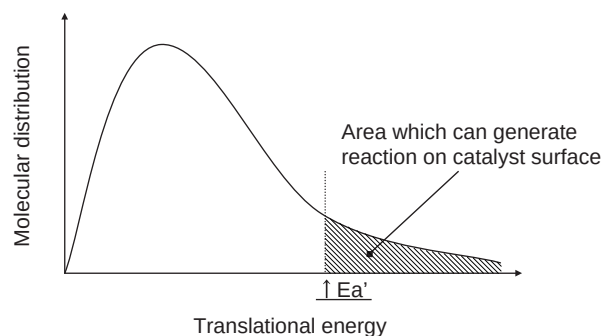
dN : Number of molecules with velocities between v and dv

This equation enables the velocity distribution of the molecules, and also the energy distribution in the gas molecules, to be obtained. Figure 3 shows the relationship between the distribution of translational energy and the translational energy equivalent to Ea' , which can enable a chemical reaction to proceed on the solid catalyst surface, in the gas molecules. As the diagram shows, although average molecular energy is low, there are molecules in the system that can enable a reaction. Based on the theory discussed above, the mechanism is as follows: Molecules with a translational energy greater than Ea' , i.e. that satisfy the conditions for a reaction, collide with the solid surface and their translational energy is transformed into surface free energy. If the conditions for a reaction are satisfied, the reaction will then proceed. It is therefore important to effectively exploit these small numbers of molecules that enable a reaction to proceed in order to ensure low-temperature reactions in the catalyst. We will look at two scenarios from the perspective of developing a method of effectively exploiting the molecules with energy levels greater than that of Ea' :

- (1) Molecular collisions cause a reaction to proceed
- (2) A reaction does not proceed despite molecular collisions

3.3.2. Scenario (1)

We will look first at scenario (1), in which the collision of molecules with a translational energy greater than Ea' satisfies the



Ea' : Translational energy distribution of gas molecules which can provide energy for reaction on catalyst surface

Fig. 3 Translational energy distribution of molecules

conditions for reaction and the reaction proceeds on the surface of the catalyst.

Exhaust gas conversion reactions are generally exothermic; as the reaction proceeds, reaction energy is imparted to surface free energy. Some of this surface free energy is lost through molecular desorption or as translational energy, but the area in which the reaction is taking place continues to possess a higher energy level than that of its surrounding areas. As the surface free energy of the solid catalyst increases, even molecules with a translational energy lower than E_a' will satisfy the conditions for reaction, enabling the reaction to propagate. In other words, maintaining the surface free energy of the area of the catalyst in which the reaction has occurred is important in enhancing light-off performance.

As time elapses, the imparted surface free energy is diminished as a consequence of thermal conductivity toward the interior of the solid. It is therefore necessary to transfer our attention from the atomic and molecular levels to that of the macro level. At this level, controlling the dispersion of heat around the surface of the solid becomes an important issue. After the reaction has commenced, heat energy is dispersed rapidly from isolated active sites, which makes it necessary to concentrate the active sites, that is, achieve a high concentration of precious metals.

3.3.3. Scenario (2)

We will go on to look at the effective use of energy in scenario (2), in which the reaction does not proceed. In this scenario, molecules satisfying the conditions for reaction impart some of their energy to surface free energy through collisions, and as a result lose their energy. Figure 4 shows the phenomenon on the solid catalyst surface considered from the macro level; molecules with a translational energy greater than that of E_a' penetrate the bottom layers of the catalysts without causing a reaction and lose energy by colliding with macro pores. This reduces the number of molecules with energy levels greater than that of E_a' at the surface

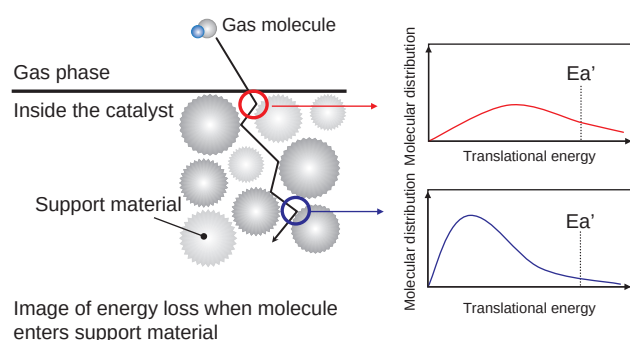


Fig. 4 Conceptual diagram of efficient usage for translational energy

of the catalyst. To prevent this, it is necessary to position the active sites as much as possible at the surface of the catalyst, in close proximity to the gas phase.

3.4. Results of Study of Catalyst Modification

The considerations discussed above indicated that the following two methods would be necessary to effectively exploit the energy of molecules to enable them to cross the energy barrier for reaction and cause a reaction to proceed:

- (1) Increasing the concentration of active sites, i.e. increasing the concentration of precious metals
- (2) Ensuring the precious metals in the top layer of the catalyst

Tests were therefore conducted to verify the effectiveness of these methods, and studies were conducted of relevant catalyst specifications.

4. Results and Discussion

4.1. Catalyst Design

4.1.1. Achievement of high concentration of precious metals in top layer

Lean air-fuel ratios (air in excess of fuel for a stoichiometric air-fuel ratio) are employed to reduce exhaust emissions at engine start, before the catalyst has warmed up. Pd is known to possess excellent HC conversion performance in a lean condition⁽⁸⁾. Pd was therefore employed in the top layer of the catalyst developed in this project.

Tests were conducted to study the effect of Pd concentration on light-off performance. Four-layer coat catalysts were manufactured as test pieces using only Pd. Figure 5 shows the structure of these catalysts. A fixed Pd loading was employed in each catalyst, but the concentration in the top layer was varied.

As Fig. 6 shows, light-off performance changes in proportion to the concentration of Pd in the top layer. The test catalyst with the highest concentration of Pd in the top layer displayed the best light-off performance.

These results verified that the concentration of Pd in the top layer contributes significantly to the light-off performance of the catalyst.

In addition, the results showed that the distribution of Pd had no effect on performance at 350 °C and above, when the catalyst is activated.

4.1.2. Reducing the concentration of precious metals in the bottom layers of the catalyst

When the catalyst is activated, changes in driving conditions

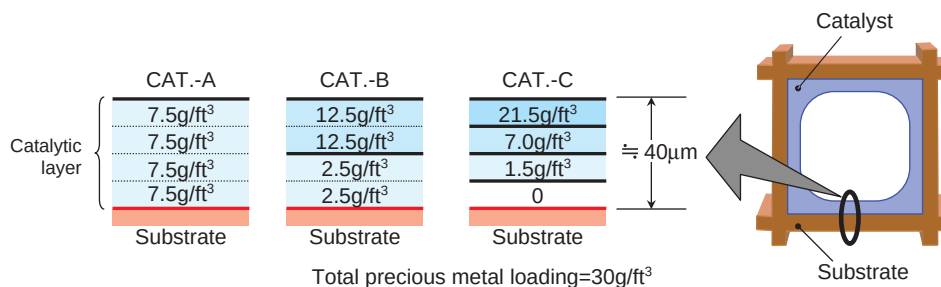


Fig. 5 Pd distribution of test piece

(acceleration, deceleration, etc.) tend to increase NO_x emissions. The use of Pt and Rh, which display excellent NO_x conversion performance⁽⁹⁾⁻⁽¹¹⁾, in the bottom layers of the catalyst developed in this project enabled the achievement of conversion performance identical to that of a conventional catalyst when activated with an 80% reduction in precious metal loading.

4.2. Achievement of a High Concentration of Precious Metals in the Top Layer - Issues and Measures

A catalyst with a high concentration of precious metals in the top layer was manufactured using a standard support material and subjected to a heat resistance test at 900 °C. After the test, the catalyst failed to display the expected light-off performance. The cause of this phenomenon was theorized as follows:

Alumina is employed as the support material for precious metals in automotive catalysts. To increase the probability of the alumina coming into contact with exhaust gas, γ alumina with a high specific surface area is used. However, when alumina is exposed to high temperatures, it undergoes a phase transition in its crystal structure from that of γ phase to α phase. This reduces the specific surface area, which in turn reduces the distance between the grains of

precious metal and causes sintering. As a result, the number of active sites drops, and light-off performance declines.

To respond to this issue, other metallic elements were added to alumina in an attempt to stabilize its crystalline structure and increase its resistance to high temperatures. The elements formed compounds with part of the alumina, generating a crystal structure that is stable at high temperatures. In addition, the modified alumina and γ alumina share a common layer in their crystal structures that makes it easy for linkages to form; the crystal structure becomes highly stable at the interface, controlling the formation of α alumina. It was hypothesized that the use of this method would ensure the stability of the crystal structure of the alumina at high temperatures.

Figure 7 shows the crystal structure change in the conventional alumina and the newly developed alumina with an X-ray diffractometer after heat resistance tests at 1 100 °C to verify the effectiveness of this method of stabilizing the crystal structure. The conventional alumina shows numerous α phase diffraction peaks, indicating a phase transition from that of γ phase to α phase. By contrast, no α phase diffraction peaks are shown for the modified alumina, indicating that the phase transition from γ to α phase has been controlled, thus verifying the effectiveness of adding metallic

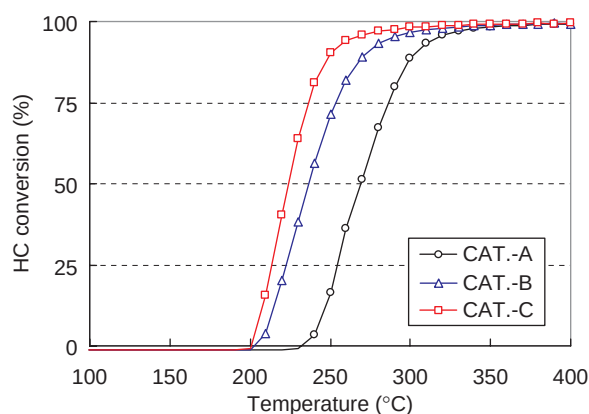


Fig. 6 Relationship between Pd concentration in top layer and HC light-off temperature

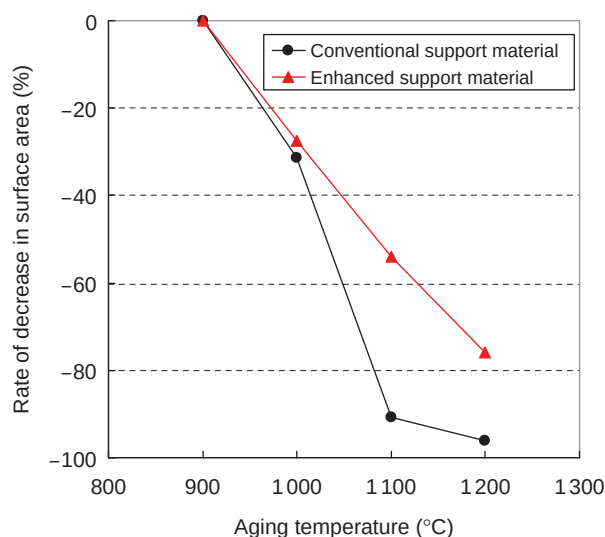


Fig. 8 Effect of enhancing support materials on specific surface area

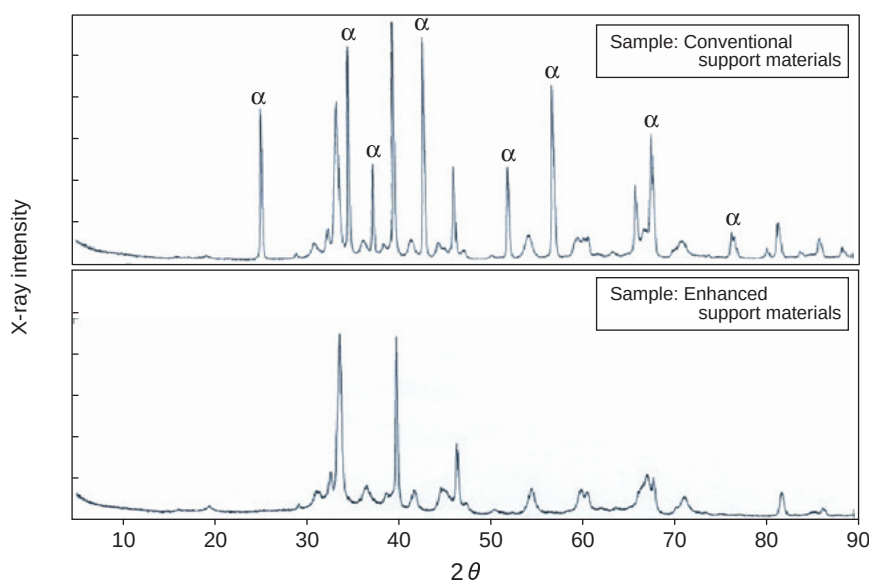


Fig. 7 XRD pattern of support materials (Aging condition:1 100°C)

elements to the alumina as a method of stabilizing the crystalline structure.

Figure 8 shows changes in the specific surface area of conventional alumina and the modified alumina after high-temperature heat resistance tests for a fixed period of time. The specific surface area of the conventional alumina changes rapidly above 1 100 °C, and declines by approximately 60% between 1 000 °C and 1 100 °C. By contrast, the specific surface area of the modified alumina is reduced by only approximately 25% between 1 000 °C and 1 100 °C, and the change above 1 100 °C is slower than that displayed by the conventional alumina. These results indicate that the reduction in specific surface area has been effectively controlled by modification of the alumina.

To verify the effectiveness of the modified alumina in controlling the sintering of Pd, catalysts were manufactured using identical amounts of Pd to support conventional and modified alumina, and subjected to heat resistance tests at 1 100 °C. Figure 9 shows the results of Transmission Electron Microscopy (TEM) of the Pd grains in these catalysts. In the catalyst employing conventional alumina, the diameter of the Pd grains varies between 30 and 300 nm, and sintering has occurred. In the catalyst employing the modified alumina, the diameter of the Pd grains is approximately 20 nm, the grains are evenly dispersed, and no sintering has occurred.

4.3. Effect of Top Layer Pd Concentration on Heat Resistance

Catalysts with top layer Pd concentrations of 12% (12%-Pd) and 18% (18%-Pd) were manufactured and subjected to heat resistance tests from 900 °C to 1 100 °C, after which light-off performance was tested in model gas tests. As Fig. 10 shows, the 18%-Pd catalyst displayed superior light-off performance to that

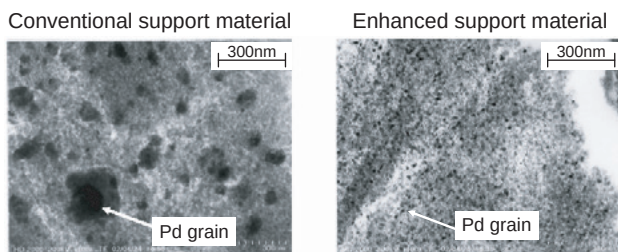


Fig. 9 TEM image of Pd grain (Aging condition: 1 000°C)

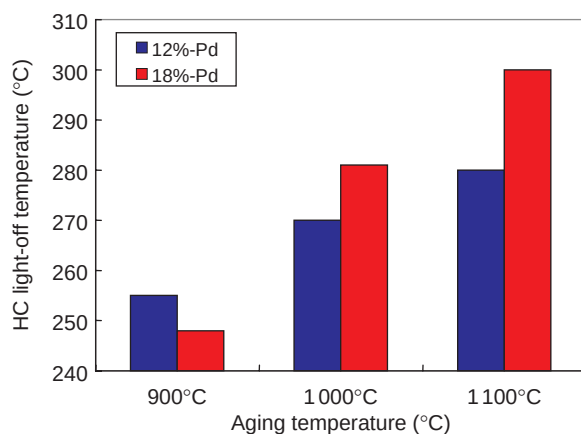


Fig. 10 Relationship between Pd concentration and HC light-off temperature

of the 12%-Pd catalyst after heat resistance testing at 900 °C. However, the effect was reversed with change in temperature, with the 12%-Pd catalyst displaying superior light-off performance after testing at temperatures of 1 000 °C and above. The catalyst with the higher concentration of Pd showed a significantly greater decline in HC light-off performance at 1 000 °C than that of the other catalyst; the light-off temperatures of the 12%-Pd and 18%-Pd catalysts increased by 15 and 33 °C respectively, between 900 and 1 000 °C. This is believed to be because the 1.5 times greater concentration of Pd in the 18%-Pd catalyst brings the grains into closer proximity, and sintering therefore occurs despite the use of the modified alumina as a support material.

These results indicate that optimal Pd concentration exists with respect to the temperature of the exhaust gas in the vehicles in which the catalyst will be employed.

4.4. Comparison of the PZEV Mass-production Catalyst and the Modified Catalyst

Based on the test results discussed above, a catalyst with half the precious metal loading of the PZEV mass-production catalyst was manufactured. Figure 11 shows the results of tests comparing the PZEV and the newly developed catalyst. The newly developed catalyst displays identical HC light-off performance and conversion performance after warm-up to the PZEV catalyst.

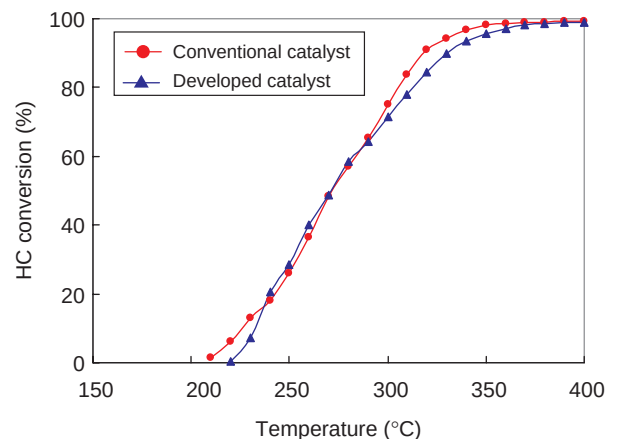


Fig. 11 Comparison of HC light-off temperature in conventional catalyst and developed catalyst

5. Conclusion

Major conclusions of the research discussed above are as follows:

- (1) Consideration of the catalyst reaction mechanism indicated that the concentration of precious metals in the top layer was an important factor in the achievement of enhanced light-off performance, and this hypothesis has been verified experimentally.
- (2) Increasing the concentration of precious metals in the top layer and reducing its thickness, at the same time reducing the concentration of precious metals in the bottom layers, has enabled a reduction in the amount of precious metals employed in each catalyst.
- (3) Sintering of the precious metals in the catalyst was controlled by increasing the concentration of precious metals and employing a high heat resistance support material.
- (4) Appropriate concentrations of precious metals exist for different temperatures to which the catalysts will be exposed.

This project succeeded in developing a catalyst with a reduced precious metal loading that displays identical performance to that of a conventional catalyst. The newly developed catalyst is being studied for use as a mass production catalyst for 2006 models.

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Syouji INOSE



Tatsuya OKAYAMA

Development of High-Pressure Die-Casting Simulation and Verification in Motorcycle Parts

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ABSTRACT

The quantitative evaluation of trapped air and strength of cast products at the completion of solidification in high pressure die-cast has been studied using material test and simulation. As a result, the prediction accuracy of trapped air analysis was enhanced by considering back pressure. In addition, by coupling the trapped air defect at the completion of flow with the solidification process, the analysis of mechanical characteristics of cast products has become possible.

1. Introduction

Motorcycle frame parts are being increasingly made of aluminum for the purposes of weight reduction. These frame parts are joined with aluminum extruded tubes, and have conventionally been manufactured by the gravity die-casting method in consideration of weldability. The high-pressure die-casting method is suitable for further weight reduction of frame parts from the perspectives of thickness, high accuracy and high productivity. However, conventional high-pressure die-casting methods have been considered unsuitable for welding.

In order to resolve these issues, efforts are being made to create die-cast frame parts using transition flow filling method die-casting⁽¹⁾. In addition, hollow structural castings that can be welded and offer high strength and rigidity have recently been manufactured by transition flow filling method die-casting and commercialized in response to increasing demands for weight reduction⁽²⁾. As an

additional method, there are also cases where the high-pressure die-casting method has been successfully applied to frame parts by increasing the vacuum inside the die cavity to reduce the amount of air trapped inside high-pressure die-cast products⁽³⁾.

Developing these cast products in a short time requires technology for predicting formability, weldability, strength, quality and other properties in the early development stages. However, it is not easy for conventional casting simulations to quantitatively analyze the amount of trapped air. On the other hand, high-pressure die-cast products do not have uniform internal quality so the strength differs depending on the product position, and developing methods for predicting the actual strength is desired to achieve further weight reduction.

This paper discusses an outline of a newly developed method for analyzing trapped air and actual strength in the high-pressure die-casting method, and reports the correlation between actual cast products and simulation results.

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2. Comparison with Conventional Technology

Recent casting simulations have often been performed with the aim of enhancing casting quality. Conventional casting simulations individually analyze the behavior of molten metal in the flow processes, changes in physical properties, or the generation of shrinkage cavities in the solidification processes, and these separate simulations have been used as necessary. It has been possible to predict defects and to qualitatively analyze casting designs for general casting, even by using these methods. However, most conventional casting simulations do not take into account the effects of back pressure inside the die cavity, so analysis was performed using constant pressure inside the die cavity. This made it unlikely to predict the amount of trapped air with sufficient accuracy, which limited these methods to qualitative evaluation. Therefore, this study took into account back pressure in addition to the conventional analysis methods and dealt with the two-phase flow of the gas and liquid phases. This made it possible to analyze the pressure gradient of the air inside the die cavity, and enhance the quantitative prediction accuracy of the amount of trapped air.

In addition, instead of analyzing the injection of molten metal from the biscuit, based on the boundary condition settings, the behavior of the molten metal inside the sleeve was also analyzed in order to perform more accurate flow simulations.

On the other hand, air that becomes trapped during the molten metal flow ultimately becomes porous within the cast product as shrinkage cavities occur during solidification. However, conventional solidification analysis is unable to account for the trapped air at the completion of filling in the initial solidification stages. This made it unlikely to accurately analyze the size and amount of porosity. These items must be taken into account, particularly for die-cast products where air becomes trapped in the filling process. The newly developed method made it possible to quantitatively analyze the size and amount of porosity at the completion of solidification by coupling the trapped air at the completion of the flow state with that of the solidification process. In addition, the strength and elongation could also be predicted by taking into account dendrite arm spacing (DAS) and segregation.

solidification time method, temperature gradient method, corrected temperature gradient method ($G/\sqrt{R}$ method), and so on⁽⁶⁾. However, these methods do not sufficiently take into account the porosity creation mechanism, so it was difficult to quantitatively predict the position and size to which porosity would be created. Therefore, simulations were conducted by modeling the principle of solidification porosity creation as shown by the images in Fig. 2. Initially, small porosities form at the root of dendrites, then the small porosities grow in size and separate from the root of the dendrites as solidification progresses. Then, as solidification progresses further and the dendrites contact each other, the porosities which have moved between the dendrites grow to compensate for solidifying contraction and become shrinkage cavities. The basic equations for these simulations are shown by (1) to (4) below⁽⁷⁾. In addition, Fig. 3 shows the flow chart for predicting defects.

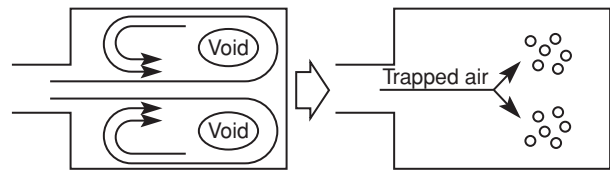


Fig. 1 Mechanism of trapped air evolution

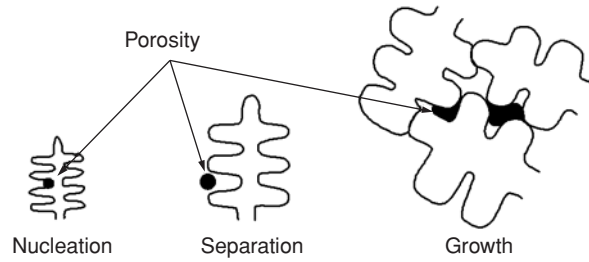


Fig. 2 Principle of porosity creation

3. Casting Simulations

3.1. Flow Simulations

Flow simulations allowed analysis by modeling the trapped air as shown by the image in Fig. 1. First, the occurrence of voids was predicted by solving the free surface by the Volume of Fluid (VOF) method⁽⁴⁾. In this case a void is a macro cavity expressed as an aggregation of multiple mesh elements. When a void occurs during the course of the flow simulation, the equation of conservation is solved. The void is then reduced to the size of a single element or less and disappears when the molten metal is filled. Trapped air the size of a single element or less is judged to have been generated at the point this macro cavity disappears. The size, gas pressure and number of trapped air locations are calculated. After that the size and gas pressure of the trapped air changes as the trapped air moves along with the molten metal flow⁽⁵⁾.

3.2. Solidification Simulations

On the other hand, methods for predicting defects in conventional solidification simulations include the isothermal method,

- (1) Equation of heat transfer

$$C\rho \frac{\partial T}{\partial t} = \lambda \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) - \rho L \frac{\partial f_L}{\partial t} \quad (1)$$

C : Specific heat, ρ : Density, T : Temperature,
 t : Time, λ : Thermal conductivity, L : Latent heat,
 f_L : Liquid phase ratio

- (2) Equation of continuity

$$\left(\frac{\rho_s}{\rho_L} - 1 \right) \frac{\partial f_L}{\partial t} - \frac{\partial f_L u}{\partial x} - \frac{\partial f_L v}{\partial y} - \frac{\partial f_L w}{\partial z} + \frac{\partial f_v}{\partial t} = 0 \quad (2)$$

ρ_s, ρ_L : Solid phase and liquid phase density
 u, v, w : Flow speed between dendrites
 f_v : Volume ratio of porosity

- (3) Equation of motion

$$u = -\frac{k}{\mu f_L} \frac{\partial P}{\partial x} \quad v = -\frac{k}{\mu f_L} \frac{\partial P}{\partial y} \quad w = -\frac{k}{\mu f_L} \frac{\partial P}{\partial z} \quad (3)$$

$$k : \text{Permeability} = \frac{f_L^3 d^2}{180(1 - f_L)^2} \quad \mu : \text{Coefficient of viscosity}$$

d : Dendrite cell size

(4) Gas volume conservation

$$[H_0] = (1 - f_L)[H_S] + f_L[H_L] + \alpha_H \frac{P_g f_v}{T} \quad (4)$$

$[H_0]$: Total gas volume, $[H_S]$: Gas volume in solid phase,

$[H_L]$: Gas volume in liquid phase, α_H : Coefficient,

P_g : Gas pressure within the porosity

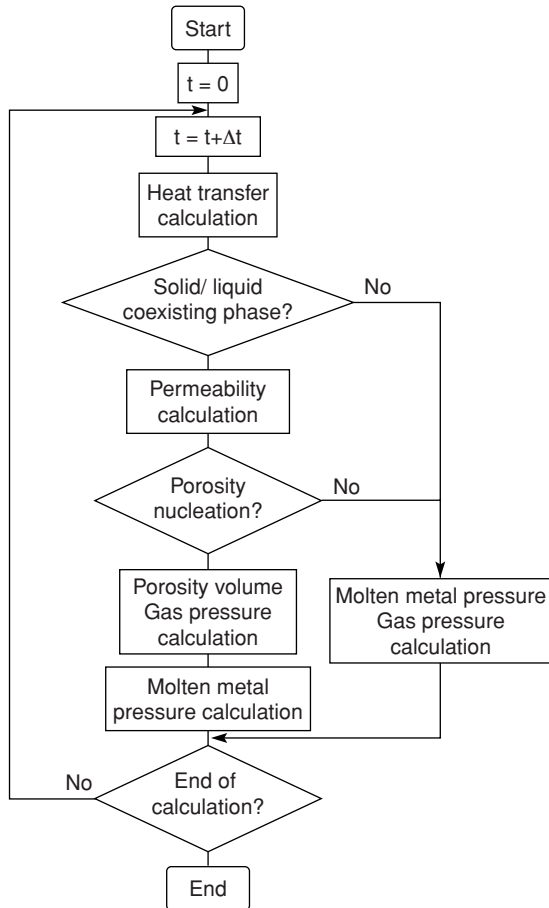


Fig. 3 Flow chart for predicting of defects

4. Comparison of Simulations and Actual Measurements

4.1. Transition Flow Filling Method Die-Cast

First, comparisons were performed for transition flow filling method die-casting which has a slower inlet velocity of molten metal than general high-pressure die-casting which also requires a weldability judgment. The flow and solidification simulation results were compared using the pivot plate shown in Fig. 4 which is a motorcycle frame part. The plunger velocity was 1.8 m/s, and overflows were set in five locations, of which three had reduced pressures. Test pieces were taken from general surfaces and used in experiments to measure various properties. In addition, the simulations used 137 285 mesh elements, and the physical property of the molten metal used the values for Al-5%Mg-0.7%Mn-0.8%Ni materials.

4.1.1. Air trap simulation

Figure 5 shows the simulation results for the amount of trapped air. (a) and (b) show different casting designs, the amount of trapped air measured was 3.6 and 2.0 cm³/100 g Al, respectively. With conventional analysis that does not take into account back pressure, these values were 1.7 and 2.4 cm³/100 g Al, respectively, also the accuracy deteriorated for both the trapped air trend and absolute values. In contrast, the new analysis method that takes into account back pressure produced the values of 3.1 and 1.9 cm³/100 g Al, respectively, indicating enhanced accuracy for both the trend and absolute values. This is thought to be because taking the back pressure into account brought the simulated molten metal filling pattern closer to the actual pattern. In addition, the casting was also found to be of a weldable level even at the last part the molten metal fills.

4.1.2. Amount and size of porosity

Figure 6 (a) and (b) show the simulation results for the amount and size of porosity, and Figs. 7 and 8 compare the simulated values and measured values. The horizontal axes show the simulated values and the vertical axes show the measured values. The measured values are slightly higher for both the amount of porosity and the porosity size, but the overall trends match well. The amount of porosity in the product was 0 to 0.02%, and the porosity size was 10 to 80 μm depending on the position. The amount and size of porosity both tended to increase in positions with large amounts of trapped air or a slow solidification rate.



Fig. 4 Pivot Plate

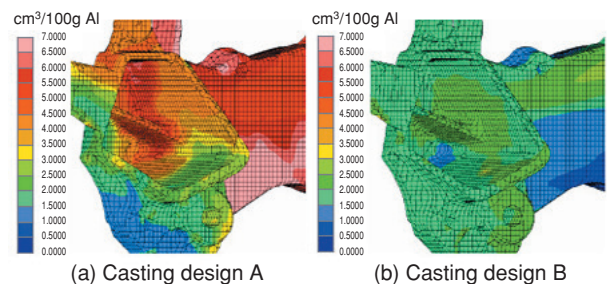


Fig. 5 Example of trapped air simulation with consideration to back pressure

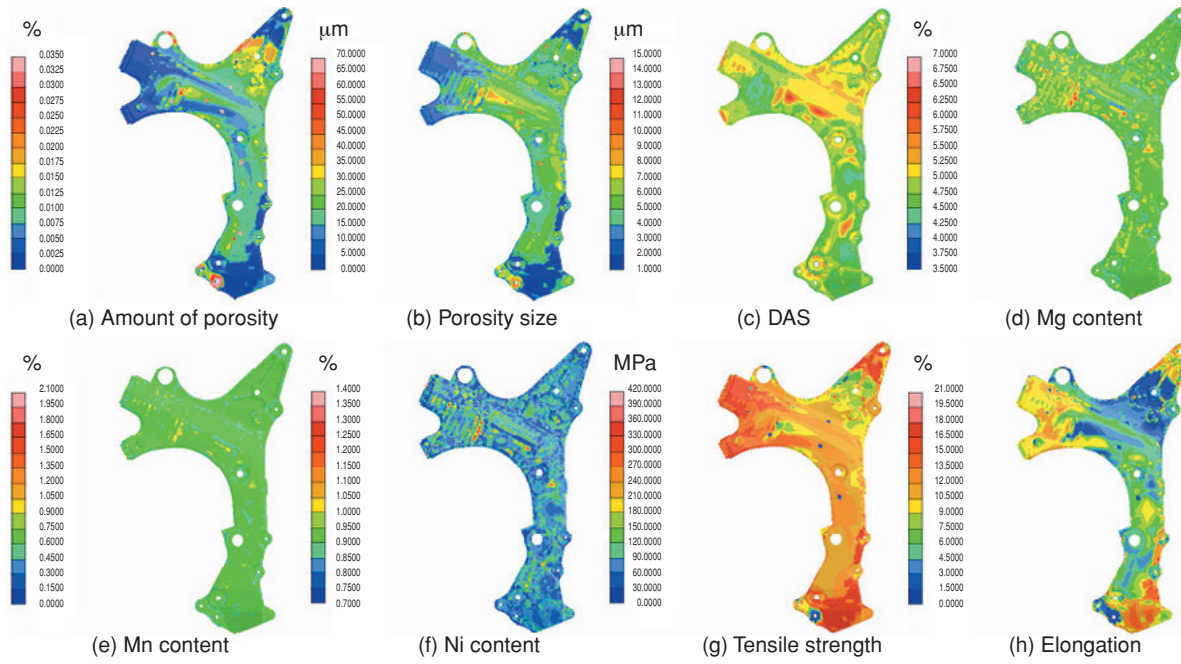


Fig. 6 Simulation results of pivot plate

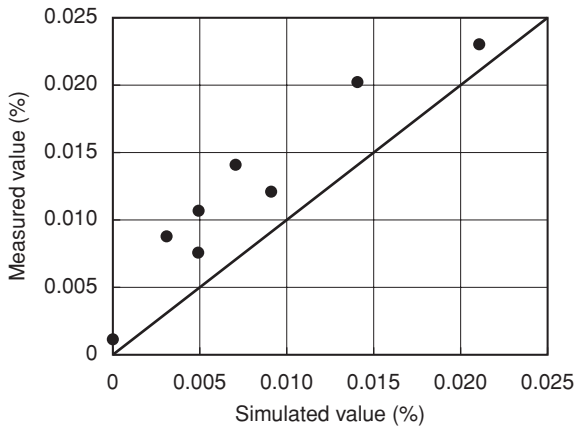


Fig. 7 Comparison of amount of porosity

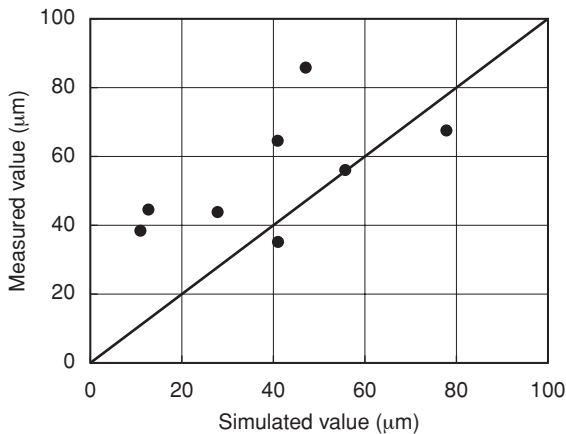


Fig. 8 Comparison of porosity size

4.1.3. Dendrite arm spacing (DAS)

Figure 6 (c) shows the DAS simulation results, and Fig. 9 compares the simulated values and measured values. This shows that the measured values and the simulated values match closely, with the DAS between 6.5 and 9 μm depending on the position. There was little difference in the DAS between positions, and the DAS was approximately the same as that of general high-pressure die-casting. This indicates that transition flow filling method die-casting provides the same cooling rate as that of general high-pressure die-casting.

4.1.4. Chemical composition

Figure 6 (d), (e) and (f) show the simulated values for each chemical composition, and Fig. 10 compares the simulated and measured chemical composition values. This shows that the measured values and simulated values match relatively well for each element. These distributions of Mg, Ni and Mn concentrations indicate that macro segregation exists throughout the entire casting. This type of macro segregation produces uneven mechanical, chemical and physical properties in die-cast products, and also often occurs in the same positions as micro-porosity⁽⁸⁾.

4.1.5. Tensile strength and elongation

Figure 6 (g) and (h) show the simulation results for tensile strength and elongation, and Figs. 11 and 12 compare the simulated values and measured values. This shows that both tensile strength and elongation match relatively well. The tensile strength and elongation were 195 to 285 MPa and 2 to 10.6% respectively. Both tensile strength and elongation tended to decrease locally in positions with high porosity. This is because an Al-Mg alloy was used. That is to say, in contrast to Al-Si alloys which have a skin-formed type solidification format, Al-Mg alloys have a mushy type solidification format. For this reason, primary crystal α -Al dendrites crystallize throughout the entire casting in the initial solidification stages. The gaps between these dendrites act as pressure transmission routes, resulting in an early loss of molten metal pressure. As a result, pressure is not sufficiently transmitted in positions removed from the gates, so the porosity size is larger and the tensile strength and elongation values are lower than in other

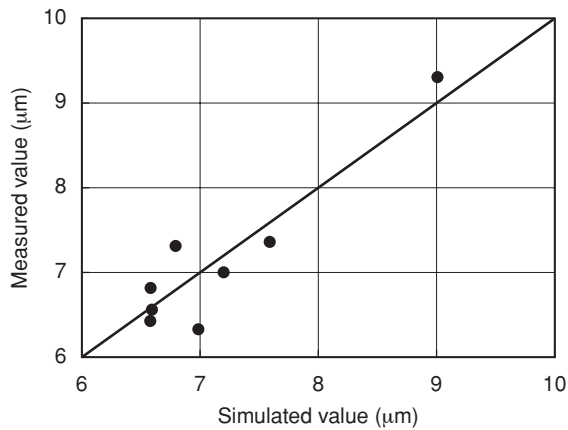


Fig. 9 Comparison of DAS

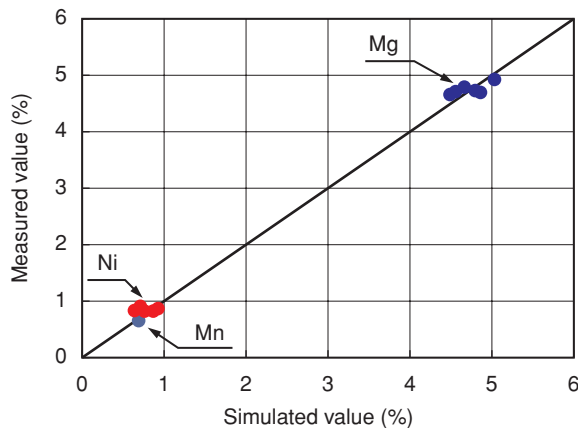


Fig. 10 Comparison of chemical composition

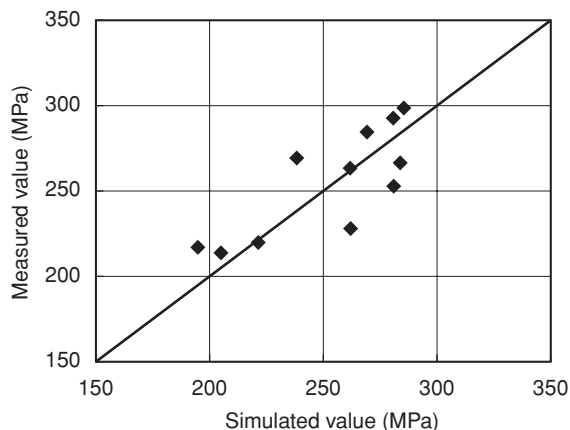


Fig. 11 Comparison of tensile strength

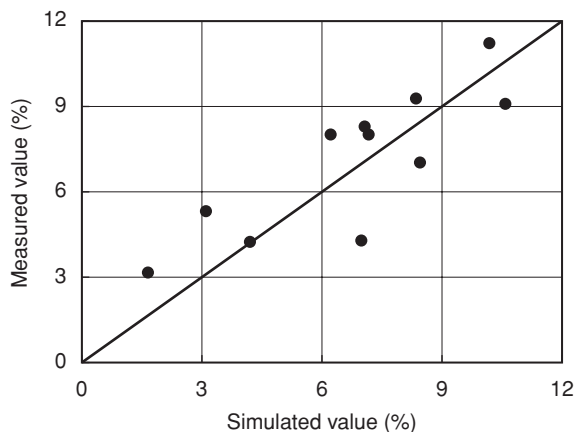


Fig. 12 Comparison of elongation

positions. These tendencies are more pronounced for Al-Mg alloys than for Al-Si alloys.

4.2. High-pressure Die-cast

Verification with transition flow filling method die-casting confirmed a correlation between the simulated and measured values, so next verification was conducted with high-pressure die-casting. The flow simulation results for high-pressure die-casting were compared using the clutch outer shown in Fig. 13. The plunger velocity was 2 m/s, and overflows were set in seven locations, of which five were connected to vents. Test pieces were taken from the fin parts and used in experiments to measure various properties. The simulations used 51 723 mesh elements, and the physical property of the molten metal used the values for JIS ADC12 equivalent materials.

4.2.1. Air trap simulation

Figure 14 (a) shows the simulation results for the amount of trapped air, and Fig. 15 compares the simulated and measured values. The horizontal axes show the simulated values and the vertical axes show the measured values. Analysis that does not take into account back pressure showed that when the air pressure inside the cavities increased, the air trapped in those positions was less than that of the actual volume. In addition, the filling pattern and the final filling position differed, so the trapped air volume also differed. However, the new analysis method that takes into account back pressure produced values that matched well in each position. The amount of trapped air inside the product was approximately 2 to 10 cm³/100 g Al. The difference in trapped air volume between positions is mainly due to the flow length, with more air becoming trapped as the flow length increases.

4.2.2. Amount and size of porosity

Figure 14 (b) and (c) show the simulation results for the amount and size of porosity, and Figs. 16 and 17 compare the simulated and measured values. The newly developed simulation method coupled the amount and size of trapped air, etc. at the completion of the flow state with the solidification analysis, so both the amount and size of porosity matched relatively well with those of the measured values. The amount of porosity in the product was 0 to 0.047%, and the porosity size was 20 to 330 μm depending on the position. The amount and size of porosity both tended to increase in positions with large amounts of trapped air or a slow solidification rate.

4.2.3. Dendrite arm spacing (DAS)

Figure 14 (d) shows the DAS simulation results, and Fig. 18 compares the simulated and measured values. In contrast to the results for transition flow filling method die-casting using an Al-Mg alloy, the measured values were larger in these results. This is due to the difference in the alloys rather than the casting methods. Al-Mg and other metal-metal element alloys are generally in

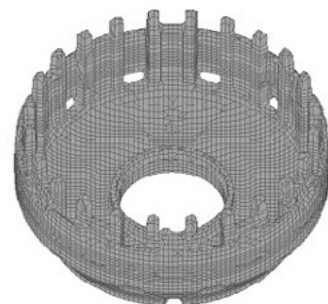


Fig. 13 Clutch outer

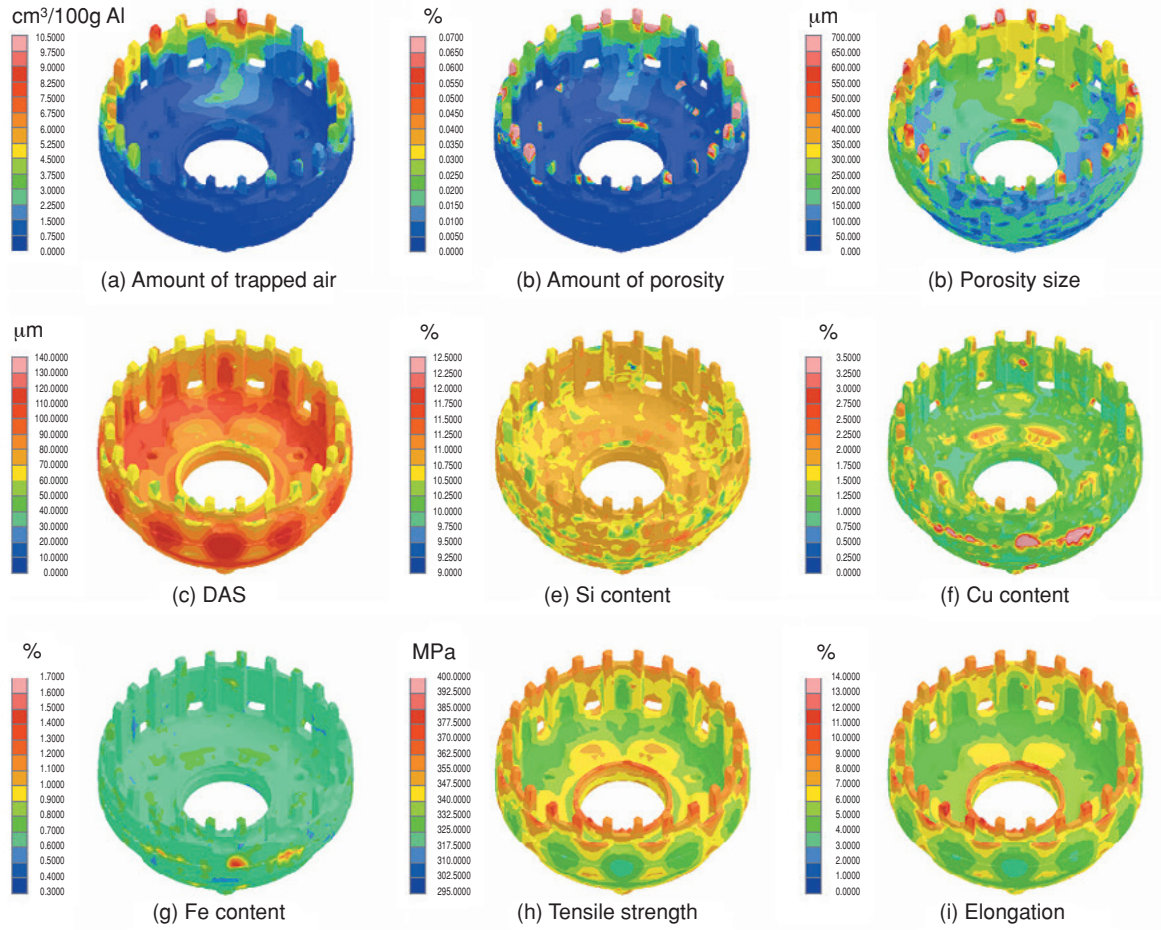


Fig. 14 Simulation results of clutch outer

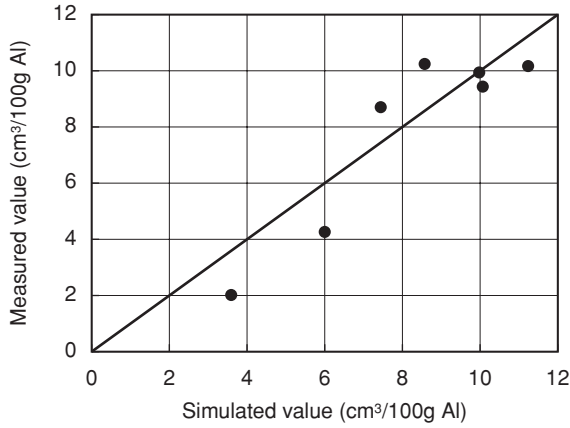


Fig. 15 Comparison of trapped air

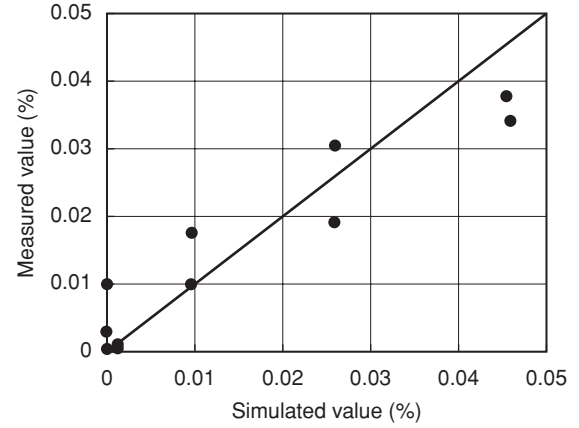


Fig. 16 Comparison of amount of porosity

equilibrium at the interface of solid phase and liquid phase, which presents no issue. However, Al-Si and other metal-metalloid element alloys undergo facet-nonfacet eutectic solidification, so a relatively large supercooling occurs at the interface of solid phase and liquid phase. This supercooling significantly affects the solidification pattern and the micro structure. The analysis program used in this study performed equilibrium solidification analysis, but it is thought that the prediction accuracy can be enhanced by performing nonequilibrium solidification analysis.

4.2.4. Chemical composition

Figure 14 (e), (f) and (g) show the simulated values for each chemical composition, and Fig. 19 compares the simulated and measured chemical composition values. The measured and simulated values match well, and these distributions of Si, Cu and Fe concentrations indicate that macro segregation exists throughout the entire casting. The ease by which macro segregation occurs is thought to be more due to permeability than the solidification rate.

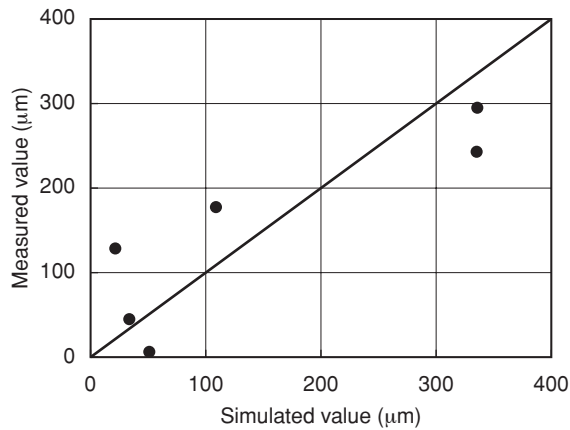


Fig. 17 Comparison of porosity size

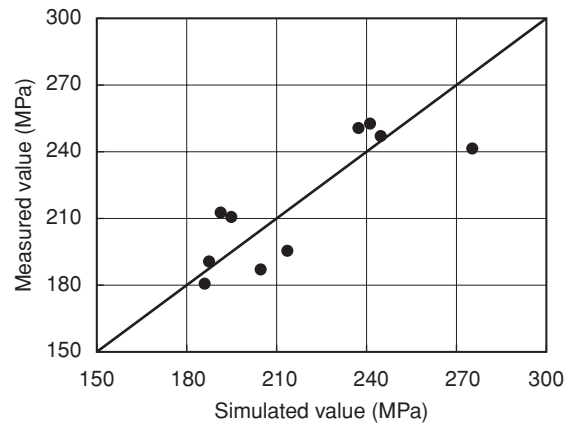


Fig. 20 Comparison of tensile strength

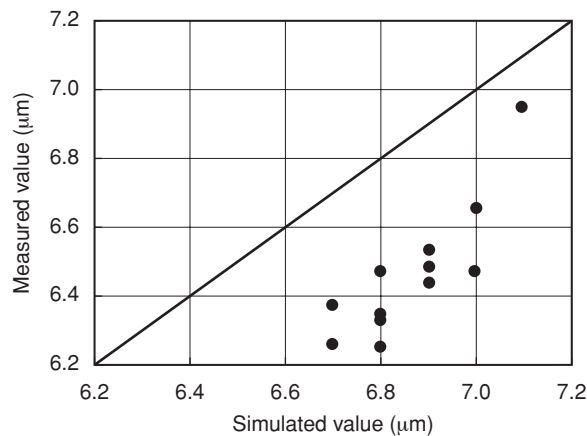


Fig. 18 Comparison of DAS

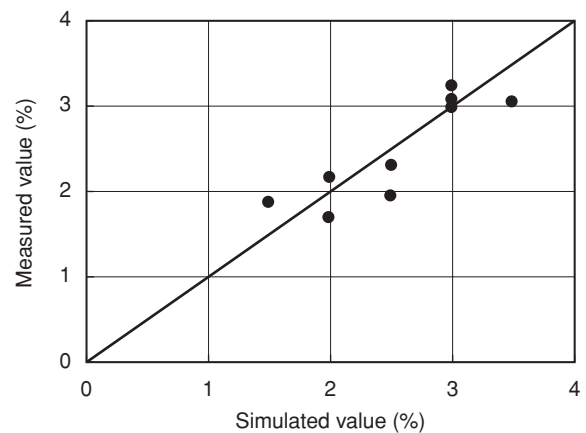


Fig. 21 Comparison of elongation

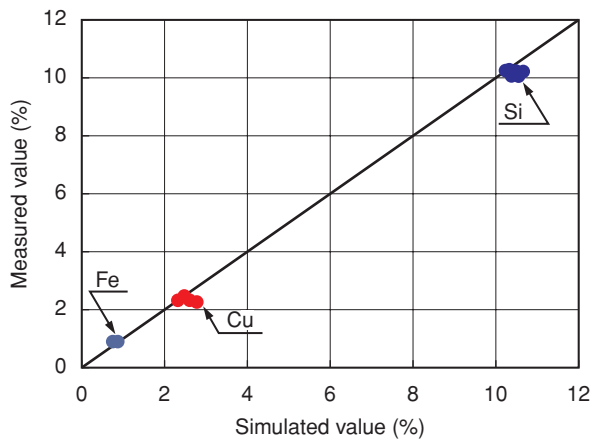


Fig. 19 Comparison of chemical composition

4.2.5. Tensile strength and elongation

Figure 14 (h) and (i) show the simulation results for tensile strength and elongation, and Figs. 20 and 21 compare the simulated and measured values. Both the tensile strength and elongation values match relatively well despite a low DAS correlation as described in 4.2.3. This is thought to be because there is almost no change in the mechanical properties due to the DAS in rapidly cooled structures 10 μm or smaller such as high-pressure die-casting.

5. Conclusion

A system for coupling the air trapped during the molten metal flow with the solidification process in casting simulations was developed, and the following results were obtained.

- (1) Analysis that coupled the process from flow analysis to solidification analysis was possible.
- (2) Taking into account back pressure in flow analysis enhanced the quantitative prediction accuracy of the amount of trapped air inside products.
- (3) Quantitative analysis of the amount and size of porosity in solidification analysis was possible.
- (4) Taking into account the amount and size of porosity in the DAS and segregation analysis results also made it possible to analyze the tensile strength and elongation.

Various casting simulation methods are currently being investigated, and the system developed in this study is thought to indicate a new direction for casting simulations. Further enhancements in accuracy and expanded applications are anticipated in the future.

6. Acknowledgements

The authors would like to express their deep thanks to engineering staff of EKKJAPAN Inc. for their enormous cooperation in the development of this new system.

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Exploratory Study of an Airbag Concept for a Large Touring Motorcycle: Feasibility Verification

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ABSTRACT

Based on the injury assessment method established by ISO/CD 13232 and including neck injury assessment, which had not been established previously, the prototype airbag system was evaluated using full scale impact tests in seven configurations and computer simulation of 200 impact configurations, which calculate the process of dummies contacting the ground.

The GL1800 was used for this study; previous studies used the predecessor large touring motorcycle (GL1500). Furthermore, in addition to the assessment specified in ISO/CD 13232, tests using other configurations and other rider conditions were conducted. It was judged from the results obtained that an airbag system for a large touring motorcycle, like the GL1800, is feasible.

1. Introduction

Honda has conducted research aimed at enhancing the protection of motorcycle riders since the 1960s⁽¹⁾. Recently, research has been directed towards the possibility of fitting airbags on motorcycles, as discussed below.

A prototype airbag system, which included a sensor system, was mounted on a large touring motorcycle, the GL1500, and its rider protection performance was evaluated based on ISO13232⁽²⁾. The results of a series of full scale impact tests and computer simulations showed, that while the system could lead to the possibility of an increase in injuries in some impact configurations, it could help reduce rider injuries overall⁽³⁾.

Concern over a possible increase in injuries was generated by analysis of measurements of neck injuries due to the airbag causing the test dummy to make contact with the ground head-first. This issue was addressed by increasing the size of the airbag⁽⁴⁾.

The majority of the differences in injury type, recorded in tests with and without an airbag, were generated when the dummy made contact with the ground. Therefore, Honda developed a computer

simulation technique covering the aspect of ground impact. A computer simulation was developed with a high degree of accuracy that is able to reproduce the motion of the dummy up to the time of its contact with the ground⁽⁵⁾. Airbag research was furthered by fitting a prototype airbag system on a large scooter-type motorcycle with a different frame design to that of the large touring motorcycle. Full scale impact tests were conducted using part of ISO 13232, and the results indicated that an airbag employing the same design concepts as the airbag used on the large touring motorcycle was also potentially feasible for the large scooter-type motorcycle⁽⁶⁾.

While these research projects on the feasibility of an airbag system for motorcycles were being conducted, a standard for evaluation of neck injuries, which had previously been unavailable, was established in ISO/CD 13232. In addition, Honda had developed a computer simulation technique enabling the range for the assessment of injuries to be extended to the point at which the dummy makes contact with the ground. These advances have made it possible for researchers to evaluate rider protection devices more comprehensively.

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This paper will discuss the results of a series of evaluations of the airbag system for large touring motorcycles, like the GL1800, based on these newly established methods.

2. Concept of Airbags for Large Motorcycles

Figure 1 shows the fatal motorcycle accidents data in Japan⁽⁷⁾. These results show that:

- The majority of fatal injuries (64%) occur in frontal impacts.
- The majority of rider injuries (87%) are caused by impact with the object the motorcycle collided with or the ground rather than with the motorcycle itself.
- The majority of fatal injuries (87%) occur to the upper half of the body.

From this statistical data, the typical fatal injury scenario can be defined as: One in which the rider separates from the motorcycle as a result of a frontal impact, strikes the opposing vehicle or the ground with his or her upper body, and sustains a fatal injury. On this basis, Honda established the following concept for its motorcycle airbag: To reduce the speed of separation of the rider from the motorcycle in the forward direction during a frontal impact by absorbing the kinetic energy of the rider, to help reduce the level of injuries sustained upon impact with the opposing vehicle or the ground. In addition, the following factors were considered based on previous research:

- Support of the airbag (upon which the rider exerts a horizontal load) by the motorcycle frame (i.e. in a frontal impact, with a passenger vehicle with a hood, the airbag should function without being supported by the hood of the opposing vehicle).
- The effectiveness of the airbag should be maintained as much as possible during a pitch, yaw and roll, typical of motorcycle motion in a frontal impact.

3. Test Motorcycles Fitted with Prototype Airbag System

Figure 2 shows a test motorcycle fitted with a prototype airbag system. The motorcycle is the GL1800. The main specifications of the motorcycle are shown in Table 1.

Table 2 shows the specifications of the airbag and the inflator. The airbag and retainer box were redesigned for use on the GL1800. The design of the prototype airbag reflects the results of previous research in a variety of areas: the capacity is larger, the design of the overall shape is superior, the shape of the rear of the airbag has been altered and a support belt has been used to connect the rear of the bag to the frame of the motorcycle. The time required for

deployment of the airbag is approximately 45 msec. A sensor system judges whether to deploy the airbag using the logic shown in Fig. 3 on the basis of the deceleration degree in the vicinity of the front wheel axis (hereinafter referred to as deceleration of the front wheel system).

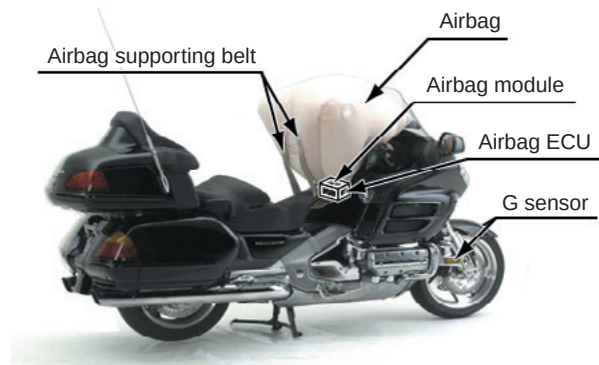


Fig. 2 Prototype airbag system installed on the GL 1800 motorcycle

Table 1 Specifications of test motorcycle

Manufacturer	Honda
Model	GL1800
Year	2000-2003
Mass (average) (empty, as tested no airbag) (kg)	367
Mass (average) (empty, as tested with airbag) (kg)	375
Overall length (mm)	2 635
Overall width (mm)	945
Overall height (mm)	1 455
Wheelbase (mm)	1 690
Type	Touring

Table 2 Specifications of prototype airbag

Volume (Net, inflated) (Liters)	157
Height × Width × Length (mm)	750 × 720 × 640
Diameter of vent holes (mm)	45 (× 2 holes)
Rear shape	Foams a concave "V"
Airbag mounting	To module box To motorcycle frame under the seat, via two connecting belts
Inflator type	Pyrotechnic; Passenger car type
Inflating pressure (kPa)	495

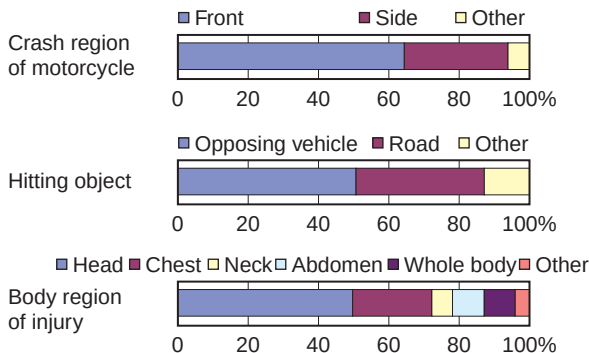


Fig. 1 Fatalities of motorcycle accidents in Japan

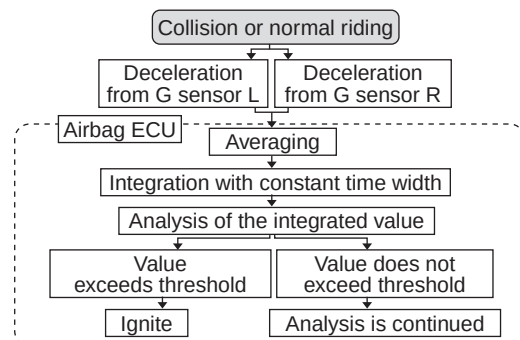


Fig. 3 Decision flow for airbag deployment

4. Sensor System Settings

4.1. Study of Impact Conditions for Deployment of Airbag

Full scale impact tests were conducted using the base motorcycle to study the conditions of frontal impacts resulting in the separation of the rider from the motorcycle in a forward direction. Each test was studied to enable deployment of the airbag during these crash events. The 1998-2001 Honda Accord 4-door with Japanese specifications, as shown in Table 3, was used as the opposing vehicle during impact testing. The rider dummy was a Motorcyclist Anthropometric Test Device (MATD) specified in ISO/CD 13232. Table 4 shows impact conditions and test results. Separation of the dummy from the motorcycle in a forward direction occurred at impact speeds of 35 km/h, but not at 25 and 30 km/h. From these results, it was judged that the airbag should deploy at 35 km/h and above, in front and side impact configurations. The degree of deceleration in the sensor unit was measured for use as base data, for developing the judgment logic and setting the threshold values as discussed below.

4.2. Study of Deceleration of Sensor Unit in Conditions for Non-deployment of Airbag

Other than in the event of an accident, the airbag should not deploy under normal running and handling conditions, including rough roads. To simulate conditions in which the airbag should not deploy, tests were conducted running the motorcycle over steps, on roads with depressions, and on rough roads. The front wheel was also dropped from a "wheelie" position. In addition, the impact on the front wheel when the motorcycle is dropped from the platform of a truck during cargo handling operations was simulated, and the front forks were subjected to impact from a hammer. The degree of deceleration in the sensor unit under these running and handling conditions was measured and used as base data in the development of the judgment logic, and in the setting of the threshold values.

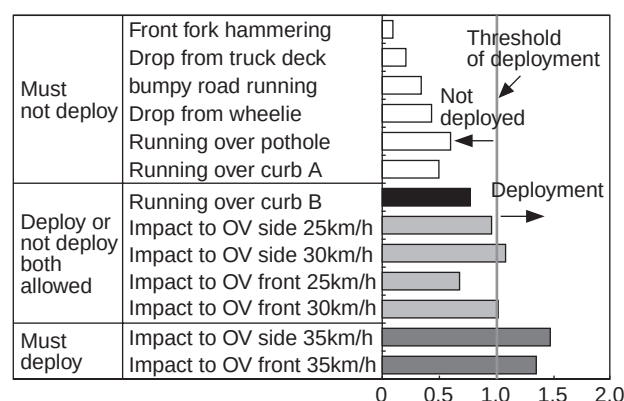


Fig. 4 Decision of deployment by integrated impact deceleration

4.3. Development of Collision Judgment Logic and Setting of Threshold Values

The collision judgment logic (shown in Fig. 3) and the threshold values were decided based from the study of data on deceleration in the sensor unit during impact conditions in which the airbag should deploy, and in running and handling conditions in which it should not deploy. Figure 4 shows the performance of the logic in determining deployment and non-deployment. This figure shows the ratio of threshold values - predetermined values for the judgment to deploy - to the maximum computed values, in each case of deployment or non-deployment. As the figure shows, the resulting system will not deploy as a result of impacts during normal running or handling, but will deploy in collisions requiring the operation of an airbag.

5. Full Scale Impact Tests Based on ISO/CD 13232

Table 3 Specifications of opposing vehicle

Manufacturer	Honda
Model	Accord, Japan
Year	1998-201
Mass (average) (kg)	1 300
Overall length (mm)	4 635
Overall width (mm)	1 695
Overall height (mm)	1 420
Wheelbase (mm)	2 665
Engine displacement (cm ³)	1 997

Table 4 Tests and results of airbag deployment impact conditions

Collision conditions	Dummy motion	Deploy or not
To Accord front	25 km/h	No separation from MC
	30 km/h	Just before separation from MC
	35 km/h	Separated from MC
To Accord side	25 km/h	No separation from MC
	30 km/h	Just before separation from MC
	35 km/h	Separated from MC

5.1. Test Method

The airbag was evaluated on the basis of ISO/CD 13232. Comparative evaluations of motorcycles with and without an airbag were conducted using the seven impact configurations shown in Fig. 5.

A 1998-2001 Honda Accord 4-door with Japanese specifications conforming to the stipulations of ISO/CD 13232 was employed as the opposing vehicle in the impact tests (see Table 3).

To enable evaluation under fairly severe conditions, the impact speed in the tests was increased by 4%, shown in parentheses in the figure, against the speed stipulated in ISO/CD 13232.

A MATD was used as the dummy, and injury level to the head, neck, chest, abdomen and leg regions were evaluated using the injury analysis method specified in ISO/CD 13232.

5.2. Test Results

5.2.1. System deployment / non-deployment, judgment timing

The results of each impact test using motorcycles fitted with the airbag system were collected and analyzed. The airbag deployed in five of the seven impact configurations and did not deploy in two.

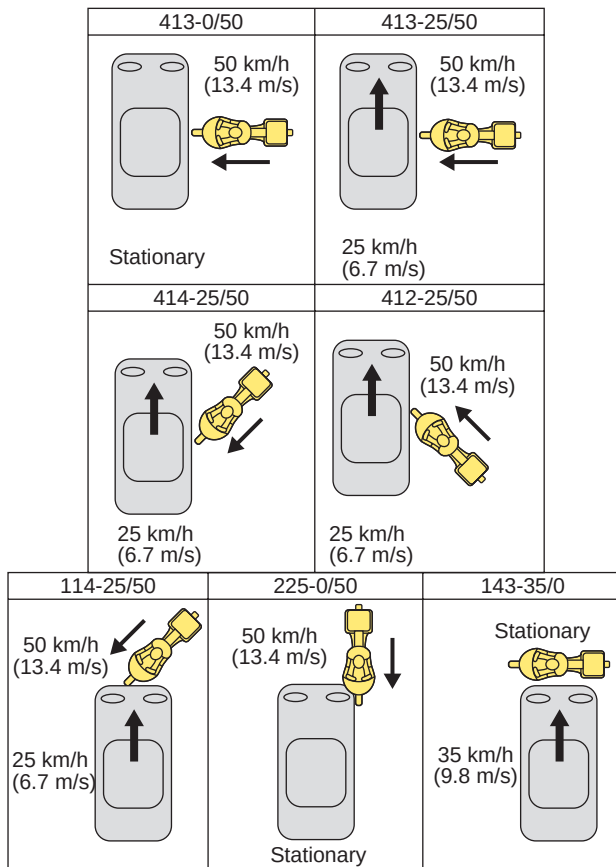


Fig. 5 Configurations of full-scale impact tests

In four impact configurations in which the airbag deployed (413-0/50, 413-25/50, 414-25/50 and 114-25/50), deployment was completed before the dummy reached the airbag. For these four impact configurations, Fig. 6 shows the time to fire, the airbag deployment time, and the time it takes the dummy to reach the airbag. To provide an example of the mode of deployment of the airbag and its effectiveness, Fig. 7 shows the situation at the start of impact ($t=0$ msec), the start of deployment of the airbag ($t=20$ msec), the completion of deployment ($t=60$ msec) and almost complete absorption of the kinetic energy of the dummy ($t=120$ msec) for configuration 413-25/50.

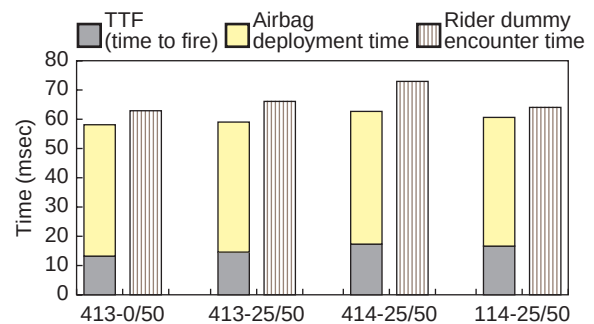


Fig. 6 Decision time of airbag deployment from full-scale impact tests

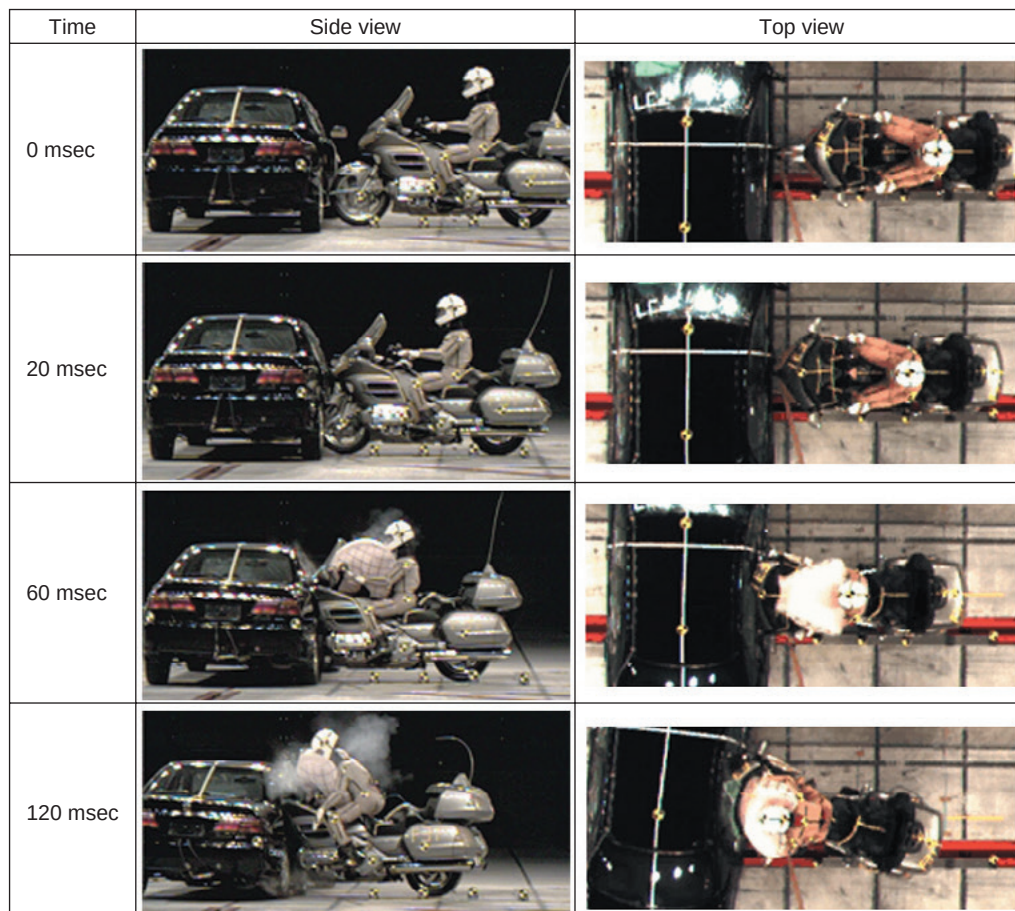


Fig. 7 Example of airbag deployment and effectiveness

In the case of configuration 225-0/50, the dummy contacted the airbag before deployment was complete, but no injury indices resulting from interference with the airbag were indicated (see the section “Results of injury analysis” below). In configurations 412-25/50 and 143-35/0, the sensor system judged that deployment was unnecessary, and therefore the airbag did not deploy. In the case of 412-25/50, although the motorcycle collided with the opposing vehicle, it did not decelerate significantly, and it was diverted along the side of the vehicle. This was not to be considered as the case of the airbag deploy because no AIS index measured (HIC 373) when the dummy head contacted the opposing vehicle. 143-35/0 is not a frontal collision, which is the objective of this airbag system, therefore, the airbag will not deploy.

5.2.2. Results of injury analysis

Changes in injury indices in the seven impact configurations, with and without an airbag, were analyzed on the basis of ISO/CD 13232. The results are shown in Table 5 in terms of changes in the Abbreviated Injury Scale (AIS) by body region, and in the Normalized Injury Cost (NIC) for the whole body. AIS=1 corresponds to minor injuries and AIS=6 to fatal injuries. NIC=0 corresponds to no injury, and NIC=1 corresponds to fatal injuries. A positive index indicates a benefit, and a negative index indicates a risk. In 412-25/50 and 143-35/0, the two configurations in which the airbag did not deploy, the airbag naturally had no effect on rider injuries.

6. Evaluation of Effectiveness in Reducing Rider Injuries by Computer Simulation

6.1. Computer Simulation

Computer simulations were employed to evaluate the effectiveness of the airbag in reducing rider injuries on the basis of ISO/CD 13232. However, ISO/CD 13232 specifies a range for evaluation up to 0.5 seconds after the motorcycle collides with the opposing vehicle (hereinafter referred to as primary impact sequence). The computer simulations employed in this research enables evaluation not only primary impact sequence but also after primary impact sequence up to the point at which the dummy makes contact with the ground (hereinafter referred to as secondary impact sequence). The degree of correlation of the computer simulation with each model and the results of impact tests was assessed on the basis of ISO/CD13232. Figures 8 to 10 show comparison data between the impact test and the computer simulation. Figure 8 shows the correlation of head HIC for primary impact sequence. Figure 9 shows the correlation of head HIC for secondary impact sequence. Figure 10 shows the correlation of chest compression for primary impact. Table 6 shows the correlation of the femurs, knees, and tibias fracture for entire impact sequence. As a result, the simulation was judged to be capable of evaluating the benefits

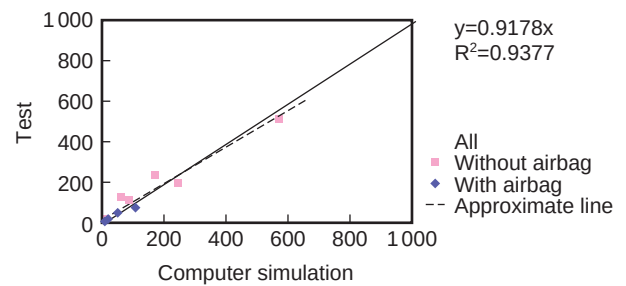


Fig. 8 Correlation of head HIC for primary impact sequence

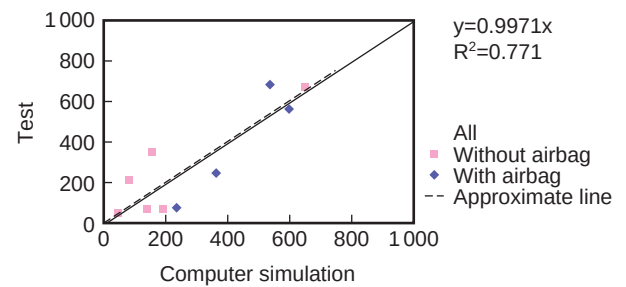


Fig. 9 Correlation of head HIC for secondary impact sequence

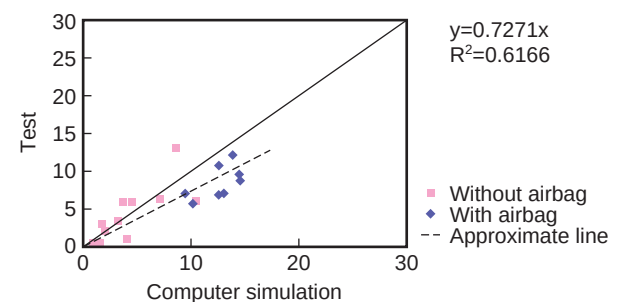


Fig. 10 Correlation of chest compression (Cmax%) for primary impact

and the risks of the airbag, in terms of rider injuries for the 200 impact configurations specified in ISO/CD 13232.

6.2. Results of Computer Simulation

In 79 of the 200 impact configurations, it was judged that the airbag would not deploy; hence, computer simulations were not conducted for these configurations. Thus, simulations were conducted with and without an airbag for 121 of the configurations, making a total of 242 cases.

The results are shown in Fig. 11. Total average benefit is 0.048 and total average risk is 0.004. The risk/benefit ratio is 0.083 and average net benefit is 0.044. These results indicate that the airbag system displays a satisfactory level of performance in reducing rider injuries.

Table 5 Airbag injury benefits and risks by impact configuration and body region

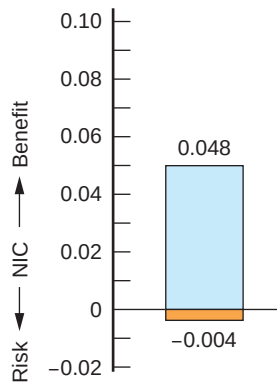
Collision configuration	AIS change					NIC change
	Head	Neck	Chest	Abdomen	Leg	
413 - 0/50	0	0	0	0	0	0
413 - 25/50	0	0	0	0	0	0
414 - 25/50	-1	0	0	0	0	-0.07
412 - 25/50	-	-	-	-	-	-
114 - 25/50	-1	0	0	0	3	0.11
225 - 0/50	0	0	0	0	3	0.16
143 - 35/0	-	-	-	-	-	-

Table 6 Correlation of leg fracture for entire impact sequence

Femurs		Full scale tests		Corrected percentage
		Fracture	No fracture	
Simulations	Fracture	0	1	96%
	No fracture	0	23	

Knees		Full scale tests		Corrected percentage
		Fracture	No fracture	
Simulations	Fracture	0	0	100%
	No fracture	0	24	

Tibias		Full scale tests		Corrected percentage
		Fracture	No fracture	
Simulations	Fracture	1	0	100%
	No fracture	0	23	

**Fig. 11** Total average benefits and risks; 200 impact configurations

7. Results of Tests under Various Conditions

In addition to the full scale impact tests in the seven ISO/CD 13232 configurations used to evaluate benefits and risks in typical collisions, tests employing other impact configurations and rider conditions were conducted to further evaluate the airbag's effectiveness.

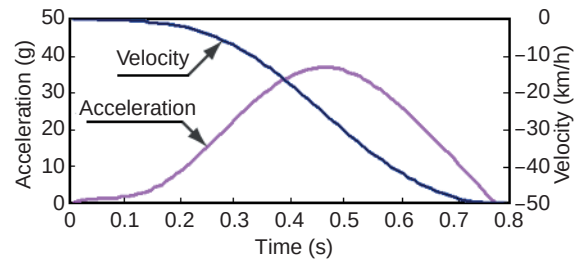
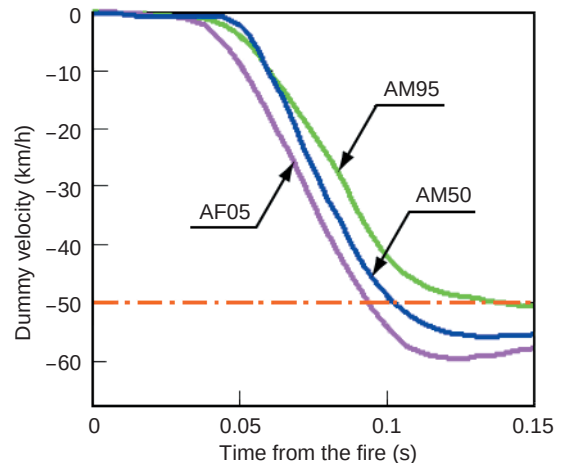
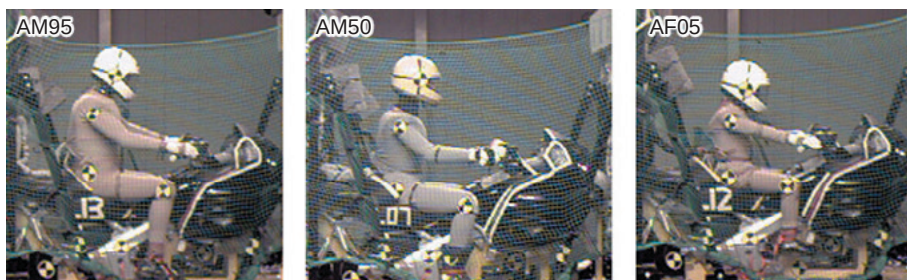
7.1. Influence of Rider Size

Sled tests were conducted with dummies of three different sizes in order to investigate the effectiveness of the system with riders of different sizes. A standard-sized dummy, AM50 percentile (H-III), a smaller-sized dummy, AF05 percentile (H-III) and a

larger-sized dummy, AM95 percentile (H-III), were employed. Injury indices for the head, neck and chest were measured and evaluated on the basis of the stipulations of FMVSS208 and related data^{(8), (9)}. The neck of the AM50 dummy was replaced with an MATD neck for measurement and evaluation based on ISO/CD 13232.

The acceleration waveform of the sled in the tests was based on the deceleration produced waveform, when a motorcycle traveling at 50 km/h collides with a stationary four-wheeled vehicle. The acceleration waveform of the sled is shown in Fig. 12. Figure 13 shows the motorcycle and the dummy when set on the sled.

The changes in the velocity of the upper half of the dummies are shown in Fig. 14. None of the three dummies were thrown forward, off the motorcycle. Figure 15 shows the absorption of the kinetic energy of the dummies by the airbag, and Table 7 shows the results of dummy injury measurements. In these tests, no injury indices in the head, neck or chest exceeding the injury criteria were recorded in any of the dummies.

**Fig. 12** Acceleration and velocity change of motorcycle in sled tests**Fig. 14** Velocity change of dummy's upper body**Fig. 13** Initial dummy posture in sled tests

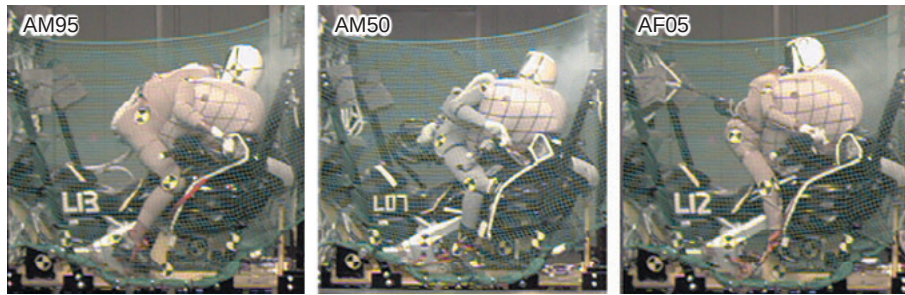


Fig. 15 Dummy posture at the moment when airbag absorbs kinetic energy

Table 7 Dummy injury indices in sled test results

Dummy size	Head HIC	Neck Nij (or NII)	Chest D max.
AM95	52 (not over 700)	0.56 (not over 1)	32 (not over 70mm)
AM50	134 (not over 700)	2.0 (not over 4)	26 (not over 63mm)
AF05	134 (not over 700)	0.73 (not over 1)	17 (not over 52mm)

Criteria are shown in parentheses.

7.2. Effect of Airbag Deployment When Rider Is Leaning forward

We also considered the fact that motorcycle riders change their posture while riding. In particular, we want to try to mitigate any injury that could occur due to the deployment of the airbag, including when a rider is leaning forward into the airbag's deployment zone. A static deployment test confirmed that the airbag would not cause a serious injury in this case. The tests assumed that the rider was leaning forward. A standard-sized AM50 percentile (H-III) dummy and a smaller-sized AF05 (H-III) dummy were used in the tests. Injuries to the head, neck and chest were measured and evaluated on the basis of the stipulations of FMVSS208. The neck of the AM50 percentile dummy was replaced with an MATD neck for measurement and evaluation based on ISO/CD 13232.

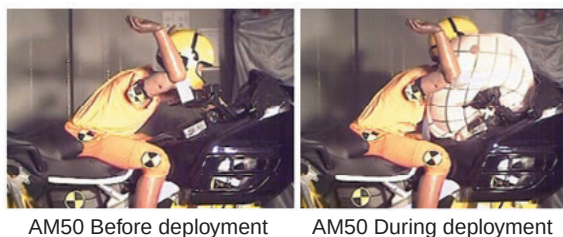


Fig. 16 Static inflation test with forward leaning AM50-dummy

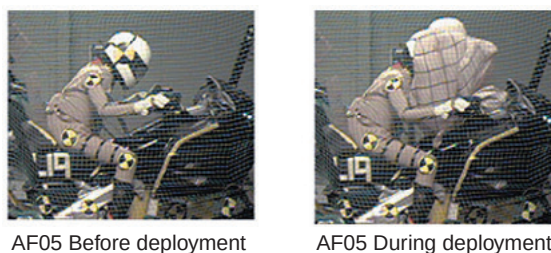


Fig. 17 Static inflation test with forward leaning AF05-dummy

The situation of the dummies before and during deployment of the airbag is shown in Figs. 16 (AM50 percentile) and 17 (AF05 percentile). In both cases, the airbag initially deployed in the region of the chest and neck, after which it contacted the head and moved the neck backwards. The results of the dummy injury measurements are shown in Table 8. In these tests, no injury indices exceeding the injury criteria were measured in either the AF05 or AM50 dummy.

7.3. Effect of Airbag on Rider in High-speed Collision with Passenger on Motorcycle

A collision test was created with the combined conditions of two riders and a higher impact speed. The focus of the test was an analysis of the results of a situation in which a passenger exerted pressure on the rider from behind, while the rider is contacting the airbag.

In the test, a Honda Accord impacted a stationary motorcycle with an airbag at 75 km/h. The opposing vehicle colliding with a stationary motorcycle is used to suppress the pitching of motorcycle body, and therefore is considered more appropriate for the purpose of the test. An MATD (AM50 percentile) was used as the rider dummy, and injuries were evaluated on the basis of ISO/CD13232. An AM95 percentile (H-III) dummy was employed as the passenger dummy to generate a greater impact with the rider dummy.

As Fig. 18 shows, during the impact test, the rider dummy was first in between the airbag and the passenger dummy. After this, the passenger dummy moved forward off the motorcycle, while the kinetic energy of the rider dummy was being absorbed by the airbag and the rider dummy remained on the motorcycle. The airbag was not damaged during this test. The results of injury measurements on the rider dummy are shown in Table 9. Injuries to the upper body were focused on, because it is in this area that the airbag will have an effect. Other than a measurement of AIS 1 in the chest, no injury index that can be represented on the AIS was measured in the region of the head, neck and chest in this test.

7.4. Under-ride Impact

In certain impact configurations, the front wheel of the motorcycle is not the first point of contact with the opposing vehicle or object. One example of this type of collision (termed "under-ride impact," below) is a collision with the platform of a truck.

Table 8 Measured dummy injury in static inflation test with forward leaning dummy

Dummy size	Head HIC	Neck Nij (or NII)	Chest D max.
AM50	168 (not over 700)	1.55 (not over 4)	5.5 (not over 63 mm)
AF05	37 (not over 700)	0.53 (not over 1)	0.4 (not over 52 mm)

Criteria are shown in parentheses.

Under-ride impact tests were conducted to determine whether the airbag system, which judges whether to deploy, and the timing of deployment on the basis of deceleration of the front wheel system, would operate and deploy in this case.

A motorcycle with an airbag was collided with an opposing trolley with a brake applied, as shown in Fig. 19, at a speed of 50 km/h. The trolley was rigged to simulate the platform of a truck. The main specifications of the trolley are shown in Table 10. An MATD was employed as the dummy in the tests, and injury indices were measured and evaluated on the basis of ISO/CD 13232.

In the test, the timing of collision judgment by sensor system was 30 msec after the initial impact. The dummy made contact with the airbag as it was inflating, and did not collide directly with the under-ride rig. The airbag absorbed the dummy's kinetic energy. Figure 20 shows the dummy and the airbag during the test, and Table 11 shows the results of injury measurements. In this test, no injury indices were measured which could be represented on the AIS.



Fig. 20 Under-ride impact test

Table 11 Measured injury in under-ride impact test

Rider dummy	Injury by body region (AIS)			
	Head	Neck	Chest	Abdomen
MATD	0	0	0	0

8. Conclusion

A GL1800 motorcycle fitted with a prototype airbag system was evaluated in full scale impact tests and computer simulations based on ISO/CD13232. Furthermore, by other tests and conditions set independently by Honda.

The following findings were obtained:

- (1) The sensor system, using a judgment logic based on detection and calculation of deceleration in the front wheel system, was capable of appropriately judging the deployment/non-deployment of the airbag in collisions and other impacts.
- (2) As the results of the evaluation based on ISO/CD 13232, the airbag system has shown an overall benefit.
- (3) 75 km/h impact tests with two riders, tests with large and small riders, tests with the rider leaning forward, as well as under-ride impact tests, were carried out to study the effectiveness of an airbag under a variety of conditions, and no unacceptable results were generated.

Given these results, an airbag system for large touring motorcycles, like the GL1800, can be judged to be feasible. Future research will consider durability, weather resistance, reliability, marketability and productivity, looking towards the practical use of the system.

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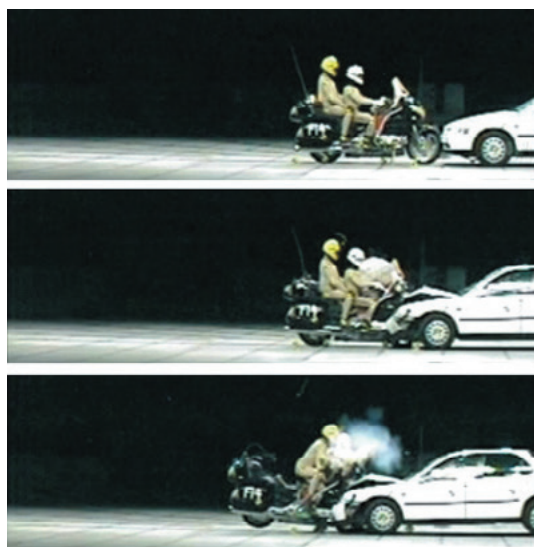


Fig. 18 75 km/h impact test with a passenger

Table 9 Measured injury of rider dummy in 75 km/h full-scale impact test

Rider dummy	Injury by body region (AIS)			
	Head	Neck	Chest	Abdomen
MATD	0	0	1	0

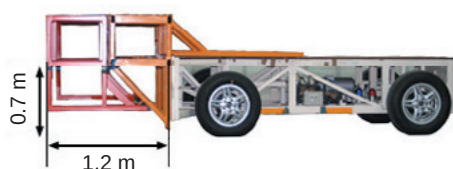


Fig. 19 Opposing trolley used in under-ride impact test

Table 10 Specifications of opposing trolley

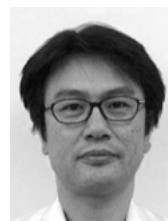
Mass	1750 kg
Overall length	4250 mm
Overall width	2080 mm
Overall height	1300 mm

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A Computer Simulation for Motorcycle Rider Injury Evaluation in Collision

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ABSTRACT

Honda has been developing computer simulation technology, capable to predict the injury levels, in tests from the start of a collision until the point where the dummy strikes the ground. This research shows the correlation between the computer simulations and full scale impact tests as specified in ISO/CD 13232. The results validate that computer simulation can predict the injury levels, from the start of collision until the point where the dummy strikes the ground. The performance and effectiveness of the airbag system for the GL1800 were evaluated in 200 impact configurations using the computer simulation specified in ISO/CD 13232. The results showed that the total average benefit was 0.048 and the total average risk was 0.004. The relative impact speed which demonstrated the highest effectiveness was 20-25 m/s.

1. Introduction

Honda has been conducting research since the 1960s⁽¹⁾ on enhancing motorcycle rider protection at the time of collision. In recent years, research has been proceeding on the feasibility of a motorcycle airbag as a way of reducing injury levels. In this research, Honda has test-produced an airbag system with impact detection sensors and mounted it on the GL1500, a large touring model. This motorcycle has been used in collision tests. Results of these tests were reported at the ESV conferences in 1998⁽²⁾ and 2001⁽³⁾. One of the findings of this research was that there are many cases in which the dummies' measured injury level influenced by airbags when the dummy struck the ground.

ISO 13232⁽⁴⁾ is for Motorcycles Test and Analysis Procedures for Research Evaluation of Rider Crash Protective Devices Fitted to Motorcycles. It specifies that evaluation of rider protection device performance shall be done by computer simulation, in addition to impact tests using actual motorcycles (hereinafter referred to as

full scale impact tests). It is specified that computer simulation shall cover a 0.5 second time period, from the start of impact to the point where the dummy hits the opposing vehicle (hereinafter referred to as the primary impact sequence). However, after consideration of the aforementioned results, it was decided that when evaluating an airbag's rider protection performance by computer simulation, it is necessary to evaluate injury indices up to the point when the dummy strikes the ground.

It was decided, therefore, to establish a computer simulation technology capable of evaluating injury indices from the start of collision to the point where the dummy strikes the ground (hereinafter referred to as the entire impact sequence). Explicit FEM software, chosen for the analysis, could easily express shapes and reproduce deformation of vehicles. Mesh sizes, which greatly affect calculation time and accuracy with FEM software, were examined and decided upon. Using the selected mesh size, models were created of the motorcycle, airbag, dummy and opposing vehicle, and computer simulation calculations were performed. From these

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calculations, the compliance of dummy motion, up to the point of striking the ground, was evaluated using dummy head velocity and torso angle.

As a result, a computer simulation technique, capable of reproducing test results with high accuracy, was successfully created. At the same time, a very effective technique for reducing calculation time was contrived. These results were reported at the 2003 Small Engine Technology Conference⁽⁵⁾. This research endeavored to use the previously described computer simulation technology to make it possible to evaluate injury indices up to the point when the dummy strikes the ground. The correlation of results of computer simulation and full scale impact tests were investigated for 12 cases in seven impact configurations, as specified by ISO/CD 13232. The results confirmed that this computer simulation technology is capable of evaluating injury indices, from the start of collision to the point when the dummy strikes the ground.

2. Evaluation of Injury Indices to When the Dummy Strikes the Ground

The research attempted to further develop the computer simulation technology capable of reproducing dummy motion, up to the point of striking the ground with good accuracy, and create a computer simulation technology that can be used for evaluating injury indices, up to the point when the dummy strikes the ground. This project, and the tests to verify accuracy, covered seven basic impact configurations specified by ISO/CD 13232. These impact configurations are shown in Fig. 1.

Simulations were performed with and without airbags. However, impact configuration No. 1 and No. 4, with airbags, shown in Fig. 1, were not included in the scope of correlation because the airbag did not deploy in the full scale tests in these configurations. Therefore, the computer simulation correlation scope included 12 cases in seven impact configurations.

Full scale impact test speeds were set somewhat higher than the speeds defined by ISO/CD 13232, and evaluation was done under these more severe conditions. The computer simulation impact speeds were the same as those of the full scale impact tests.

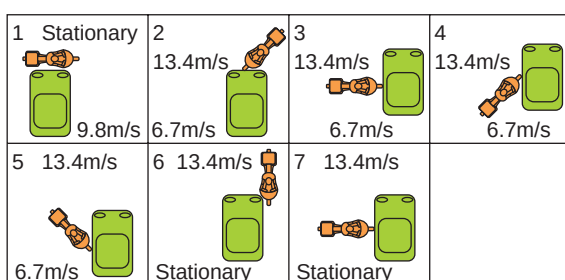


Fig. 1 Full scale impact test configurations

3. Model Creation Methods

3.1. Motorcycle Model

The motorcycle model previously used to validate the motion of dummies striking the ground was the GL1500, a large touring motorcycle. In subsequent airbag research, the GL1800 replaced the GL1500 as the motorcycle model. Therefore, a GL1800 motorcycle model was also created for the computer simulation. The model was created using a procedure reported in 2003 SETC⁽⁵⁾. The motorcycle model was to be entirely deformable, in order to enhance calculation accuracy. The motorcycle model is shown in Fig. 2.

3.2. Opposing Vehicle Models

Models of the opposing vehicle were additionally created in accordance with the diversity of impact configurations. Because local rigidity of the opposing vehicle model is very effective for reducing calculation time, rigidity was given to parts that would not come into contact with the motorcycle or dummy, and would not deform. This resulted in the creation of four different opposing vehicle models, each with different rigid parts. The model used was chosen according to the impact configuration. The opposing vehicle models created are shown in Figs. 3 to 6. The 1998 - 2001 Honda Accord four-door sedan with Japanese market specifications was used as the opposing vehicle model; Table 1 gives the specifications.

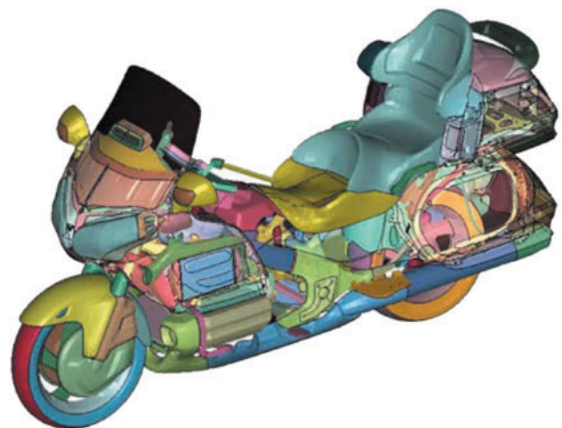


Fig. 2 Motorcycle model

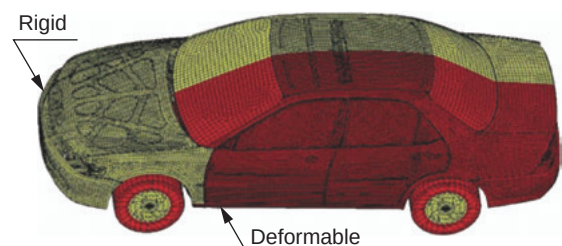


Fig. 3 Opposing vehicle model for side impact case



Fig. 4 Opposing vehicle model for front-side impact case

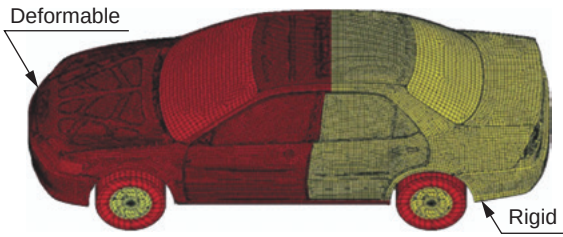


Fig. 5 Opposing vehicle model for front impact case

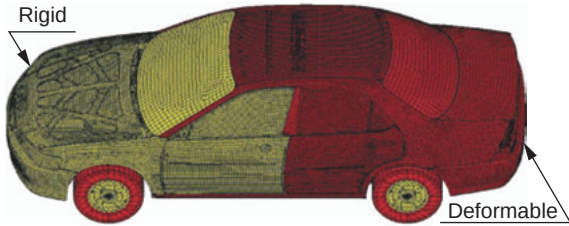


Fig. 6 Opposing vehicle model for rear impact case

Table 1 Specifications of opposing vehicle

Manufacturer		Honda
Model		Accord, Japan
Year		1998-2001
Mass (average)	(kg)	1300
Overall length	(mm)	4635
Overall width	(mm)	1695
Overall height	(mm)	1420
Wheelbase	(mm)	2665
Engine displacement	(cm ³)	1997

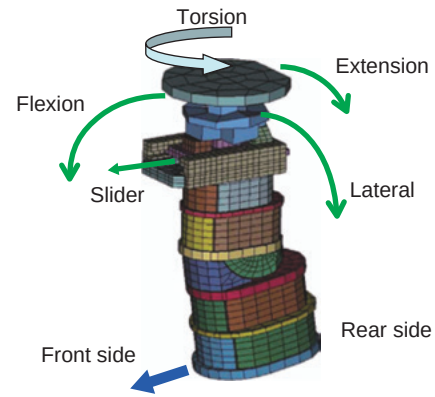


Fig. 7 Neck model of MATD (ISO/CD13232)

Table 2 Neck static characteristics of MATD model

Characteristics	Load	Required value	CAE model
Flexion angle	20 kgf	17.6 ± 2.6 deg	15.9 deg
Slider displacement	20 kgf	14.0 ± 3.0 mm	13.8 deg
Extension angle	20 kgf	30.9 ± 4.6 deg	27.7 deg
Lateral angle	20 kgf	28.7 ± 4.3 deg	25.8 deg
Torsion angle	3.2 kgf	41.5 ± 6.2 deg	43.6 deg

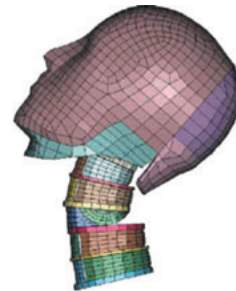


Fig. 8 Dynamic extension position

3.3. Dummy Model

The dummy model was based on the hybrid III 50th - percentile model for LS-DYNA⁽⁶⁾ and conformed to the MATD dummy specifications as defined in ISO 13232. The neck injury evaluation method, undefined in ISO 13232 (1996), has been defined in ISO/CD 13232 (2002), so the model followed ISO/CD 13232 (2002). The model created is shown in Fig. 7. The bending characteristics of the neck model were conformed to both the static and dynamic characteristics defined in ISO/CD 13232 (2002). The results of static correlation are shown in Table 2. Figures 8 to 18 show the correlation results for each of the dynamic characteristics, which are extension, flexion, lateral and torsion. Based on the above results, it was decided that the computer simulation neck model created for this research was usable.

3.4. Airbag Model

An airbag model was created based on the airbag designed for the GL1800. The V-shape of the back of the airbag to hold the rider, and the connecting and supporting belts from the back of the airbag to the motorcycle frame, were modeled. The computer simulation model of the airbag mounted on the GL1800 is shown in Fig. 19.

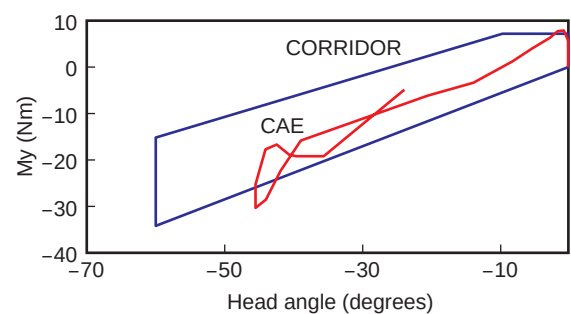


Fig. 9 Dynamic extension moment vs. head angle

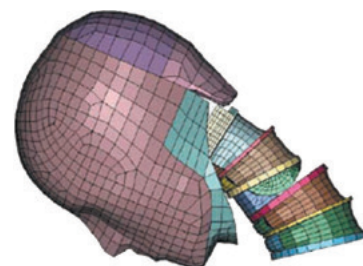


Fig. 10 Dynamic flexion position

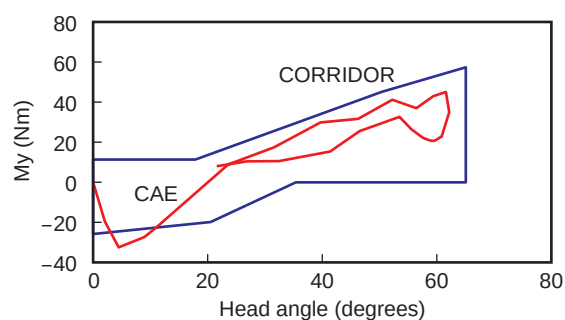


Fig. 11 Dynamic flexion bending moment vs. head angle

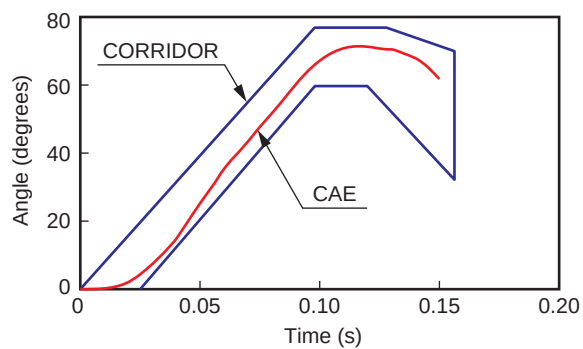


Fig. 15 Dynamic lateral head angle vs. time

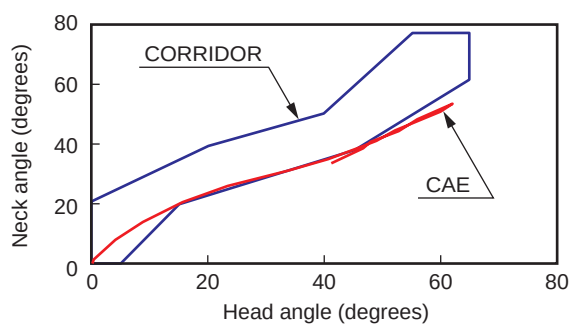


Fig. 12 Dynamic flexion neck angle vs. head angle

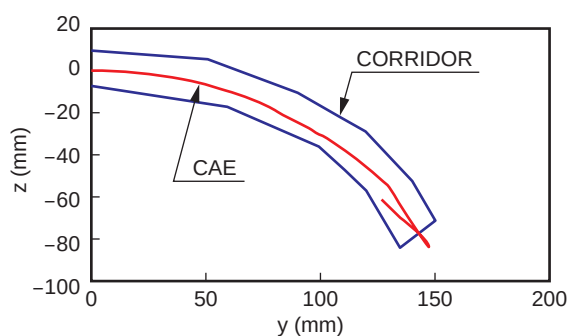


Fig. 16 Dynamic lateral head center of gravity position

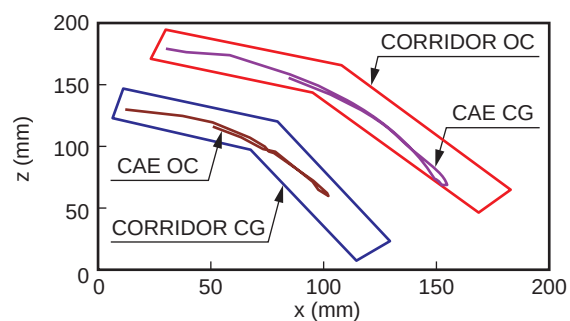


Fig. 13 Dynamic flexion occipital condyle and head center of gravity position

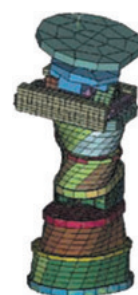


Fig. 17 Dynamic torsion position

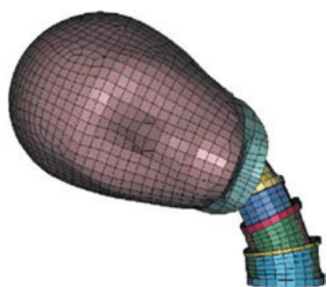


Fig. 14 Dynamic lateral position

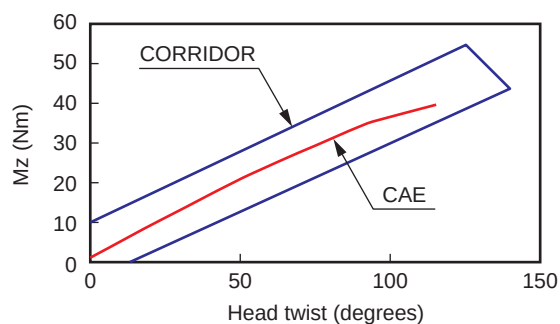


Fig. 18 Dynamic torsion stiffness

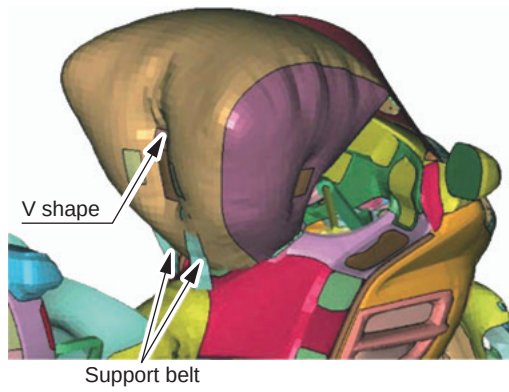


Fig. 19 Airbag model

4. Methods of Enhancing Model Accuracy

Each model created requires high accuracy to enable the evaluation of injury up to the point where the dummy strikes the ground. As the first step in this process, parts which can be predicted to deform during collision were faithfully modeled as far as possible. As the second step, tests were conducted to determine characteristics under simple conditions, such as the unit test defined in ISO/CD 13232 (2002), and rigid barrier impact tests using motorcycles and opposing vehicles. Simulation models and some calculation factors were adjusted so that the simulations would conform to the actual tests with high accuracy.

As the final step of validation, model accuracy and some calculation factors were modified when confirming the correlation with the test results in the 12 test cases using seven impact configurations defined in ISO/CD 13232. Some factors not included in the simple phenomena of the unit tests, or the rigid barrier tests, were included among the actual vehicle collision phenomena, which are more complex. These factors include friction properties as well as detailed structural parts of the motorcycle and the opposing vehicle, which had been eliminated in prior simulation, in order to reduce calculating time. The accuracy with respect to these factors was enhanced in the process of confirming correlation with the seven impact configurations.

The accuracy of component models of the side of the opposing vehicle and frontal part of the motorcycle were enhanced in the impact configurations No. 3, No. 5 and No. 7, shown in Fig. 1, in which the motorcycle collides with the side of the opposing vehicle. Next, using impact configuration No. 2, shown in Fig. 1, in which the motorcycle collides with the front of the opposing vehicle, the accuracy of the model of the front part of the opposing vehicle was enhanced. In impact configurations No. 4 and No. 6, shown in Fig. 1, the accuracy of models of the front side part of the motorcycle and the opposing vehicle was enhanced, while in impact configuration No. 1, the accuracy of the model of the side of motorcycle was enhanced.

5. Correlation of Evaluated Injury Indices

The correlation was conducted using the dummy injury indices of 12 test cases of seven impact configurations. Figure 20 shows the head HIC injury index values in the primary impact sequence, while Fig. 21 shows the HIC values more than 0.5 seconds after the start of collision, up to the point when the dummy strikes the ground (hereinafter referred to as the secondary impact sequence). The r^2 correlation coefficient in the primary and secondary impact

sequences was 0.94 and 0.77, respectively.

Figure 22 shows the maximum compression ratios on the chest in the primary impact sequence. Because the airbag takes the primary impact, this is an important factor. The r^2 correlation coefficient was 0.62.

Table 3 shows a comparison between the simulation results and the full scale impact test results for leg fractures. The conforming rate of the simulation results and the test results was 96% for the femurs and 100% for the knees and tibias.

From the results above, it was decided that this simulation technology enables one to evaluate the dummy injury indices, from the start of impact to the point in time when the dummy strikes the ground. Therefore, we considered that this computer simulation makes it possible to evaluate the benefits and risks of the airbag system in the 200 impact configurations defined in ISO/CD 13232.

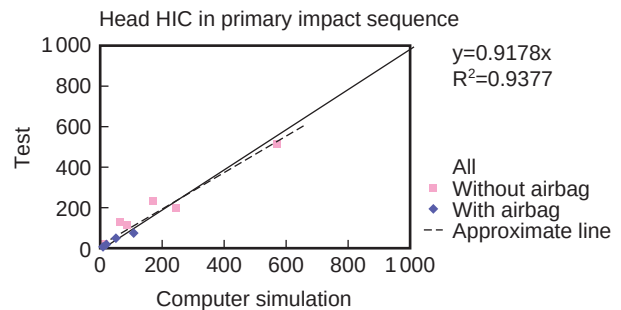


Fig. 20 Correlation of HIC at primary impact sequence

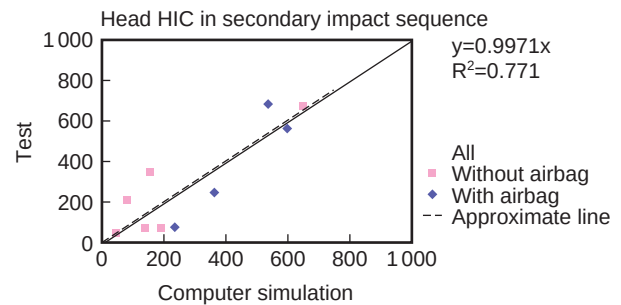


Fig. 21 Correlation of HIC at secondary impact sequence

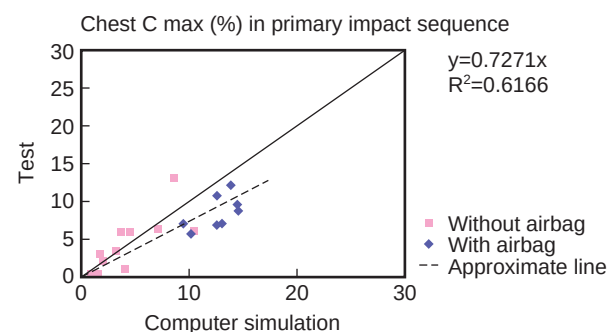


Fig. 22 Correlation of chest sternum compression at primary impact sequence

Table 3 Correlation of leg injury for entire impact sequence

Femurs		Full scale tests		Corrected percentage
		Fracture	No fracture	
Simulations	Fracture	0	1	96%
	No fracture	0	23	

Knees		Full scale tests		Corrected percentage
		Fracture	No fracture	
Simulations	Fracture	0	0	100%
	No fracture	0	24	

Tibias		Full scale tests		Corrected percentage
		Fracture	No fracture	
Simulations	Fracture	1	0	100%
	No fracture	0	23	

6. Evaluation of Injury Reduction Effectiveness of Airbag for GL1800

ISO/CD 13232 defines the methodology for conducting the computer simulation of 400 impact cases, which are 200 impact configurations with and without the proposed rider crash protective device, and thereby determining the benefit and risk of that device.

Figure 23 shows 25 impact configurations defined by different combinations of relative heading angles and contact points. Additionally, the differences in impact speeds are defined in ISO/CD 13232. In combination, the number of impact configurations grows to 200.

Those of the 25 basic configurations that are marked with an X were omitted from simulation because it was judged that the airbag would not deploy, or it would deploy but did not influence the motion of the rider or the injury indices. Those configurations marked with a triangle were omitted from simulation because it was judged that the airbag would not deploy because of the impact speed. In the omitted configurations, the injury indices were defined as being the same, whether or not there was an airbag. Consequently, the final calculations used 121 impact configurations and 242 simulation cases.

7. Results of Airbag System Evaluation by Computer Simulation

Computer simulations were run for 121 impact configurations and 242 simulation cases.

The airbag's injury reduction performance was evaluated based on the methodology of ISO/CD 13232. The results quantified and characterized the effectiveness of the airbag system at reducing injury level in the various impact configurations.

7.1. Rider Injury Reduction Effectiveness

Figure 24 shows the influence the airbag has on injury during the primary and secondary impact sequences. The vertical axis represents NIC values, where an NIC of 0 indicates that there is no injury and 1 is the equivalent of one fatality. Figure 24 shows the total average benefit and total average risk in the primary impact

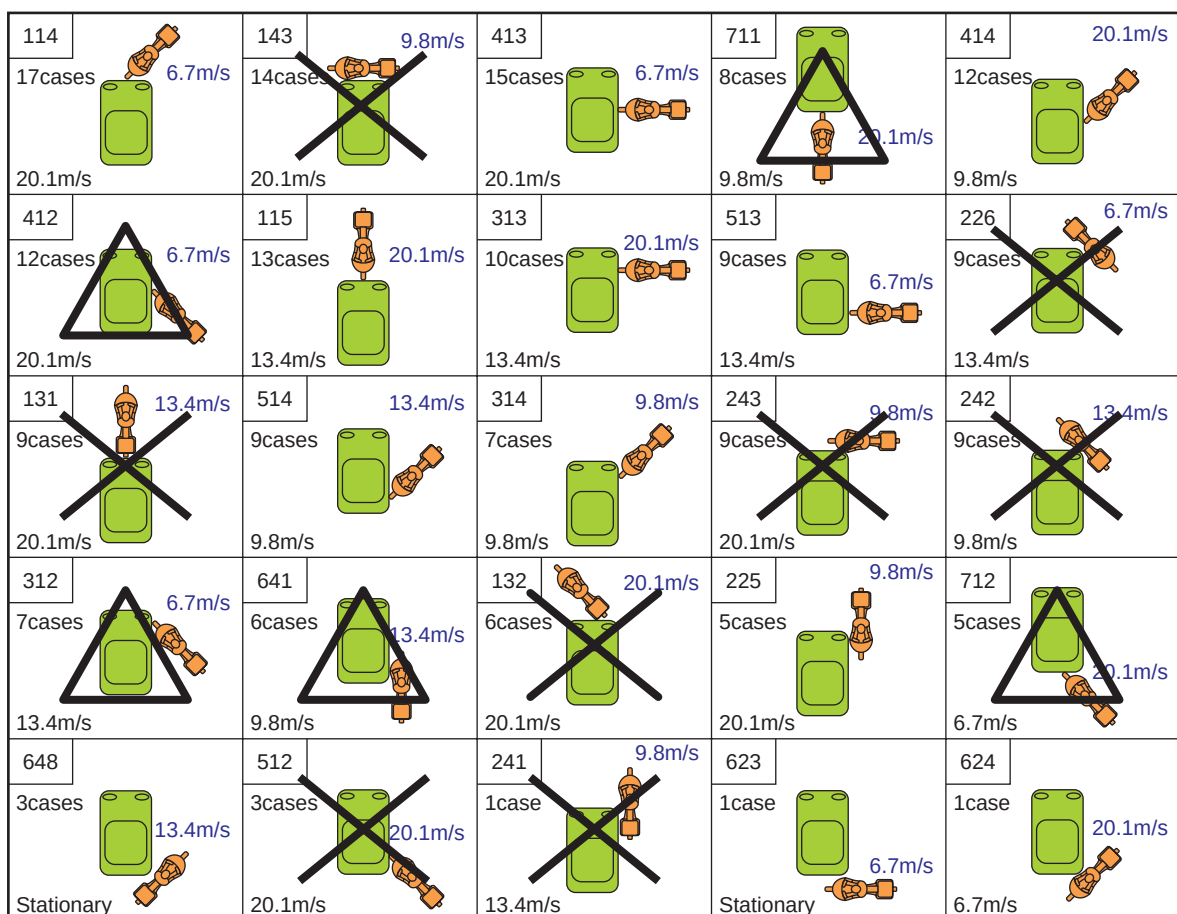


Fig. 23 Impact configurations for computer simulation

sequence. The total average benefit is 0.038 and the total average risk is 0.001. The ratio of risk to benefit is 0.026. The average net benefit is 0.037.

Figure 24 shows the total average benefit and total average risk in the secondary impact sequence. The total average benefit is 0.022 and the total average risk is 0.011. The ratio of risk to benefit is 0.500. The average net benefit is 0.011.

Thus, the injury reduction effect from the airbag is greater when the rider impacts the opposing vehicle than when the rider strikes the ground.

Figure 25 shows the total average benefit and total average risk in the entire impact sequence, encompassing both the primary and secondary impact sequences (i.e. from the start of collision until the point where the dummy strikes the ground). The total average benefit is 0.048 and the total average risk is 0.004. The ratio of risk to benefit is 0.083. The average net benefit is 0.044. From these results, it was judged that the airbag system has appropriate injury index reduction performance.

7.2. Influence of Impact Speed

The average net benefit values were compared by impact speed. Figure 26 shows the results. Impact speed in this case refers to relative speed. When the motorcycle collides with the side of the

opposing vehicle, the speed of the motorcycle is used as the relative speed. When it collides head-on with the front of the opposing vehicle, the combined speed of the two vehicles is used as the relative speed. When it collides with the rear of the opposing vehicle, the speed of the motorcycle minus the speed of the opposing vehicle is used. When colliding with the front or rear of the opposing vehicle, and the impact configuration is angled, the impact speed of the opposing vehicle is reduced in proportion to the angle of impact.

For instance, in configuration No.2 of Fig.1, the motorcycle collides with the front of the opposing vehicle at an angle of 45 degrees. The relative impact speed in this case was calculated as “13.4 m/s (motorcycle speed) plus 6.7 m/s (opposing vehicle speed) times $\cos 45^\circ$.”

Figure 26 verifies the benefit of the airbag for the GL1800 from the low to high speed ranges, demonstrating that the airbag was effective. The highest benefit appears at the relative impact speed range of 20 - 25 m/s. Therefore, the effect of injury reduction is significant at a high-speed range.

Figure 27 gives an example of an impact configuration where the effectiveness of airbags in a high-speed range is clearly demonstrated, as based on the above results. Dummy motion during collision of the baseline motorcycle (i.e. the one without an airbag) is shown in Fig. 28. Dummy motion during collision of an airbag-equipped motorcycle is shown in Fig. 29. In the case of the baseline motorcycle without an airbag, the head of the dummy impacts the roof of the opposing vehicle hard. In contrast, in the case of the airbag-equipped motorcycle, the head of the dummy softly impacts the opposing vehicle. The injury indices are shown in Table 4.

The dummy injury index is shown as AIS. Injury indices are expressed on a six-level scale, with AIS 1 being a minor injury level and AIS 6 a fatal injury level.

With the baseline motorcycle without an airbag, AIS for the dummy's head was calculated as 6, i.e. fatal, whereas the calculated AIS for the head with the airbag-equipped motorcycle was representative of no injury. Similarly, the AIS for the dummy's neck with the baseline motorcycle without an airbag was calculated as 4, whereas the AIS for the neck with the airbag-equipped motorcycle was equivalent to no injury.

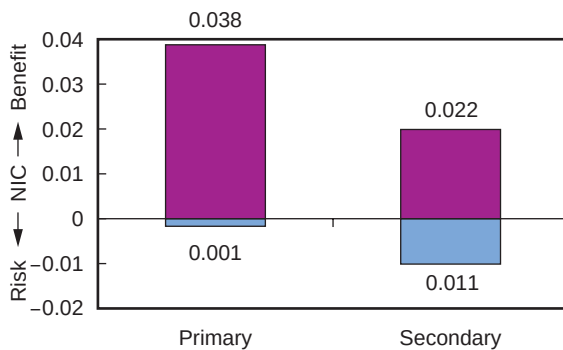


Fig. 24 Total average benefits and risks, primary and secondary impact sequence

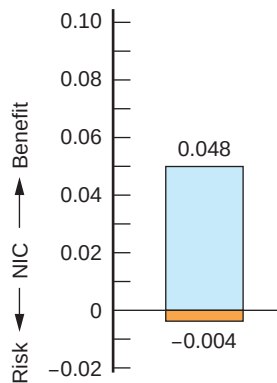


Fig. 25 Total average benefits and risks in the entire impact sequence

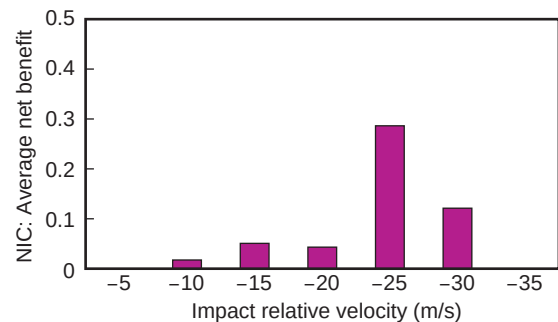


Fig. 26 Total average net benefits, by relative impact speed

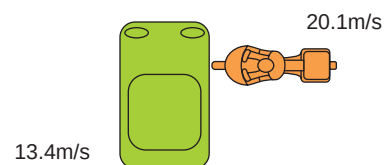


Fig. 27 Benefit impact configuration

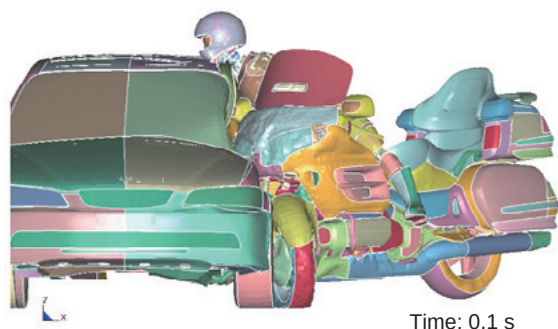


Fig. 28 Baseline impact case

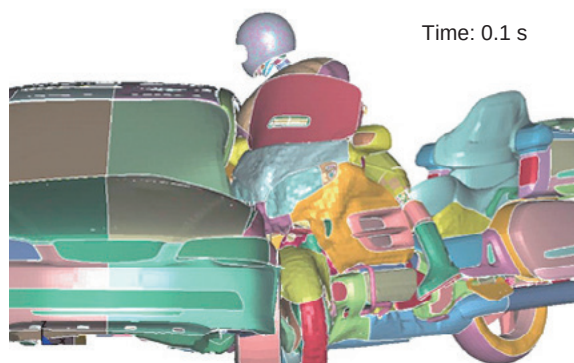


Fig. 29 With airbag impact case

Table 4 Comparison of AIS between baseline and with airbag

	Base line	Airbag
Head	6	0
Neck	4	0
Chest	1	1
Abdomen	0	0
Leg	3	3

8. Conclusion

- (1) Computer simulation using explicit FEM software enables the evaluation of the dummy injury index during a motorcycle collision up to the point where the dummy strikes the ground.
- (2) Using this simulation method, the injury reduction performance of an airbag mounted on the GL1800 was evaluated based on ISO/CD 13232.

The following conclusions regarding injury reduction are drawn from this research:

- The total average benefit of the entire impact sequence was 0.048 and the risk was 0.004. The performance of the injury reduction system is therefore appropriate.
- The highest average net benefit appears at the relative impact speed range of 20 - 25 m/s (72 - 90 km/h).
- The injury reduction effect is greater when the rider impacts the opposing vehicle than when the rider strikes the ground.

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Enhancement of Motorcycle Conspicuity Using Brain Activities Induced by the Facial Image Stimuli

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Masahiro KIMURA**

ABSTRACT

Humans have a special ability for recognizing the face. We have proposed a new motorcycle design technique to enhance search performance using this ability. This technique uses shape recognition and light positioning.

Some experiments were carried out to measure brain activity while showing facial image stimuli. Results showed significant activity in the lateral fusiform gyrus (FFA). This implies that the brain analyzes our facial image stimuli as a human face.

1. Introduction

Extraction and analysis of figures for fatal motorcycle accidents (in which automobiles are the principal party) from the Annual Traffic Accident Statistical Database, 1998 Edition⁽¹⁾, showed that failure by drivers of automobiles to notice motorcycles was, together with the misperception of approach speed, a primary factor in accidents (Fig. 1).

Enhancing the visibility of motorcycles should be effective in helping prevent the failure of noticing motorcycles. There is concern, however, that the growing use of headlights during the daytime by trucks and buses will result in a relative lowering of motorcycle visibility. The formulation of rules for the daytime use of headlights by motorcycles is thus being promoted in different countries as a way of dealing with that situation.

Meanwhile, reports from research on the brain in recent years indicate that there are parts of the brain (neurons) that react very sensitively to specific patterns. Reports on "hand neurons"⁽²⁾ in macaques in 1968 were followed by reports on "face neurons" in 1981⁽³⁾.

This present research proposes a means for raising the visibility of motorcycles by making use of the human capability to process

facial images. It also demonstrates, using brain measurements, that it is possible to quantitatively compare human psychological perceptions of facial qualities.

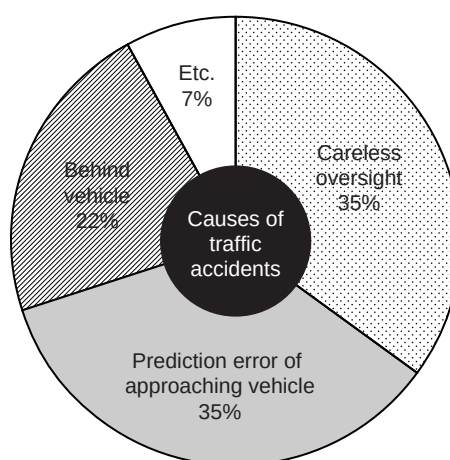


Fig. 1 Analysis of traffic accidents

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2. Proposition

2.1. Human Capability for Facial Image Processing

The technology for noninvasive brain measurement by means of functional Magnetic Resonance Imaging (fMRI) has made advances. This has made it possible to observe brain activity without harming the person being studied, and it has been found that there is a part of the brain in human beings, as in macaques, that handles the processing of facial images. The latest research has confirmed that the face area of the fusiform gyrus (Fusiform Face Area, FFA) in the temporal lobe, and the face area of the inferior occipital gyrus (Occipital Face Area, OFA), have high receptivity to facial images⁽⁴⁾⁻⁽⁶⁾.

These parts of the brain also demonstrate a capability to search for facial images. The human being is easily able, for example, to find the faces in scenes such as seen in Fig. 2. Lewis and others have reported that the facial image pops out from groups of different patterns when a human being is searching for it⁽⁷⁾.

It should be noted that these human capabilities are possessed by human beings without any particular training. Exercise of these capabilities does not require the imposition of any particular psychological burden.

2.2. Motorcycle Design Proposal

Bruce and others showed images of faces and of faces with different parts removed, such as eyes, mouth, and so on, to macaques, and recorded the neuron activity in their inferior temporal lobes⁽³⁾. The results show that a significant decrease in neuron activity was observed with images that had the eyes or mouth concealed. It has been reported that, when measuring human brains by means of fMRI, FFA activity becomes more intense when the facial images being shown are expressing emotion (joy, anger, sadness, pleasure, etc.)⁽⁸⁾. According to experiments by Fox and others⁽⁹⁾, who measured differences in face searching efficiency by differences in facial expression, searches for angry faces were faster than searches for faces with other expressions. These findings were used to create a design that would stimulate the human brain more



Fig. 2 Hidden face images



Fig. 3 Design proposal

efficiently, such as the proposed motorcycle design shown in Fig. 3. This design was configured on the front cowl of the Honda ASV test vehicle. Lamp shapes that would add emphasized facial shapes were devised, with the main lamp forming the mouth, and the directional indicators and position lamps forming the eyes.

The concept is that motorcycles designed in this way might be easier to recognize in a complex traffic environment by means of the human capability for sophisticated facial pattern processing. If so, this might also be expected to raise motorcycle visibility relative to automobiles.

3. Experiments

3.1. Purpose of Experiments

The purpose of the brain measurement experiments was to utilize fMRI to:

- (1) Compare activated regions of the brain to confirm whether the proposed design stimulates the facial image processing part of the brain in the same way that a human face does; and
- (2) Measure brain activity to perform a quantitative comparison of the facial qualities of the proposed design, and of human faces and ordinary motorcycle designs.

3.2. fMRI

Functional MRI (fMRI) is unlike the anatomical MRI, which measures the living structure, and instead measures brain activity and function. It can make measurements noninvasively, so less burden is placed on experimental subjects, even in repeated experiments.

3.3. Experimental Subjects

The subjects for these experiments were six healthy men and women recruited within the company. The measurements were conducted on these experimental subjects (5 men, 1 woman, ages ranging from 20s to 40s).

The experimental subjects were given full explanations of the purpose of the experiments and of ethical and safety aspects. They then gave their informed consent to the experiments, and signed releases accordingly, before the experiment proceeded.

3.4. Overview of Experiments

For these experiments, the subjects were placed in an MRI apparatus (Fig. 4), laying on their backs and holding a pushbutton, by which to make their responses. The stimulating image would be displayed upon a screen placed in the device, reflected in a mirror. The task of the experimental subjects was to look at a series of



Fig. 4 MRI apparatus

images and indicate by pushbutton responses whether each image matched the immediately preceding image or not (1-Back Task).

The angular field of view of the stimulus image, as shown in Fig. 5, was 13° horizontal and 10° vertical, and the image size was 650 x 500 pixels.

3.5. Images Used in the Experiments

The experiments made use of six categories (Fig. 6). Each category was made up of 20 different grayscale images (256 gradations).

1. FBK (facial motorcycle designs)
2. BK (normal motorcycle designs)
3. NFBK (non facial motorcycle designs)
4. HF (human faces)
5. NHF (faces in outline only)
6. OBJ (objects unrelated to faces)

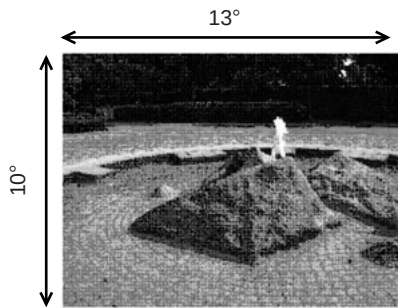


Fig. 5 Stimulation size

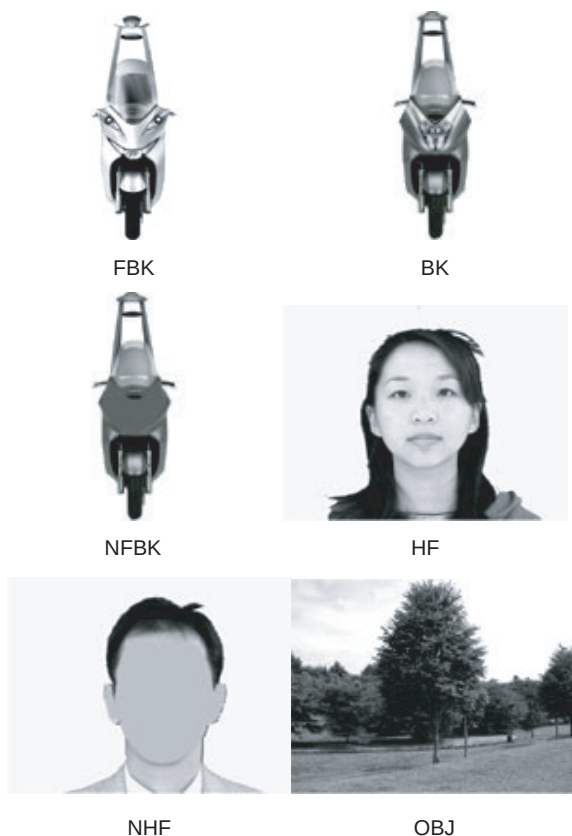


Fig. 6 Examples of stimuli

3.6. Experimental Design

The design used in the present experiments was a block-type task design, which follows a basic experimental protocol for measuring brain activity by means of fMRI. This is considered to produce highly reliable results, even with a relatively small number of experimental subjects.

The tasks were divided among six sessions. Each session was approximately four minutes long. Considering the effects of the order of stimuli, the six categories were presented in a random order during that period. At the beginning and end of each session, and in between categories, a fixation image (a white cross shape on a black background) was shown (Fig. 7).

The stimuli in a session were made up of the basic stimulus image, which lasted one second (a category image for 725 ms and a fixation point for 275 ms) (Fig. 8).

The stimuli in each category were made up of 16 basic stimulus images. In order to keep the experimental subjects aware and hold their attention on the stimulus images, the same image would be randomly repeated in series, and the subjects were assigned the task of indicating by pushbutton responses whether each image matched the immediately preceding image or not (1-Back Task) (Fig. 9).

Each experimental subject went through six-session experiments, two times. The net time taken for each experiment (of six sessions) was approximately 23 minutes (Fig. 10). With preparation included, the time taken was approximately 40 minutes.

3.7. fMRI Imaging and Analytical Method

The equipment used was a MAGNEX ECLIPSE 1.5T Power Drive 250 scanner, manufactured by Shimadzu-Marconi, at the ATR Brain Activity Imaging Center (BAIC).

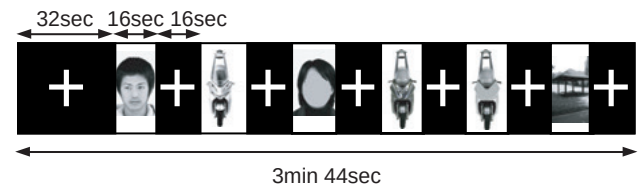


Fig. 7 Session of stimuli

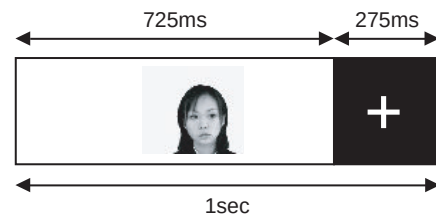


Fig. 8 Element of stimulus

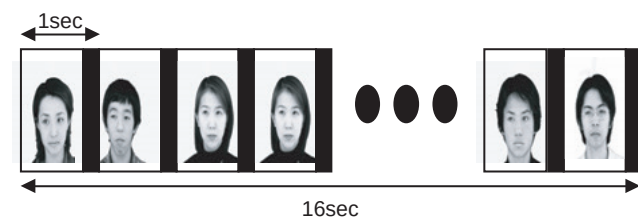


Fig. 9 Category of stimuli

For the present experiments, it was necessary first of all to determine which part of the brain was activated by the given stimulus images. Therefore, the imaging parameters were set to scan the entire human brain (Table 1).

Six experimental subjects did not form a large enough group for the analysis of total group tendencies in these experiments. Therefore, analysis was done for each individual subject.

The analysis was performed using Statistical Parametric Mapping (SPM2), from the Wellcome Institute of Cognitive Neurology, London, UK⁽¹⁰⁾. This is a standard software package for statistical analysis of fMRI data. The analytical process was composed of a part in which the fMRI data is pre-processed in a series of steps, and a part in which hypotheses as to whether brain activity is significant or not are tested statistically.

Pre-processing involved: Realignment, in which position changes caused by head movement were corrected; normalization, which is linear and non-linear conversion to a standard brain; and smoothing, which is 10-mm half-bandwidth Gaussian filtering.

In the part of the analysis that statistically tested hypotheses, the risk rate of testing T was considered to be of $P < 0.05$ (FWE corrected). Active parts were confirmed by conversion of Voxel coordinates (X, Y, Z) to Talairach coordinates (X', Y', Z') and checking against the atlas of Talairach and Tournoux⁽¹⁰⁾. The conversion formula was as follows:

$$\begin{aligned} \text{For } Z \geq 0 \\ X' &= 0.9900 X \\ Y' &= 0.9688 Y + 0.0460 Z \\ Z' &= -0.0485 Y + 0.9189 Z \end{aligned}$$

$$\begin{aligned} \text{For } Z < 0 \\ X' &= 0.9900 X \\ Y' &= 0.9688 Y + 0.0420 Z \\ Z' &= -0.0485 Y + 0.8390 Z \end{aligned}$$

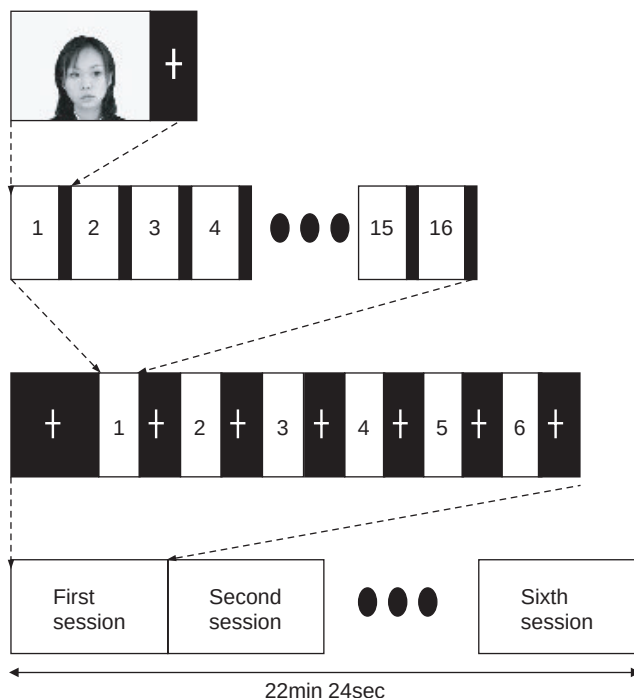


Fig. 10 Experimental design

Table 1 Parameters of fMRI

TR	4 000 ms
TE	50 ms
Flip angle	90°
FOV	192 mm
Slice thickness	3 mm (1 mm gap)
MATRIX size	64 × 64
Slice number	30

The parts of the brain confirmed were the FFA and OFA, which process facial images. Those locations are shown below:

Fusiform Face Area (FFA)

— The middle lateral fusiform gyrus

— Brodmann Area 37 (BA 37)

Occipital Face Area (OFA):

— Inferior occipital gyrus

— Brodmann Area 19 (BA 19)

In order to analyze the amount of brain activity, values conforming to the Blood Oxygen Level Dependent (BOLD) Signal Change were used (calculated from contrast estimates and 90% C.I. in SPM2). BOLD Signal Change values indirectly show changes in nervous metabolism and brain blood flow that accompany brain activity. They show the rate of change of oxygen consumption in the blood⁽¹²⁾.

4. Results and Discussion

4.1. Confirmation of Activated Parts of the Brain

Brain function images (hereafter, contrast images) obtained through analysis are given in Figs. 11 and 12 as representative examples of individual analysis. Figure 11 is a contrast image of HF and OBJ, and Fig. 12 of FBK and OBJ. Activation of the FFA was confirmed at the locations marked with black circles.

Table 2 shows the coordinates for FFA activation (Talairach coordinates), the risk rate (Voxel level) for testing, and the size of the area (Cluster level).

In addition, contrast between FBK and OBJ in OFA activation on the right side was confirmed in three experimental subjects (Subject B, Subject C, and Subject D) (Table 3).

These results yielded data suggesting that FBK and HF are processed generally by the same brain activity in four of the six experimental subjects. Statistically significant differences in activation were not confirmed in the remaining two experimental subjects.

Statistically significant activation of FFA and OFA was not observed in the contrast images for BK and OBJ, and for NFBK and OBJ, in any of the experimental subjects.

4.2. Analysis of the Amount of Brain Activity

The amount of brain activity across the six categories for the four experimental subjects in whom FFA and OFA activation was confirmed, as described in section 4.1, was analyzed and the results

Table 2 Right fusiform face area activated (SubjectA) activated (SubjectA)

Contrast	Subject	Talairach coordinates			Voxel-level	Cluster-level
		X'	Y'	Z'	$P_{FWE-CONT}$	K_E
HF-OBJ	A	48	-57	-11	0.000	25
FBK-OBJ	A	50	-57	-11	0.001	15

Table 3 Right occipital face area activated (SubjectB, SubjectC, Subject D)

Contrast	Subject	Talairach coordinates			Voxel-level	Cluster-level
		X'	Y'	Z'	$P_{FWE-CONT}$	K_E
FBK-OBJ	B	42	-72	-11	0.023	3
	C	55	-67	-13	0.031	3
	D	44	-66	-7	0.013	6

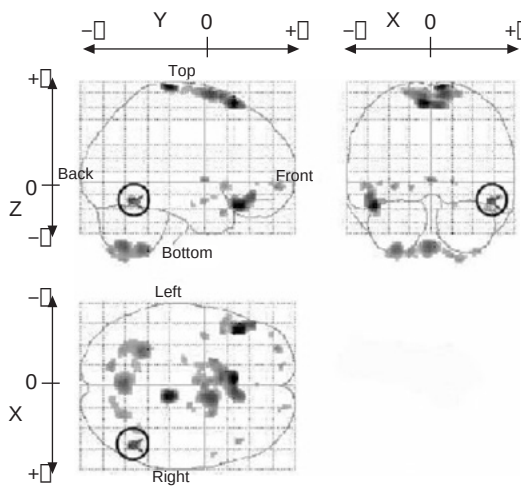


Fig. 11 Contrast image (Subject A: HF - OBJ)

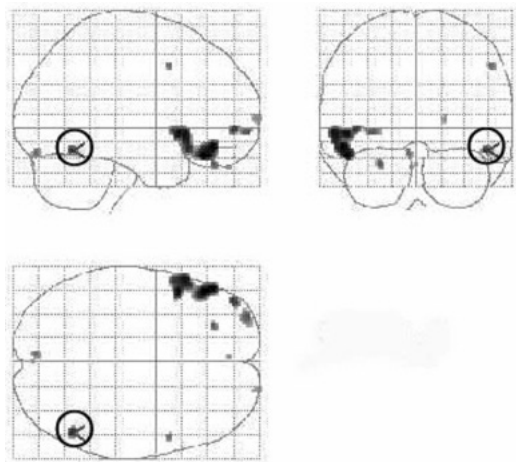


Fig. 12 Contrast image (Subject A: FBK-OBJ)

are shown here (Fig. 13, Fig. 14, Fig. 15, Fig. 16). The error bars in each graph show the Standard Error (SE).

The four experimental subjects all showed about the same level of activity for FBK as for HF. The level of activity tended to be lower for BK and NFBK than for FBK. In all four experimental subjects, the amounts of activity were in this order: HF > FBK >

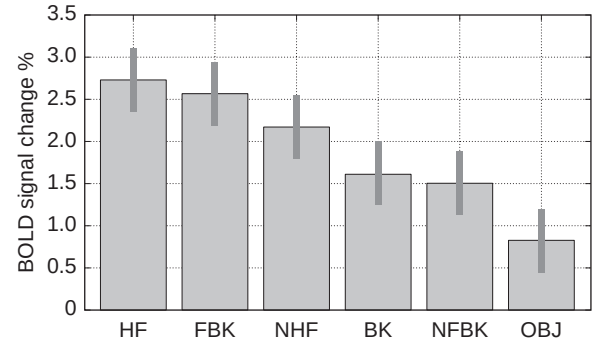


Fig. 13 BOLD signal change (Subject A)

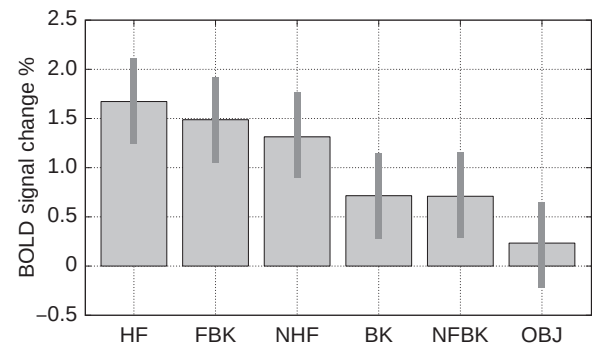


Fig. 14 BOLD signal change (Subject B)

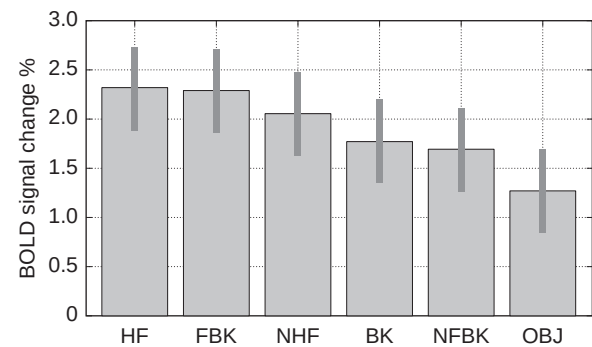


Fig. 15 BOLD signal change (Subject C)

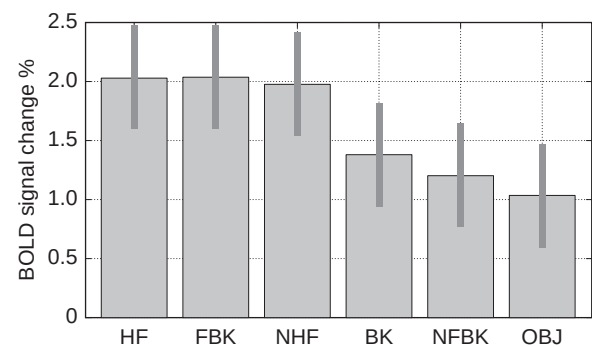


Fig. 16 BOLD signal change (Subject D)

NHF > BK > NFBK > OBJ.

The above results demonstrate experimentally that the FBK design presented here stimulates the same level of brain activity as a human face. This suggests that the BOLD Signal Change values for FFA and OFA, which were employed as quantitative indicators of brain activity for these experiments, could possibly be used as evaluation indices for facial qualities.

5. Conclusion

Styling design technology that enhances motorcycle visibility by making use of the human ability to process facial images was proposed. In order to evaluate its effect, measurements of brain activity were made by means of fMRI. As a result, it was confirmed that the proposed design stimulates the part of the brain that processes facial images, just as the human face does. This provided experimental demonstration of new possibilities for enhanced visibility. These experiments also demonstrated the possibility of quantitative comparison of facial qualities, as sensed psychologically by humans. Efforts will be made to make measurements of more experimental subjects in the future, in order to verify the validity of this proposal, and to take steps toward verification of its effectiveness with actual vehicles in actual traffic environments.

The facial data in this paper are used by permission of Softopia Japan, Research and Development Division, HOIP Laboratory. It is strictly prohibited to copy, use, or distribute the facial data without permission.

Finally, the authors express their gratitude to Hideaki Kawabata, Assistant Professor, Faculty of Education, Kagoshima University, for providing pertinent advice during the conduct of this research.

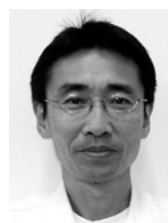
Faces Detected More Efficiently?, Cognition & Emotion, Vol. 14, p. 61-92 (2000)

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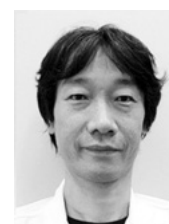
Yoshiyuki MATSUMOTO



Masahiro KIMURA



Kazumitsu KUSHIDA



Yutaka MURATA

Technical Papers Sumbitted to Out-of-company: January, 2005 to June, 2005

(English Papers only)

Date	Conference	Title	Author	Organization
2005. 1.19~22	ARAI&SAE INDIA/Symposium on International Automotive Technology SIAT 2005 (Pune, India)	Research into In-line Fuel Feed Pump Module for Fuel Injection System of Two-wheel Small Vehicle	Toichiro Hikichi Mitsuo Nakagawa	Asaka R&D Center Asaka R&D Center
2005. 1.30	JSAE/Review of Automotive Engineering Vol.26 No.1	Development of flywheel battery and ANG storage system	Shoei Abe Hiroshi Ogasa Yukihiko Itai	Tochigi R&D Center Tochigi R&D Center Wako Research Center
		Study of Hydro Bush for Reduction of Harshness and Road Noise	Yasuo Miyamoto Sou Nakamura	Tochigi R&D Center Tochigi R&D Center
2005. 2. 2~4	MAGNA STEYR/6th All-Wheel Drive Congress Graz (Graz, Austria)	Development of SH-AWD (Super Handling-All Wheel Drive) System	Yoshihiro Atsumi	Tochigi R&D Center
2005. 2. 9~10	SAE/SAE Hybrid Vehicle Technologies Symposium – Today and Tomorrow (Costa Mesa, USA)	ACCORD HYBRID Provide superior dynamic performance while achieving adequate fuel economy enhancement	Naritomo Higuchi	Tochigi R&D Center
2005. 2.16~18	Japanese Society of Tribologists (JAST)/The 2005 International Symposium on Tribology of Vehicle Transmissions (Tsukuba, Japan)	Development of Highly Efficient CVT for Upper Class Compact Vehicle	Akihito Ohashi Yoichi Sato Kaoru Kajikawa Shinji Sakuma Junji Urano Masaaki Yamaguchi Akihiro Shibahara	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
2005. 3.13~17	American Chemical Society/2005 ACS National Meeting (San Diego, USA)	First-Principles and Tight Binding quantum chemical Molecular Dynamics Approach to Vibrationally Excited States Dynamics of Water	Masahiro Hirota Kazuyoshi Akutsu Masayoshi Kitada Hajime Hirata Ewa Broclawik Momoji Kubo Akira Miyamoto Keiko Chiba Yumiko Sasaki Abdul Rajjak Shaikh Qiang Pei Hideyuki Tsuboi Michihisa Koyama	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tohoku University Tohoku University Tohoku University Tohoku University Tohoku University Tohoku University Tohoku University Tohoku University Tohoku University
		TIGHT-BINDING QUANTUM CHEMICAL MOLECULAR DYNAMICS APPROACH TO THE FORMATION DYNAMICS OF HYDROGEN BY THE CHEMICAL REACTIONS OF VIBRATIONALLY EXCITED WATER ON SI SURFACES	Masahiro Hirota Kazuyoshi Akutsu Masayoshi Kitada Hajime Hirata Momoji Kubo Ewa Broclawik Akira Miyamoto Keiko Chiba Yumiko Sasaki Abdul Rajjak Shaikh Qiang Pei Hideyuki Tsuboi Michihisa Koyama	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tohoku University Tohoku University Tohoku University Tohoku University Tohoku University Tohoku University Tohoku University Tohoku University Tohoku University
2005. 3.16	SAE/2005 SAE Colloquia/Fuel & Lubes Asia Conference (China)	Mobility 2030	Takanori Shiina	Tochigi R&D Center

2005. 4. 2~6	European Electric Road Vehicle Association (AVERE)/EVS21 (The 21st International Battery, Hybrid and Fuel Cell Electric Vehicles Symposium & Exposition) (Montecarlo, Monaco)	Development of 2005 ACCORD Hybrid System	Masaaki Kaizuka	Tochigi R&D Center
2005. 4. 4~8	The Institute of Electrical Engineers of Japan (IEEJ)/IPEC (International Power Electronics Conference) (Niigata, Japan)	High Speed Analysis of 3-phase High Frequency Voltage Input Sensorless Control Fed by PWM Inverters with Magnetic Saturation	Nobuyuki Imai Shigeo Morimoto Masayuki Sanada Yoji Takeda	Tochigi R&D Center Osaka Prefecture University Osaka Prefecture University Osaka Prefecture University
		Development of an Electric Motor for EV/HEV/FCV	Keiichi Yamamoto	Tochigi R&D Center
2005. 4.11~14	SAE/SAE 2005 World Congress (Detroit, USA)	Introduction of Direct Injection System to 4-Stroke Spark Ignition Engine for Motorcycle Application	Masatoshi Suzuki Kaoru Hanawa Junichiro Suzuki	Asaka R&D Center Asaka R&D Center Asaka R&D Center
		Development of Fuel Injector and Fuel Pump for a Fuel Injection System to Use in Small Motorcycles	Tatsuya Ujiie Hidetoshi Saito Minoru Ueda Shunji Akamatsu Akira Hayashi Yoshikatsu Nakano	Keihin Corp. Keihin Corp. Asaka R&D Center Asaka R&D Center Asaka R&D Center Tohoku University
		Rapid Engine Speed Control for AMT Using Two-Degree-of-Freedom Sliding Mode Algorithm	Yuji Yasui Kanao Shimojo Mitsunobu Saito Yutaka Tamagawa	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Research into Surface Improvement for Low Friction Pistons	Hidemi Ogihara	Tochigi R&D Center
		New Concept Sliding Mode Control for AMT	Kanao Shimojo Yuji Yasui Eijiro Shimabukuro Yoshihisa Iwaki	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		PCCI Operation With Early Injection of Conventional Diesel Fuel	Tomohiro Kanda Takazo Hakozaiki Tatsuya Uchimoto Junichi Hatano Naoto Kitayama Hiroshi Sono	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Development of the Intelligent Power Unit for the V6 Hybrid Midsize Sedan	Noboru Yusaku Hiroshi Kosaka	Tochigi R&D Center Tochigi R&D Center
		Development of a Hybrid System for the V6 Midsize Sedan	Kohei Hanada Masaaki Kaizuka Toshitaka Imai Susumu Ishikawa Hideki Matsuoka Hiromitsu Adachi	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Development of the IMA Motor for the V6 Hybrid Midsize Sedan	Akira Kabasawa Kazunari Takahashi	Tochigi R&D Center Tochigi R&D Center
		Development of Hexavalent Chromium-free Surface Treatment	Tsutomu Miyadera Kazunori Takikawa	Tochigi R&D Center Tochigi R&D Center
		Development of Topology Optimization Method for Reduction of Transmission Housing Weight	Yuji Ogata Satoru Suzuki Masami Hiraoka	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center

		Development of High-Efficiency CVT for Luxury Compact Vehicle	Akihito Ohashi Yoichi Sato Kaoru Kajikawa Shinji Sakuma Junji Urano Masaaki Yamaguchi Akihiro Shibahara	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Development of Direct Electromagnetic Clutch System	Rikiya Kunii Akihiro Iwazaki Shigenobu Sekiya Yutaka Kanno Katsuhiko Kitamura Yasuji Shibahata	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
2005. 4.30	JSAE/Review of Automotive Engineering Vol.26, No.2	Accurate Engine Speed Control Using Adaptive Disturbance Observer	Yuji Yasui Kanako Shimojo Mitsunobu Saito Yutaka Tamagawa	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Predictions of the Temperature Dependency of Braking Distance Using Bench Tests	Ayao Kawakami Hiroyuki Hase Tetsuya Mizone Hideki Hirai Kenji Horinouchi Yoichi Mizuno Haruyuki Suzuki	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Falken Tires, Ltd. Sumitomo Rubber Industries, Ltd. Sumitomo Rubber Industries, Ltd.
		Effects of Vehicle Motion on Cabin Thermal Environment	Akihiro Fujita Tsunehiro Saito Toru Takabayasi	Tochigi R&D Center Asahi Glass Co., Ltd. Asahi Glass Co., Ltd.
		Engine Evolution of the 4-Stroke Road Racer RC211V	Koji Minami Yoshihisa Yamaguchi	Asaka R&D Center Asaka R&D Center
2005. 5.24~25	Massachusetts Institute of Technology/MIT Consortium Program Review Meeting SEOUL (Seoul, Korea)	NV Countermeasure Technology for a Cylinder-On-Demand Engine — Development of Active Control Engine Mount	Hiroto Nemoto Hideki Matsuoka Tetsuo Mikasa	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		ROADMAP OF AUTOMOTIVE ELECTRIC POWER SYSTEMS, PART1: AUTOMOTIVE	Tadashi Fujiwara Tatsuo Teratani Tetsuro Sakamoto Masayuki Saito	Tochigi R&D Center TOYOTA MOTOR CORPORATION NISSAN MOTOR CO., LTD. YAZAKI PARTS CO., LTD.
2005. 5.29~6.2	Japanese Society of Tribologists (JAST)/ITC KOBE 2005 (International Tribology Conference) (Kobe, Japan)	Technology for Applying Solid Lubricant to the Piston Skirt of Internal Combustion Engine and Its Effects	Hidemi Ogihara	Tochigi R&D Center
2005. 5.31~6.2	UK & International Press Events/VEHICLE DYNAMICS EXPO 2005 (Stuttgart, Germany)	Development of SH-AWD System	Yoshihiro Atsumi	Tochigi R&D Center
2005. 6. 6~9	NHTSA/DOT/19th. ESV Conference (19th International Technical Conference on the Enhanced Safety of Vehicles)(Washington DC, USA)	Exploratory Study of an Airbag Concept for a Large Touring Motorcycle: Further Research Second Report	Takeshi Kuroe Hideo Namiki Satoshi Iijima	Asaka R&D Center Asaka R&D Center Asaka R&D Center
		A Computer Simulation for Motorcycle Rider Injury Evaluation in Collision	Hideo Namiki Toyokazu Nakamura Satoshi Iijima	Asaka R&D Center Asaka R&D Center Asaka R&D Center
		Development and Full-Scale Dummy Tests of a Pop-Up Hood System for Pedestrian Protection	Kaoru Nagatomi Ken Hanayama Tatsuya Ishizaki Sakae Sasaki Kazuo Matsuda	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center

		Development of a finite element model for Flex-PLI	Yuji Kikuchi Yukou Takahashi Fumie Mori	Tochigi R&D Center HONDA R&D AMERICAS, INC. OHIO CENTER PSG Co., Ltd.
		Biofidelity Evaluation for the Knee and Leg of the POLAR Pedestrian Dummy	Yukou Takahashi Yuji Kikuchi Akihiko Akiyama Masayoshi Okamoto Johan Lvarsson Dipan Bose Jeff Crandall Damien Subit JaeHo Shin	HONDA R&D AMERICAS, INC. OHIO CENTER Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center University of Virginia Center University of Virginia Center University of Virginia Center University of Virginia Center University of Virginia Center
		Development of Performance Specifications for a Pedestrian Research Dummy	Akihiko Akiyama Jeff Crandall Douglas Longhitano Ken Wiley	Tochigi R&D Center University of Virginia Center HONDA R&D AMERICAS, INC. OHIO CENTER Dynamic Research Inc.
		Effectiveness Estimation Method for Advanced Driver Assistance System and its Application to Collision Mitigation Brake System	Youichi Sugimoto Craig W. Sauer	Tochigi R&D Center Dynamic Research Inc.
		Study of load cell MDB crash tests for evaluation of frontal impact compatibility	Satoshi Takizawa Eisei Higuchi Tatsuo Iwabe Takayuki Kisai Takayuki Suzuki	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center PSG Co., Ltd. PSG Co., Ltd.
		Determining Accurate Contact Definitions in Multi-body Simulations for DOE-type Reconstructions of Head Impacts in Pedestrian Accidents	Yasuhiro Dokko Robert Anderson Alexandra Long Jack McLean	Tochigi R&D Center The University of Adelaide The University of Adelaide The University of Adelaide
		ANALYSIS OF SUDDEN NATURAL DEATHS WHILE DRIVING WITH FORENSIC AUTOPSY FINDINGS	Yasuki Motozawa Tomoko Yokoyama Masahito Hitosugi Shogo Tokudome	Tochigi R&D Center Dokkyo University School of Medicine Dokkyo University School of Medicine Dokkyo University School of Medicine
		First Results from the JAMA Human Body Model Project	Tomiji Sugimoto Kunio Yamazaki	Tochigi R&D Center Japan Automobile Research Institute
2005. 6. 7	Proceedings of GT2005/ASME Turbo Expo 2005/ASME (Reno-Tahoe, Nevada, USA)	Three Dimensional Aerodynamic Optimization for an Ultra-Low Aspect Ratio Transonic Turbine Stator Blade	Martina Hasenjager Bernhard Sendhoff Toyotaka Sonoda Toshiyuki Arima	Honda Research Institute Europe GmbH Honda Research Institute Europe GmbH Wako Nishi R&D Center Wako Research Center
2005. 6. 7~9	Society for Experimental Mechanics/2005 SEM Annual Conference & Exposition on Experimental and Applied Mechanics (Portland, Oregon, USA)	Application of Elastoviscoplastic Constitutive Equation with Craze to Three-Dimensional Issue	HiroYuki Mae	Tochigi R&D Center
2005. 6.27~30	International Conference on Computational and Mathematical Methods in Science and Engineering/CMMSE-2005 Alicante/American Institute of Mathematics & University of Alicante (Spain)	Computation of Air Flows and Motion of Environmental Pollutants over Complex Geographical Topographies	Toshiyuki Arima Yoshinori Matsuura Shinnosuke Oharu	Wako Research Center Japan Coast Guard Academy Chuo University

Editor's Postscript

I have submitted technical papers to the *Honda R&D Technical Review* several times in the past, but recently my opportunities for contact have been mostly as a peer reviewer. I have also been involved on the editing side. Now, with this issue, I am handling the Editor's Postscript. The *Honda R&D Technical Review* has become something quite familiar to me.

What I would like to do here, therefore, is to introduce my recent style in peer reviewing: (1) I read just the abstract and conclusion, but very thoroughly, and clearly identify any questions. (2) Then I have a face-to-face talk with the author (two-way communication), in which we go over and revise the abstract in detail. (3) I finalize the structure. (4) After this, I start work on the fine points of the paper. I have come to think that step (2), talking with the author, is an extremely important part of this process. This is the point where I can continue asking the author one question after another to get a clearer picture of the basic technology involved. If this can be described precisely, then the readers will be able to gain a proper understanding, as I have come to see it. What I hope to achieve by all this work is to have as many engineers as possible build up their technical, logical, objective, and universal way of thinking.

All this may seem to be placing undue emphasis on me, but I seem to be the one who has the most to learn in this process, not just in terms of technical knowledge, but also in terms of human skill. It is not just the author who learns and grows in this *Honda R&D Technical Review*. It is the peer reviewer, as well. I would like to express my gratitude for the Three Joys of the *Honda R&D Technical Review*, which are a joy to write, a joy to peer review and edit, and a joy to read.

(Shiomi)

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