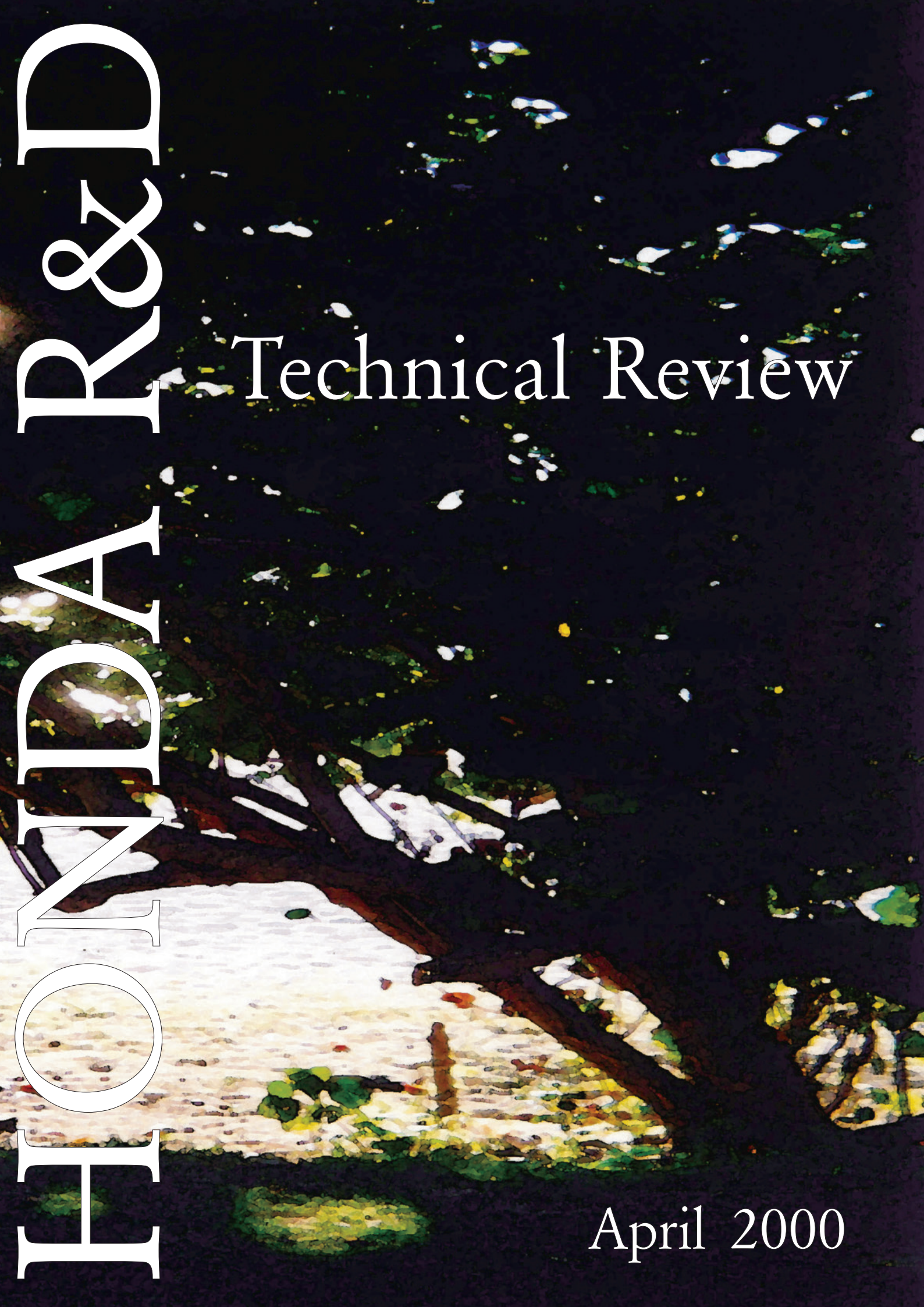




HONDA

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== COMMENT ON COVER DESIGN ==

I express infinite possibilities of new ideas and technologies with the image of beautiful and brilliant sunlight streamed through leaves.



Wako R&D Center
Daiju Nakagawa

Development of an Automatic Charger for ICVS City Pal

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Kenji UCHIBORI* Hiromichi IMAI* Toshihiro SONE*

ABSTRACT

An inductive charger and automatic connecting unit have been developed to charge the two-seater electric vehicle “ICVS-City Pal”. Inductive charge coupling is accomplished in just one action, that is, by placing a convex cylinder shaped inductive charge coupler into a concave shaped inlet. The automatic connecting unit employs a multi-joint robot arm structure to smoothly connect the unit to the City Pal regardless of slight variations in stopping position.

1. Introduction

Tests of the futuristic Honda ICVS⁽¹⁾ joint utilization-type regional transportation system began in October 1998 at the “Twin Ring Motegi” speedway located in the town of Motegi in Tochigi prefecture. During the course of that tests, the City Pal, a two-passenger electric vehicle, was driven with the aim of becoming a next-generation commuter vehicle.

This City Pal allows automatic parking and pickup before leasing and after return within an unmanned driving area⁽²⁾. In addition, as the battery capacity of the City Pal decreases, it automatically moves to a stopping position for charging to be charged by an automatic charging unit. These units are installed and operated at two locations, the south gate port within the Twin Ring Motegi speedway and in front of the Hotel Twin Ring. Fig.1 illustrates the inductive charger and automatic connecting unit.



Fig.1 Inductive charger and automatic connecting unit

This paper provides an explanation of the overall configuration of the automatic charging system, the structure and functions of the inductive charger and automatic connecting unit, and the operating sequence. This paper also clarifies the performance of the inductive charger and automatic connecting unit.

2. Automatic Charging Technology

2.1. Automatic Charging of the City Pal

The automatic charging technology for the City Pal was designed to attain a high charging rate and to operate unmanned. Described next are the automatic charger and connecting system for the automatic charging unit.

2.2. Automatic Connecting

In order to charge an electric vehicle (EV), it is necessary to connect the charging port (charging inlet) of the EV with the charge coupler on the charger.

Automatic charging requires the charging inlet and charge coupler to be moved automatically within the connectable range.

The City Pal can move to a stopping position within an unmanned port automatically and achieve consistent stopping position accuracy. In addition, the charge coupler can move within a three-dimensional envelope defined by a robot arm. After the City Pal has stopped, the charge coupler moves to a position close to the charging inlet and is then connected, after which charging of the City Pal begins.

2.3. Structure and Characteristics of the Automatic Charging Unit

2.3.1. Arrangement of Charging Inlet and Automatic Charging Unit

- Although conventional automatic charging frequently involves connecting beneath the floor,⁽³⁾ the City Pal has

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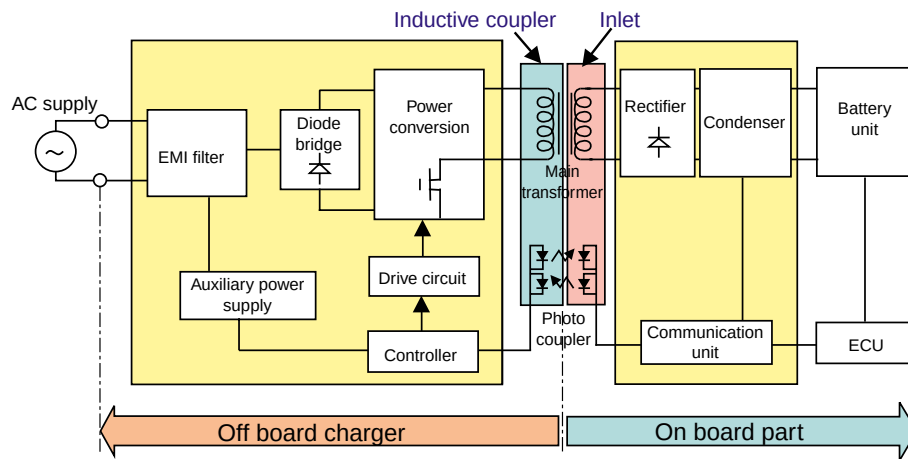


Fig.2 Inductive charging structure

a charging inlet on the side of the vehicle, thereby facilitating manual charging as well.

- The automatic charging unit can be installed above the road, enabling construction costs to be reduced by eliminating the need for embedding work.
- Since arranging the charging inlet on the side of the vehicle instead of underneath the floor results in less risk of contamination by rain and mud, the performance is stable and reliability is high.

2.3.2. Structure of the Automatic Connecting Unit

- The automatic connecting unit comprises a biaxial horizontal multi-joint mechanism, a vertical movement mechanism and a balancer mechanism (see Fig.7), thus eliminating the need for a position holding force. This structure enables the unit to follow changes in a vehicle's stopping position and maintain the connection during charging. Further, the automatic connecting unit can be easily disconnected during a power outage, malfunction and so forth.
- Connection can be performed with the driving force of a small servo motor.
- The latching mechanism on the charge coupler conserves energy, even when charging for several hours, because no energy is required to maintain the holding position.

2.3.3. Horizontal Multi-joint Structure

- Two axes of movement are provided for the horizontal multiple joints of the robot arm to enable the charge coupler to move freely within the horizontal plane, making it possible to set a wide operating range for connection.
- The joints in the robot arm allow it to be folded into a compact shape when stowed, enabling the automatic charging unit to be installed in parking lots where there is little free space.
- The charging cable runs along the robot arm, so the problem of cable slack does not occur.

2.3.4. Inductive Charging System

The charger employs an inductive charging system. As shown in Fig.2, this inductive charging system uses

electromagnetic induction to transfer electrical power through a magnetic coupling between a primary coil at the end of the charging cable and a secondary coil in the charging inlet.⁽⁴⁾ Since there is no direct connection of electrical contacts with this system, the charge coupler and charging inlet can be easily connected and made waterproof for outdoor use.

3. Automatic Charging Unit for the City Pal

3.1. Installation Sites

As shown in Fig.3, inductive chargers and automatic connecting units were installed at the South Gate of the Twin Ring Motegi speedway and in front of the Hotel Twin Ring. Automatic parking at return and delivery to the collection point as well as platooning⁽²⁾ of the City Pal mainly took place at the South Gate, with automatic charging was also provided there. Views of the installation sites are shown in Figs.4 and 5.

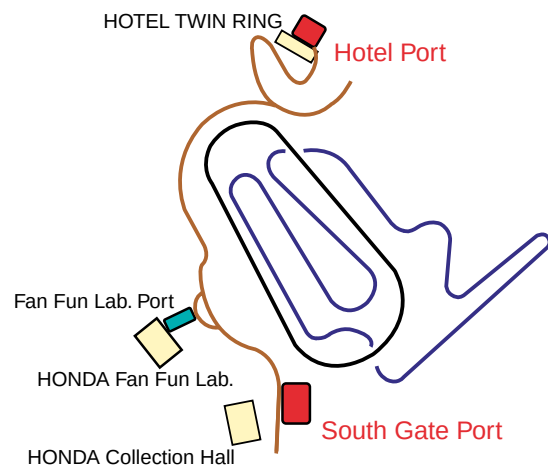


Fig.3 Automatic charger installation locations



Fig.4 South Gate Port



Fig.5 Hotel Port

3.2. Design Concept

The ICVS design concept is based on a circle. Since the ICVS project involves numerous sectors throughout the company, an overall design was employed that strongly reflects the design image of each sector. The inductive charger and automatic connecting unit also incorporate this design concept of a circle.

The housing of the arm covers the complex moving joints and cable. Although the cable is completely covered, there is sufficient free space for its movement within the housing.

In order to foster a sense of integration with the connecting unit and other ICVS, the automatic charger was designed while taking into consideration the height of the connecting unit in the standby condition. The shape is based on a circle motif, and a gray housing makes the exterior of both the automatic charger and connecting unit appear smaller. The result emphasizes that the two machines are part of a single system. Fig.6 shows image sketches of the automatic charger before development.

During the course of this development, studies were



Fig.6 Sketched image of automatic charger

conducted on the component shapes and internal arrangement of the automatic charging unit based on these image sketches. The inductive charger made it possible to save space when arranging the internal components. The automatic connecting unit was given a long vertical shape for stacking the internal components that closely matches the sketched image.

3.3. System Configuration

A block diagram of the ICVS automatic charging system is shown in Fig.7.

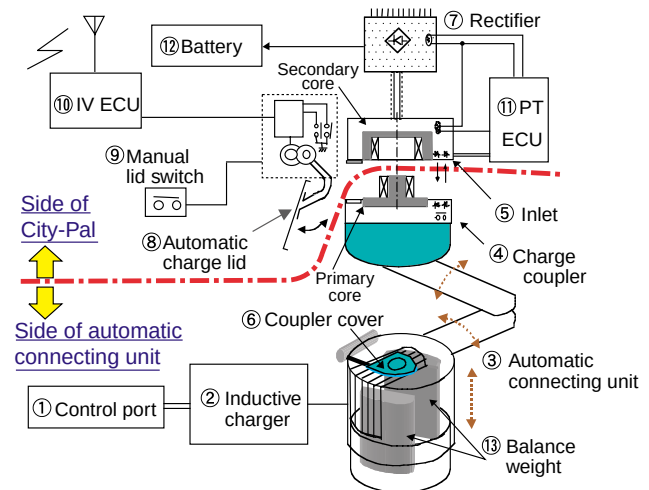


Fig.7 Automatic charging system structure

3.4. System Components

A description of the system components and their functions follows.

(1) Port Controller

The port controller controls the opening and closing of the lid, the starting of the connection and the interruption of charging, and it sends out status request commands when a vehicle is being automatically charged at the port.

(2) Inductive Automatic Charger

The inductive automatic charger is connected to the port controller by an RS422A line resistant to external interference, and performs charging according to charging commands from the port controller. The power circuit converts AC 200 V to 100 kHz high-frequency power to generate an output of 6 kW. This charger contains controllers for charging and for the connecting unit.

(3) Connecting Unit

The connecting unit is a horizontal biaxial and vertical uniaxial horizontal multi-joint robot with a charge coupler mounted on the end of the robot arm. This unit has an ultrasonic sensor for detecting the presence of a vehicle and measuring the relative distance, and a balance weight for maintaining position when the servo is off.

(4) Charge coupler⁽⁴⁾

As shown in Fig.8, the charge coupler has a primary coil around the outside of a cylindrical ferrite core. An infrared photoelectric sensor for detecting the location of the charging inlet on the side of the vehicle, and a light-receiving element for two-way optical communication

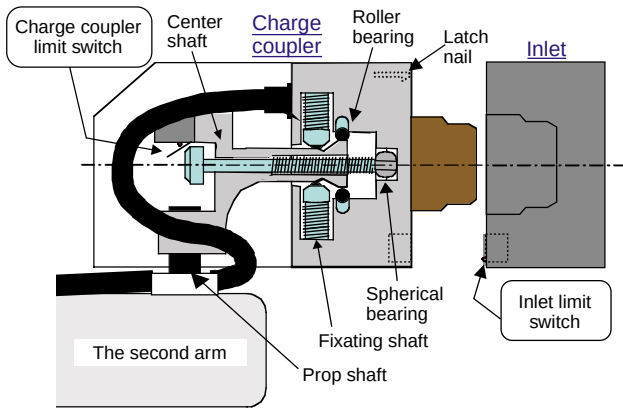


Fig.8 Set of charging coupler and inlet

between the charger and vehicle are mounted on the surface of the charge coupler. The charge coupler contains a small motorized fan that provides forced air cooling from the charge coupler to the charging inlet. A microswitch for detecting connection with the charging inlet is installed on the rear end of the charge coupler support shaft. The periphery of the charge coupler is provided with a latch to prevent disconnection by external factors during charging.

(5) Charging Inlet

The charging inlet has a secondary coil around the inside of the core. During connection, the overlapping coil arrangement results in a large coupling coefficient between the primary and secondary sides. The power transfer efficiency is 99.6%. A reflecting plate for the infrared photoelectric sensor, a light-receiving element for optical communication, and a microswitch to detect the connected condition are installed on the surface of the charging inlet.

(6) Coupler Cover

This covers the charge coupler when not connected to protect it from the effects of water and dust. During the connecting operation, the cover is open and closed by the coupler cover drive motor that operates through a crossed helical gearbox. It also has a retracting function.

(7) Rectifier

The rectifier rectifies the high-frequency power supplied through the charge coupler and charging inlet, and converts it to direct current power for charging the battery.

(8) Motorized Charging Lid

The lid protects against the ingress of rain and mud, damage caused by flying stones and tampering. This lid is motorized, and is automatically opened and closed according to a command from the IV ECU in coordination with the start and end of charging.

(9) Manual Lid Switch

The lid can be opened and closed during manual charging by the manual lid switch located in the floor of the trunk.

(10) IV ECU (Intelligent Vehicle ECU)

This controls all of the vehicle operation systems, including automatic charging. It also controls motorized lid opening and closing commands as well as charging request commands sent to the port controller.

(11) PT ECU (Power Train ECU)

This performs overall control of the EV drive system and battery management. When the residual battery charge

becomes low, a command prompting charging is sent to the port controller through the IV ECU. In addition, this also checks the temperature of the charge coupler and rectifier to monitor any abnormalities.

(12) Batteries⁽⁵⁾

Twenty-four 50 Ah Ni-MH batteries are used to generate a total voltage of DC 288 V. The total battery weight is 250 kg.

(13) Balance Weight

The weight of the arm and charge coupler are balanced by two equal weights installed inside the connecting unit to enable a small servo motor to be used.

3.5. Operating Sequence of the Automatic Connecting Unit

A detailed description of the operating sequence of the connecting unit during automatic charging is provided next.

3.5.1. Measurement of the Relative Distance

- The charging lid opens in response to a charge request from the port controller.
- The automatic connecting unit rises from the standby position (see Fig.9) to the operation reference position.
- The relative distance between the vehicle and the connecting unit is then measured at that position by an ultrasonic sensor.



Fig.9 Stand-by posture

3.5.2. Opening Operation of the Coupler Cover

- The coupler cover that protects the charge coupler during standby opens automatically.
- The coupler cover can be retracted by a crossed helical gear, and it is housed in the upper portion of the connecting unit.

3.5.3. Search Operation for the Charge Coupler

- The charge coupler moves to the charging inlet search starting position according to the measured relative distance between the vehicle and the connecting unit.
- The charge coupler begins a search sweep at the rate of 200 mm/s in front and behind the vehicle while maintaining a fixed distance between itself and the vehicle so as not to interfere with the charging lid (see Fig.10). The approximate location of the charging inlet reflecting plate is detected by an infrared photoelectric sensor installed on the front of the charge coupler.
- The charge coupler then moves to a position roughly corresponding to the location of the reflecting plate while maintaining a fixed distance from the vehicle.
- A precision search sweep is then performed at the rate of 20 mm/s to determine the planar coordinates of the reflecting plate.



Fig.10 Search action

3.5.4. Connecting Operation

- After the charge coupler position has been corrected by the amount of offset from the center line between the location of the reflecting plate and the charging inlet, it approaches the charging inlet at the rate of 100 mm/s.
- The charge coupler slows down to a speed of 20 mm/s about 50 mm in front of the position where it connects with the charging inlet, after which the charge coupler and charging inlet are connected.
- This action activates the connecting switch on the charging inlet to inform the vehicle that connection has been completed.

3.5.5. Swinging of the Charge Coupler and Completion of Connection

- When the charge coupler is pushed in towards the charging inlet by the servo motor (see Fig.11), the bearing of the charge coupler is disengaged from the fixing shaft to enable the charge coupler, which had previously been fixed on the arm, to move freely.
- The freed charge coupler is then connected at the correct angle to the vehicle and securely locked to the charging inlet.
- At the same time, the center shaft that has been pushed in by the charge coupler depresses the charge coupler limit switch, and the resulting signal interrupts movement of the charge coupler while simultaneously freeing up the servo.
- The charge coupler is finally stabilized by the resistance of the robot arm while inserted into the charging inlet. This completes the connecting operation and charging then starts.



Fig.11 Connecting action

4. Performance of the Automatic Charging Device

4.1. Inductive Charger

Fig.12 shows the charging output characteristics of the inductive charger.

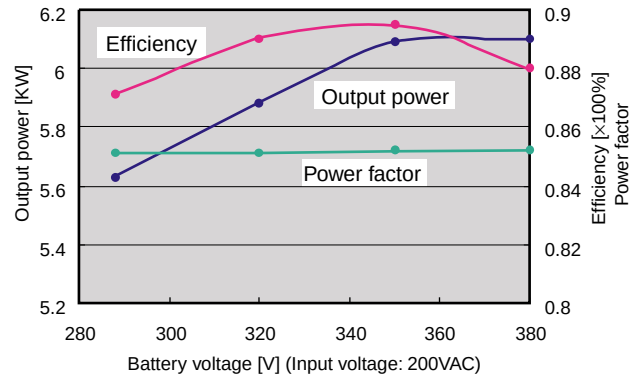


Fig.12 Characteristics of inductive charger for City Pal

A charging output of 6 kW is achieved with a battery voltage of DC 335 V or higher. As a result of this high output, the capacity of the City Pal Ni-MH batteries can be charged from 20% to 100% in roughly 2 hours. A high vehicle operating efficiency can be achieved as a result of this short charging time. The charging efficiency is 87% or better over nearly the entire range of the battery working voltage, which is higher than the figure for conventional on-board chargers.

On the other hand, the power factor is of the order of 0.85, which is somewhat lower than the figure for conventional chargers. This has resulted from the priority placed on minimizing the size of the charger, but can be improved by adding a power factor improvement circuit in the future.

4.2. Automatic Connecting Unit

Charging by the automatic charging device can be performed when the charging inlet installed inside the motorized lid of a City Pal parked at the automatic charging port is within range of the defined three-dimensional stop position. The conventions front and rear of the vehicle, left and right, up and down, and the pitch, yaw and roll angles are defined as shown in Fig.13.

The allowable ranges for the relative position and connection angle between the charge coupler and charging inlet within which automatic connection is possible are shown in Table 1 and Fig.14. The actual measured values are also shown. This device has been demonstrated to have a level of performance that is more than adequate for practical use based on the allowable ranges and measured values.

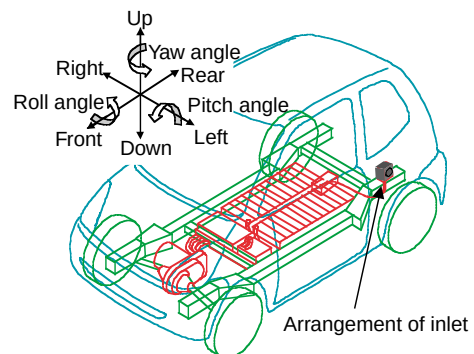
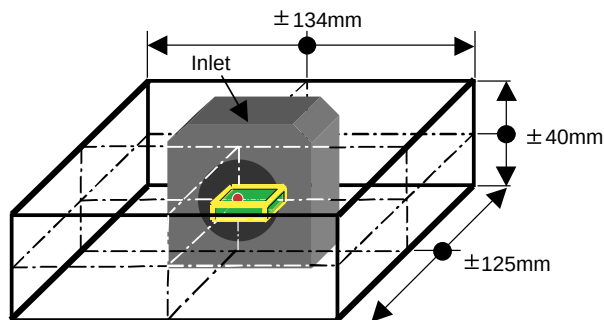


Fig.13 Definitions of directions and angles

Table 1 Test data of automatic connecting unit

Items		Connectable range of automatic connect unit itself	Automatic stoppable range of City-Pal
Difference of relative position	Front-rear	$\pm 134\text{mm}$	+7 mm -20
	Left-right	$\pm 125\text{mm}$	+5 mm -37
	Up-down	$\pm 40\text{mm}$	+4 mm -2
Difference of connected angle	Roll angle	$\pm 5^\circ$	+0.5° -2.5°
	Pitch angle	$\pm 5^\circ$	+0.5° -1.5°
	Yaw angle	$\pm 5^\circ$	+0.16°



Black frame: Connectable range of automatic connecting unit itself
 Yellow frame: Automatic stoppable range of City-Pal
 Red point: Ideal center point of connect

Fig.14 Test results for connectable range

5. Conclusion

An automatic charging system was developed that uses an inductive charger and automatic connecting unit. The following results were achieved through this development.

- The use of an inductive charging system enabled a cylindrical shape to be adopted for the charge coupler and charging inlet. This allows connection to be achieved in a single operation to provide contact between the charge coupler and charging inlet.
- The use of a horizontal biaxial and vertical uniaxial horizontal multi-joint robot arm for the automatic connecting unit allows connection to be made regardless of slight variations in stopping positions of the City Pal without causing practical problems.

These achievements have made it possible to minimize the charging time for the City Pal, maximize operating efficiency and allow unmanned operation.

6. Prospects

When considering the ideal form of automatic charging in the future, there are still many problems that must be overcome to adapt the method to manual operation for home use due to the allowable range of stopping position.

When considering the development of electric vehicles in the near future, it is predicted that the charging equipment will use a higher power level and be larger in size to provide high-speed charging of vehicles with a greater travel range,

and there will likely be a need for automatic charging based on these requirements. It will be necessary to conduct more studies on practical automatic charging systems that are compatible with manual operation.

Acknowledgments

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Development of a Motor Assist System for a Hybrid Car – INSIGHT

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ABSTRACT

A motor-assist system has been developed and employed for the ‘Insight’ hybrid car. The system consists of an internal combustion engine as a primary power source, and a motor placed around the engine crankshaft. This construction is highly compact and offers more flexibility for a power plant layout.

The system’s functions include absorption of braking energy, idling-stop, driving power assistance, and power supply for the 12V electrical system. A proper energy management method for various driving modes has been established by combining these functions and fuel economy is significantly improved as a result. Other control features include, an active motor vibration control, which compensates for idling vibration that is unique to 3-cylinder engine, and sophisticated transient torque control between assist and regenerating modes, which harmonizes the characteristics of the two power sources to ensure better driving comfort.

1. Introduction

Automotive emissions reduction as a means of reducing the greenhouse gas carbon dioxide (CO₂) is an important issue facing the automobile industry. One approach taken to achieve this goal is to improve fuel economy with hybrid power plants, which combine an engine with an electric motor. During braking, a conventional gasoline engine wastes the vehicle’s motion energy by converting it to heat energy via the brakes. Fuel economy is also lowered by fuel consumption during idling.

A hybrid system, in contrast, is capable of:

- Recovering braking energy
- Idle-stopping and restarting with the motor
- Highly efficient recharging and assistance

Various types of hybrid systems have been proposed, such as the series type, which, based on the electric automobile, uses the engine to generate electric power; the parallel type, which centers on the engine and uses a motor for auxiliary power; and, other designs that strive for an intermediate approach⁽¹⁾⁽²⁾.

2. The Goals of Development

Although capable of using energy effectively as stated above, a hybrid car can also have lower fuel economy due to increased system weight.

To achieve a proper balance between the effective use of energy and weight, the authors developed a hybrid system dubbed Integrated Motor Assist (IMA).

Other objectives are to ensure smooth driving by harmonizing the two driving power supplies, achieve a

superior idle-stopping feature that ensures smooth restarting, and constructing system control that makes maximum use of energy.

3. System Configuration

In developing the system, three major points were emphasized regarding system configuration, component layout, and component design.

First, because of body shape restrictions, a compact component layout and thorough system streamlining were needed to achieve the enhanced aerodynamic performance necessary to improve fuel economy.

Second, decreasing the weight of the components themselves was necessary to improve fuel economy.

Third, the high-voltage system necessitated measures that would assure safety (e.g., prevent electrocution) not only during normal use but also during maintenance and in the event of collision.

To meet these requirements by keeping wiring as short as possible and thereby assuring high-voltage safety, the Integrated Power control Unit (IPU), as shown in Figs.1 and 2 was placed outside of the crushable zone, concentrating the high-voltage components in the vehicle rear.

The cooling system chosen for inside the IPU employs a lightweight forced-air cooling system rather than water cooling, and features cooling passages that are designed to minimize passage resistance and to prevent the entry of water and such.

The motor developed is a thin motor designed to be placed around the crankshaft, between the engine and the transmission.

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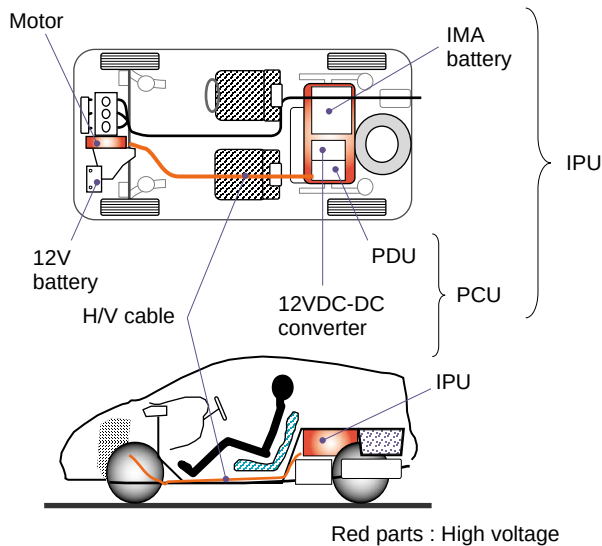


Fig.1 IMA package

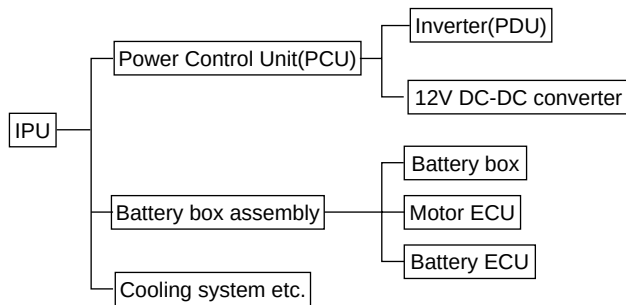


Fig.2 IPU component

1000 rpm and maximum power of 10 kW at 3000 rpm (Table 1). To fit inside the engine room, the engine was made more compact, while the motor was equipped with high-thermal-resistance neodymium magnets to fit in the 60-mm-wide space between the engine and transmission - highly severe thermal conditions. Natural air cooling is used.

Table 1 IMA specifications

Engine specifications	
Construction	Liquid-cooled 3-cylinder in-line
Valve train	Chain-driven SOHC 2intake/2 exhaust valves
Capacity (cc)	995
Bore x Stroke (mm)	72.0 × 81.5
Compression ratio	10.8
Max. power (PS/rpm)	70/6700
Max. torque (kgm/rpm)	9.4/4800
Motor specifications	
Electric current	AC
Rated voltage (V)	144
Max. power (kW/rpm)	10.0/3000 (MT)
	9.2/2000 (CVT)
Max. torque (Nm/rpm)	49.0/1000

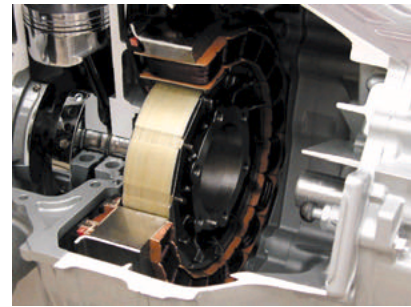


Fig.3 IMA motor

3.1. The Engine

As the primary power source, the engine incorporates numerous new technologies designed to improve fuel economy⁽³⁾.

This lean-burn engine has a 1.0-liter displacement. A 3-cylinder design was chosen in order to reduce mechanical loss and such. To improve combustion efficiency, existing VTEC lean-burn technology was given a further evolved combustion chamber shape.

Other weight reducing technologies adopted include a magnesium oil pan, the use of resin in such parts as the intake manifold and headlight cover, and an aluminum exhaust manifold integrated with the cylinder heads.

3.2. The Motor

An alternating current synchronous (DC brushless) motor was chosen for its superior efficiency. Bearings were eliminated from the design to reduce mechanical loss, and the motor was placed around the engine crankshaft to ensure compactness.

To achieve power performance comparable to that of a 1.5-liter gasoline engine-equipped automobile, motor output in the assist mode provides a maximum torque of 49 Nm at

3.3 The Battery

The battery used is a custom Ni-MH battery for hybrid vehicles (Table 2 and Fig.4). It is designed for use under various conditions, and was provided with the optimum battery capacity for the electric energy of motor assistance and regeneration.

The fan motor used to cool and warm up the battery is a brushless type that assures maintainability and reduces onboard noise levels. This switchable motor has two speeds: high and low.

Mounted atop the battery box is an Electronic Control Unit (ECU) that controls the motor and monitors remaining battery capacity.

Table 2 Battery specifications

Type	Nickel metal hydride (Ni-MH)
Cell voltage (V)	1.2
Quantity (cells)	120 (6cells × 20modules)
Connecting method	In series
Capacity (Ah)	6.5 (3-hour discharge rate)

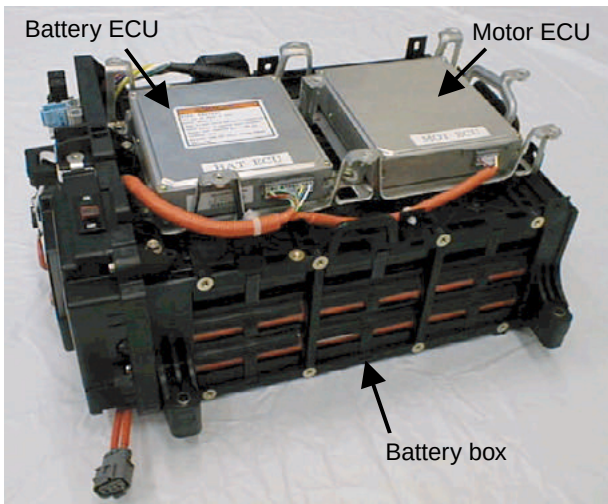


Fig.4 Battery box assembly

3.4 The Power Control Unit

The power control unit comprises a high-efficiency drive and regeneration Power Drive Unit (PDU) and a DC-DC step-down converter for 12V power supply generation, both located on either side of the cooling heat sink case (Fig.5). This forced air cooled structure enables a compact design and high-efficiency cooling.

The PDU is also made even more compact and lightweight by integrating six motor power switching elements into a single module.

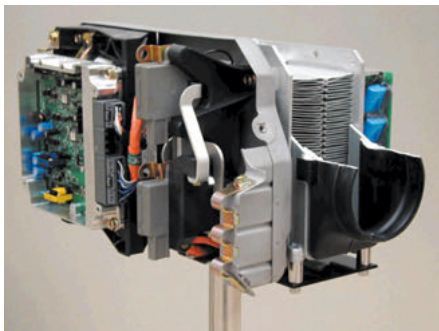


Fig.5 PCU (Power Control Unit)

3.5. ECUs

Enhancements such as a magnesium frame make the motor ECU and battery ECU approximately 50% lighter than an ECU with an aluminum frame.

Together, this compact component layout, decreased weight component enhancements, and cooling system streamlining make possible the package shown in Figs.6 and 7. This package helps lower the weight of the entire IMA system to less than 80 kg, or under 10% of vehicle weight. Further, concentrating the IPU package in the rear creates a structure that assures high-voltage safety, even in the event of collision.



Fig.6 IMA power unit



Fig.7 Rear layout of IMA system

4. Control System

4.1. Required Functions

The basic functions required by this system are energy absorption during deceleration, idle-stopping, motor assist to supplement engine output during acceleration, and generation of a 12-volt system's load.

Other requirements include motor operation that seems natural to the driver, and the ability to disengage the system to permit driving on engine power alone in the event of failure, for instance.

The minimum system configuration needed to achieve these functions is shown in Fig.8.

The functions of each ECU are listed below.

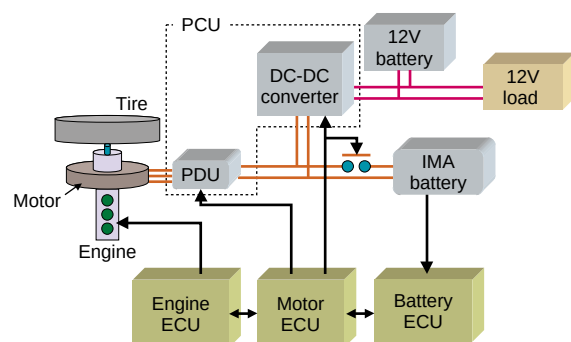


Fig.8 IMA System block diagram

1) Engine ECU

Engine control, control mode determination, assist/regeneration level calculation, idle-stopping determination, failure detection, etc.

2) Motor ECU

Motor control (assist/regeneration, engine starting), control mode determination, high-voltage power distribution control, battery protection control, idling vibration damping control, control status indication.

PDU cooling, DC-DC converter cooling, failure detection, etc.

3) Battery ECU

State Of Charge (SOC, i.e., charge level) calculation, battery protection requirement output, temperature control, failure detection, etc.

4.2. Driving Conditions and the Flow of Energy

The IMA system determines the IMA control mode and the motor's assist or regeneration level according to engine speed, vehicle speed, throttle opening, gear position, SOC and other vehicle information.

The flow of energy in the IMA system comprises a total of six directions (Fig.9).

- 1) Motor assist
- 2) IMA battery charging
- 3) 12V load generation
- 4) Supplying the 12V load from the IMA battery
- 5) Supplying the 12V power with the DC-DC converter

6) Supplying the 12V power from the 12V battery

These six directions of energy flow correspond to six energy patterns, according to driving mode and SOC.

The list below explains which energy pattern corresponds to which energy flows.

Pattern 1: Driving the motor with the IMA battery, e.g., assist during acceleration and engine starting after idle-stopping.

..... Energy flows (1), (4), (5)

Pattern 2: Absorption of deceleration energy and charging during cruising and idling.

..... Energy flows (2), (3), (5)

Pattern 3: Supplying only the DC-DC converter power (12V system load) when charging of the IMA battery is unnecessary (e.g., at maximum SOC).

..... Energy flows (3), (5)

Pattern 4: Supplying the 12V system load with the IMA battery during idle-stopping.

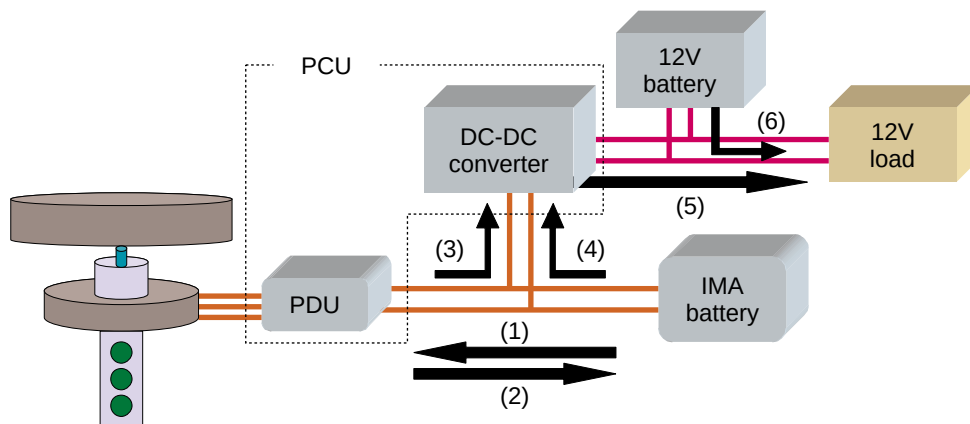
..... Energy flows (4), (5)

Pattern 5: Supplying the 12V load with the 12V battery when starting the engine with the IMA motor.

..... Energy flows (1), (6)

Pattern 6: Starting the engine with the starter (as in an ordinary gasoline engine-equipped vehicle) in cases where starting with the IMA motor is not possible, e.g., at extremely low temperatures.

..... Energy flow (6)



IMA mode	Condition	(1) Assist	(2) IMA battery Charging	(3) 12V from Motor	(4) 12V from IMA battery	(5) 12V from DC-DC	(6) 12V from 12v Battery	Energy pattern
1. Assist	Normal	●			●	●		Pattern 1
	Low SOC			●		●		Pattern 3
2. Decelerate	Normal		●	●		●		Pattern 2
	High SOC			●		●		Pattern 3
3. Cruise	Normal		●	●		●		Pattern 2
	High SOC			●		●		Pattern 3
4. Idling	Normal		●	●		●		Pattern 2
	High SOC			●		●		Pattern 3
5. Idling stop					●	●		Pattern 4
6. Engine start	Ignition ON	●					●	Pattern 5
	Idling stop	●			●	●		Pattern 1
	Starter						●	Pattern 6

Fig.9 Energy flow of the IMA system

The energy flow as determined by the driving condition is displayed on the IMA meter (Fig.10).

The SOC meter displays the battery's SOC with a 20-segment display, and the assist/charge meter communicates the IMA's operational status to the driver by displaying the assist or charging power in real time.

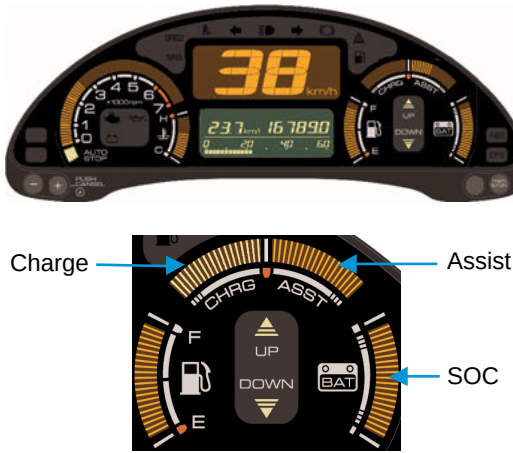


Fig.10 Hybrid meter

4.3. Energy Management

A hybrid car's SOC must be maintained within a certain range in order for motor driving and regeneration to function. To this end, energy management based on driving conditions is incorporated as described below to maintain the proper balance between SOC and driving power/regeneration power control.

1) Assist/deceleration modes

In the assist and deceleration modes, in which the vehicle's driving and braking power is emphasized, power is provided exactly as requested by the PDU's DC unit, regardless of the 12V load level.

2) Cruising/idling modes

In the cruising and idling modes, the battery power balance is more important than motor output. Because SOC variation must be kept to a minimum, the battery input element's DC power is controlled. This minimizes battery charge/discharge loss so that the motor's regenerated power can be used for the vehicle system's 12V load. In the example in Fig.11, the charging requirement is 2 kW and DC-DC converter consumption is 0.5 kW. In this case, feedback is provided so as to maintain the battery side's power at 2 kW, effectively enabling the motor to generate 2.5 kW of power and thereby providing the power consumed by the DC-DC converter.

3) Idle-stopping mode

While the vehicle is stopped, unnecessary fuel consumption is avoided by stopping the engine rather than idling. In this idle-stopping mode, the IMA battery provides the power needs for the 12V load through the DC-DC converter.

4) Engine starting mode

In this mode, the engine is started by the IMA motor

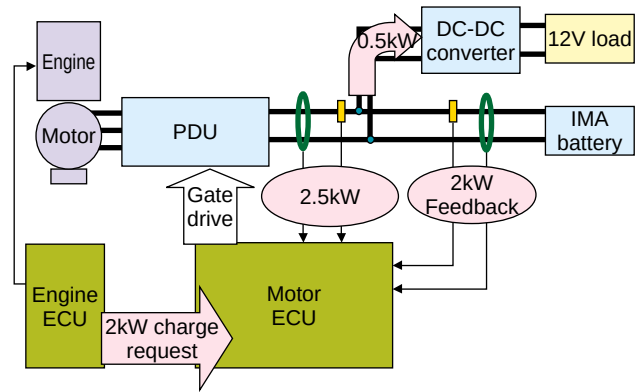


Fig.11 Energy flow for conventional electric load

using the regeneration energy that is generated during deceleration. More durable than the starter motor, the IMA motor assures faster starting and lower levels of noise and vibration. The torque pattern in Fig.12 was adopted to enable the engine to quickly achieve idling speed. Figs.13 and 14, which show how engine speed rises after a cold start with

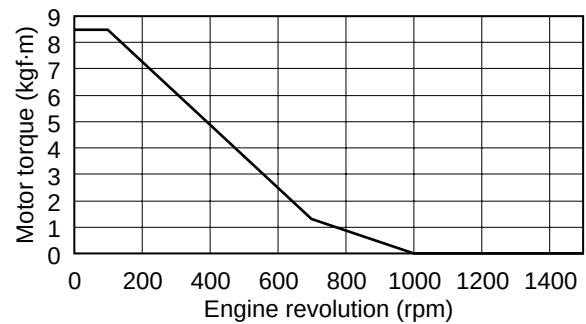


Fig.12 IMA motor torque at engine start

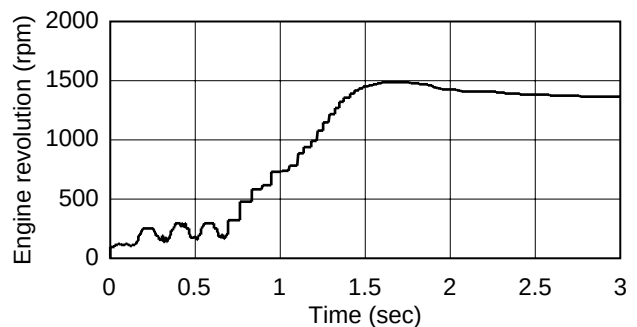


Fig.13 Engine revolution chart (starter motor)

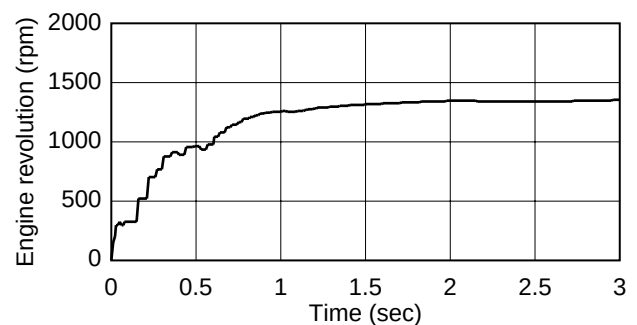


Fig.14 Engine revolution chart (IMA motor)

the starter motor and IMA motor, demonstrate how the IMA motor is capable of smooth engine restart after idle-stopping.

4.4. Motor Output Control

Improving the controllability of vehicle movement requires sufficient responsiveness and stability in motor output control. Although required output depends on such factors as vehicle speed, engine speed, and SOC, the actual command value is calculated by switching between open control and feedback control, taking into account the fidelity of the motor's actual output.

As Fig.15 shows, when the difference between current actual output and required output is large, open control is initiated to provide a larger change in torque per unit time. This is continued until the predetermined differential is reached, after which power feedback control, which also includes the DC-DC converter, is carried out until the target value is achieved.

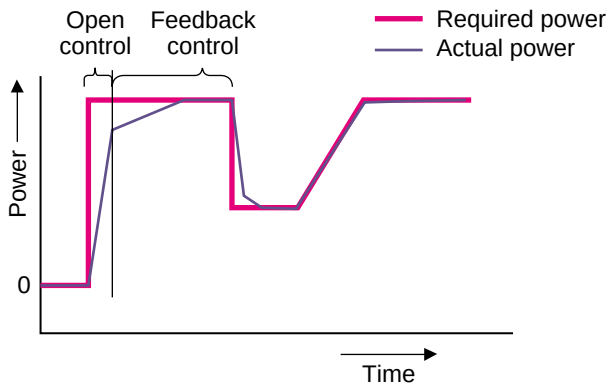


Fig.15 Method of torque feedback

In feedback control, feedback gain is also changed according to the degree of change in the requested output. A large change is interpreted as a transient state, and feedback gain is increased to improve responsiveness.

In times other than when the transient state has been identified, feedback processing functions to adjust feedback gain according to the control mode (e.g., deceleration, assist, cruising, idle) as shown in Table 3, thus stabilizing vehicle movement.

Table 3 Feedback gain of control mode

Control mode	Deceleration	Assist	Cruise	Idling
Gain				
Gain of proportional control	Kp1	Kp2	Kp3	Kp4
Gain of integral control	Ki1	Ki2	Ki3	Ki4
	High		Middle	Low

5. Using Motor Control to Damp Idling Vibration

The Insight's 3-cylinder engine generates vibration of n th order $\times 1\frac{1}{2}$ of engine speed. Therefore, the IMA system, taking advantage of the fact that the motor is placed around the engine crankshaft, counters this 1.5-order vibration, which is the primary component of vibration, by applying a torque whose phase is reversed relative to this torque, thereby damping vehicle vibration during idling (Fig.16). This process is referred to as damping control.

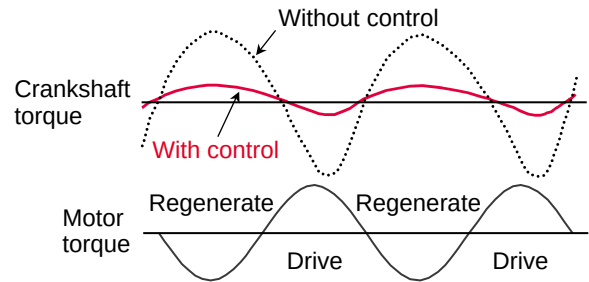


Fig.16 Principle of damping control

The example in Fig.17 gives actual engine torque, its 1.5-order component, and the torque to apply during damping control. This demonstrates how it is possible to use motor torque to cancel out 1.5-order vibration, which is the primary component of 3-cylinder engine vibration.

Damping control determines torque phase according to the parameters of engine load and speed, using the engine's top dead point signal as a benchmark. The amplitude of the torque to apply is similarly determined according to the parameters of engine load and speed.

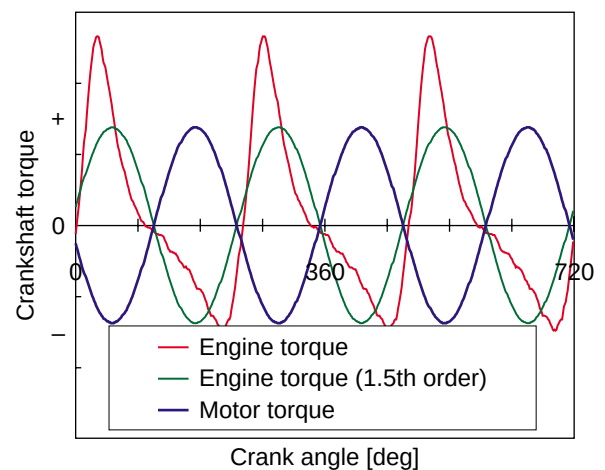


Fig.17 Example of engine and motor torque

Generating the waveform of the torque to apply requires crankshaft angle information, which is obtained from the top dead point signal.

The torque used in damping control and the regeneration torque used for battery charging are superimposed when they are output. This relationship is shown in Fig.18; the effects of damping control, in Table 4.

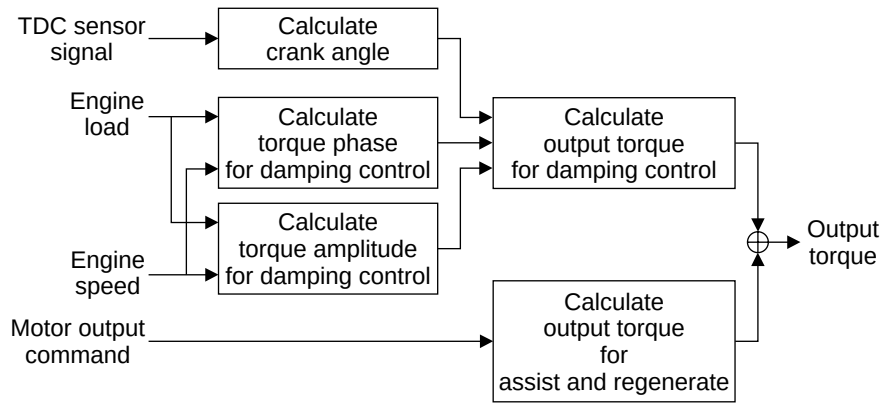


Fig.18 Block diagram of damping control

Table 4 Effect of damping control

Measuring point	Effect of damping control
Vibration of steering wheel	-3dB
Vibration of seat	-5dB
Sound pressure of vehicle compartment	-5dB

6. Ensuring Battery Longevity

A battery's SOC, voltage, and temperature must be maintained within their predetermined limits during actual use in order to ensure battery longevity.

The IMA system uses current sensors, battery voltage sensors, and battery box internal temperature sensors to:

- 1) Keep the SOC within the appropriate range
- 2) Keep input and output power within the upper and lower limits for battery voltage (voltage power save feature)
- 3) Restrict input and output power according to battery temperature (temperature power save feature)

6.1. Voltage Power Save

When the battery voltage exceeds the upper or lower voltage thresholds, which are determined by battery current and battery temperature, the IMA system reduces the motor's current output command value gradually, so that vehicle motion does not change abruptly. Once battery voltage returns to within the predetermined range, the restricted output is gradually returned to its original level. Fig.19 gives a block diagram of this.

6.2. Temperature Power Save

To prevent overheating and assure sufficient accuracy in SOC determination at extreme temperatures, motor output limits are controlled according to battery temperature to keep battery power below its predetermined upper limit (this is the batter temperature power save feature). With the settings shown in Fig.20, output control makes it possible to keep battery temperature within an appropriate range.

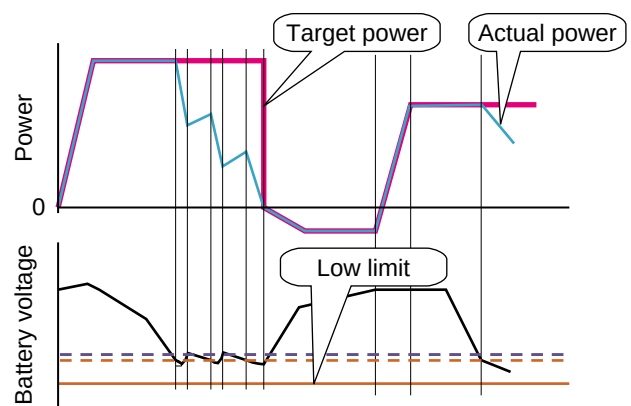


Fig.19 Power save control by battery voltage

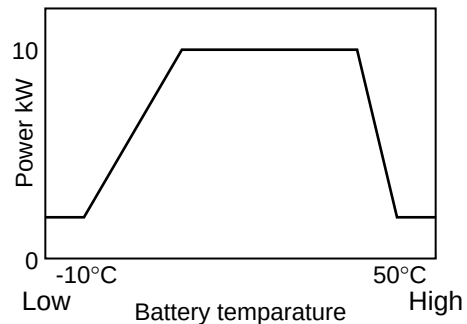


Fig.20 Power save action by battery temperature

7. Conclusion

The following achievements were made with a hybrid system whose motor is designed to function especially for engine assistance during acceleration and regeneration braking during deceleration:

- Energy management based on driving conditions and battery SOC helped improve fuel economy.
- A compact packaging designed for high-voltage safety and superior mountability was achieved.
- An assist/regeneration system that harmonizes engine

driving power with motor driving power was achieved.

- Motor-based damping control of idling vibration was shown to damp vibrations of -3 dB to -5 dB.
- The motor assist system was shown to be capable of smooth restarting after idle-stopping.

Acknowledgments

The authors wish to express their deepest gratitude to all those within and outside the company who provided invaluable assistance with the development of this system.

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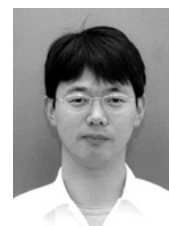
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Development of the Ultra-thin DC Brushless Motor for a Hybrid Car – INSIGHT

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ABSTRACT

We developed an ultra-thin DC brushless motor for use as the auxiliary drive of the ultra-low-fuel-consumption hybrid car, “INSIGHT”.

Use of an ultra-thin design for this motor overcomes restrictions of the layout construction between the engine and the transmission. The ultra-thin design was achieved using a concentrated winding for the salient-pole of the split stator and a centralized distribution bus ring. Utilization of these technologies resulted in a maximum output performance of 10 kW at a maximum torque of 83.3 N-m while keeping the width of the electromagnetic motor circuits to a mere 60 mm. In addition, since the use of this arrangement for the motor subjects it to the effects of water and dirt just as with the clutch compartment, a joint chamber and labyrinth construction is used to form a drain. Meanwhile a stator cover and disk sensor are employed to prevent foreign matter from entering. This design achieves a motor capable of withstanding a harsh ambient environment.

1. Introduction

With the growing importance of environmental issues involving air quality, the INSIGHT ultra-low-fuel-consumption hybrid car offering a fuel consumption of 35 km/liter (10-15 mode) has been developed as one solution to these problems. The Integrated Motor Assist (IMA) system installed in this vehicle employs a parallel hybrid system which uses the engine as the main power source and an electric motor as an auxiliary power source, making it possible to reduce the size and weight of the entire system, including the electric motor⁽¹⁾.

The objectives that were set for the development of this motor involved reducing its thickness to allow transverse mounting of the power train of the IMA system in the engine compartment of a 1.5-liter class vehicle while satisfying required vehicle performance, and ensuring environmental resistance equivalent to that of the engine. This report describes how these objectives were met.

2. Motor Functions and Objectives

Fig.1 shows the engine and electric motor of the IMA system, while Fig.2 shows the system configuration. Since this system is of the parallel hybrid type, the motor has the functions of assisting the engine that serves as the main power source according to the performance being demanded, regenerating braking energy in the form of electrical power, and generating the 12 V load component. In addition, the motor also restarts the engine after it has stopped idling⁽²⁾.



Fig.1 IMA power unit

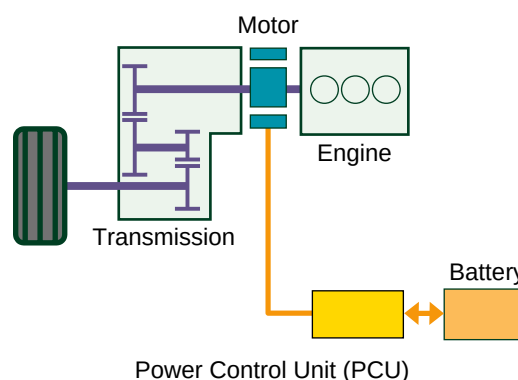


Fig.2 IMA system construction

* Tochigi R&D Center

It was necessary to reduce the size and weight of the entire power train to realize a vehicle capable of fulfilling these functions and of achieving a fuel consumption of 35 km/liter (10-15 mode). In order to accomplish this, the motor is coupled directly to the engine and positioned between the engine and transmission. During the course of motor development, emphasis was placed on a thin design that would provide good resistance to the operating environment, leading to the following technical objectives.

- (1) Compact size, requiring the development of a thin motor that allows the power train to be installed transversely in the engine compartment of a 1.5-liter class vehicle.
- (2) Environmental resistance that ensures durability in the operating position between the engine and transmission.

3. Technology for Developing the Ultra-thin Motor

Fig.3 shows the construction of the stator. The stator is a salient-pole, concentrated-winding split type, incorporating a centralized distribution bus ring and stator holder, and this construction formed the technological basis for developing the ultra-thin motor.

Fig.4 shows examples of a conventional type of winding and the newly developed type of winding. Generally, there are many cases in which the stator is formed into an integrated unit, and wave winding or salient-pole concentrated winding is typically used. There are problems

associated with developing an ultra-thin motor incorporating either of these types of winding.

Wave winding requires an unacceptably wide area for the winding transfer portion and a long time for the winding process. In the case of an integrated stator with salient-pole concentrated winding, it is necessary to ensure adequate space for the winding nozzle to move as shown in Fig.5, otherwise the range over which windings can be provided is limited. This problem was solved by the construction described next.

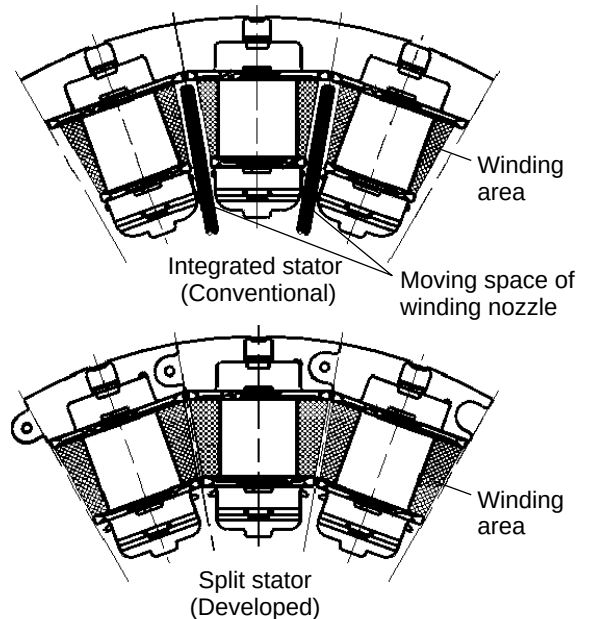


Fig.5 Comparison of winding area

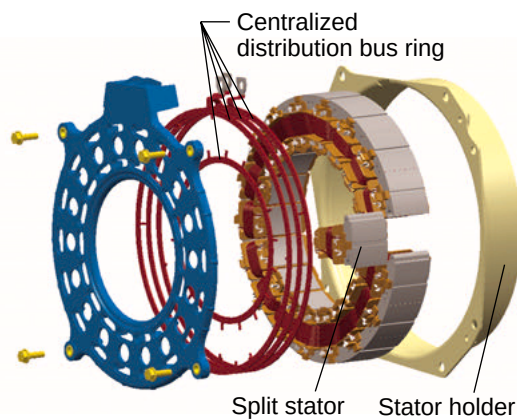


Fig.3 Stator construction

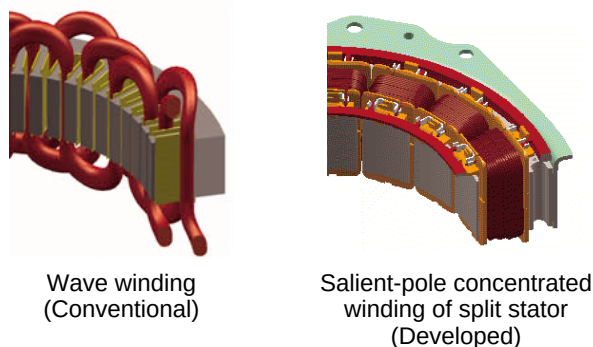


Fig.4 Comparison of winding

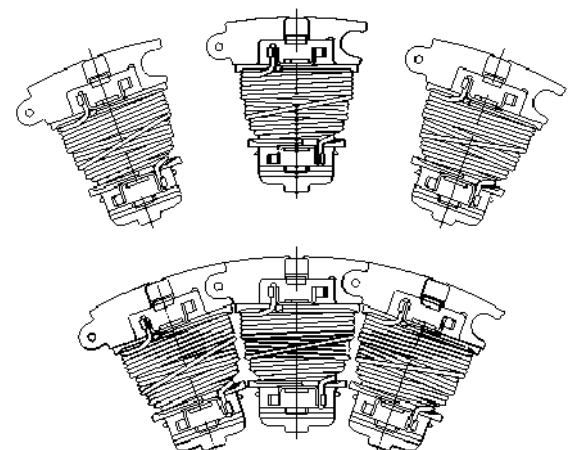


Fig.6 State of winding with split stator

that of an integrated stator. Moreover, by maintaining the alignment with salient-pole concentrated winding, increasing the space factor of the windings and combining with a centralized distribution bus ring, the width of the electromagnetic motor circuits could be reduced without compromising the vehicle torque demand.

This type of stator also offers better manufacturability than an integrated stator, starting with the yield of the silicon steel plate. In order to improve the yield, the stator was split and blanked as shown in Fig.7. The lamination and fixing processes are also relatively simple due to the use of internal die caulking. In addition, since the split stator allows winding work to be done for each pole, the size of the equipment can be minimized.

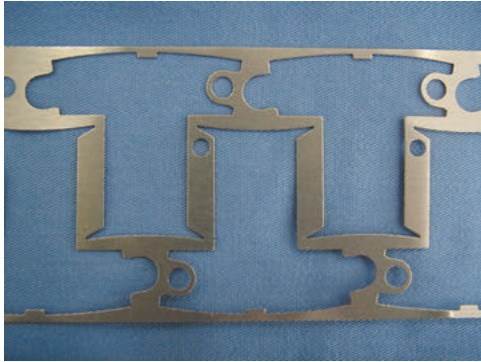


Fig.7 Blanking of flat rolled silicon steel

3.2. Centralized Distribution Bus Ring

This 3-phase motor uses six windings connected in parallel for each phase as shown by the equivalent circuit in Fig.8. The red portion of this figure corresponds to the centralized distribution bus ring. This component is stamped from a copper plate, and each winding is then connected. Since the six windings would need to be fixed with the wiring in the form of a crossover for each phase in a conventional motor, this would result in manufacturing difficulties. However, the use of a centralized distribution bus ring enables the work to be simplified by fusing only at the terminal portion. Fig.9 shows a comparison of these wiring diagrams with and without a centralized distribution bus ring.

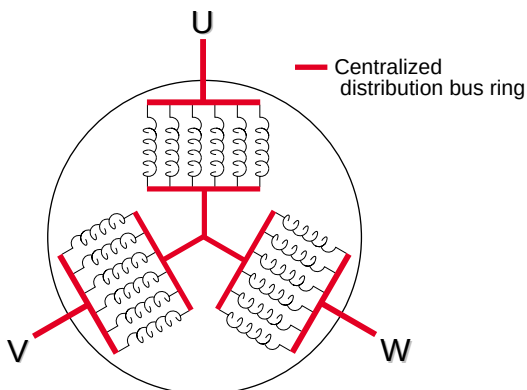


Fig.8 Equivalent circuit of the motor

The crossover portions can be seen to be much simpler as a result of employing the centralized distribution bus ring.

The three-phase centralized distribution bus ring is stacked as shown in Fig.10 to minimize the thickness. Although insulation between the phases could present a problem with this stacked structure, the required level of resistance in the operating environment in terms of heat, vibration and dust required fluorine to be selected for the insulation coating material. The thickness of the fluorine coating was determined from the results of durability and reliability tests, using the data in Fig.11 as the evaluation criteria. The results of these tests under the harsh conditions of hot and cold, vibration and dust clearly proved that the fluorine coating would be optimally suitable. These technological solutions made it possible to reduce the

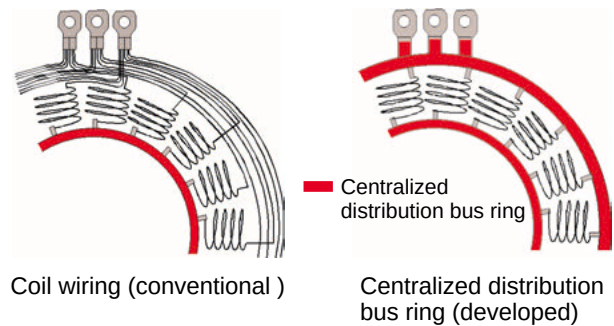


Fig.9 Comparison of wiring

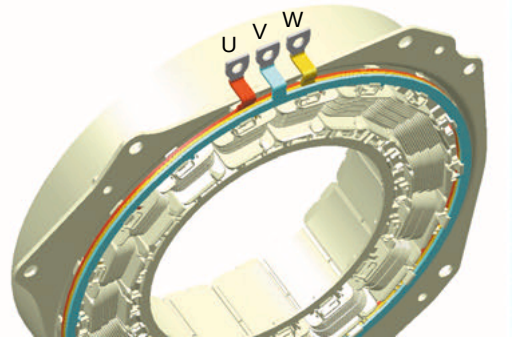


Fig.10 Layout of centralized distribution bus ring

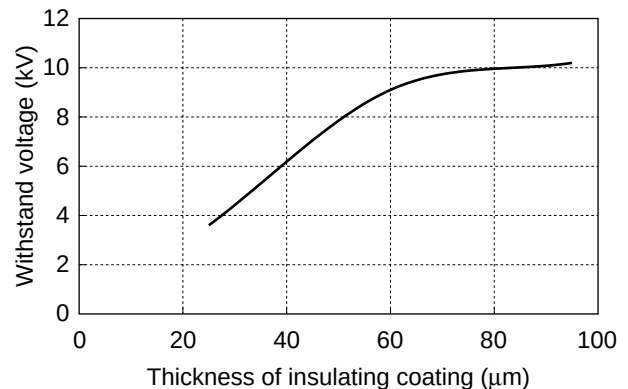


Fig.11 Isolation performance curve

thickness of a conventional motor by approximately 40% to produce an ultra-thin motor having a thickness of only 60 mm. Fig.12 shows the hybrid power train featuring this new electric motor, while Table 1 and Fig.13 show the motor specifications.

4. Durability in a Harsh Operating Environment

Since the motor is located inside the clutch housing between the engine and transmission, it is difficult to create a watertight operating environment. Thus, a design was

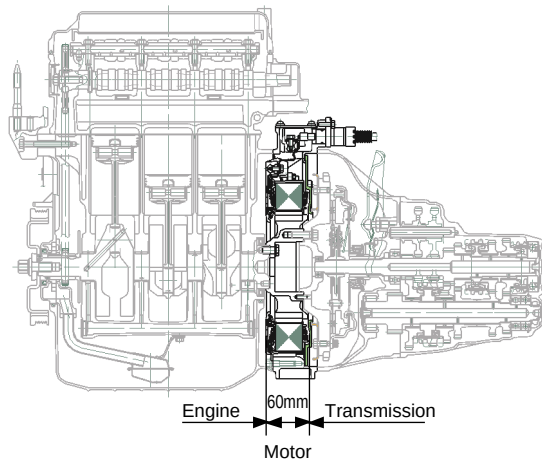


Fig.12 Cross section of hybrid power train

Table 1 Specifications of motor

Motor type	DC brushless	
Number of phase	3 phases	
Number of pole	12 poles	
Number of slot	18 slots	
Weight	18.6kg	
Maximum output power	Starter	83.3N-m/2kW
	Assist	49.0N-m/10kW
	Regeneration	83.3N-m/10kW

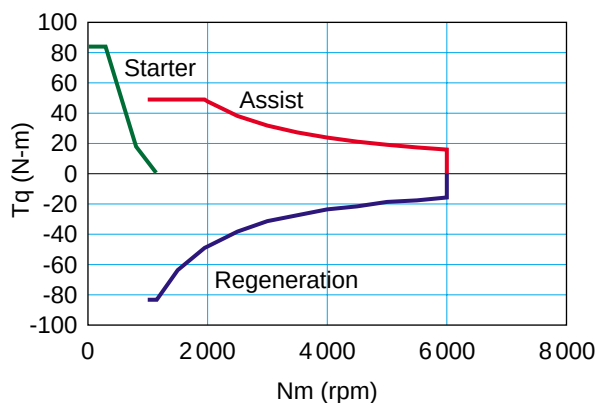


Fig.13 Motor power performance curve

developed to make it difficult for water, oil, etc. to enter this housing, while also allowing rapid drainage in the case of ingress. It was also necessary to prevent any contamination of the motor electromagnetic circuits by foreign matter, despite the fact that the motor would need to tolerate clutch lining dust and other magnetic substances. These problems were solved in the manner described next.

4.1. Water Sealing and Drainage

The motor is provided with a drain and breather to minimize the effect of water and oil ingress. Particular attention was focused on drainage, since the bottom of the motor is at nearly the lowest point of the vehicle in its mounted position. In the case of traveling in wet conditions, water is sprayed from the tires on to the motor housing. This spray will penetrate into the motor if drainage is provided simply through holes in the motor housing.

Fig.14 shows the drainage method employed in the early stages of development. Although a pipe was inserted into the motor housing and this pipe discharged away from the direction of vehicle travel, the high-pressure water spray penetrated the motor through this drain pipe. The design shown in Fig.15 resolved this problem by incorporating a baffled plenum chamber between the drain outlet and the hole leading into the motor.

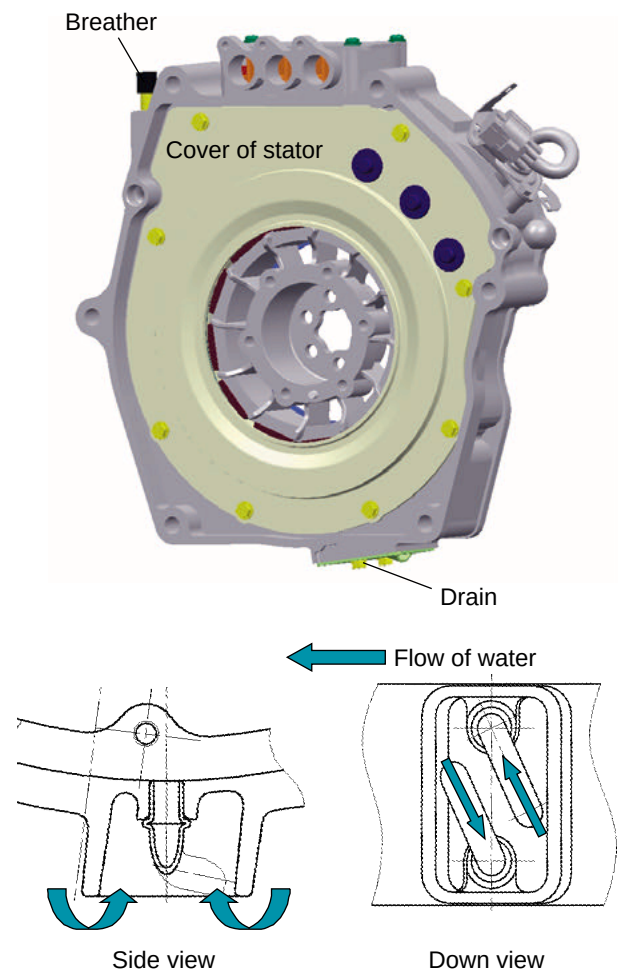


Fig.14 Prototype of drain construction

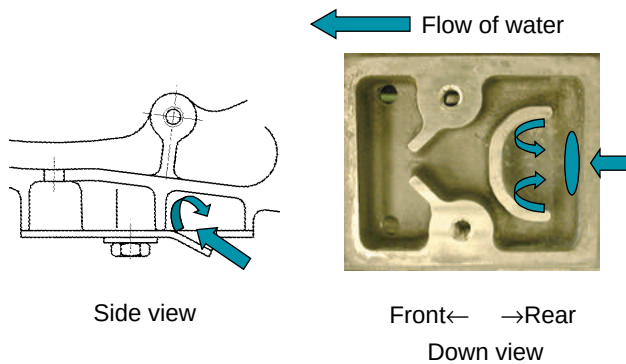


Fig.15 Production model of drain construction

4.2. Sealing Against Clutch Lining Dust, Metallic Particles and Other Foreign Matter

The clutch housing in which the motor is located is basically open to the atmosphere. Accordingly, external dust and other foreign matter is able to enter the clutch chamber. In addition, routine operation of the clutch generates clutch lining dust and metallic particles within the clutch housing. Any substantial amount of this dust and foreign matter accumulating on the windings and magnets of the motor would have a detrimental effect on its performance. A labyrinth seal is therefore provided between the stator cover and commutation sensor disk as shown in Fig.16 to inhibit the entry of dust and foreign matter from the clutch housing. This sealing system has been rigorously tested over the service life of the vehicle.

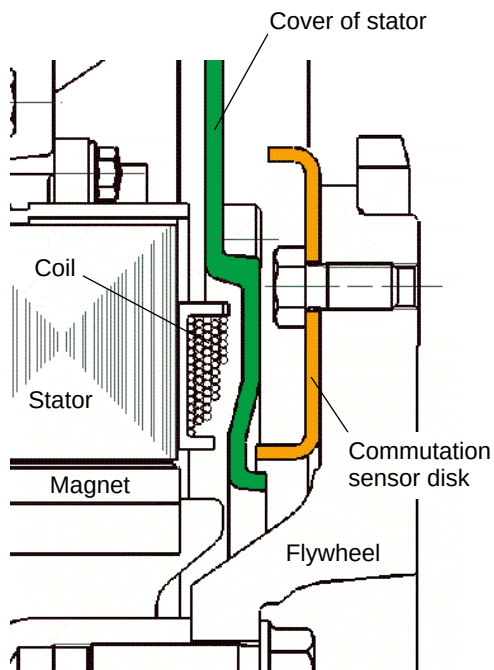


Fig.16 Enlarged cross section of motor & transmission

5. Conclusion

Novel approaches were taken to develop an ultra-thin motor that would operate reliably under harsh conditions to produce a compact hybrid power train.

(1) Salient-pole, concentrated-winding, split stator

The use of this split stator results in an approximately 70% increase in winding range when compared to a conventional integrated stator.

(2) Centralized distribution bus ring

The conventional wiring and anchoring processes used for winding crossovers have been simplified by only needing to perform fusing work.

(3) Effective water sealing and drainage

The use of a chamber structure and labyrinth structure are combined.

(4) Sealing against clutch lining dust and foreign matter

A labyrinth seal is provided between the stator cover and commutation sensor disk.

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Development of a 4-wheel Electric Wheelchair, MonPal ML100

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ABSTRACT

The MonPal ML100 is a 4-wheel electric motor-driven wheelchair developed with an aim toward achieving personal mobility while at the same time providing fun, freedom from care and comfort for those with restricted walking capabilities. It is characterized by a rearward tapering design intended to prevent the rear wheels from scraping and snagging.

The power train, which is an integrated transmission and motor type, and the control system were specially developed to provide improved slope-climbing capabilities.

Various features are incorporated in the operating system. These include an emergency stop function whereby if a driver grips the drive lever firmly, the vehicle will automatically come to stop; a sudden start prevention function that is employed to prevent mis-operation related incidents; and, a buzzer notification function.

1. Introduction

The number of elderly people continues to grow in Japan. It is predicted that in the year 2000, one of every six persons in Japan will be 65 years of age or older, and by the year 2010, this ratio will have increased to one in every five persons.

Communication with society is necessary for leading a vibrant life, even after we reach an old age and are no longer able to walk freely. Providing a means to move freely is important for achieving this communication.

Handle-type electric wheelchairs have begun to be used in recent years as vehicles for the elderly. According to the Road Traffic Law, electric wheelchairs are included in the category of wheelchairs for physically challenged persons. Since the driver is treated as a pedestrian, a driver's license is not required. In addition, according to the Japan Industrial Standards definitions, electric wheelchairs are classified into standard types and handle types primarily according to their appearance and application. Handle types are further classified into three wheel types and four wheel types. Since their appearance on the market in 1985, the number of these handle-type wheelchairs has increased each year. Despite this increase, however, the rate of their proliferation is still low in comparison with latent needs.

As a possible reason for this, apart from the present state of the transportation infrastructure, there are many cases in which persons decide not to purchase an electric wheelchair because they feel embarrassed, even though they would like to have one.

The "MonPal ML100" four-wheel handle-type electric wheelchair was developed to eliminate this feeling of

embarrassment and to provide support for persons with restricted mobility thereby encouraging them to go out into society.

2. Aim of the Development

During the course of development of the MonPal ML100, emphasis was placed on the following points in order to realize enjoyable personal mobility and to provide peace of mind, comfort and simplicity:

- (1) A design that offers both fun and peace of mind
- (2) A power train that provides reliable power even on slopes
- (3) A smooth and comfortable travel control system

3. Overview

The appearance of the MonPal ML100 is shown in Fig.1, while the design concept is shown in Fig.2 and the major specifications are in Table 1.

The key styling concept of "enjoyable personal mobility" is aimed at proposing a new style that is able to bring out the enjoyment and fun of the vehicle, while also enabling anyone to use it with peace of mind. The essential features of the design are as listed below.

- A shape that provides a wide step and minimizes the height of the front end of the body.
- A hood design that reliably makes the driver aware of the front of the vehicle, while providing a sense of stability for the legs and feet.
- Emphasis on driving enjoyment by employing a seat with three-dimensional curvature for better holding and large rear tires.

* Asaka Higashi R & D Center



Fig.1 External view of MonPal ML100



Fig.2 Design concept

Table 1 Major specifications

Items		Specifications
Overall length	(mm)	1190
Overall width	(mm)	680
Overall height	(mm)	1080
Tread	(mm)	Front: 590/Rear: 530
Curb weight	(kg)	94
Motor		DC brush type
Batteries (rated at 5 hours between recharges)		12V 35AH × 2 (sealed)
Tire sizes		Front: 3.00-4 Rear: 70/100-8
Drive systems		Direct 2-rear-wheel drive system (with differential)
Brake systems		Regenerative brake Electromagnetic brake Mechanical brake
Maximum speed	(km/h)	6
Maximum continuous Running distance (6km/h, flat road)	(km)	25

The overall layout of the MonPal ML100 is shown in Fig.3. The chassis dimensions comply with the Japan Industrial Standards classification for “electric wheelchairs” and are compatible with the National Public Safety Commission model type certification. Driver space has been ensured by arranging the control unit at the front of the vehicle, and the battery, charger and power train at the rear of the vehicle. The floor is low (150 mm above the ground) and flat for easy accessibility and ample driver space. A rearward tapering design is employed in which the rear tread is narrower than the front tread to reduce the occurrence of scraping and snagging of the rear wheels when traveling in confined spaces.

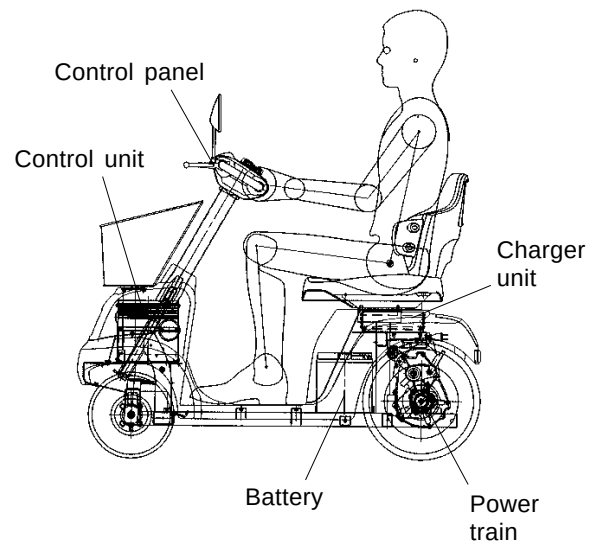


Fig.3 Layout of MonPal ML100

The power train was specially developed to be housed within the rear tread, and is an integrated single unit comprised of a flat motor and transmission.

The seat is specially shaped to support the lumbar region from both sides to provide secure holding against centrifugal force generated during turning and for the easy access required by persons with a weak lower back and legs. All controls are centrally located on the handle, and are arranged for convenient operation within easy reach of the driver's hands. The drive control in particular employs a system that allows traveling and stopping to be performed with a single lever preventing fatigue even when the lever is held for long periods during travel. In addition, a function is also provided to stop the vehicle when the drive lever is gripped firmly by the driver. Full consideration has been given to ensure peace of mind while driving the vehicle by providing a function against operation error that prevents the vehicle from suddenly starting and informs the driver when an error has been made, and one that slows the vehicle down automatically during turning.

4. Component Technology

4.1. Power Train

The appearance of the power train is shown in Fig.4, while its construction is shown in Fig.5.

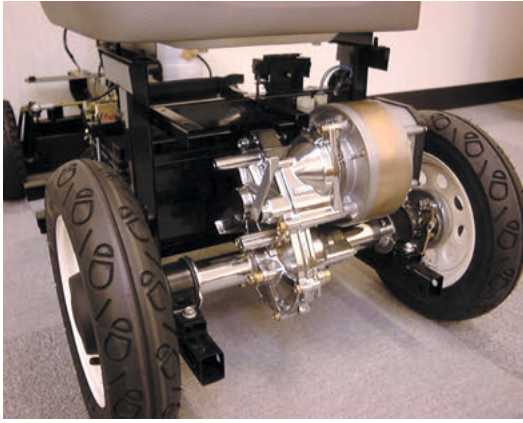


Fig.4 Power train

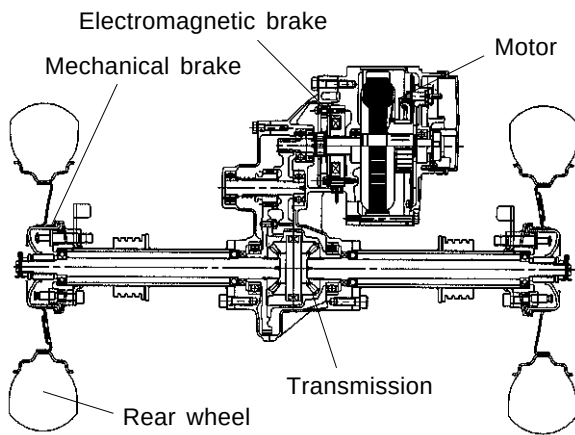


Fig.5 Layout of power train

The power train is a three parallel shaft type which integrates a flat motor into a single unit with the transmission and accommodates the primary reduction gear on the motor shaft. The braking system incorporated in the power train consists of an electromagnetic brake in the primary reduction gear unit that is activated automatically when the vehicle is stopped, a mechanical brake on each end of the rear axle that is operated manually, and a regenerative brake in the form of the motor itself that maintains a constant braking effect when descending a slope.

4.1.1. Transmission

A new low-noise, lightweight transmission was developed that is ideally suited for use in electric wheelchairs. The reduction gearbox incorporates a three-shaft secondary speed reducer to ensure that drive power is efficiently transferred to the large-diameter tires. The drive gear is formed on the input shaft itself as shown in Fig.6 with 10 helically shaped gear teeth. The transmission specifications are listed in Table 2.

The gearbox noise level and wear have been minimized by optimizing the tooth shape and gear trace of each gear. In particular, rather than being formed by a normal grinding process, the motor input shaft gears are subjected to two processes: induction hardening and cutting by and ultra-hard cutter (skiving).

The gear case is produced by “waved rib forming” to



Fig.6 Input shaft and gear

Table 2 Specification of transmission

Type	3 parallel shaft reduction gear
Input revolution (rpm)	Forward: 0~2 832 Reverse: 0~983
Reduction ratio	29.3
Clutch type	Constant mesh and manual separation
Dry weight (kg)	7.8

minimize the weight and provide high rigidity with the superior stress flow that results. The standard wall thickness of the case is only 2 mm, resulting in a 20% reduction in weight compared with previous types. This waved rib shape is shown in Fig.7.

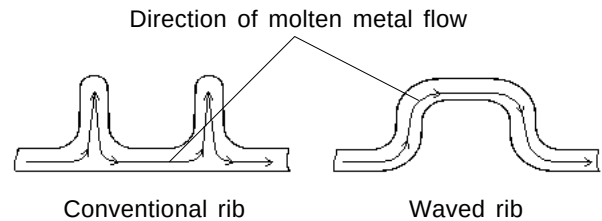


Fig.7 Rib shape

The clutch mechanism incorporates an inverse taper formed by a special rolling process on the clutch engagement portion to ensure reliable transfer of motive power during normal travel. The clutch release mechanism employs a cam and detent, which provides a feeling of shifting continuity operating on a common axis to minimize operating friction. The construction of the clutch is shown in Fig.8.

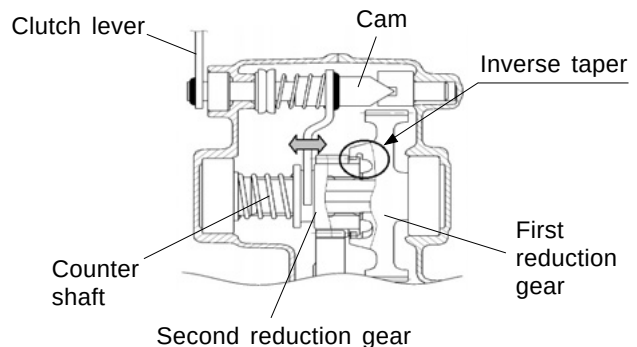


Fig.8 Clutch

In terms of production technology, various methods were employed as processing reduction mechanism. These include: the integration of three components in one model through the use of an optimum die construction and flow simulation; the adoption of a “differential automatic alignment mechanism” that utilizes the meshing alignment force of the pinion gear; and directly inserting the shaft into the die cast skin. The structure of the differential is shown in Fig.9.

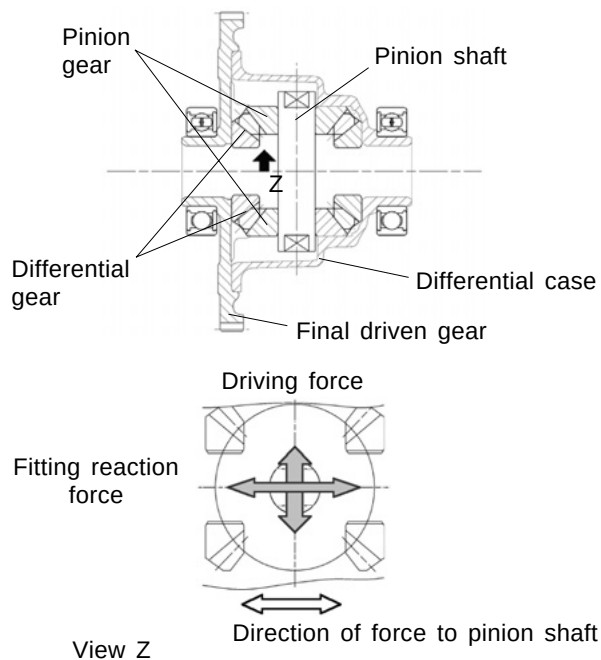


Fig.9 Differential

4.1.2. Motor

The flat disc motor has a flatness ratio of 1:8.75 in contrast to the ratio of cumulative thickness to armature outer diameter of 1:1 in typical motors. The motor specifications are shown in Table 3.

Table 3 Specifications of motor

Motor	Rated power (V/W)	24/360
	Armature (mm)	Ø140×16 (31 slots)
	Winding	Wave class B
Sensor	Revolution	2-phase magnetic pickup
	Temperature	Thermistor

The relatively large diameter of the motor makes it possible to incorporate 31 poles, enabling the voltage between segments to be maintained at a low level.

In addition, the use of a large-diameter armature provides good heat dissipation and high overload characteristics to ensure adequate power even when climbing slopes.

4.2. Control System

A 16-bit micro-controller is integrated into a single unit with the motor driver. The control unit is composed of the core “travel control unit,” the “indicator unit” that provides the interface with the driver, and the “power control unit” that is linked with the charger and is responsible for overall system management. The system diagram is shown in Fig.10, while the inside of the control unit is shown in Fig.11.

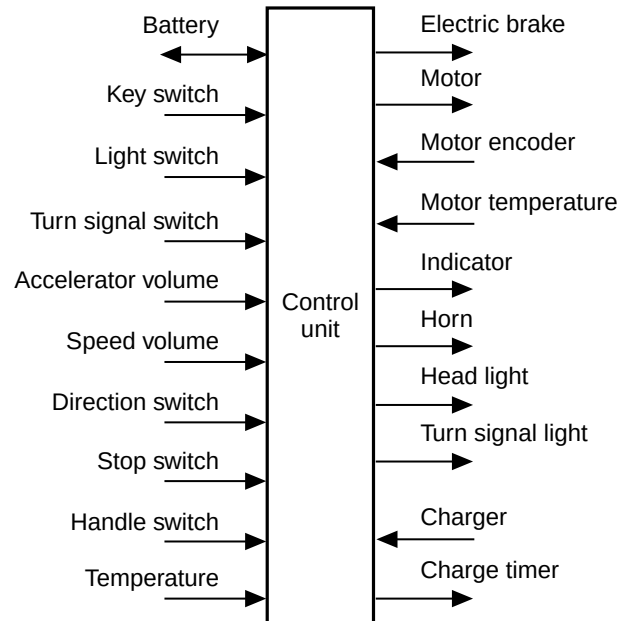


Fig.10 System diagram

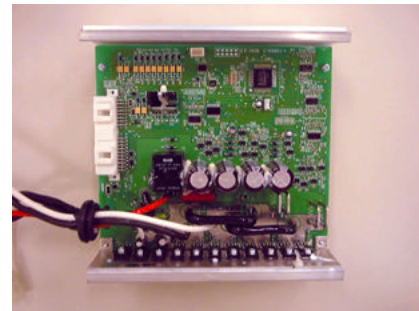


Fig.11 Control unit

4.2.1. Motor Driver

The motor driver employs a full-bridge configuration consisting of two parallel FETs to control forward and reverse rotation of the motor.

A finned heat sink is incorporated into the extruded aluminum case to provide high cooling efficiency and thereby gain maximum motor output.

4.2.2. Control Unit

(1) Travel Control

1. Constant Speed Control

Digital PI control is used in which the rotational speed and rotation direction are fed back from a Hall IC rotation sensor contained in the motor.

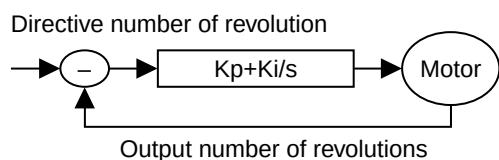


Fig.12 Speed control system

2. Optimization of the Acceleration and Deceleration Patterns

The profiles of the acceleration pattern to the target speed as set with the speed setting knob, and the deceleration pattern when the drive lever is opened, are set to 5 and 10 steps, respectively, for smooth starting and gentle deceleration.

3. Feel-Easy Functions

The travel control system incorporates the feel-easy functions required for use primarily by the elderly.

- The automatic stop function is activated by tightly gripping the drive lever. This quickly stops the vehicle whenever the driver wants to stop suddenly.
- The sudden start prevention function prevents the vehicle from moving forward suddenly when the power is turned on while gripping the drive lever.
- The automatic deceleration when turning function automatically slows the vehicle down to 4 km/h when the steering handle is turned.

4. Alarm and Protective Functions

Various alarm and protective functions are provided, including “overload detection” and “motor overheating detection” that can occur when climbing steep slopes, “motor lock detection” if the motor becomes locked such as when traveling over a bump, and “low battery detection” that prevents excessive discharging of the battery.

(2) Control Indicators

Various types of information are conveyed to the driver by 5 LEDs installed on the control panel.

1. Battery Power Level Indicator

This calculates the residual battery capacity from the battery voltage and current and displays the information in six levels. This indicator returns to the fully charged display when charging complete.

2. Charging Indicator

This indicates the progress of charging in terms of the battery voltage and elapsed time.

3. Error Mode Indicator

This indicates fifteen types of errors that can occur, including a defective rotation sensor and defective charger.

4. Maintenance History Indicator

This provides a history of the five most recent malfunctions that occurred to facilitate dealer inspections.

The travel distance, number of times charged and charging time are also indicated by LEDs.

5. Headlight Indicator

The left LED flashes to indicate that the headlight is on. This prevents the battery from being discharged

due to inadvertently leaving the headlight on.

Examples of these indications are shown in Table 4, while the appearance of the control panel is shown in Fig.13.

(3) Power Control

Table 4 Indication

Condition	Indication	Comment
Battery power	●●●●●●	
Charging	●●●○●○	
Error mode	●○●○●●	Motor lock
Maintenance history	●○●○●○	Charge cycle: under 50
Headlight	○●●●○●	Lighting on

● Lighting on ○ Lighting off ◎ Flashing



Fig.13 Control panel

1. Charger

The charger is a constant voltage, constant current type. The charging voltage range is 29.6-32.3 V, and the current is 5 A. The output voltage is set according to the outside air temperature to enable reliable charging under all temperature conditions. The charger has been certified by the Electrical Appliance and Material Control Law.

2. Charger Management

A system is employed that allows the selection of three charging modes corresponding to the battery status to prevent battery deterioration caused by overcharging from a low level of discharge. These charging modes are shown next.

*Normal charging mode

Charging is performed for 5 hours under timer control after the set voltage has been reached.

*Quick timing mode

Charging is performed for 2.5 hours under timer control in case the set voltage is reached within 2 hours.

*Overcharging prevention mode

Charging is quickly terminated the charging current falls to 1 A or lower within 1 minute. The set voltage that operates the charging timer is corrected for temperature based on an outside air temperature sensor.

Additional power saving features have been adopted such as an alarm function that operates when the power switch is on and a function that automatically turns off the headlight or turn signals when either has been inadvertently left on.

3. Travel prevention when charging

A function is provided that activates a buzzer alarm and prevents the vehicle from being moved forward by operating the drive lever while the power plug is inserted into an AC power supply and the power switch is turned on.

4. Battery

A lead-acid battery is used that is ideally suited for recharging. The battery is sealed, making it maintenance-free.

5. Conclusion

The MonPal ML100 is an electric wheelchair possessing functions that make it well-suited to areas having many slopes. It supports the leading of an enjoyable life by users with otherwise-impaired mobility. Future products will evolve in response to the changing needs of society.

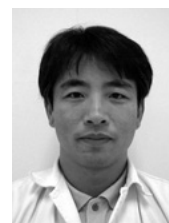
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Development of a Closed Crankcase System Engine for Motorcycles

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Nobuaki INUMARU*

ABSTRACT

A compact and lightweight single cylinder engine was developed for the off-road motorcycle XR650R. The engine features a closed crankcase. This engine system addresses the issue of deteriorated oil discharge prevention performance caused by pressure fluctuations in the crankcase, an issue common to single piston, large displacement engines. A construction was adopted whereby a bulkhead divides the crank chamber and the transmission chamber housed in the same housing. The pressure generated by the reciprocating motion of the piston discharges the engine oil to the transmission chamber. As a result, the pressure fluctuations in the transmission chamber were reduced, and, subsequently, so was the amount of blow-by gas discharge. Further, it was possible to reduce the breather chamber volume to 1/5 that of a conventional engine. Moreover, effects from reduced engine weight and friction loss were obtained.

1. Introduction

The strong demand for a lightweight, compact engine in our XR series off-road sports motorcycles has resulted in all models in this series being equipped with single-cylinder engines having a displacement in the range from 50 to 600 cm³. A single-cylinder engine requires a large-volume breather chamber to avoid oil discharge via the crankcase breather that is caused by large pressure fluctuations inside the crankcase. There is also a friction loss by oil agitation due to the wide variations in oil level inside the engine.

Four-stroke engines are typically provided with a deep oil pan under the crankshaft to reduce this oil agitation loss and to provide an oil level that does not result in immersion of the crankshaft in oil. However, large-displacement off-road motorcycles require a high minimum ground clearance beneath the engine, which must also be mounted as low as possible in the frame, thus precluding the use of a deep oil pan. This brings back the issue of oil friction loss.

We have addressed these issues during the course of implementing a full model change for the XR600R, the motorcycle in this series having the largest engine displacement, by developing a new engine with a sealed crankcase. The fluctuations in crankcase pressure that are caused by the reciprocating motion of the piston are accommodated within the crank chamber, rather than being transferred to the transmission chamber, due to the use of a sealed bulkhead between the two chambers. This new engine is used in the XR650R model, and this new system is called a closed crankcase system.

2. Engine Overview

The two objectives that had to be achieved in the

development of the XR650R were as follows:

- High torque at low engine speeds and high power at high engine speeds
- A significant reduction in engine weight over the previous model

These objectives were achieved by increasing the displacement by 50 cm³ to improve the torque characteristics at low engine speeds, while the change from air cooling to water cooling and an increased compression ratio improved the power output at high engine speeds. The issue of breather performance is usually encountered when attempting to increase engine displacement. The closed crankcase system was employed to address this issue. The shape and material of every engine part was reviewed with the objective of

Table 1 Engine specifications

Model	XR650R	XR600R
Engine stroke	4-stroke	4-stroke
Number of cylinder	1	1
Cooling system	Liquid-cooled	Air-cooled
Lubricating system	Dry sump	Dry sump
Bore × Stroke (mm)	100 × 82.6	97 × 80
Displacement (cm ³)	649	591
Compression ratio	10.0	9.0
Maximum engine output (kW/ r/min)	41/6750	31/6500
Maximum torque (Nm/ r/min)	64/5500	52/5000
Valve train	SOHC 4-valve	SOHC 4-valveRFVC
Fuel supply system	Carburetor	Carburetor
Engine dry weight (kg)	40	45
Crank case	Closed	Conventional

* Asaka R&D Center

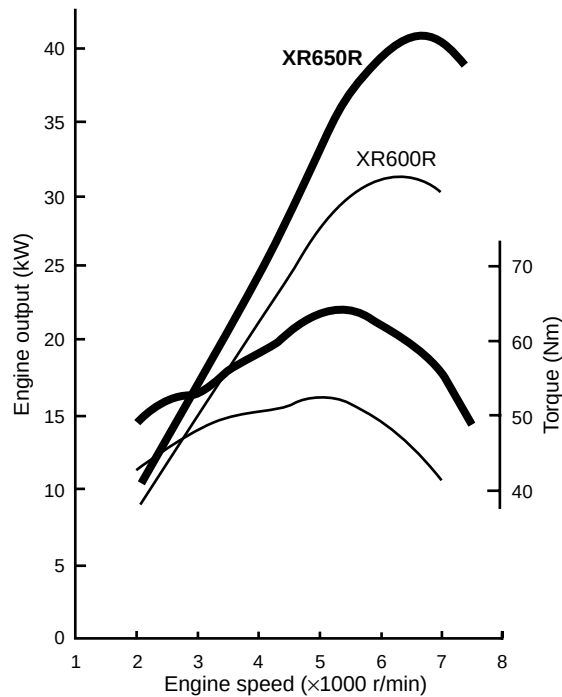


Fig.1 Engine performance curve

reducing engine weight, and a closed deck-type aluminum alloy cylinder with a nickel-cadmium-zirconium-plated bore was adopted. These changes have increased the maximum power output by about 30%, improved the low-speed torque by about 13%, and reduced the engine dry weight by about 5 kg. Even when such added parts as the water cooling radiator are included, a weight reduction of about 1 kg has

been achieved. The engine specifications are shown in Table 1, while the output characteristics are shown in Fig.1.

3. Improvement of the Breather Performance

The oil discharge prevention performance of the breather (breather performance) in a single-cylinder engine can be improved by enlarging the breather chamber or increasing the volume of the crankcase. However, both of these methods conflict with the need to reduce engine weight and size. Typical concerns that can be encountered when applying a method to separate the crank chamber and transmission chamber include crank chamber lubrication oil processing, and the effect on pressure loss inside the closed crank chamber. This method, however, is advantageous in terms of reducing the overall size of the engine. In addition, just as pumping loss is minimal for the deactivated cylinder in engines that stop operation when the intake valve and exhaust valve are closed and that have a variable number of cylinders⁽¹⁾, so is the pumping loss that occurs in a closed crank chamber. A comparative study was therefore conducted on the following concern items:

- (1) Function and settings
 - Oil discharge function
 - Pumping loss
 - Transmission chamber pressure
 - Amount of oil supplied to the crank chamber
- (2) Effects
 - Breather performance
 - Scavenging performance
 - Friction loss

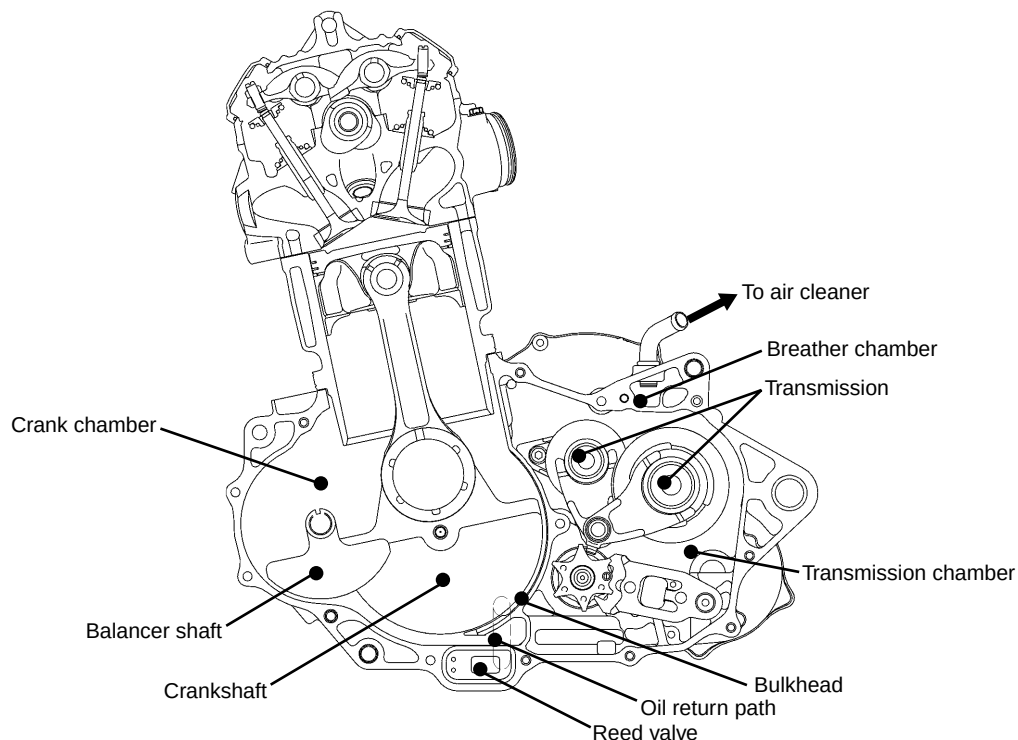


Fig.2 Section of closed crankcase

4. Construction and Action of the Closed Crankcase System

4.1. Construction

A typical motorcycle engine has the crankshaft and transmission housed within a single crankcase and lubricated with the same oil. As shown in Fig.2, the closed crankcase system (subsequently referred to as the closed type) is characterized by having the crank chamber and transmission chamber separated by a bulkhead with no pressure leakage between them. The crankshaft and primary balancer are arranged inside the crank chamber, and a bulkhead to separate the crank chamber from the transmission chamber is provided around the outside of the crank weight. An oil seal is fitted on the outside of the crankshaft bearing to maintain pressure tightness within the crank chamber.

Oil is pumped to lubricate the connecting rod big end and crankshaft bearings, and the piston is cooled by spraying oil on its underside. The oil that flows into the crank chamber is discharged into the transmission chamber from a discharge port provided beneath the bulkhead through a reed-operated check valve. A breather chamber is provided above the transmission chamber, and after condensing the oil, the blow-by gas is directed to the air cleaner.

4.2. Oil Return System

A schematic diagram of the oil return system is shown in Fig.3. During the time the piston is rising as shown in (A), a negative pressure is generated in the crank chamber, and the reed valve closes to prevent the return flow of oil and air from the transmission chamber. During the time the piston is falling as shown in (B), the crank chamber volume decreases and the pressure increases. The reed valve is opened by this pressure, allowing the oil that has flowed into the crank chamber to be returned to the transmission chamber.

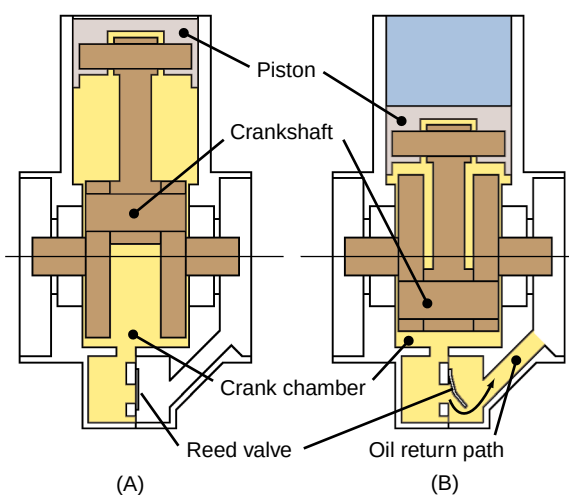


Fig.3 Schematic drawing of oil return action

4.3. Pumping Loss

Although the crank chamber is closed, it is still necessary for the oil that has been supplied to the crank chamber to be

returned to the transmission chamber. The pressures within the crank chamber and transmission chamber, and the opening and closing of the reed valve when the engine is running are shown in Fig.4.

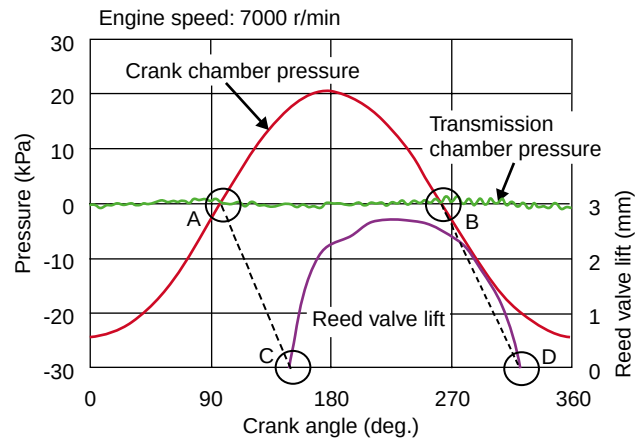


Fig.4 Pressure in crank chamber and reed valve lift

The pressure within the crank chamber exceeds that in the transmission chamber between the crank angles of 95° and 265° (A-B). This difference in pressure provides the pumping action to return the oil in the crank chamber to the transmission chamber. However, there is a time delay in the opening and closing of the reed valve with respect to the changes in pressure (A-C, B-D). The reason for this delay is believed to be the viscosity and inertia of the oil. Consequently, there is a potential concern of the back flow of oil and air into the crank chamber between the crank angles of 265° and 320° (B-D) when the pressure in the transmission chamber is higher than that in the crank chamber and the reed valve is open.

A study was made of the pumping loss that occurs in the crank chamber to identify the effects of this back flow. Supplied lubricant oil and blow-by gas that passes the piston rings flow into the crank chamber. Their combined volume is only several cm^3 for each revolution of the crankshaft, making it extremely small relative to the swept volume. Consequently, if an amount is discharged that is equivalent to the volume flowing in with each cycle, there would be hardly any change in the amounts of oil and air in the crank chamber, resulting in an extremely low pumping loss. However, as the degree of back flow increases, an excessive volume is pumped, and this is thought to contribute to an increased pumping loss. Fig.5 shows a P-V diagram for the crank chamber, with the crankcase volume indicated on the horizontal axis. For the sake of comparison, an engine was constructed without the bulkhead between the crank chamber and transmission chamber (subsequently referred to as the conventional engine type), and the same measurements were performed on that engine.

The pumping loss in the crank chamber is represented by the hatched area that is bounded by the P-V plots during the compression and expansion strokes. In contrast to the pressure difference that occurs between the compression stroke and expansion stroke in the conventional engine, no

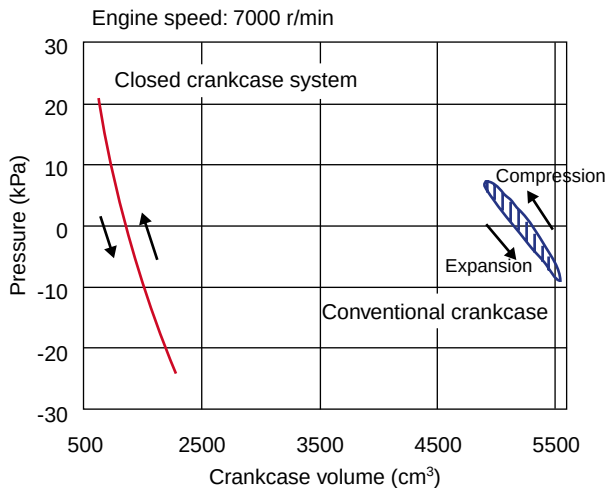


Fig.5 P-V diagram of crankcase

such pressure difference exists in the closed crankcase system engine. The pumping loss in the conventional engine is due to a cycle in which air is taken in from outside the engine when the piston rises, and then discharged with the blow-by gas when the piston falls. In contrast, there is hardly any risk of back flow from the oil discharge port in the closed crankcase system engine and, since the combined volume pumped from the crank chamber to the transmission chamber is low, the pumping loss was confirmed to be low.

4.4. Reduced Pressure Fluctuations in the Transmission Chamber

Since a very low volume is pumped between the crank chamber and transmission chamber, there were lower pressure fluctuations in the transmission chamber than that in the conventional engine. These results are shown in Fig.6. In this way, the pressure fluctuations in the crankcase of a single-cylinder engine that result from the reciprocating motion of the piston were limited to the crank chamber only. This effect was confirmed in the closed crankcase system engine as no pressure fluctuations were transferred to the

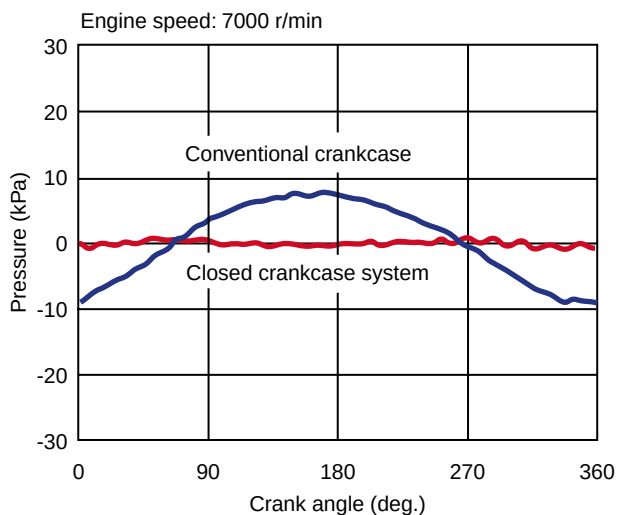


Fig.6 Comparison of transmission chamber pressure

transmission chamber. As a result of verifying the effects of engine operating conditions on pressure fluctuations, it was found that there were no differences between the intake, compression, expansion and exhaust strokes, or due to differences in engine load. Further, we verified that there were no differences between firing and motoring conditions.

4.5. Oil Feed Volume

Since oil agitation loss caused by the crankshaft tends to increase as the amount of oil fed in to the crank chamber is increased, a study was conducted to determine the optimum oil feed volume. In the case of a conventional engine, oil is pumped to the connecting rod big end, and the oil splashed from the big end is then used to cool the piston. Therefore, there is the possibility that the oil supplied could exceed the required volume for both applications. After providing a piston cooling jet to evaluate the minimum amount of oil required for both the big end bearing and for piston cooling, we modified the lubrication system. We determined the required volume of oil fed into the crank chamber to make the crank pin and piston temperatures comparable to those in the conventional engine, and thereby reduced the total feed volume by 52% as shown in Fig.7.

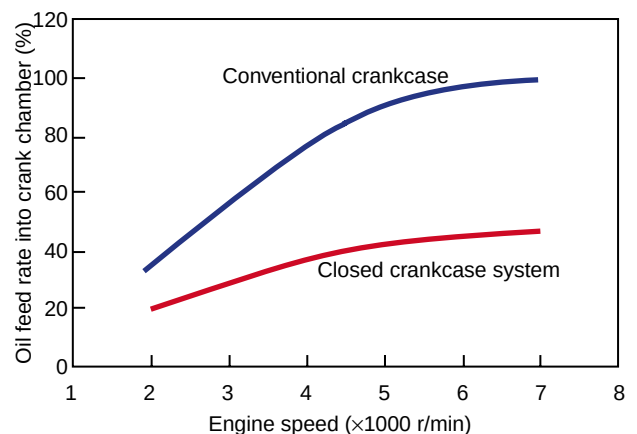


Fig.7 Oil feed rate into crank chamber

5. Closed Crankcase System Engine Performance

The performance of the closed crankcase system engine was evaluated against a conventional engine without the bulkhead between the crank chamber and transmission chamber.

5.1. Improved of Breather Performance

Since the closed crankcase system engine produced a lower pressure fluctuation in the transmission chamber, a small breather chamber with a volume in the order of 60 cm³ situated at the top of the transmission chamber was able to achieve adequate performance. In a comparative test, oil was discharged from the conventional engine when it was fitted with this small-volume breather as a result of the breathing circuit being closed. Therefore a larger breather was necessary. The breather chamber volume in the

XR600R, the previous model employing a conventional breather, was 330 cm³. This is roughly five times larger than that needed in the closed crankcase system engine of the XR650R. These figures indicate how the closed crankcase system contributed to crankcase size reduction. A comparison between the breather chambers is shown in Fig.8.

5.2. Improved Scavenging Performance of the Dry Sump System

The XR650R employs a dry sump lubrication system. An oil outlet port is provided in the base of the transmission chamber to discharge oil into an external oil tank situated within the main frame through a scavenging pump. The reduction in pressure fluctuations in the crank chamber and enhanced stability of the transmission chamber affected the scavenging performance. Since a dry sump lubrication system involves pumping oil under pressure from the external oil tank to each part of the engine, it is necessary to stabilize the oil level in the tank. The closed crankcase system engine, with its small fluctuations transmission chamber pressure, maintained a stable oil level on the outlet side when the engine was running. Further, the amount of oil that could be drawn up into the external tank was greater than that in a conventional engine. As a result, the volume of on-board oil could be decreased to 300 cm³ leading to a consequent reduction in vehicle weight. The amount of oil in the external tank with the engine running for a total oil volume of 2000 cm³ is shown in Fig.9.

5.3. Reduced Friction Loss

A comparison of the friction loss is shown in Fig.10. The friction loss mean effective pressure (Pmf) is represented on the vertical axis, and the pumping loss of the crank chamber is also shown as represented by mean effective pressure (defined as PmfP*). There are two reasons for the decrease in friction loss resulting from the closed crankcase system engine: the first is the lower pumping loss in the crank chamber as previously mentioned, and the second is the lower oil agitation loss.

In a conventional crankcase, PmfP* increases due to the movement of large volumes of air at the blow-by discharge port accompanying the reciprocating motion of the piston. In the closed crankcase system, however, PmfP* is low since no back-flow occurs to accompany the discharge of oil from the crank chamber. In addition, the oil level in a conventional

engine rises due to deteriorated dry sump scavenging performance. Pressure fluctuations also increase and, since the oil level changes, there is considerable friction loss by oil agitation. In the closed crankcase system, in addition to optimizing the oil volume, a long oil scraper that protrudes over the crankcase along the crank weight is provided as shown in Fig.11. In addition, after optimizing the clearance for the crank weight, various configuration related methods

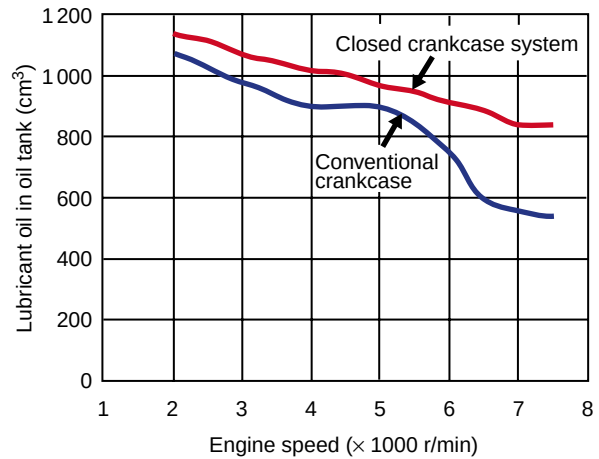


Fig.9 Lubricant oil in oil tank

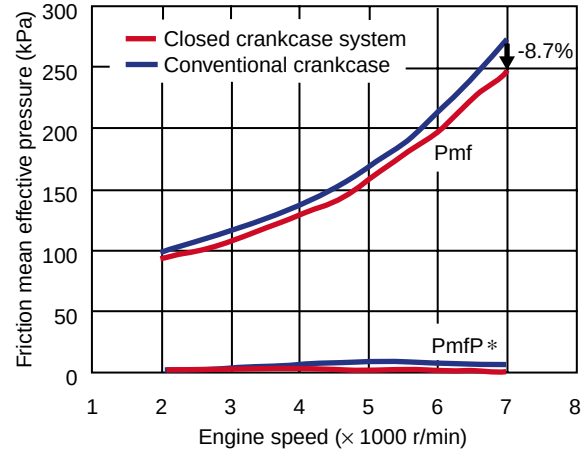


Fig.10 Friction mean effective pressure

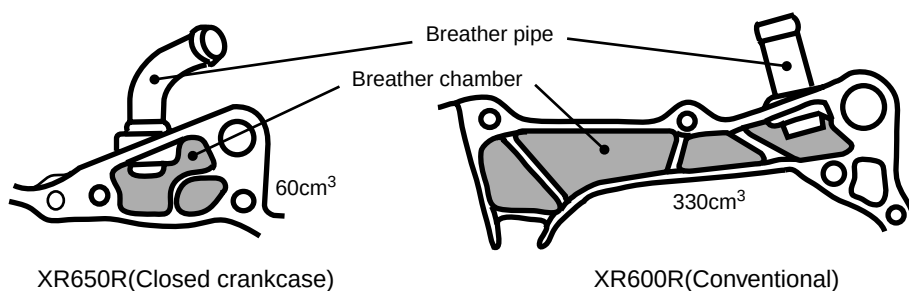


Fig.8 Comparison of breather chamber volume

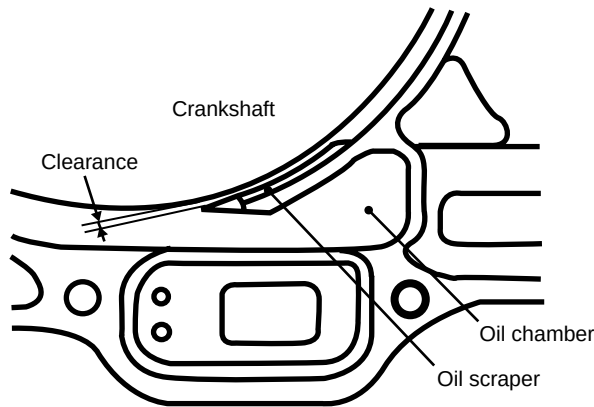


Fig.11 Detail drawing of sump part

to decrease friction loss by oil agitation were investigated including the temporary accumulation of oil in a rear oil chamber. The overall effect was to reduce the friction loss by 8.7% at an engine speed of 7000 rpm, resulting from a reduction in the pumping loss of 2.0% and a reduction in the agitation loss of 6.7%.

6. Conclusion

A closed crankcase system was devised, by which the crank chamber and transmission chamber are separated, in a 650 cm³ single-cylinder engine for use in off-road motorcycles. This system prevents the crankcase pressure fluctuations that are characteristic of a single-cylinder engine from being transferred outside the crank chamber, thereby resulting in the following advantages:

- (1) The breathing performance and scavenging performance, when applying a dry sump lubrication system, were both improved due to the reduced pressure fluctuations in the transmission chamber. This enabled the volume of the breather chamber to be reduced by 80% in comparison with the previous model, and the on-board oil volume to be reduced by 300 cm³. Both of these factors contributed to a reduced engine weight.
- (2) The friction loss by oil agitation was decreased by optimizing the lubricating oil volume, the shape of the sump part, and by stabilizing the oil level in the engine. In addition, the pumping loss due to oil being discharged from the crank chamber was reduced to a level lower than that of a conventional engine in which there is extensive movement of air at the blow-by gas discharge port. These factors combined to reduce friction loss by 8.7% at an engine speed of 7000 rpm.

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Effects of Connecting Rod Big End Stiffness on Oil Film Pressure

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ABSTRACT

A new thin-film sensor that enables accurate measurement of the oil film pressure of compact bearings for passenger cars was developed. Using this new sensor, the oil film pressure distribution was measured for a variety of connecting rod big end stiffness levels, and the effect of those changes was investigated. These results confirmed that the oil film pressure distribution changes according to variety of the stiffness. Whereas the oil film pressure occurs with one peak in the load direction for a rigid connecting rod big end, the oil film pressure for an elastic connecting rod big end is dispersed and the direction of the peak does not match that of the load. In addition, elastohydrodynamic calculations were performed with the same conditions as the measurements, and the reasonableness of the measurements and calculations were confirmed.

1. Introduction

The width of connecting rod big end bearings has tended to become smaller to reduce bearing friction in response to recent environmental issues, making it more difficult to ensure durability. Bearings in particular, which determine the engine framework, are being required to have their performance predicted to a level of precision that is approaching the limits of feasibility from the perspective of shortening development time.

Engine bearings are lubricated by fluid in the form of an oil film, and measurement of this oil film pressure is an important factor for evaluating durability. However, it is extremely difficult to directly measure the oil film pressure between sliding parts, and there are few cases of this having been successfully carried out.⁽¹⁾ Although the authors had previously applied a method in which a diaphragm-type sensor was embedded in a shaft, this method had the following drawbacks:

- (1) Restrictions on the space required for embedding the sensor in the case of a small engine bearing for a passenger car engine
- (2) Restrictions on the heat resistance of the semiconductor gauge
- (3) Measurement under high-pressure conditions
- (4) The effects of irregularities in the sensor pressure-receiving portion

We therefore decided to develop a thin-film sensor⁽²⁾⁽³⁾ that would be nearly able to completely overcome these problems and make it possible to measure the pressure between sliding parts that has been considered difficult to achieve in the past. In addition, oil film calculations (elastohydrodynamic calculations; EHD) are attempted that take into consideration the rigidity of the connecting rod big end. Such calculations have become practical with the

dramatic improvement that has taken place in the calculation environment in recent years, and several calculation results have been reported.⁽⁴⁾⁻⁽⁷⁾ Based on the combined results of the calculations and measurements, a discussion is presented about the mechanism by which the annular rigidity of the connecting rod big end affects the oil film pressure.

2. Measurement Method

2.1. Construction of a Conventional Diaphragm-type Pressure Sensor

Fig.1 shows the appearance of a diaphragm-type pressure sensor and the cross-sectional construction of the sensor unit. A sensor unit 14 mm in diameter and 6 mm thick, in which a semiconductor strain sensor is attached to a diaphragm having a diameter of 1.0 mm, is embedded in a hole provided in the shaft. Electron beam welding is used to secure the sensor unit in position.

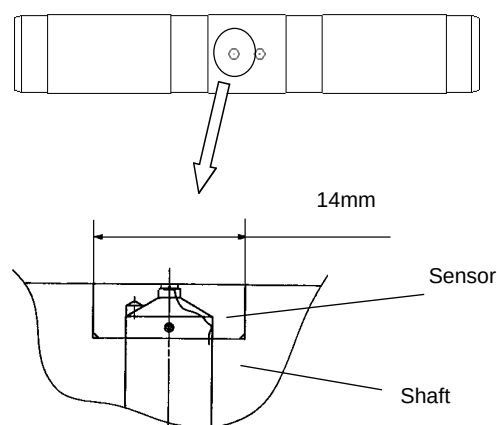


Fig.1 Diaphragm sensor

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2.2. Construction of the New Thin-Film Pressure Sensor

The appearance of the new thin-film pressure sensor is shown in Fig.2, while the cross-sectional construction of the sensor unit is shown in Fig.3. Manganin, whose resistance is sensitive to pressure, was used for the sensor material, and a pressure-receiving area measuring 1 mm in diameter and 20 μm in radial width was formed on the shaft sliding surface by sputtering. The total thickness of the sensor, resulting from the combined thickness of the insulating and protective films, is about 7.2 μm . The chemical ingredient of manganin is shown in Table 1.

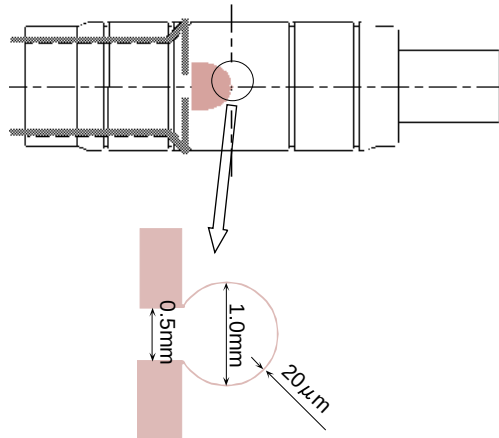


Fig.2 Thin-film sensor

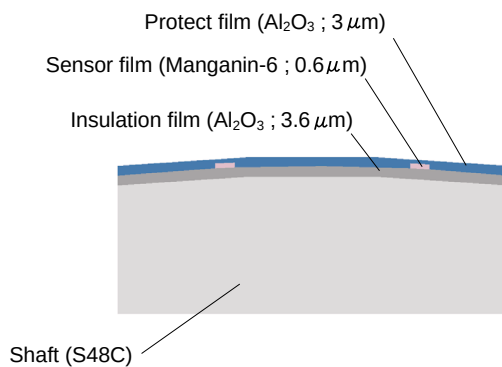


Fig.3 Sensor part section

Table 1 Chemical ingredient

	Cu	Mn	Ni
Manganin-6	87.6	9.2	3.2

Fig.4 shows a plot of the pressure versus percentage resistance change. This graph demonstrates the excellent linearity and low hysteresis performance of the sensor over a broad pressure range of 0-100 MPa.

The area of the shaft where the sensor is formed is ground by the amount of sensor thickness prior to processing, and a small circular shape with an extremely low level of irregularities is maintained even after the sensor is formed.

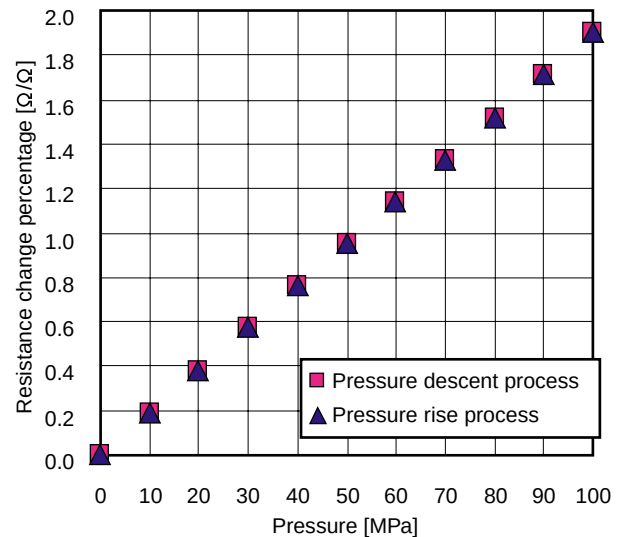


Fig.4 Thin-film sensor characteristic

2.3. Overview of the Experimental Device

Fig.5 shows the experimental device used in this study for measuring the oil film pressure. Roller bearings were used to support the shaft in order to minimize the thermal effects of the support bearings themselves on the test zone. A vertical hydraulic load was applied downward to the connecting rod. The output from the sensor is led outside the rotor by a slip ring, and a Wheatstone bridge is set up in front of the slip ring to increase the S/N ratio. An electromagnetic clutch is installed on the motor side to protect the apparatus from an overload torque in the event of bearing seizure or other high torque-generating events.

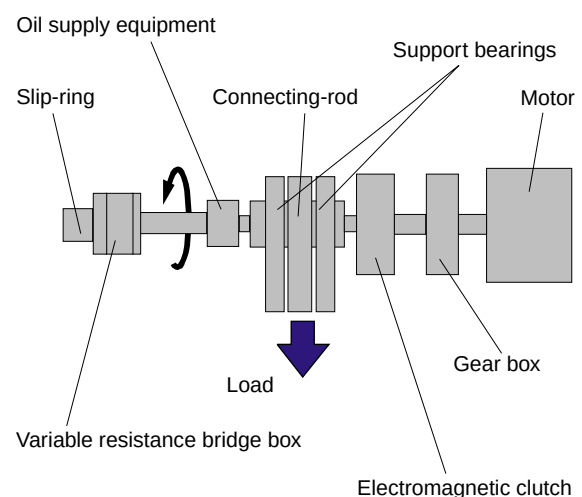


Fig.5 Assembled experimental device

2.4. Measurement Conditions

Fig.6 shows the appearance and Table 2 shows the dimensions of the test connecting rods. The two types of connecting rods were tested to identify any difference attributable to the annular rigidity of the big end in the distribution of oil film pressure. The Type A connecting

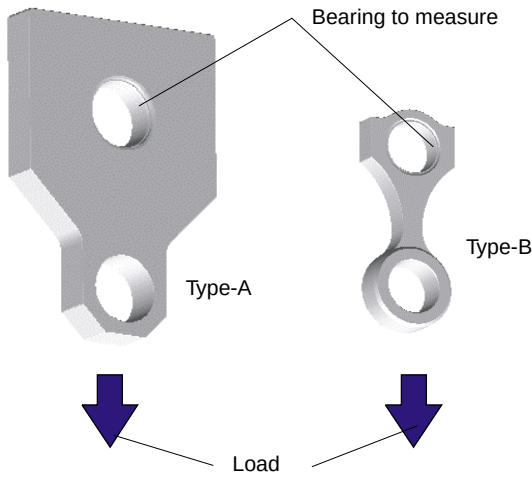


Fig.6 Connecting rod

Table 2 Test specification

Type		A	B	Unit
Conrod	Length	138.5		mm
	Big-end axial width	30	24	mm
	Material	SUS304		-
Bearing	In-diameter	50		mm
	Mean-width	20		mm
	Material	Cu + Pb		-
Oil		SAE10W-30		-

rod has a big end which is equivalent to a rigid body, while Type B, in which elastic deformation can occur more easily, has a shape resembling that of the big end used in passenger car engines.

The test connecting rods were each in one piece, without a separate big-end cap, and a completely circular bearing was gently press fitted into the big end.

3. Calculation Method

Commercially available software was used for the EHD calculations. The oil film characteristics were determined by using Reynold's equation, the fundamental equation of fluid lubrication. The elastic deformation of the connecting rod big end was incorporated into Reynold's equation as a mass-rigidity matrix contracted from an FEM model. Calculations were made by assuming the density, temperature and viscosity to remain constant at all times within tolerable limits. The connecting rod was used as the constraint in the calculations, the load being input from the shaft. The internal roundness of the bearing after it had been press fitted into the big end was measured and expressed in simple terms in the FEM model for input into the calculations to improve the overall accuracy. The FEM model used in the calculations is shown in Fig.7.

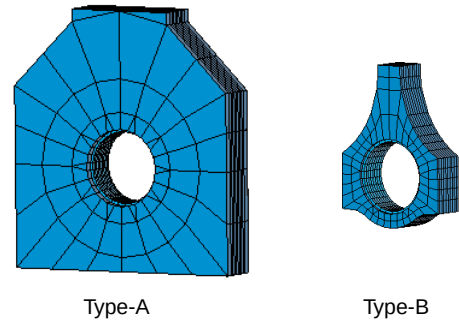


Fig.7 FEM model

4. Comparison Between the Measured and Calculated Results

4.1. Measured Results of the Oil Film Pressure

Fig.8 shows the results of measuring the distribution of oil film pressure around the bearing circumference at 3000 rpm for the Type A and Type B connecting rods.

The oil film pressure in the rigid body connecting rod big end of Type A was distributed over a narrow range around the bearing circumference, and increased with increasing load, while demonstrating a single peak coinciding with the direction of the load.

In the case of the Type B big end which can undergo

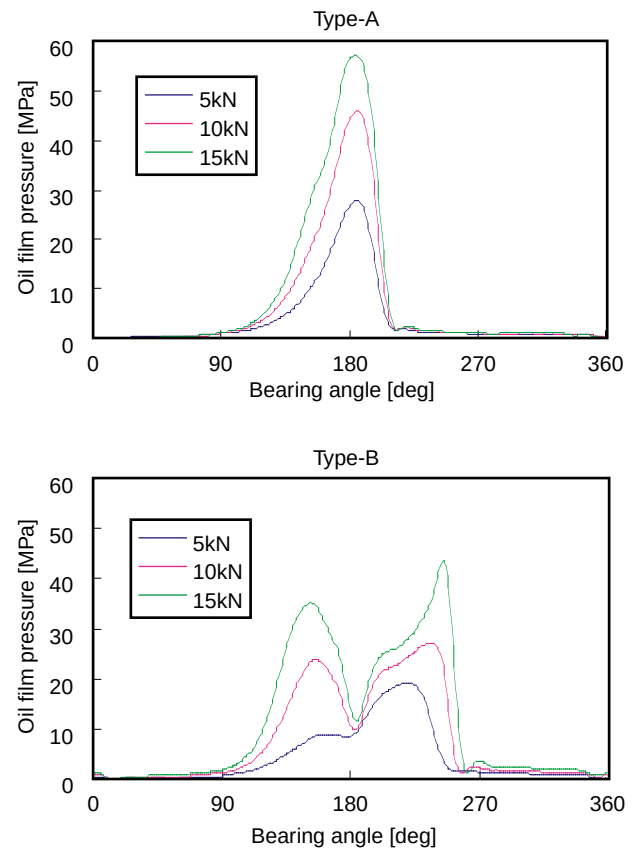


Fig.8 Measurement oil film pressure at 3000rpm

elastic deformation, the oil film pressure was distributed over a wider range around the bearing circumference than the case of Type A. As load was increased, the broad single peak was divided into two peaks, one either side of the direction of the load. In comparison with Type A, the increase in oil film pressure with increasing load was less pronounced.

4.2. Comparison Between the Measured Results and EHD- calculated Results for the Oil Film Pressure

Fig.9 shows the measured results and EHD-calculated results for the oil film pressure at 2000 rpm for the Type A and Type B big ends. The distribution of oil film pressure around the bearing circumference could be simulated with considerable accuracy for both the Type A and Type B big ends. These results made it possible to confirm the integrity of both the thin-film sensor measurements and EHD calculations.

In order to assess the feasibility limit for predicting bearing performance, it was thus determined that it is necessary to use EHD calculations that incorporate bearing elasticity, instead of using conventional oil film calculations that assume the connecting rod big end to be a rigid body.

4.3. Discussion on the Mechanism for the Change in Oil Film Pressure

We confirmed from the measured results of the thin-film sensor that the distribution of oil film pressure changed

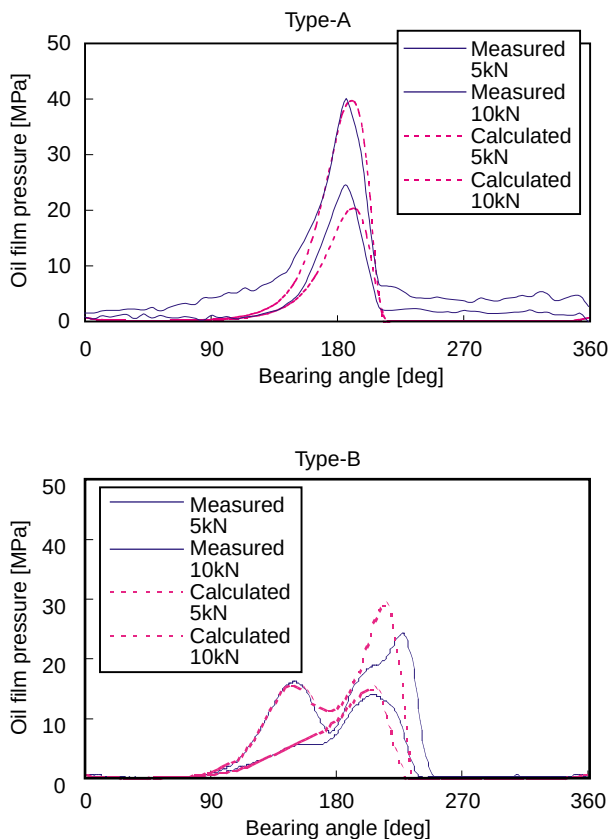


Fig.9 Comparison measurement and calculation at 2000rpm

according to the rigidity of the connecting rod big end. This is believed to have resulted from the load causing contraction of the big end diameter in the direction perpendicular to the load (referred to as close-in) when there was elastic deformation in the big end, while a corresponding increase in bearing clearance in the direction of the load occurs; this resulted in a reduction in the rate of increase in oil film pressure in the direction of the load, thereby leading to a corresponding increase in the distribution of oil film pressure in order to maintain the balance.

The mechanism for the change in oil film pressure can be deduced from the results of the EHD calculations. Fig.10 shows a 3D contour diagram of the oil film pressure distribution based on the calculated results, while Fig.11 shows the distribution of oil film thickness also based on the calculated results. In the case of the Type A big end, at the same time as the oil film pressure increases in the direction of the load, the oil film thickness can be seen to become thinnest at nearly the same location. This can be said to have resulted from no increase in clearance in the direction of the load due to the absence of close-in since the big end did not deform.

In the case of the Type B big end, however, oil film pressure in the direction of the load is relatively low, and oil film is consequently relatively thick. Locations where the

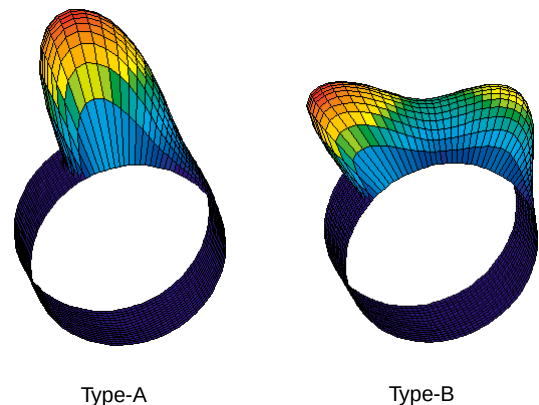


Fig.10 3D contour of oil film pressure distribution

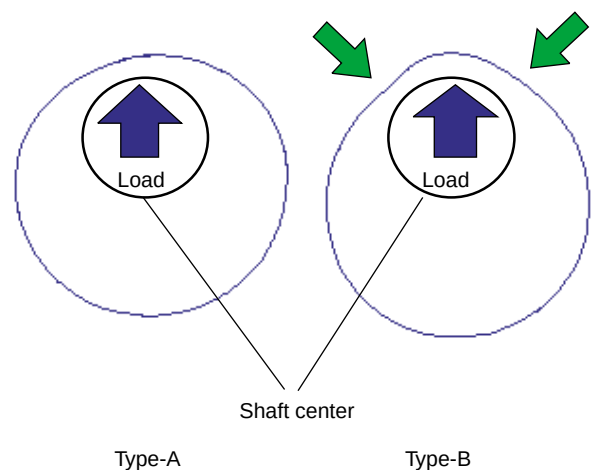


Fig.11 Oil film thickness distribution

oil film is relatively thin can be seen to exist on both sides. Increased clearance in the direction of load can be said to have occurred due to the close-in caused by elastic deformation of the big end.

Fig.12 shows a schematic representation of the mechanism for the change in distribution of oil film thickness.

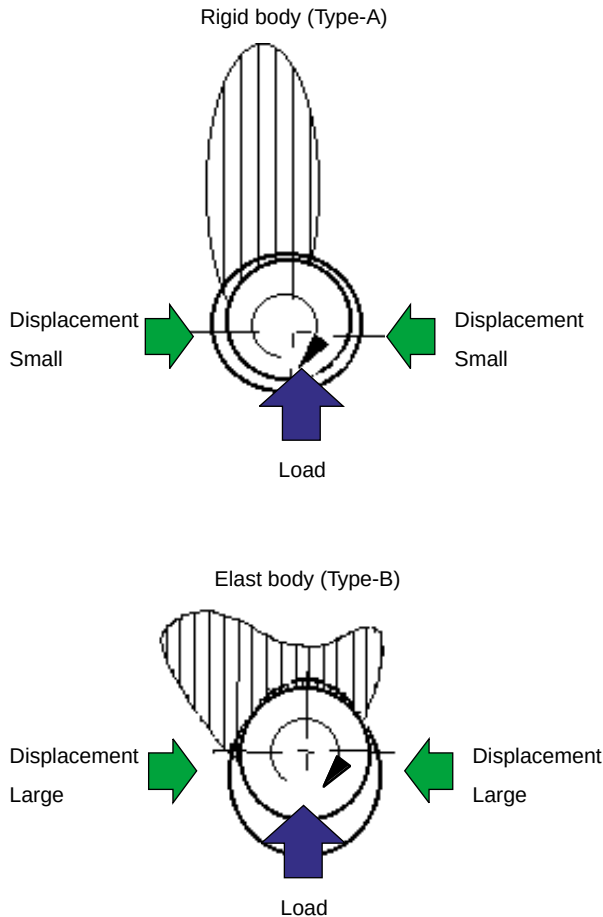


Fig.12 Mechanism of oil film pressure distribution

5. Conclusion

- (1) The development of a small thin-film pressure sensor made it possible to directly measure the bearing oil film pressure in gasoline engines used in passenger cars.
- (2) The distribution of the oil film pressure around the bearing was confirmed to be considerably affected by the rigidity of the connecting rod big end. The oil film pressure distribution in a big end with high rigidity demonstrated a narrow peak in the direction of the load, and the pressure increased with increasing load. On the other hand, in the case of a big end with relatively low rigidity like that used in passenger car engines, two lower peaks in the oil film pressure distribution resulted, one being each side of the load, while the increase in pressure was more gradual with increasing load.
- (3) Good correlation was obtained between the measured

results and those obtained by EHD calculations, verifying the integrity of both. It was confirmed that EHD calculations that allow for elastic deformation of the big end, and not conventional oil film calculations that assume the big end to be a rigid body, will be necessary in the future for precisely predicting bearing performance.

6. Acknowledgments

The authors would like to express their deep appreciation to Professor Yuji Mihara of the Musashi Institute of Technology and his staff for their generous cooperation in conducting this study.

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Bearing Tester Analysis of the Factors in Engine Plain Bearings Seizure

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ABSTRACT

A bearing tester that overcomes the problems of a conventional bearing tester was fabricated and experimental modes were selected in order to clarify factors determining the seizure limit of engine plain bearings. Using this tester, seizure tests were performed on copper-lead bearings with an overlay and aluminum alloy bearings, both of which are used as automotive engine bearings. Bearing hydrodynamics calculation was performed based on the obtained experimental data. These results clarified that the seizure limit for copper-lead bearings with an overlay is determined by the occurrence of plasticity flow of the overlay due to the maximum oil film pressure, and that the seizure limit for aluminum alloy bearings is determined by the minimum oil film thickness.

1. Introduction

Requirements more stringent than ever before have been placed on the plain bearings of automobile engines in recent years, due to such factors as the reduction of friction level, more compact engine sizes, reduced width due to improved crankshaft rigidity for improving NVH, and a reduced amount of lead in the fuel to minimize the burden on the environment.^{(1)~(5)}

It consequently became necessary to clarify the mechanism for bearing seizure in order to identify the parameters for judging the suitability of a plain bearing.

However, since conventional plain bearing testers seem unable to completely break away from the level of a relative comparison of bearing performance, it is necessary to employ a quantitative evaluation in order to enable a design study of bearing suitability. In order to achieve this quantitative evaluation, it was necessary to devise a new type of plain bearing tester and the most appropriate experimental procedure. This report presents the mechanism for seizure as determined by this new bearing tester with the experimental procedure that was used.

2. Overview and Problems with Conventional Bearing Testers

Fig.1 shows a system diagram of a conventional testing system, Fig.2 shows the conventional method for measuring the temperature of the test bearing and sliding parts in the direction of the load, and Fig.3(a) shows the conventional load pattern.⁽⁶⁾

This conventional test employs a method in which the static load generated by hydraulic pressure is applied to a round shaft supported by bearings. The test is stopped when the indicated temperature at the back of the test bearing, or the bearing friction torque, reaches a prescribed value. The

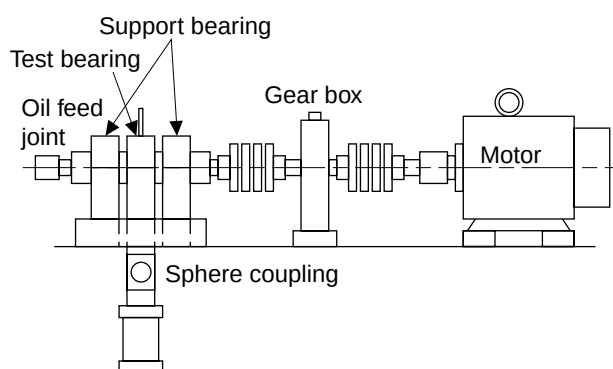


Fig.1 Conventional system of the equipment

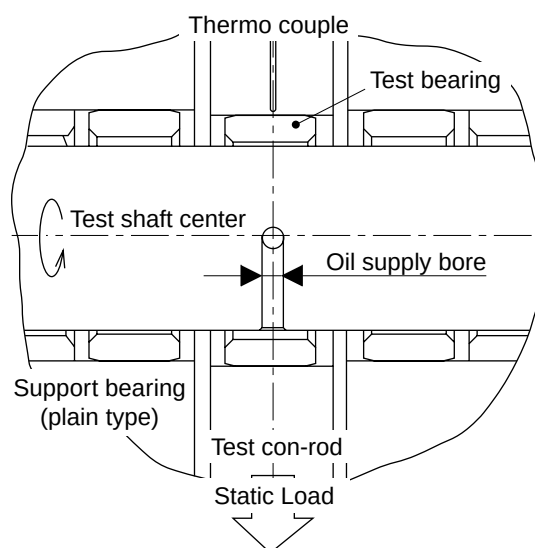


Fig.2 Conventional bearing part

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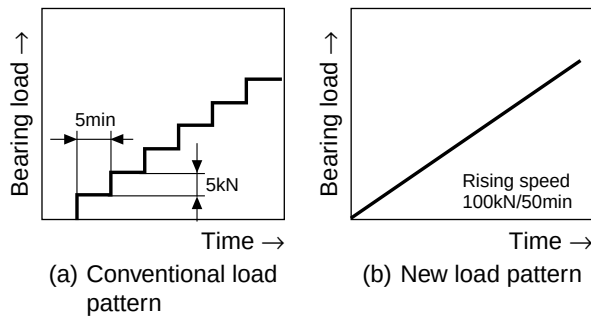


Fig.3 Bearing load pattern

reason for stopping the tester before the bearing has completely seized is to prevent damage to the tester components that would be caused by the sudden stopping torque that is generated when seizure occurs. Problems associated with this testing system are as follows:

- (1) The tester must be stopped at a stage before the bearing completely seizes, despite the purpose of the test being to analyze seizure.
- (2) The temperature of the support bearings has a significant effect on the temperature at the back of the test bearing.
- (3) The method for assessing contact between the shaft and bearing involves only visually examining the bearing contact state after testing.

3. New Bearing Tester for Seizure Analysis

Fig.4 shows a system diagram of the newly devised bearing tester, while Fig.5 illustrates the method used to measure the temperature of the test bearing and load sections.

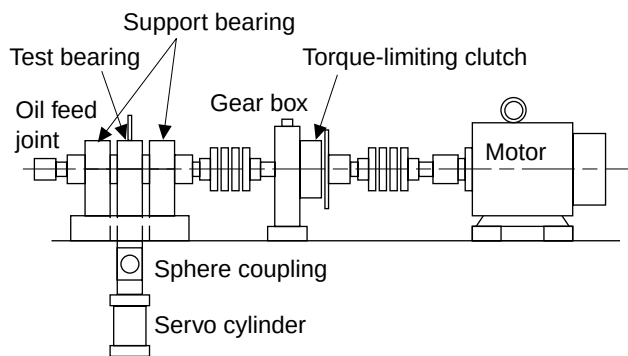


Fig.4 New system of the equipment

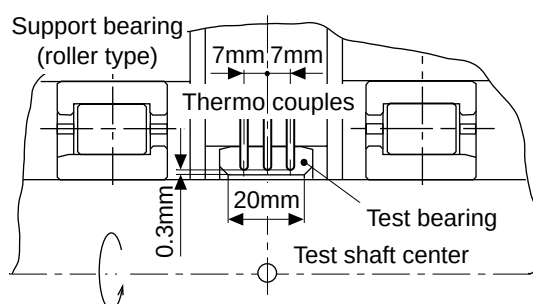


Fig.5 Developed bearing part

This tester has the following characteristics:

- (1) By providing a torque-limiting clutch between the motor and gear box, testing can continue until the bearing has completely seized (until the bearing surface melts or the shaft and bearing seize solid).
- (2) A roller bearing that generates very little heat is used for the support bearings to minimize the effects of the support bearing temperature.
- (3) An additional location for temperature measurement is provided across the bearing width as a method for assessing whether the shaft and bearing are making contact. Contact between the shaft and bearing can be detected by monitoring the temperature balance. In addition, a design is employed in which sheathed thermocouples are embedded in holes at the back of the bearing surface so that the temperature close to the bearing sliding surface can be measured to monitor the phenomena that occur on the sliding surface. The use of this method makes it possible to accurately adjust the shaft inclination without having to view the contact state of the bearing.

Instead of the conventional method of applying a load in steps, a linearly increasing load is applied as shown in Fig.3(b). Since a rapid temperature rise and seizure tended to occur when the load was increased in steps as shown in Fig.3(a), or in other words, since it was determined that the seizure load varied according to the load increment at each step and the time the load was held constant, the load is increased along a linear gradient to avoid the effect of increasing the load in steps.

Fig.6 shows the results of repeatability testing by this method.

Although there is a difference of about 10% in the load during seizure occurrence, the plots during the course of the temperature rise and those of the secondary rise in temperature are nearly identical.

A list of test conditions applied is shown in Table 1.

Tests were conducted on copper-lead bearings and aluminum alloy bearings that are typically used in automotive applications.

The structure of an automotive plain bearing consists of a soft material having outstanding sliding properties supported on a strong backing plate. The sliding materials

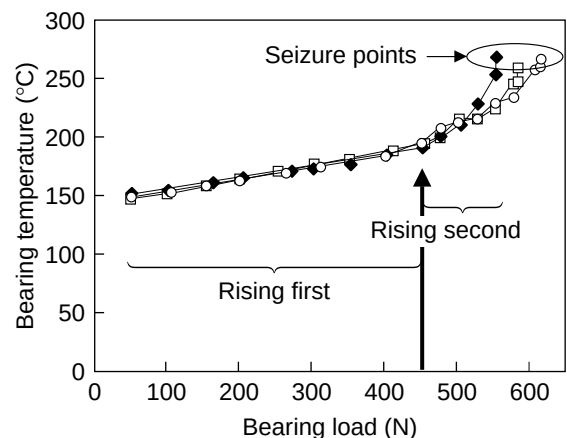


Fig.6 Confirmation test of the repeatability

Table 1 Tested condition

Shaft rotation speed	3000 ~ 6000 rpm
Feed oil temperature	35 ~ 140 °C
Feed oil pressure	300 kPa
Radial clearance	13 ~ 20 μm
Oil SAE grade	10 W-30
Shaft surface roughness	0.8 S (Rmax)
Oil feed method	Shaft feed

can be broadly classified into those made of lead alloy and those made of aluminum alloy, each of these being further classified into several subcategories according to the performance. In contrast to using an overlay of soft lead on the sliding surface of copper-lead bearings to achieve uniformity, many aluminum alloy bearings use a single alloy layer having aluminum as its main component.

4. Results of Seizure Tests for Individual Bearing Materials

4.1. Seizure Characteristics of Copper-Lead Bearings

Fig.7 shows measured results for the sliding surface temperature up to seizure relative to the load for a lead alloy bearing.

Fig.8 is a schematic drawing of the contact characteristics of the bearing surface that were observed when the tester

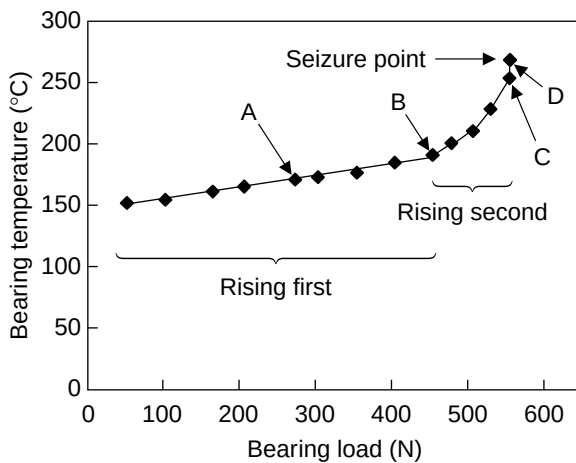


Fig.7 Temperature process of the Pb bearing

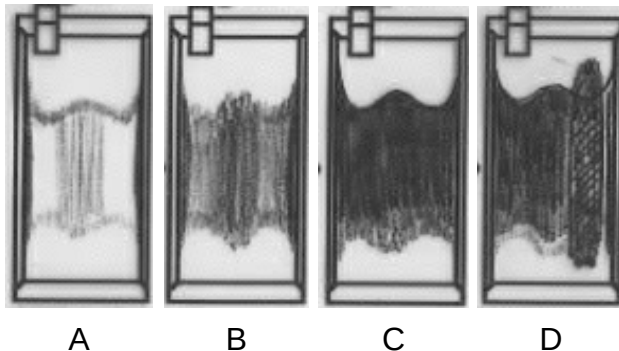


Fig.8 Characteristic of bearing surface of Pb bearing

was stopped at each of the points indicated in Fig.7.

Characteristics at each test point are described next.

A: The state during which the temperature linearly increases: Slight contact is apparent between the shaft and bearing.

B: The state near the end of the linear rise in temperature: Substantial contact occurs overall.

C: The state during which there is a secondary rise in temperature: Although greater overall contact occurs, in contrast to the contact in states A and B which is apparent as scratches both upstream and downstream, plastic flow of the overlay in the downstream contact portion is clearly apparent. This phenomenon occurs prior to reaching the melting temperature of the overlay, and indicates flow due to physical plasticity.

D: The state after the overload torque clutch has operated: Flow occurs due to melting of the overlay.

Based on these findings, over the linear range of the temperature increase, metal contact on the bearing sliding surface progresses, and the temperature increases linearly relative to the load. A secondary temperature rise is then initiated at point B, and plastic flow of the overlay is thought to occur.

Changes in the sliding surface temperature relative to the load are shown in Fig.9 when changing the oil temperature, bearing radial clearance and shaft rotational speed to identify the onset of seizure under the various conditions.

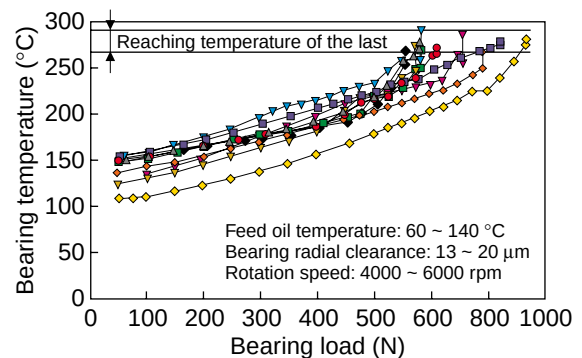


Fig.9 Parametric study of oil temperature, radial clearance and rotation speed on Pb bearing

The measured temperatures were influenced differently by each of these parameters under the same load. Although the load at which seizure occurred is different in each case, the ultimate temperature can be seen to be within a fixed range under all the conditions. This is believed to correspond to the melting point of the surface layer.

4.2. Seizure Characteristics of Aluminum Alloy Bearings

Fig.10 shows the measured results of the sliding surface temperature leading up to seizure of the aluminum alloy bearing. Fig.11 is a schematic drawing of the contact characteristics of the bearing surface that were observed in the same manner as that for the copper-lead bearing.

A: The state near the end of the linear rise in temperature: Substantial contact is apparent throughout, which is

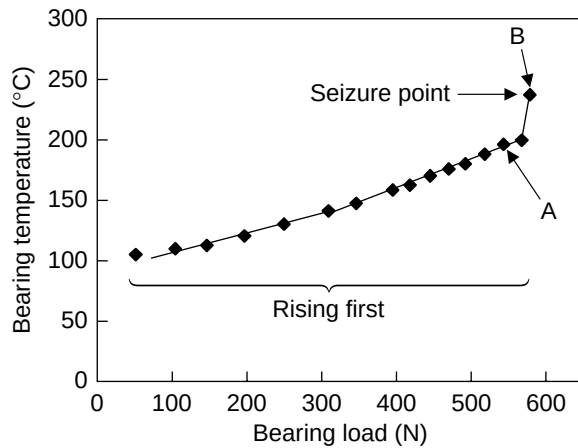


Fig.10 Temperature process of the Al bearing

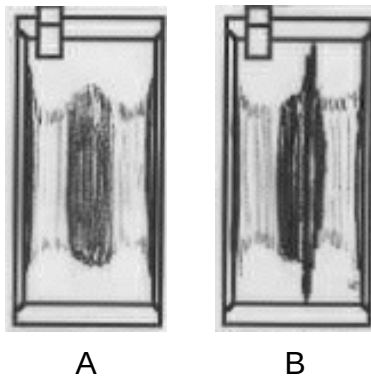


Fig.11 Characteristic of bearing surface of Al bearing

similar to the case of the copper-lead bearing.

B: The state after the overload torque clutch has operated: In contrast to the apparent melting of the surface overlay in the case of the copper-lead bearing, flow occurs due to adhesion of the surface of the aluminum alloy that has been picked up from the area with the vertical scratches at the center.

Fig.12 illustrates the process leading up to seizure when the parameters of rotational speed, oil temperature and bearing radial clearance were changed.

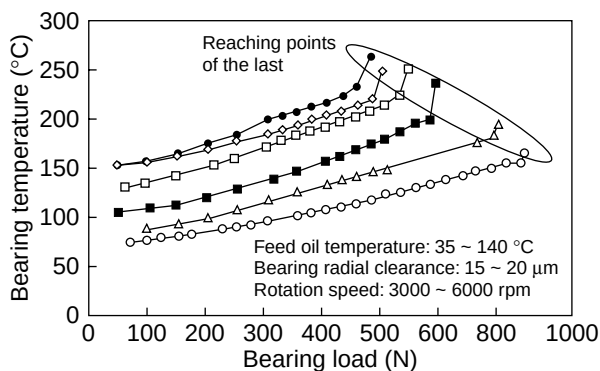


Fig.12 Parametric study of oil temperature, radial clearance and rotation speed on Al bearing

The aluminum alloy bearing was determined to have the following characteristics in comparison with the copper-lead bearing:

1. Seizure occurred immediately after completion of the primary temperature rise.
2. Adhesion within the bearing occurred regardless of the melting point of the aluminum alloy.
3. There was no correlation between the seizure threshold load and the sliding surface temperature.

5. Analysis of the Cause of Seizure

5.1. Calculation Conditions

Bearing hydrodynamic calculations were made using the test conditions as input values (load and sliding surface temperature) to analyze the cause of seizure.

The mobility method⁽⁷⁾ used in rigid body bearing calculations was applied. The shaft oil feed holes, lemon shape and relief shape were not considered in the calculations.

Fig.13 indicates the sampling points for data input into the calculation.

The value at which the temperature started to rise prior to the secondary rise in Fig.13(a) was used for the lead alloy bearing, this value being selected to determine the cause of the secondary rise in temperature.

The value just before the sudden rise in temperature and the onset of seizure was used for the aluminum alloy bearing.

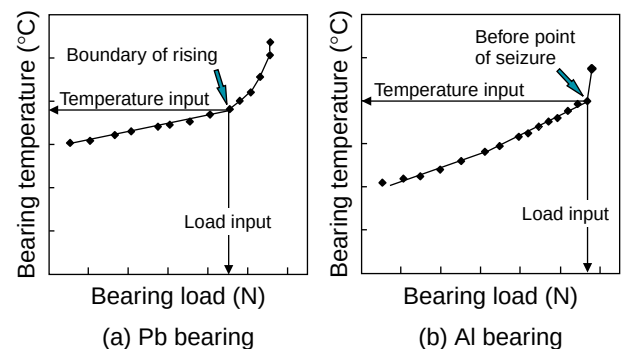


Fig.13 Definition point of calculation

5.2. Calculation Results

Fig.14 shows the results of the bearing hydrodynamic calculations. The graph also shows two types of bearing material. The first is the standard lead alloy material. The second is a lead + indium (Pb + In) material in which the overlay is softer than the standard lead alloy as a result of the added indium.

Table 2 shows the hardness of the bearing surfaces used in these tests.

When the minimum oil film thickness was plotted on the horizontal axis, and the maximum oil film pressure was plotted on the vertical axis, the lead alloy bearing demonstrates a horizontal distribution, while the aluminum alloy bearing can be seen to demonstrate a vertical distribution.

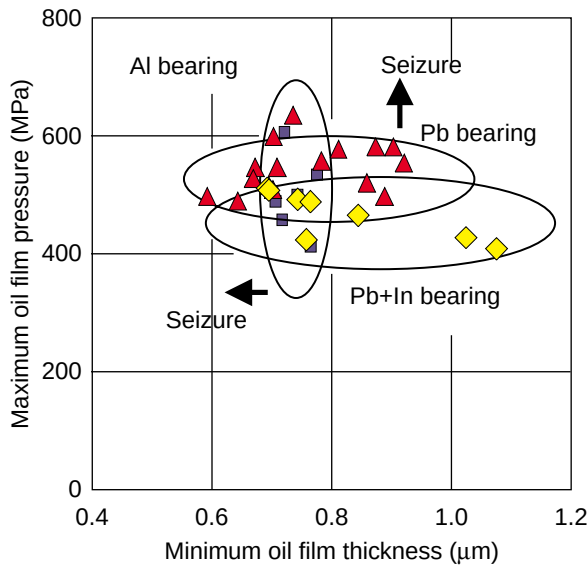


Fig.14 Calculation result

Table 2 Bearing surface hardness

Material	Scale	Value
Pb bearing	HMV10g	13 ~ 17
Pb + In bearing	↑	10 ~ 13
Al bearing	HV5g	42

This indicates that, in the case of the lead alloy bearing, when the maximum oil film pressure becomes excessively high, plastic flow of the overlay occurs, resulting in a secondary rise in temperature that eventually leads to seizure.

In addition, the plastic flow of the overlay at a lower maximum oil film pressure, as with the lead-indium bearing, also serves to support the above-mentioned cause of seizure.

6. Conclusion

One aspect of the mechanism for bearing seizure was quantitatively clarified by solving the inherent problems of conventional bearing testers and improving the test procedure.

1. The plastic flow of the overlay that occurred with the copper-lead bearings material is thought to have led to seizure due to the high oil film pressure that was induced between the shaft and bearing.
2. Since aluminum alloy bearings are not provided with a soft overlay on the bearing surface in the manner of copper-lead bearings, there is a critical limit to the minimum oil film thickness rather than a critical limit to the oil film pressure, and this is thought to lead to seizure.

The demonstration of this critical limit to the maximum oil film pressure over a wide range of oil film thicknesses has been clearly shown to be an indicator for defining the minimum oil film thickness and seizure threshold when designing bearings.

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Analysis of the Accessory Belt Running Off of the Pulley in Automotive Engines

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ABSTRACT

Having established the measurement technology to measure accessory belt behavior using laser displacement measurement methods, we were able to clarify the mechanisms causing the belt to run off of the pulley by analyzing the correlation between the belt's behavior and fluctuations in input to the belt and by performing belt behavior simulation. Based on our findings, we were able to create a design tool. We clarified the fact that the belt runs off of the pulley if a resonance occurs due to the accessory belt's natural frequency equaling 1/2 of the frequency of belt length fluctuation in the belt tension direction caused by crankshaft torsional vibration and if the length fluctuation in the belt tension direction exceeds a threshold value. We also found that lateral vibration occurred due to the existence of a lateral movement natural vibrational mode in the belt itself. Based on our findings, we created a belt movement limit map and constructed a belt run off simulation enabling us to propose design tools to prevent the belt from running off of the pulley. In addition, we provided an example of a method used to investigate the prevention of the belt running off the pulley by presenting the effects of design parameters for crankshaft related parts.

1. Introduction

Increased interest in promoting global environmental protection over the years has led to calls for the development of automotive engines that are more environmentally sound. This movement has been paralleled by a demand for high engine speed, high output, the most basic aspects of engine performance. Therefore, research is needed to resolve the various problems that arise from engine vibration at high engine speeds.

High speed engines tend to increase the circumferential speed of an automotive engine's accessory belt (i.e., V-ribbed belt) and can potentially result in degraded belt performance. In fact, problems such as the belt running off of the pulley have not been uncommon in actual engine development. However, the vibrational mode and mechanism that are behind the causes of a belt running off of the pulley have not been determined and so have resulted in enormous amounts of time and effort to develop countermeasures. V-ribbed belts used as accessory belts create a spring/mass vibration system between the crankshaft pulley and the various accessories (e.g., alternator, air conditioning compressor, and power steering pump) such that torsional vibration occurring in the crankshaft is transmitted from the crankshaft pulley to the accessories via the belt. Various

vibrational modes can be exhibited at this time depending on such factors as the accessories' inertial moment and belt elasticity. Research on belt behavior and rotational fluctuation in accessory components has not been extensive, however, due to the lack of quantifying technologies. Today, however, advanced instrumentation and sophisticated technologies for data processing make it possible to investigate multiple phenomena simultaneously and in real time. Thus, analysis of the behavior of high-rpm pulleys and accessory belts is now possible.

This paper discusses a measurement method for the behavior and quantitative displacement of accessory belts in high speed engines, the mechanism of a belt running off of a pulley, how the related quantitative limits are established, and the process of defining specifications for components related to the accessory belt.

2. Analysis of the Mechanism of a Belt Running Off of a Pulley

2.1. Overview of the Accessory Belt System

Research on why an accessory belt runs off of a pulley began with fixed-tension, biaxial systems having one accessory (Fig.1) but has expanded to include systems with up to two accessories and three axes. This paper, however,

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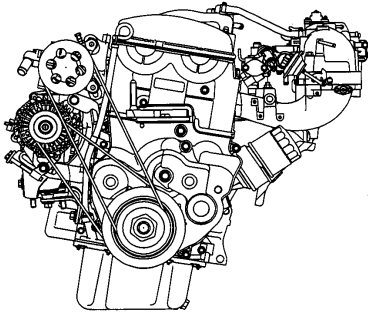


Fig.1 Structure of accessory belt system

deals with biaxial systems. The engine used was the 2.3-liter DOHC VTEC (Valve Timing and lift Electronic Control) engine shown in Fig.1.

2.2. Factors that Degrade Belt Behavior

The authors have investigated the factors that degrade belt behavior by comparing the fluctuations in angular velocity and axis direction, etc., that occur in the two pulleys and the belt behavior in the portion not in contact with the pulleys. This research assumes that the belt portion around the crankshaft and accessory pulley functions integrally, with no slippage or lift relative to the pulley.

Figs.2-a and 2-b show the variables used as influential factors in the analysis, along with the symbols for those

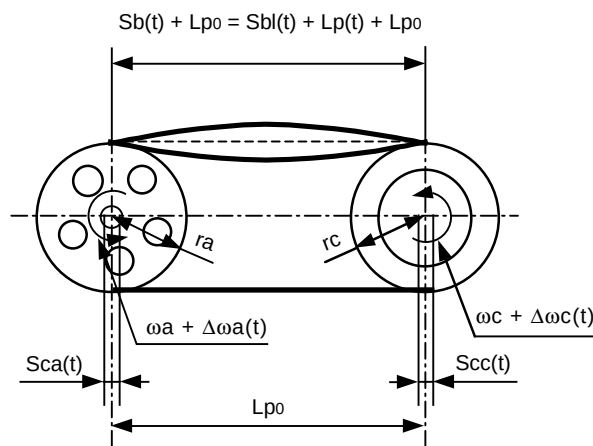


Fig.2(a) Factors in the changing length of the accessory belt between the accessory pulley and the crankshaft pulley

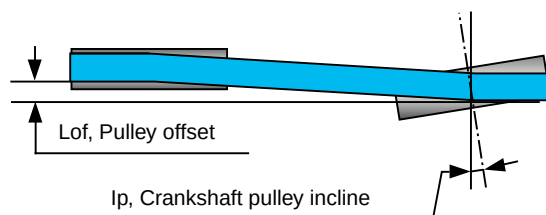


Fig.2(b) Factors in crankshaft axial direction fluctuation

variables. The two factors contributing to the belt running off of the pulley are described below.

- (1) Belt length fluctuation in the belt tension direction ($S_b(t)$, mm): Fluctuation in the length of the belt portion not in contact with the pulleys.

This is due to fluctuation in pulley circumferential movement length caused by fluctuation in the angular velocity of the crankshaft and accessory pulley (hereinafter referred to as crankshaft pulley circumference fluctuation and accessory pulley circumference fluctuation, expressed as $S_{pc}(t)$ and $S_{pa}(t)$, respectively), and to fluctuation in the belt length of the portion between but not in contact with the crankshaft pulley and accessory pulley axes due to pulley run-out ($L_p(t)$). Thus, $S_b(t)$ is expressed with the following equation.

$$S_b(t) = S_{bl}(t) + L_p(t) = S_{pc}(t) + S_{pa}(t) + S_{cc}(t) + S_{ca}(t) \quad (1)$$

As expressed in equations (2) and (3) below, each pulley's circumferential fluctuations $S_{pc}(t)$ and $S_{pa}(t)$ can be determined by integrating angular velocity fluctuations ($\Delta\omega_c(t)$ and $\Delta\omega_a(t)$) and multiplying this by the radius of each pulley (r_c and r_a). The angular velocity fluctuation is found by subtracting the average angular velocity ($\omega_c(t)$ and $\omega_a(t)$) from the crank pulley and accessory pulley angular velocity.

$$S_{pc}(t) = r_c \int \Delta\omega_c(t) dt \quad (2)$$

$$S_{pa}(t) = r_a \int \Delta\omega_a(t) dt \quad (3)$$

$\Delta\omega_c(t)$: Crankshaft pulley angular velocity fluctuation
 $\Delta\omega_a(t)$: Accessory pulley angular velocity fluctuation
 r_c : Crankshaft pulley radius
 r_a : Accessory pulley radius

- (2) Fluctuation in the thrust direction ($S_{bs}(t)$, mm): $S_{bs}(t)$ is defined as fluctuation in the pulley axial thrust direction occurring between the pulleys (i.e., in the portion not in contact with the pulleys) due to axial incline in the crankshaft pulley and accessory pulley, and to pulley offset (L_{of} , mm), shown in Fig.2-b.

Thus it is assumed that length fluctuation in the belt tension direction ($S_b(t)$) involves the belt running off of a pulley due to flexure caused by an increase in belt length between the pulleys; and, that fluctuation in the thrust direction ($S_{bs}(t)$) involves belt lateral offset caused by belt fluctuation in the lateral direction during operation.

2.3. Measurement Methods

2.3.1. High-Speed Camera Method

High-speed cameras (NAC model HSV-500) were used to visually capture belt behavior leading up to the belt running off of the pulley. The two cameras were positioned facing the back surface and a side surface of the belt (Fig.3-a), and were used to videotape the belt at a rate of 250 frames per second. Fig.3-b is an example of the images that were taken. The backface of the belt was painted with ladder-like markings in order to facilitate analysis of belt movement.

2.3.2. Laser Displacement Measurement

A laser sensor (Keyence LX2-03; Fig.3-a) was used to

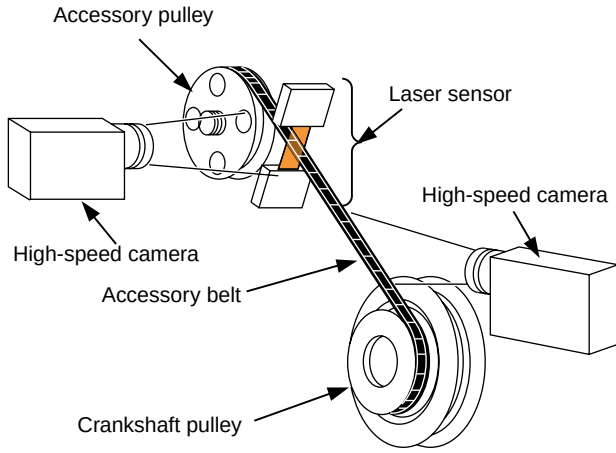


Fig.3(a) Measurement methods of accessory belt movement using a high-speed camera and laser sensor

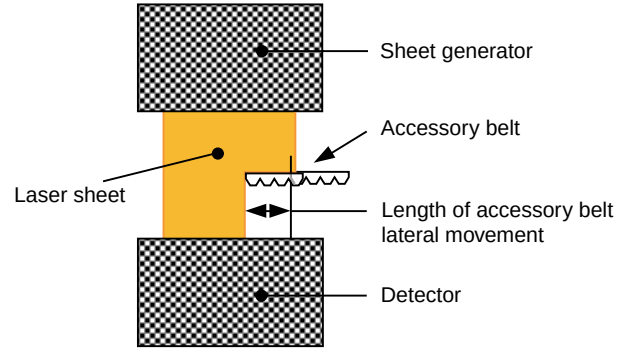


Fig.4 Structure of laser sensor

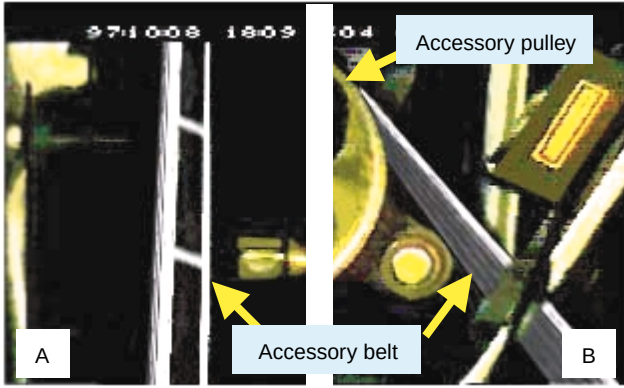


Fig.3(b) Images of the belt's back-surface (A) and side-surface (B) taken with high-speed cameras

precisely quantify belt behavior in the lateral direction after capturing overall belt behavior using the high-speed cameras. This sensor uses a band-shaped laser beam and a linear sensor containing a charged-coupled device (CCD), and provides an output that is proportional to the amount of laser light that is blocked (Fig.4). The setup shown in Fig.4 therefore yields an output that is proportional to belt displacement in the lateral direction. The laser sensor has a response frequency of 256 Hz and a spatial resolution of 0.5 mm, and can measure engine speeds of up to 7700 rpm.

2.3.3. Pulley Circumference Fluctuation

Crankshaft pulley circumferential fluctuation ($S_{pc}(t)$) and accessory pulley circumferential fluctuation ($S_{pa}(t)$) were calculated from the angular velocity fluctuation of each pulley axis ($\Delta\omega_c(t)$ and $\Delta\omega_a(t)$) using equations (2) and (3), and were determined as described below.

First, an accelerometer was mounted to each axis by means of a jig to measure acceleration ($g_c(t)$, $g_a(t)$) at the given mounting radii (crankshaft pulley = r_{ac} , accessory pulley = r_{aa}). These were integrated to determine angular

velocity fluctuation ($\Delta\omega_c(t)$ and $\Delta\omega_a(t)$). The reason for using two accelerometers was to cancel out components other than angular acceleration fluctuation that can exist when pulleys undergo eccentric motion in opposing sensitive directions.

$$\Delta\omega_c = \frac{1}{r_{ac}} \int g_c(t) dt \quad (4)$$

$$\Delta\omega_a = \frac{1}{r_{aa}} \int g_a(t) dt \quad (5)$$

2.3.4. Pulley Fluctuation

Fluctuation due to pulley rotation most likely includes eccentricity of the pulley axis and pulley axis incline relative to the belt line (Fig.2-a). The former was determined by measuring the distance between pulley periphery and two gap sensors positioned at a 90° phase angle relative to each pulley (Fig.5-a).

In the latter, pulley axis mounting and flexion related to belt tension are believed to be factors, as the pulley side surfaces do not move in the same plane. This fluctuation was measured with gap sensors positioned as shown in Fig.5-b.

3. Experimentation: Results and Observations

3.1 Belt Behavior

Observation of belt behavior using the high-speed cameras and laser sensors revealed four vibrational modes (Fig.6), and showed that the phenomenon of a belt running off of the pulley is caused by lateral movement vibration (Fig.6-d). Fig.7 illustrates positional fluctuation in lateral movement at belt cross-section C-C indicated in Fig.6-d. This positional fluctuation is belt movement in the thrust direction, and also indicates movement of the belt back surface to the outside due to torsion. The images in Fig.3-b clearly show this torsional state: in the left-hand image, belt torsion has caused the markings to slant, whereas in the right-hand image belt torsion is causing the cross-sectional width to change in the manner illustrated in Fig.7. Belt movement in the thrust direction is as great as 30 mm, which is equivalent to two times the pulley width.

To identify the causing behind belt lateral movement vibration, the authors used gap sensors to measure pulley offset, pulley eccentricity, and pulley side surface incline,

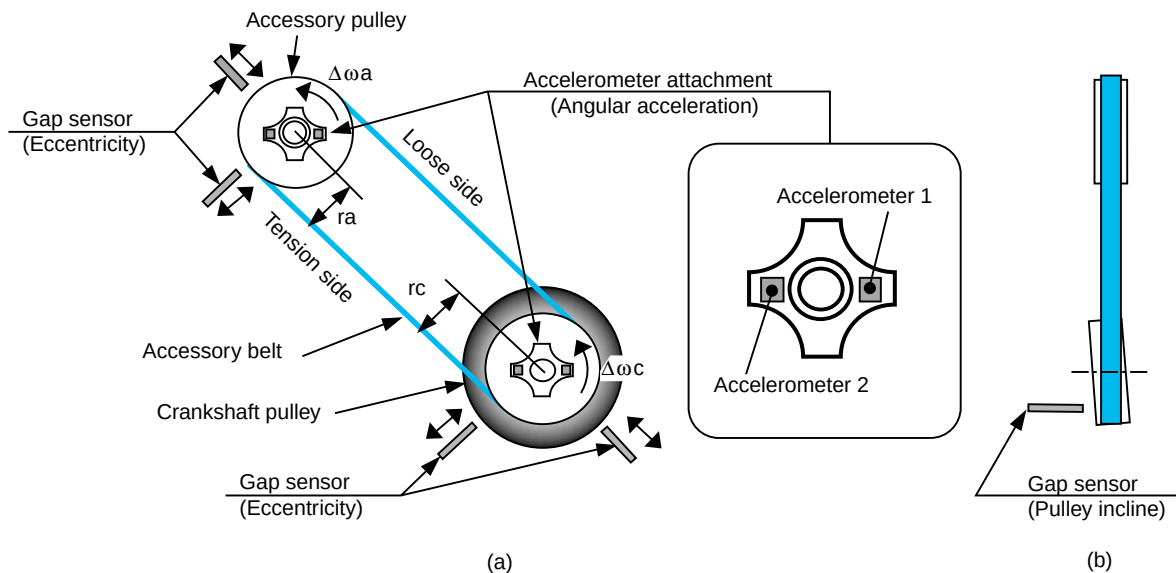


Fig.5 Measurement methods of fluctuation input to accessory belt

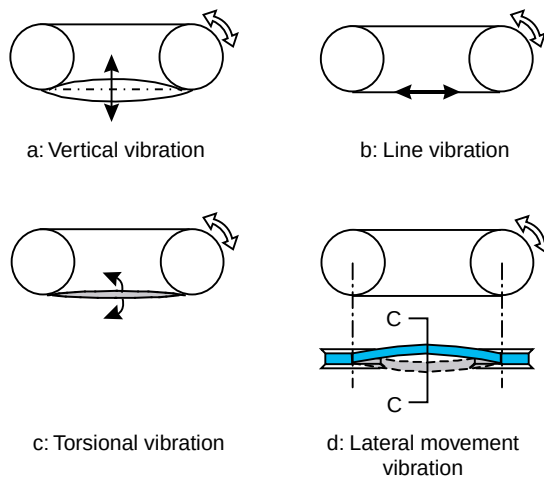


Fig.6 Vibration patterns

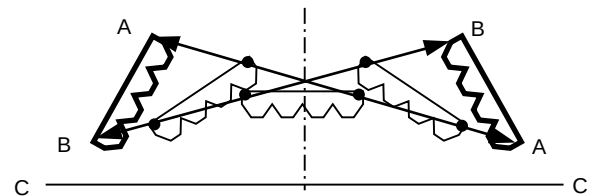


Fig.7 Lateral movement vibration at cross section C-C in Fig.6. Arrows show movement of both sides of the belt

Table 1 Effect level of fluctuation input to belt on belt behavior

Direction of fluctuation	Items	Influence
Crankshaft axis direction	Offset of pulley alignment	Negligible
	Pulley incline	Negligible
Belt tension direction	Pulley eccentricity	Negligible
	Fluctuation on pulley circumference (by revolution)	Strong

but the resultant measurements were not significantly large. This was true with measurements taken up to the maximum allowable values for mass production control. Thus, no causal relationship was detected. The results, shown in Table 1, suggest that belt lateral movement vibration is largely influenced by belt length fluctuation in the belt tension direction, which was therefore made the focus of analysis in subsequent experimentation.

3.2. Belt Length Fluctuation in the Belt Tension Direction and Lateral Movement Fluctuation

Pulley circumference fluctuation ($S_{pc}(t)$, $S_{pa}(t)$) and pulley axis run-out ($S_{cc}(t)$, $S_{ca}(t)$) are possible factors in belt length fluctuation in the belt tension direction ($S_b(t)$). As pulley axis run-out has been verified in ordinary engines as being only one-tenth of pulley circumference fluctuation, it was decided that focus would be placed only on pulley circumference fluctuation. Fig.8 gives the waveform of crankshaft pulley circumferential fluctuation ($S_{pc}(t)$) and accessory pulley circumferential fluctuation ($S_{pa}(t)$)

calculated with the accelerometer output shown in Fig.5-a for engine speeds of 2000 rpm and 7500 rpm, which are the experimental conditions used to represent stable movement and lateral movement vibration, respectively, in an accessory belt. In the former case, the circumferential fluctuation waveform of both the crankshaft pulley and accessory pulley exhibit the same phase, whereas in the latter case, there is a phase difference of approximately 80° relative to the crankshaft angle (hereinafter " $80^\circ CA$ "). Pulley circumferential fluctuation, which is a second order component relative to the crankshaft angle, is such that the

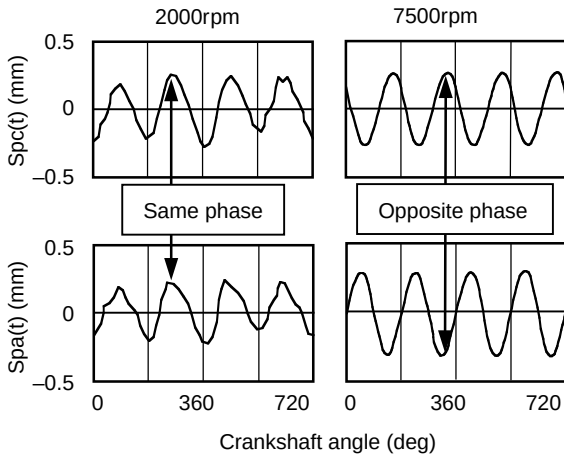


Fig.8 Wave fluctuation on pulley circumference

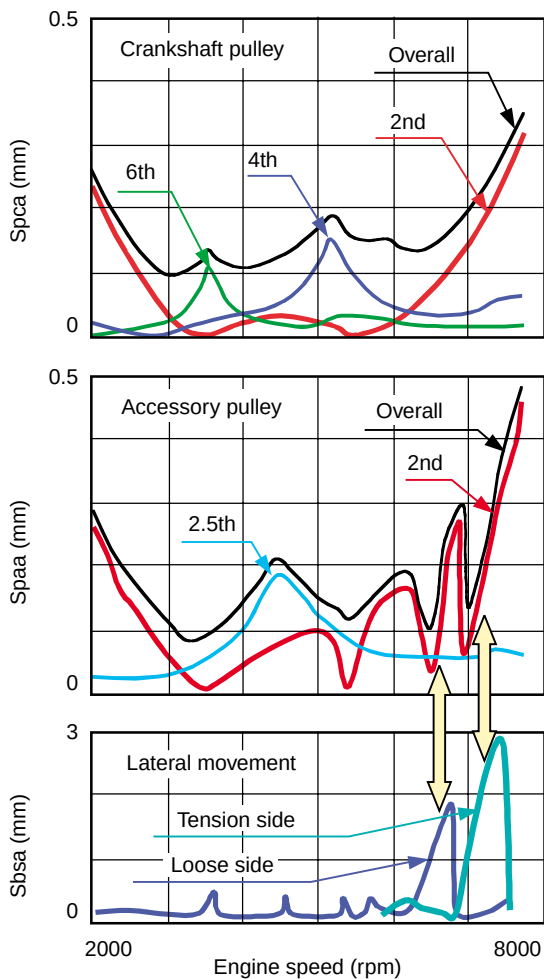


Fig.9 Characteristics of Spca, Spaa and Sbsa vs. engine speed

80°CA phase difference between the two pulleys results in approximately opposite phases. Fig.9 shows the results of using fast Fourier transform analysis (FFT) to track the crankshaft pulley and accessory pulley circumferential fluctuation ($Spc(t)$ and $Spa(t)$) for the given engine speeds; the trough-to-peak amplitude of belt lateral movement is also shown for comparison. The vertical axis shows, from the

top down, the amplitude of crankshaft pulley circumferential fluctuation ($Spca$, mm), amplitude of accessory pulley circumferential fluctuation ($Spaa$, mm), and amplitude of accessory belt lateral movement ($Sbsa$, mm). Note that the amplitude of both the crankshaft pulley and the accessory pulley circumferential fluctuations have a second order component that rises in the high-rpm zone, and that the amplitude of belt lateral movement also tends to rise along with it. The second order component of $Spaa$ in Fig.9 suddenly begins to converge at an engine speed of 6300 rpm, then rises sharply at 6500 rpm. This corresponds with the changes in loose-side lateral movement amplitude ($Sbsa$), indicated by arrows in Fig.9. Similarly, the sudden convergence and sharp increase in the second order component of $Spaa$ beginning at 6800 rpm corresponds with the fluctuation in $Sbsa$ on the tension side.

Next, Fig.10 shows the second order components of the amplitude of length fluctuation in the belt tension direction, which is a composite value that accounts for the phases of $Spcc(t)$ and $Spca(t)$, as well as the amplitude of belt lateral movement. As engine speed rises from 2000 to 7700 rpm, the waveforms of the crankshaft pulley and the accessory pulley circumferential fluctuations go from having the same phase to opposite phases (Fig.8). Consequently, the amplitude of length fluctuation in the tension direction is added on rather than being cancelled out, showing an increasing value that results in a marked lateral movement amplitude in the area where belt length fluctuation in the

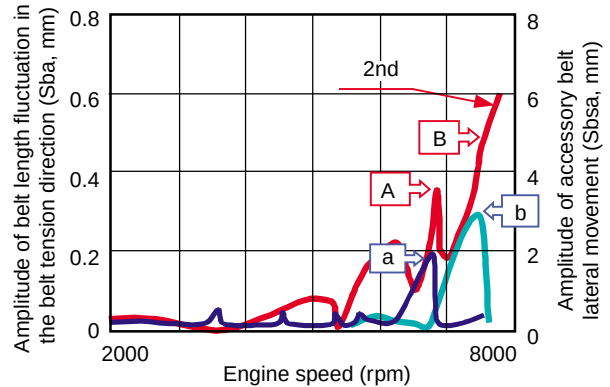


Fig.10 Characteristics of Sba and Sbsa vs. engine speed

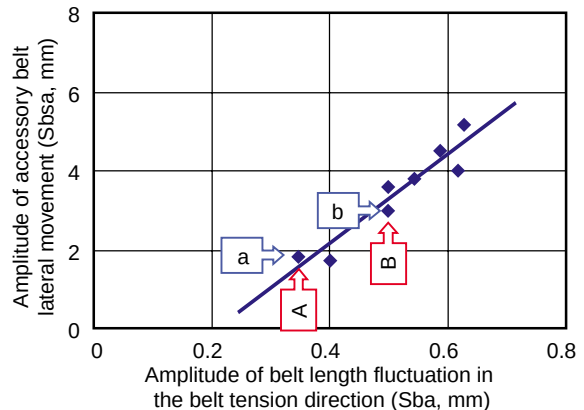


Fig.11 Relationship between Sba and Sbsa

belt tension direction (Sba) exceeds 0.2 mm. Fig.11, in which the amplitude of belt lateral movement from Fig.10 is the vertical axis and the amplitude of belt length fluctuation in the belt tension direction is the horizontal axis, adds the tension-side and loose-side points from Fig.10 (A, B, a, b) and measurements of maximum value for different initial belt tension settings. This graph indicates good correlation between the belt's loose and tension sides, as evinced by the linear distribution of both. Fig.12 shows engine speed and the lateral movement vibration frequency (Hz) at which the accessory belt exhibited lateral movement at different initial tension settings. The one-to-one correlation between the two frequencies suggests that the belt vibration frequency is the belt's natural frequency for each initial tension. The reason that engine speed at peak lateral movement vibration is lower on the loose side in Figs.9 and 10 is that because belt tension is lower on the loose side than on the tension side due to the accessory's rotational resistance, the belt's natural vibrational frequency is also lower on the loose side. Combining the results from Figs.10 and 11 shows that belt lateral movement vibration occurs when engine speed and the belt's natural vibrational frequency corresponds in an engine speed area in which the amplitude of belt length fluctuation in the belt tension direction is 0.2 mm or greater.

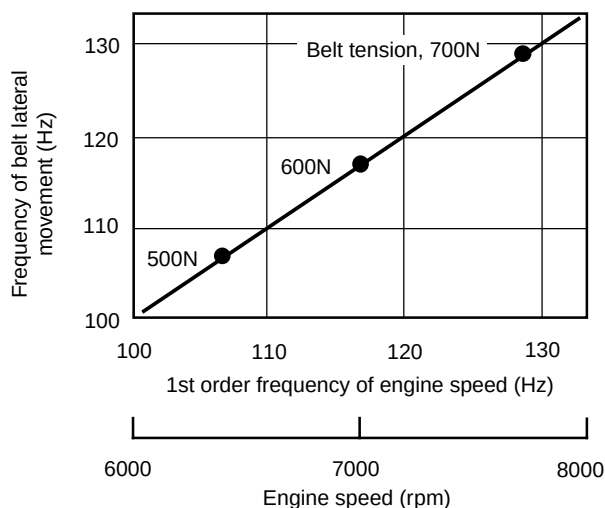


Fig.12 Relationship between engine speed and frequency of belt lateral movement when varying to belt tension

3.3. Analyzing the Mechanism of Belt Lateral Movement

3.3.1. Lateral Movement Vibrational Modes

The preceding has shown that when engine speed is high enough to create large belt length fluctuation in the belt tension direction and also matches the belt's natural vibration frequency, the belt resonates and undergoes vibrational movement in the lateral direction. In this section, the mechanism of this lateral movement is considered. Assuming one factor behind this lateral movement is the fact that the belt itself has a natural vibrational mode at which lateral movement occurs, the authors used a finite element model

to perform nonlinear natural vibrational mode analysis.

The one-dimensional string model generally used for approximation in belt vibration analysis, although capable of predicting string vibrational mode, could not predict the vibrational mode of the type of lateral movement seen in actual engines. This is believed to be because this technique cannot account for the asymmetry of mass and stiffness in a belt cross section, which is determined by the core position and rib height. For this reason, the authors performed analysis with a finite element method based on a three-dimensional belt model having the belt shape shown in Fig.13. This model was comprised of a high-stiffness core portion which primarily bears the tensile load, and a rubber portion, with a 100:1 ratio between the stiffness of the former and that of the latter, and an overall belt spring constant of 530 N/mm. The belt length and width used in the calculations were the same as those of the actual belt (352 mm and 14 mm, respectively). The calculations sought to determine the natural vibrational mode with one end of the belt anchored and the other end subjected to a tensile load in the belt tension direction.

Fig.14 shows the results of this analysis. Figs.14-a and 14-b, which view the belt obliquely from above and slightly from above in the belt's longitudinal direction, respectively, show lateral slip to the right and left. The lateral movement from the center line in Fig.14-b and the upward movement of one end in Fig.14-a conform to the lateral movement

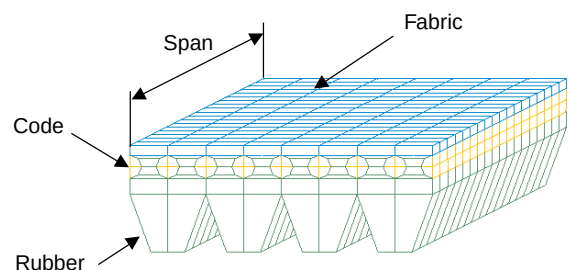


Fig.13 Belt model for FEM

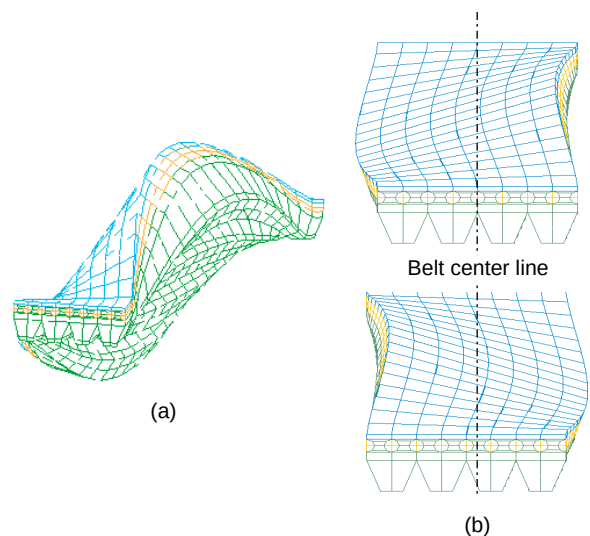


Fig.14 Vibrational mode analysis results of the accessory belts lateral movement

observed in an actual belt. This corroborates the aforementioned findings that lateral movement is a result of the accessory belt's natural vibrational mode in the direction of lateral movement.

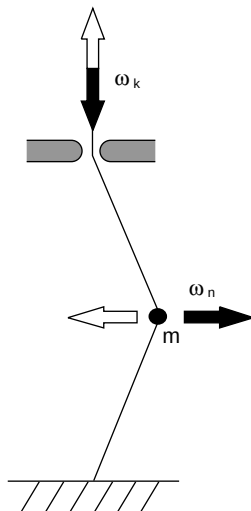
3.3.2. Lateral Movement Resonance

Belt vibration, which is a manifestation of changes in tensile load due to fluctuation in pulley angular velocity at the ends of the belt, can be approximated as vibration of a string having mass, as it can be thought of as vibration in a variable elastic system. The vibrational model is shown in Fig.15. The Mathieu equation can be used to judge whether a variable elastic system is stable or unstable (i.e., undergoing resonance). As a precise solution is difficult to arrive at, the authors focused instead on whether the system is stable or unstable at a given time. If stable, then the system is undergoing periodic motion when a disturbance is present. If unstable, then its amplitude increases with time, eventually resulting in resonance. Using the stability criteria of the Mathieu equation, and given a frequency of belt tension fluctuation of ω_k and a belt vibration frequency of ω_n , as in Fig.15, then motion becomes unstable when⁽¹⁾

$$\omega_n = 1/2\omega_k, \omega_k, 3/2\omega_k, \dots$$

If ω_n is substituted with the natural frequency of belt lateral movement (ω_s) and ω_k with the frequency of belt length fluctuation in the belt tension direction (ω_b), then the first solution ($\omega_n = 1/2 \omega_k$) suggests that resonance occurs when the natural frequency of belt lateral movement ω_s equals one-half of the frequency of belt length fluctuation in the belt tension direction (ω_b). As the frequency of belt length fluctuation in the belt tension direction (ω_b) is equal to twice the engine speed as in Fig.8, this means that $1/2 \omega_b$ is equal to the engine speed, thus agreeing with the observed results in Fig.12.

When applied to lateral movement vibration in an actual engine, this means, in regards to resonance, that belt tension decreases as mass point m moves away from the center but



ω_k ; Frequency of belt tension fluctuation
 ω_n ; Frequency of belt vibration

Fig.15 Vibration model of an accessory belt

increases as mass point moves toward the center, with an amplitude that is larger than in the latter states and which most likely increases with time. Moreover, this state of resonance is believed to actually occur as the lateral movement vibration mode in Fig.14, and to be what causes the belt to run off of the pulley.

3.4. Criteria for Belt Running Off of the Pulley

In the analysis described in the preceding section (3.3), belt lateral movement was analyzed with the specifications listed as "Sample 1" in Table 2. Next, the authors sought to develop criteria for the belt running off of the pulley based on belt specifications by conducting an experiment with different combinations of belt ribs and belt string length (Table 2). Fig.16 shows, for each number of ribs, the belt length fluctuation in the belt tension direction that caused a belt of the given length to run off of the pulley. This shows that a greater belt length or smaller number of ribs causes the belt to run off of the pulley even with a low belt length fluctuation in the belt tension direction. Thus, when belt length and belt length fluctuation in the belt tension direction are below the line for a given number of ribs, then the belt will not run off of the pulley, and the design can be judged appropriate.

Although belt length fluctuation in the belt tension direction and belt lateral movement are both possible variables for criteria for belt running off of the pulley, the

Table 2 Variations of the tested belt

	Number of rib			Belt string length (mm)		
	4	5	6	282	317	352
Sample 1	○					○
Sample 2		○				○
Sample 3			○	○		
Sample 4		○		○		
Sample 5			○		○	
Sample 6		○			○	

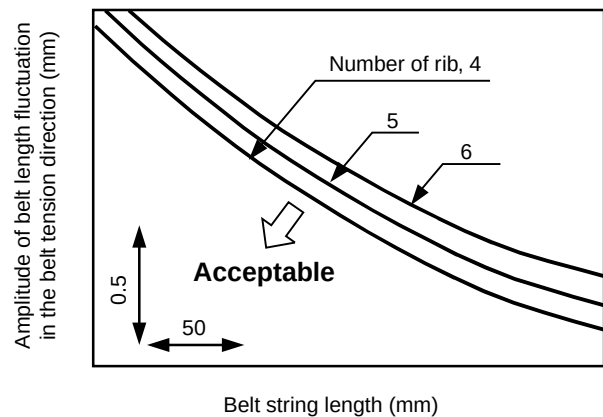


Fig.16 Judgement graph for the accessory belt running off of the pulley

authors selected the former because of its good reproducibility and used it in simulation of a belt running off of the pulley, described in the following section.

4. Design Tools

This section uses the analysis results described in the previous section to describe “simulation of the accessory belt running off of the pulley”. This is a design tool that can be used to predict and prevent an accessory belt running off of the pulley when designing components related to the belt. This process is described using an example application.

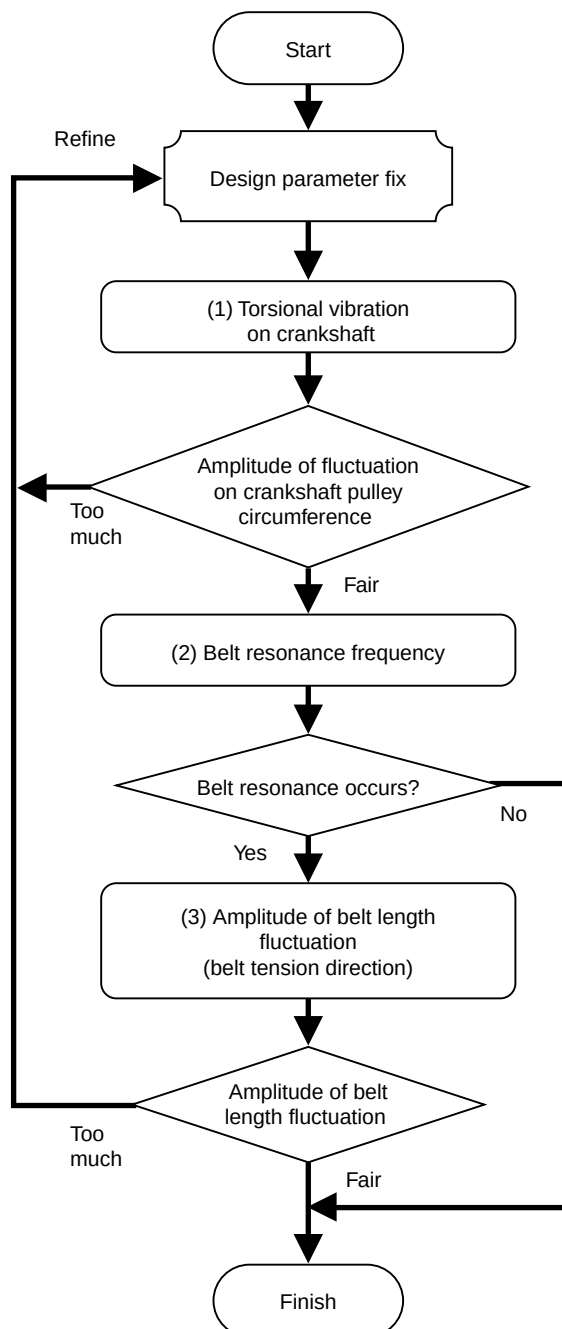


Fig.17 Design flow

4.1. Overview of Simulation

As the flowchart in Fig.17 illustrates, the simulation of an accessory belt running off of the pulley comprises these primary parts: (1) crankshaft torsional vibration simulation, (2) belt natural frequency simulation, and (3) simulation of belt length fluctuation in the belt tension direction, the input/output unit, and the judgment unit.

(1) Crankshaft torsional vibration simulation

Crankshaft torsion analysis entails calculating (with the Holzer equation) the angle of torsion from crankshaft dimensions and combustion pressure.

(2) Belt natural frequency simulation

Belt vibration analysis entails calculating the belt's natural frequency from belt specifications and accessory layout to determine whether the belt natural frequency falls within the engine's operating speed. If it does not, then belt vibration will not occur, and so design review is concluded at this point.

(3) Simulation of belt length fluctuation in the belt tension direction

Analysis of belt length fluctuation in the belt tension direction entails using the analysis model in Fig.18 to perform transient response analysis to calculate a fluctuation response that accounts for inertial mass on the accessory pulley side by defining excitation frequency as the crankshaft-side fluctuation obtained with crankshaft torsional vibration simulation. This yields the time series fluctuation shown in Fig.19. Taking phase difference in this fluctuation into account results in the

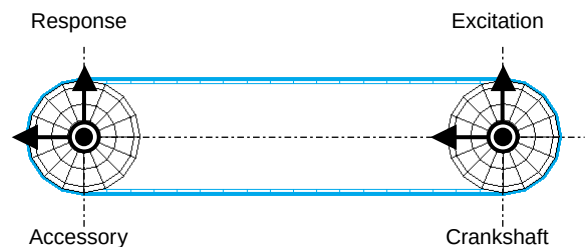


Fig.18 FEM model

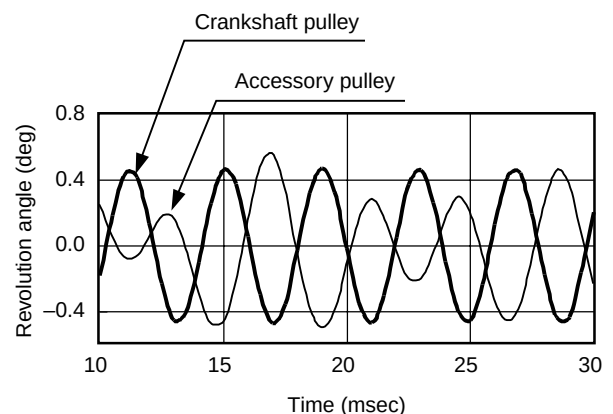


Fig.19 Simulation results of the crankshaft and accessory pulley angle of fluctuation during rotation

belt length fluctuation in the belt tension direction shown in Fig.20. This value is then evaluated with the aforementioned judgment map for belt length fluctuation in the belt tension direction (Fig.16).

4.2. A Review of an Example Using the Method for Preventing Belt Running Off of the Pulley

Fig.21 lists the parameters for assessing effectiveness in preventing a belt from running off of the pulley. Fig.22 shows those parameters in which improvement was clear. Accessory pulley hub inertial mass, crankshaft pulley damper frequency characteristics, and crankshaft pulley inertial mass were examined beyond the range of mass-production engines, but there was no significant improvement: belt lateral movement was roughly 90% that of the base engine in all cases. In contrast, substantial improvements were achieved for metal clearance (only 76% of the base engine at minimum tolerance), which affects crankshaft torsion at angle, and for journal diameter, which affects crankshaft rigidity: belt lateral movement was halved by increasing the diameter from 50 mm to 55 mm. In the latter case, the absolute value of belt lateral movement was reduced

substantially, from 4.2 to 2.4 mm, while a journal diameter of 55 mm resulted in no cases of the belt running off of the pulley despite its frequent occurrence with a 50 mm diameter journal in the experimentation (Fig.23).

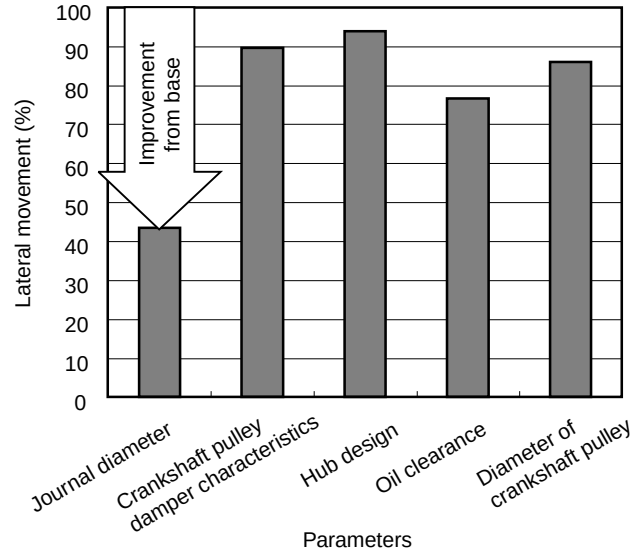


Fig.22 Effects of design parameters on accessory belt lateral movement

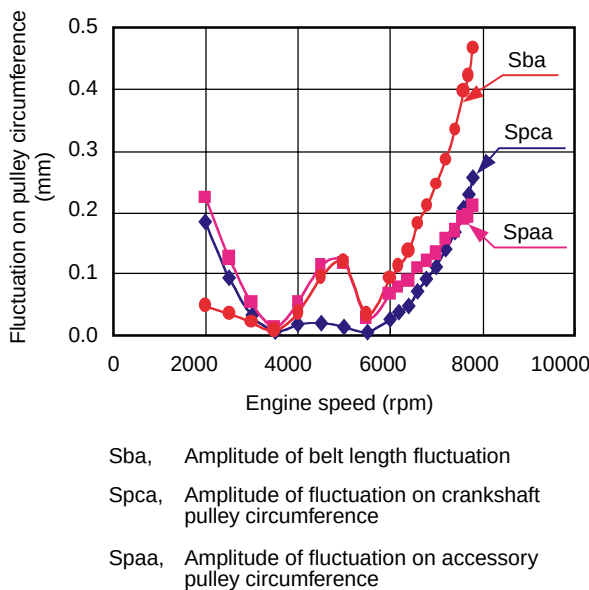


Fig.20 Simulation results of amplitude of fluctuation on pulley circumference

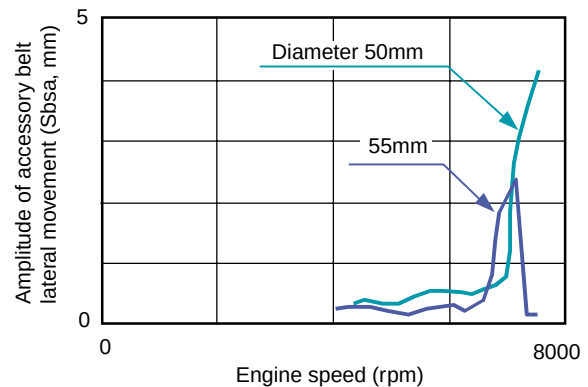


Fig.23 Effect of journal diameter size increase on accessory belt lateral movement

Design parameters for belt lateral movement in an engine

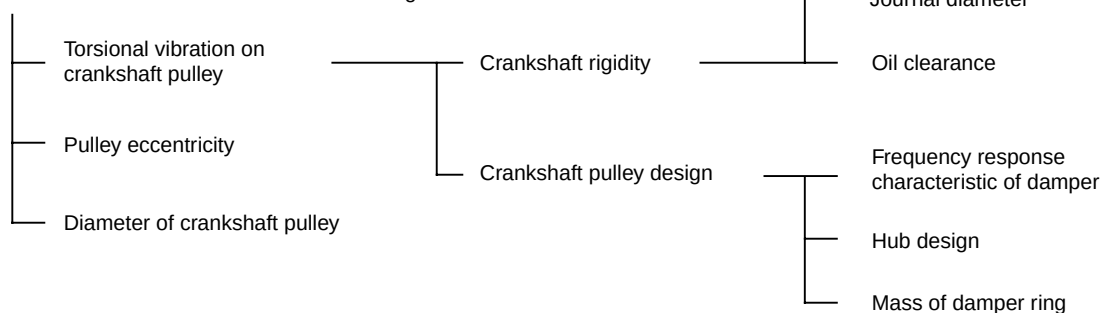


Fig.21 Design parameter tree

5. Conclusion

The authors established a laser sensor-based method for measuring and quantifying accessory belt behavior and elucidated the mechanism of a belt running off of the pulley by measuring belt movement and belt input, and by performing belt vibrational mode simulation, thus determining the following:

1. Phenomena leading up to accessory belt running off of the pulley
 - (1) Resonance occurs when the accessory belt's natural frequency equals 1/2 of the frequency of belt length fluctuation in the belt tension direction, said fluctuation caused by crankshaft torsional vibration.
 - (2) A correlation exists between belt lateral movement and belt length fluctuation in the belt tension direction, with the belt running off of the pulley when the threshold value is exceeded.
 - (3) Vibration in the lateral direction occurs as the accessory belt itself has a natural vibrational mode for lateral movement.
2. Achievements made possible by an understanding of the phenomena
 - (1) A "graph for use in judging the limit at which the belt runs off of the pulley" was created.
 - (2) Simulation of an accessory belt running off of the pulley was devised as a design tool.
 - (3) The above was shown to be effective in preventing the belt from running off of the pulley by improving design parameters for engine crankshaft components.

6. Acknowledgments

For their assistance with this research, the authors wish to express their appreciation to Mr. Hajime Maeda and Mr. Kenji Nakajima of the Tochigi R & D Center.

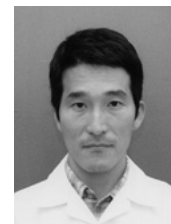
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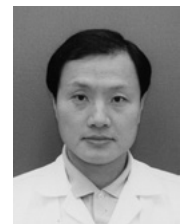
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Knocking Quantification and the Development of a Knock Intensity Meter

Teruyoshi MORITA* Masahiro SATO*

ABSTRACT

An algorithm which processes pressure signals in cylinders with a first frequency element and quantifies it based on its maximum value and statistic processing, and a real time measuring system utilizing the algorithm were established. To attain high applicability this system employs not only generally applied pressure sensors but also plug washer type sensors that can be applied to each cylinder for pressure measurements. Further, nearly all of the processing is performed using software applications. Rewriting of the software is made possible due to the simplified operations and the application of a VME board equipped with a high performance MCU processing built-in flash ROM. In consideration of expanded application in the future, the system, further, utilizes a special interface to enable automatic operation with the bench top automatic measuring system.

1. Introduction

Numerous manufacturers, primarily automobile manufacturers, are continually striving to develop revolutionary new technologies for decreasing fuel consumption, including the development of fuel-injected gasoline engines and hybrid automobiles, in an attempt to accommodate the environmental issues that have surfaced over the years. It is also important to decrease fuel consumption and reduce exhaust emissions by improving the basic combustion characteristics in the case of ordinary engine development.

In this race to develop new technology, efforts are also being made to develop complex control systems and high-precision measurement supporting techniques that are different from those in the past. We have developed a system for measuring knocking which is generally considered one of the limiting factors for improving the combustion characteristics of an internal combustion engine. This new system provides greater measurement accuracy than is possible with conventional methods by applying real-time measurement techniques that make it applicable to automated operation. A description of this new system is reported here.

2. Objectives of Knocking Quantification

The traditional assessment of knocking has relied largely on the sensory perception (hearing and vision) of the person performing the measurement. Errors consequently occur due to variations among individuals performing the measurement, differences among testing laboratories, and differences due to weather conditions. The result is an inadequate level of measurement accuracy.

Knocking has been quantified in the past by using

cylinder pressure sensors, acceleration sensors and other devices, but the complexity of the processing method and the need for a sophisticated CPU with a very large memory have made these methods unsuitable for quantifying and evaluating knocking on a real-time basis. In addition, since the number of parameters that need to be set during the course of engine development is becoming extremely large to match the increasing complexity of engines, testing will have to provide even greater efficiency in the future.

In consideration of these factors, the following four objectives were established for this development:

- 1) A method enabling the quantitative determination of knocking
- 2) Improved precision for knocking measurement
- 3) Hardware allowing the real-time evaluation of knocking
- 4) Automatic operation having an interface with a benchtop automatic measurement system

3. Quantification Method

3.1. Quantification Sensor Selection

It is necessary to select the most suitable sensor as the first step in quantifying knocking. Previous methods, which measured the cylinder pressure directly, used an acceleration sensor and evaluated sound. We evaluated the change in output of each type of sensor in the presence and absence of knocking for various ignition timing settings and analysis frequencies in order to select the most suitable sensor to quantify knocking. Figs.1, 2 and 3 show the results of the frequency analysis. Either of the results is an average of multiple cycles.

Each sensor demonstrated a change in output according to the ignition timing at a frequency of 7-8 kHz corresponding to the main component of knocking. In

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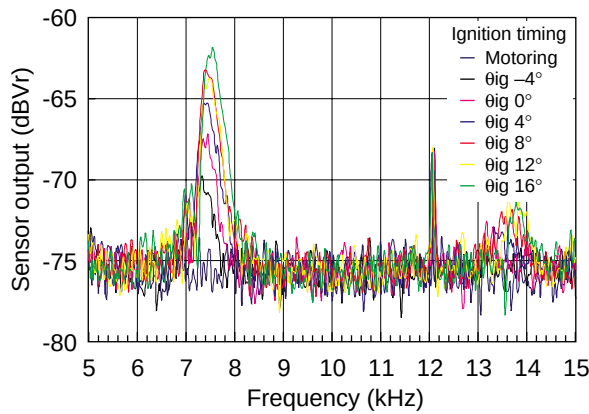


Fig.1 Frequency analysis by cylinder pressure sensor

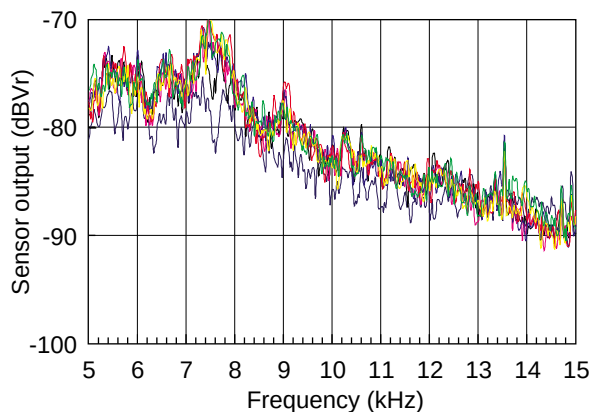


Fig.2 Frequency analysis by air-borne noise

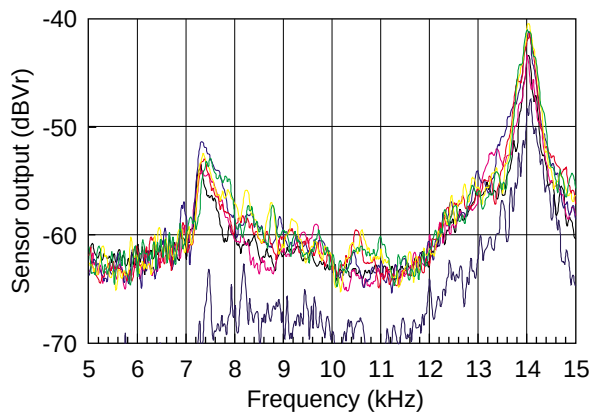


Fig.3 Frequency analysis by plug washer type sensor

addition, a similar change was observed in the frequency range of 13-15 kHz corresponding to the secondary component of knocking. Based on the results of the frequency analysis, attention was focused on the 7-8 kHz band. The results summarizing the maximum values of sensor output relative to the ignition timing are shown in Fig.4.

In the method whereby cylinder pressure is directly measured, as ignition timing advances, so does sensor output. However, the correlation between ignition timing and sensor

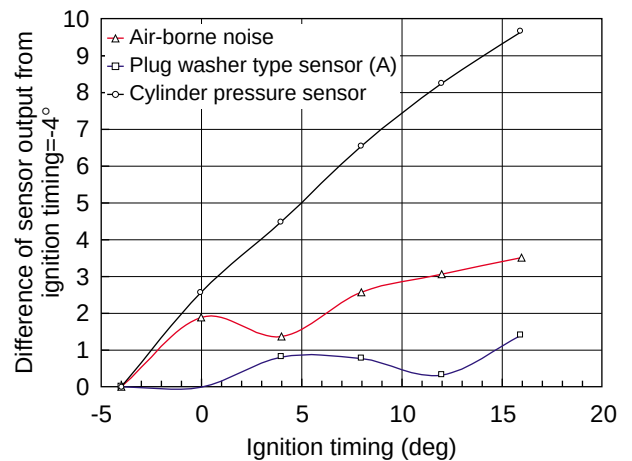


Fig.4 Sensor linearity by ignition timing

output is low with other sensors. In addition, the cylinder pressure sensor can be expected to contribute to improved measurement accuracy due to the large change in output relative to the degree of change in ignition timing. On the basis of these results, the method of directly measuring the pressure was selected for the knocking quantification.

3.2. Quantification Method

Examples of conventional methods that have been reported to quantitatively determine knocking by monitoring the cylinder pressure include one that integrates the heat generation rate and one that integrates this heat generation rate after filtering the cylinder pressure.^{(1)-(3), (5), (6)} However, these methods have the disadvantage of requiring considerable processing time, and from the viewpoint of real-time monitoring, cannot be considered suitable. We therefore decided to employ a method for quantifying knocking that involves filtering the cylinder pressure before statistical processing. Fig.5 shows how this analysis method is applied.

When focusing on the waveform after filtering, examples of the analysis method involve processing the maximum value, the amplitude, and the range between the maximum value and minimum value as shown in Fig.6. When varying ignition timing according to these methods, the output demonstrates a uniform relationship with the ignition timing with both of these methods as shown in Fig.7. Therefore all three methods are able to quantitatively identify knocking.

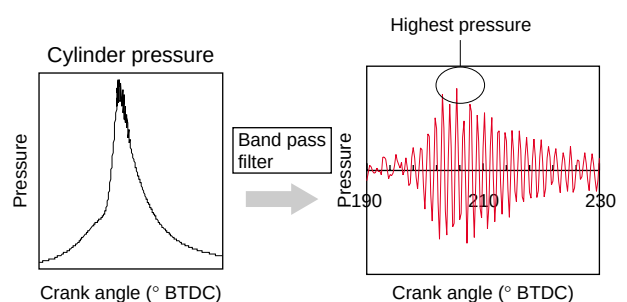


Fig.5 Analysis flow

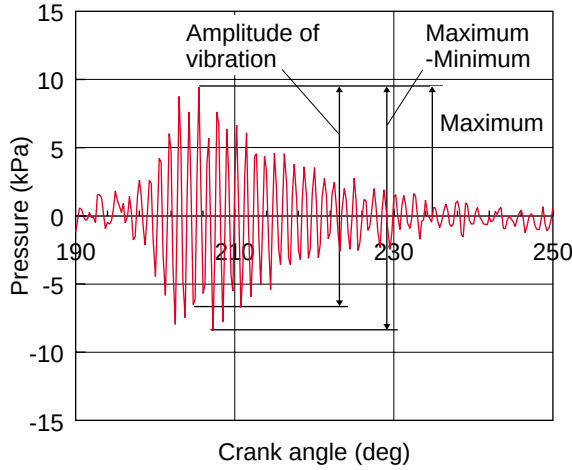


Fig.6 Parameters of knocking identification

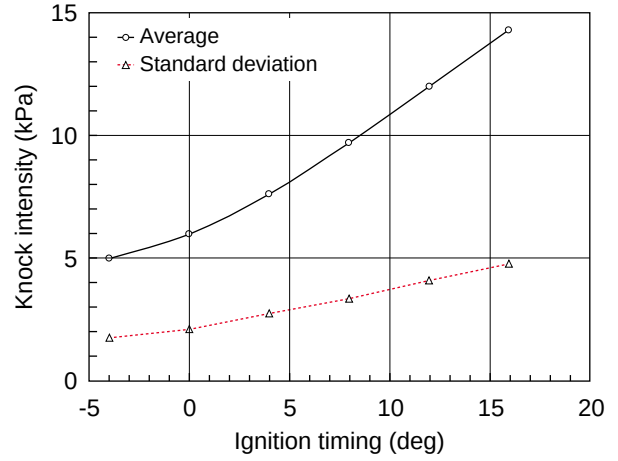


Fig.8 Average standard deviation of knock intensity

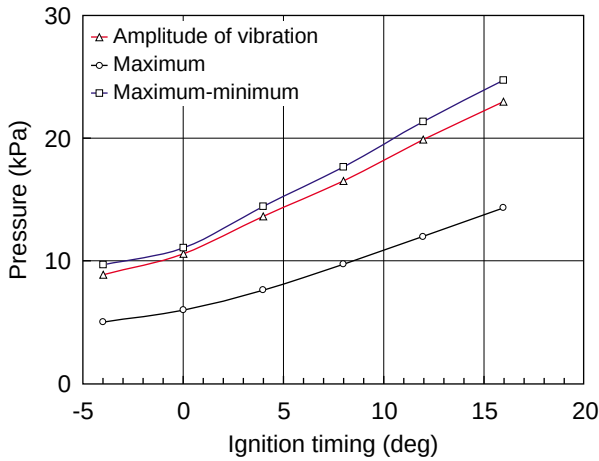


Fig.7 Comparison of analysis methods

We decided to apply the method based on the maximum value output by the cylinder pressure sensor because it requires the simplest processing. Since this maximum value is considered to represent the intensity of knocking, this value is defined as the knocking intensity.

3.3. Knocking Intensity and Frequency

While it is possible for a person to perceive knocking, this perception cannot discriminate between a single heavy knock and a series of continuous mild knocks. It is thus necessary to consider both the intensity and frequency to quantify knocking. Fig.8 shows the average value and standard deviation of the knock intensity according to the degree of ignition advance in the case of sampling 100 cycles. When the ignition timing is advanced, the average value of the knock intensity increases. This means that intense knocking caused by advanced ignition timing is occurring. Similarly, when focusing on the standard deviation of the knock intensity, the value increases with an increasing degree of ignition advance in the same manner as that of the average value.

Fig. 9 indicates the frequency distribution when the knock intensity from Fig.8 is plotted on the horizontal axis. The number of cycles for which the knock intensity exceeds the threshold level increases with increasing ignition advance. This means that the frequency at which knocking occurs (number of cycles) is increasing. On the basis of the foregoing data, knocking can be evaluated by processing the cylinder pressure through a band pass filter and focusing on the maximum value (knock intensity).

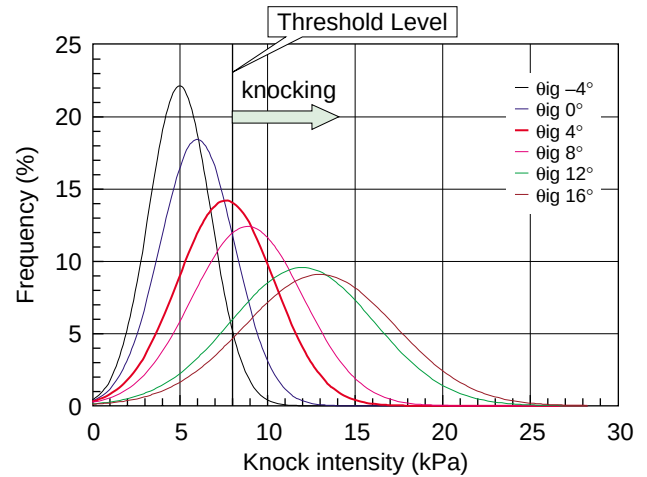


Fig.9 Distribution of knock intensity

3.4. Establishment of the Knocking Threshold Level

Since the conventional evaluation of knocking has been dependent on the sensory perception (hearing and vision) of the tester, it is necessary to maintain a correlation with the conventional method even when applying a new method for evaluating knocking. Fig.10 shows the knocking threshold level for various engine speeds. Although the threshold level rises with increasing engine speed, this is due to the increase in background noise accompanying higher engine speeds as a result of maintaining the correlation with hearing.

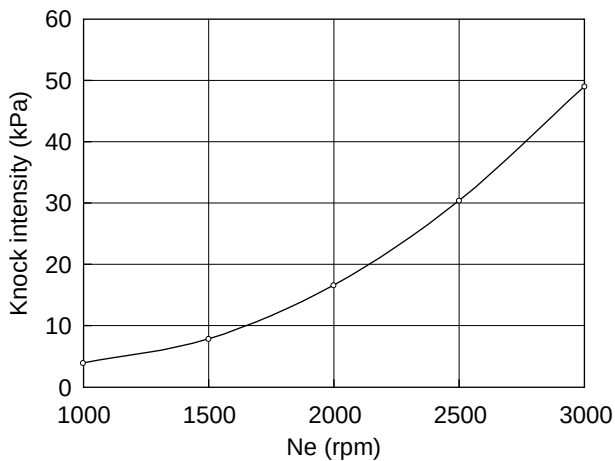


Fig.10 Threshold level table

3.5. Verification of the Quantification Method

The quantification method was verified by quantifying knocking according to the knock intensity as already described. The verification method consisted of changing various parameters, including the engine specifications, type and operating conditions. An example is shown in Fig.11. This method was applied to verify the difference in ignition timing at which knocking occurred when A/F was changed at Ne = 1500 rpm with a wide-open throttle (WOT). Although a difference of 2° occurred due to the difference in A/F at the threshold level as shown in the figure, this coincided with the difference in knocking that was determined by hearing. This quantification method thus demonstrated a high correlation with the conventional evaluation method, and can therefore be used with confidence to evaluate knocking.

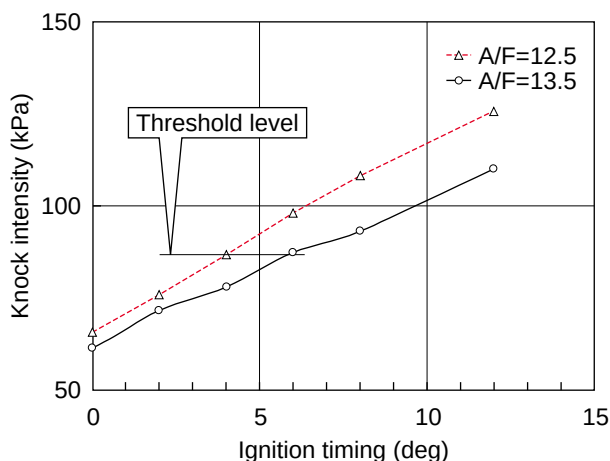


Fig.11 Verification of quantification methods

4. Evaluation of Knocking by a Spark Plug Washer Sensor

It is necessary to install a cylinder pressure sensor in every cylinder in order to perform a simultaneous on-board

evaluation of knocking for all cylinders. However, it is difficult to install a cylinder pressure sensor in every cylinder of a multi-cylinder engine due to the use of multiple valves and the closeness of adjacent cylinders. Therefore, a study was conducted using spark plug washer sensors. This method, however, involves the problem of the change in output according to the ignition timing not being clearly defined due to the poor S/N ratio of this type of sensor, even if the same evaluation method as that used for the cylinder pressure sensor is applied. Therefore, when using spark plug sensors, instead of the averaging multiple cycles, which was performed when using cylinder pressure sensors, individual cycles were examined and the number of cycles for which the knocking intensity exceeded a constant value was counted. The data were evaluated according to that frequency. A comparison between these two methods according to the knock intensity is shown in Fig.12. The evaluation method based on frequency can be seen to give a uniform relationship relative to the ignition timing, thereby allowing knocking to be evaluated.

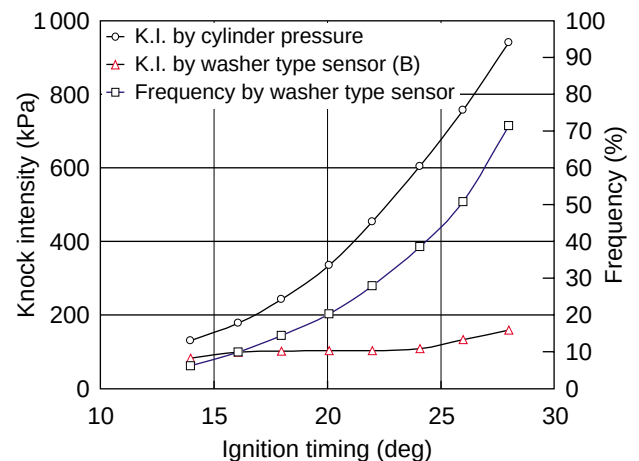


Fig.12 Comparison of quantification methods

5. System Configuration

5.1. Overview

The system block diagram is shown in Fig.13. The majority of the functions required by the system were achieved with suitable software in accordance with the previously stated objectives. In addition, Fig.14 indicates a timing chart for a given cycle starting from the ignition signal and sensor signal through to the determination of measurement timing, band pass filtering and detection of the maximum value (knock intensity).

5.2. Hardware

The basic configuration of the hardware is shown in Fig.15. This system was realized with a VME board containing the functions in the figure in a double-height, 2-slot package. Four of these boards were housed in a VME rack, a single board corresponding to the measurement of one cylinder, so that this system is capable of measuring four cylinders. The appearance of the knock intensity meter is shown in Fig.16.

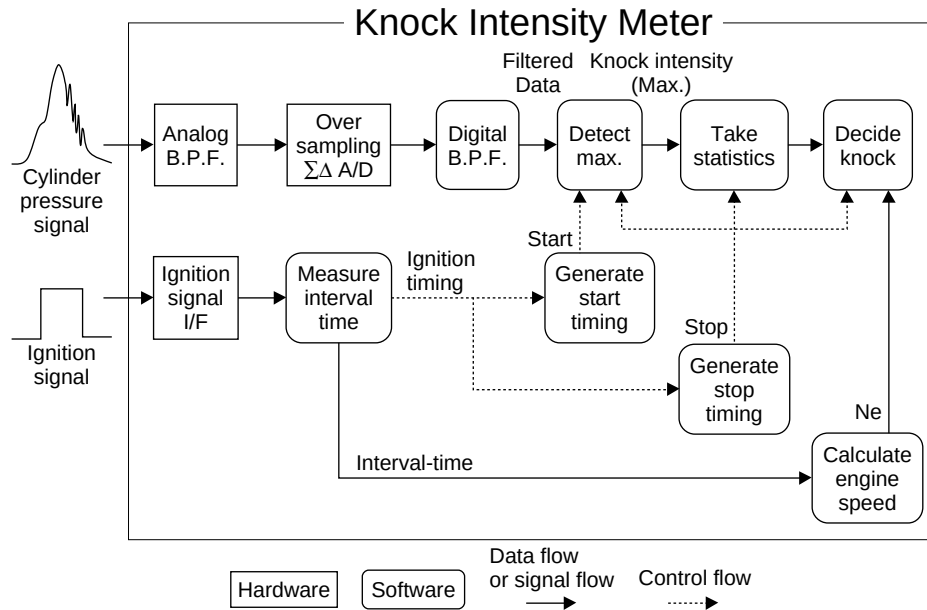


Fig.13 System diagram of knock intensity meter

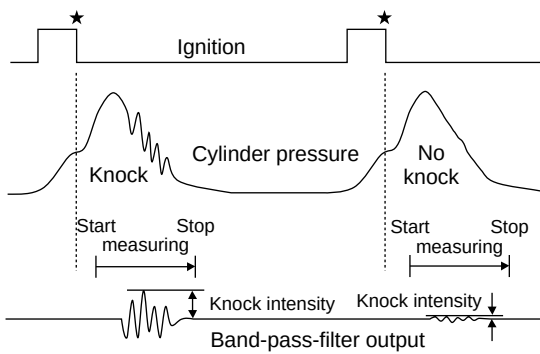


Fig.14 Timing chart of measuring signals



Fig.16 Appearance of knock intensity meter

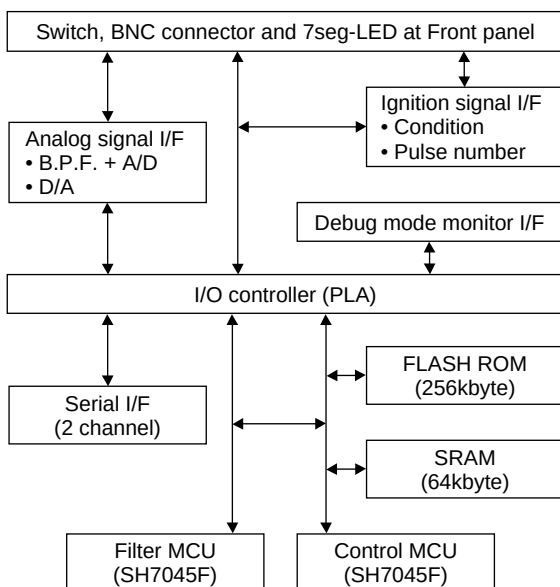


Fig.15 Constitution of hardware

5.2.1. Ignition Signal Interface

The signal level and pulse number can be selected to accommodate differences in the ignition system and distribution system. The following selections are available for the signal level:

- Battery voltage level (ignition edge: falling)
- Battery voltage level (ignition edge: rising)
- TTL level (ignition edge: falling)
- TTL level (ignition edge: rising)

The following selections are available for pulse number:

- Individual cylinder trigger (DLI or secondary side pickup of distributor power distribution); 1 pulse/720°
- Simultaneous firing ignition system trigger; 2 pulses/720°
- Simultaneous firing ignition system trigger (opposite

side cylinder); 2 pulses/720°

- Two-stroke engine trigger; 1 pulse/360°

In addition, the pulse number setting can be recognized by a processor for use in calculating the engine speed and other parameters.

5.2.2. Sensor Signal Interface

The analog filter consists of a high pass filter (1 kHz and above) for canceling the signals according to the combustion cycle, and a low pass filter (up to 100 kHz) to prevent aliasing distortion, these being connected in a cascade. Both filters are achieved with a low order CR filter and can be canceled by a jumper switch. In addition, a variable-gain (0.1-10) amplifier is installed between these two filters. The AD converter employs an oversampling SD modulation system that allows a relatively simple anti-aliasing filter to be used. This AD converter has a precision of 16 bits, an input range of -2 to +2 V, and the sampling rate can be switched between 25 kHz and 50 kHz. In addition, the sampling rate setting is recognized by a processor to perform coefficient switching of the digital filter.

5.2.3. Data Display Interface

The system is equipped with a 7-segment LED for stand-alone measurement. A single data group prepared in advance (such as the knock intensity or knock frequency) is selected, after which the physical quantities are displayed.

5.2.4. Knock Status Interface

A 12-bit D/A converter (output range: 0-5 V) is provided to give a TTL output and LED output corresponding to the results of the knock evaluation, as well as an analog output corresponding to the knock intensity, for connection to existing equipment and instruments. As a result, this can be used as an interlock, etc. during automatic operation.

5.2.5. Serial Communication Interface

A two-channel serial communication interface is provided for downloading the measurement results to a personal computer.

Since one of the serial communication interface channels allows connection to a network, the measurement results from multiple knock intensity meters can be incorporated with a single communications terminal.

5.2.6. Switch Signal Interface

The input from the sensor-type selection switch is identified by a processor as being either a charge amplifier output or a spark plug washer sensor output. This enables switching of the pass range of the digital filter and the knock evaluation algorithm.

The external switch input is recognized by the processor as a trigger in the case of rewriting or initializing (returning to the initial settings) the knock threshold value.

5.2.7. Processor

The system is equipped with two ordinary microprocessors (identical MCUs) containing built-in flash ROM. The functions are divided between the subprocessor that performs digital filtering and detects the maximum value of knock intensity, and the main processor that performs all the other tasks. All programs are assigned to the flash ROM contained within the processor, and both the main and subprocessor allow program rewriting by a flash writer. The main and subprocessor are connected in parallel and perform mutual monitoring. In addition, assigning knock threshold

values, filter coefficients and other constant data to external flash ROM makes it possible to accommodate rewriting and initialization of the knock threshold values by using an external switch for the trigger, as well as changing arbitrary constant data by connecting a personal computer for maintenance purposes.

5.3. Software

This section gives an overview of each process, including software processing that was not shown in Fig.13. In addition, it is necessary for this system to synchronize and process multiple external events, and it is equipped with a real-time OS complying with μ ITRON specifications for performing task management and other processing operations.

5.3.1. Pulse Processing

Pulse processing generates the timing (starting and ending) of the calculation and measurement of engine speed in synchronization with the ignition trigger. The timing for starting measurement detects the passage of a prescribed amount of time after the ignition trigger has been generated, while the timing for ending measurement detects the passage of a prescribed crank angle after the ignition trigger has been generated. Processing to generate the timing for ending the calculation and for measuring the engine speed is performed in consideration of the crank angle corresponding to one cycle of the ignition pulse according to the pulse set value of the ignition signal interface. However, in the case of actually generating the timing for starting and ending measurement, a delay in signal processing must be taken into consideration. Since the effect of this delay increases with increasing engine speed, the timing is generated by adding this delay time to both starting and ending points. Fig.17 shows a comparison between the ideal and actual measurement timing, which takes into consideration the delay, that are required as functions of the software.

5.3.2. Digital Signal Processing

Digital signal processing performs FIR filtering of a maximum of about 63 orders and updating of the maximum values of knock intensity at a sampling frequency of 25 kHz or 50 kHz. The filter coefficient is selected from four types

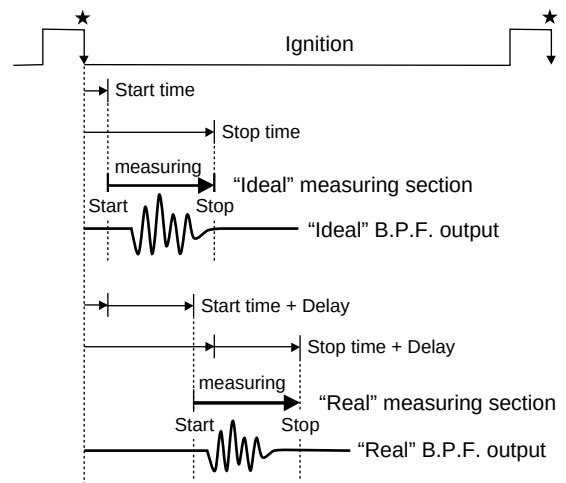


Fig.17 Measuring section

set in advance by combining the sampling frequency and sensor-type selection switches.

5.3.3. Statistical Processing and Knock Evaluation

Statistical processing (including the average value, standard deviation, peak value and knock frequency) is performed at a prescribed cycle interval for the above-mentioned maximum values. Moreover, a final evaluation of knocking is made according to the knock threshold value and the calculated results from statistical processing.

5.3.4. Processing of Each Output Means

The operation targets consist of the data display interface and knock status interface. The data display interface is used to select and display one of the following parameters corresponding to the display data selection switch:

- Calculation results of statistical processing
- Knock threshold value
- Engine speed

The knock status interface performs D/A output of the knock intensity and TTL output corresponding to the knock evaluation results. These interfaces are able to communicate with an automated measurement system to accommodate automatic operation.

5.3.5. Rewriting of the External Flash ROM

The following two methods are available for rewriting external flash ROM containing constant data:

- External switch
- Connection to a personal computer

In the case of rewriting the external flash ROM by an external switch, the knock threshold values are updated during normal operation or initialized (returned to the initial settings) during startup. For example, if the external switch is turned on during normal operation, the existing knock threshold values are replaced by newly calculated threshold values.

In the case of rewriting the external flash ROM by connecting to a personal computer, rewriting of all constant data used for maintenance is performed. For example, in the case of changing the knock resonance frequency as a result of changing the engine being measured or replacing the sensors, the newly set FIR filter coefficient can be updated using the personal computer.

compatible with automatic operation and automatic mapping.

Acknowledgments

The authors would like to take this opportunity to express their appreciation to Adtech System Science Co. Ltd. for their cooperation in the development of the measurement system.

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6. Conclusions

- 1) Knocking was quantified by passing the signal from a cylinder pressure sensor through a band pass filter and defining the maximum value as the knock intensity.
- 2) The quantified knocking data was correlated with test results obtained by hearing to confirm that the quantified data could be used for evaluating knocking.
- 3) A real-time measurement system was developed for quantifying and evaluating knocking that focuses only on the maximum values to achieve simple processing. In addition, this measurement system is compatible with various ignition signals and other sources as the timing signals, and it can be used with cylinder pressure sensors and spark plug washer sensors, giving it a high degree of flexibility.
- 4) The system has an interface for outputting knocking intensity and knocking evaluation flags, making it

■ author ■



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Masahiro SATO

Analysis of CVT and Automated MT Influence on Fuel Economy, and the Estimating Improvements in Fuel Economy

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Takashi AOKI*

ABSTRACT

The effects on mode fuel economy by different types of transmissions, CVT and Automated Manual Transmission, were investigated using the same vehicle and engine specifications while varying transmission ratio range, friction, and similar. The results are as follows: In the case of the CVT, it was confirmed that a wider transmission ratio range is effective when driving at relatively high speeds with fewer stop-and-go situations, and friction reduction is effective when driving at relatively low speeds with frequent stop-and-go situations. On the other hand, in the case of Automated Manual Transmission, it was confirmed that a wider transmission ratio range is effective when driving at relatively high speeds with fewer stop-and-go situations, which is the same as the CVT, while a combination of a wider transmission ratio range, an increase in shift steps, and an appropriate distribution of transmission ratios is necessary when driving at relatively low speeds with frequent stop-and-to situations in order to achieve high efficiency power plant operation.

1. Introduction

In the field of transmission technology, efforts to raise fuel economy have focused not just on raising transmission efficiency but also widening the transmission ratio range with larger numbers of shift steps. One example of this is the use in a wide range of mass-market and compact cars of continuously variable transmissions (CVT), which permit a comparatively wide transmission ratio range and high thermal efficiency during engine operation. Automated manual transmissions (AMT), which have roughly the same mechanism as a manual transmission but which perform clutch operation and shifting automatically, have been widely adopted for compact buses and trucks but have seldom been used in passenger cars because of the unnaturalness drivers experience during starting and shifting. Now, however, AMT technology is once again the focus of attention as its high transmission efficiency can potentially improve fuel economy.

There are reports of research that used simulation to determine which transmission type and which specifications would improve fuel economy. However, past simulation, while using identical vehicle and engine specifications, has not been able to define transmission ratios or shift schedules with a single method, and so could not effectively compare the effects on the fuel economy of CVT and AMT. The authors, however, have presented a control system that allows the power plant (including both engine and CVT) to operate at operating points of minimum fuel consumption, along with

a computational algorithm (Gf bottom calculation) for calculating the engine and CVT operating points that achieve minimum fuel consumption⁽¹⁾⁻⁽³⁾. Applying this computational algorithm to AMT permits fuel economy simulation using the same transmission ratios and shift schedules.

This paper, using identical methods for establishing vehicle and engine specifications, transmission ratios, and shift schedules, describes the fuel economy potential of a CVT, which is slightly less efficient but which permits a wide transmission ratio range, and of an AMT, a stepped-ratio transmission that, although more efficient, has a narrower transmission ratio range, and proposes how to utilize the mechanical advantages of both to further improve fuel economy.

2. Simulation

2.1. The Conventional Approach

Simulation-based comparative research has long been carried out to determine optimum transmission type and specifications with respect to fuel economy. However, comparative research on different transmission types has employed trial and error to determine the shift schedule and transmission ratio combination, which has a major impact on fuel economy, and so could not determine whether the fuel economy achieved with the decided-on shift schedule and transmission ratio combination was the best possible fuel economy. In short, shift schedule conditions were not

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the same, and so true comparative research under identical conditions was not possible.

2.2. The Authors' Approach

The authors proposed a computational algorithm for a method of determining a shift schedule that achieves the best fuel economy for a CVT, but did not mention its application for stepped-ratio transmissions⁽¹⁾. Determining the best fuel economy for the AMT, as it is a stepped-ratio transmission, requires that the right combination of shift schedule and transmission ratio be determined. The authors therefore expanded the above-mentioned CVT shift schedule determination method, to enable the determination of the shift schedule and transmission ratio combination that results in the best fuel economy for an AMT vehicle.

Moreover, because the same algorithm is used to determine the right shift schedule and transmission ratio combination, CVT vehicles and AMT vehicles can be compared using identical conditions on the assessment axis for minimizing fuel consumption.

2.2.1. Shift Schedule

The computational algorithm for determining the best fuel economy shift schedule for a CVT vehicle is based on the following logic.

The fuel consumption (Gfuel) of a vehicle traveling at a given speed (V) and driving force (F) is expressed as the integrated value of the engine's net fuel consumption rate (BSFC), engine power (Pse), and time (Hr), with Pse expressed as the sum of the tire edge drive power (Psp) and drivetrain lost work (Psf) needed to drive at a given speed and a given driving force. This is expressed in equation (1).

$$\Sigma G_{fuel} = \Sigma \{ BSFC \times (P_{sp} + P_{sf}) \times Hr \} \quad (1)$$

The authors, seeing Gfuel, located on the left side of equation (1), as the performance function, concluded that one has only to determine the combination of BSFC and Psf that minimizes Gfuel.

Because BSFC can be changed according to throttle opening and transmission ratio and Psf according to transmission ratio and engine speed, etc. while maintaining the same engine output, the left side of equation (1) can be minimized by selecting the appropriate throttle opening and transmission ratio.

Using these computations to determine the transmission ratio for V and F for every vehicle operating point yields the shifting schedule that achieves the operating points of minimum fuel consumption.

The transmission ratio shown in Fig.1 (marked "R-AMT" in the diagram) was used in order to apply this method to stepped-ratio AMT transmissions.

If the engine and transmission operating points can be controlled according to the shift schedule that was determined with this algorithm, then minimum fuel consumption can be constantly maintained during vehicle operation.

2.2.2. Transmission Ratio Combinations

The AMT transmission ratio was determined as described below:

- The transmission ratio was determined by using the shift schedule computed as described in section 2.2.1,

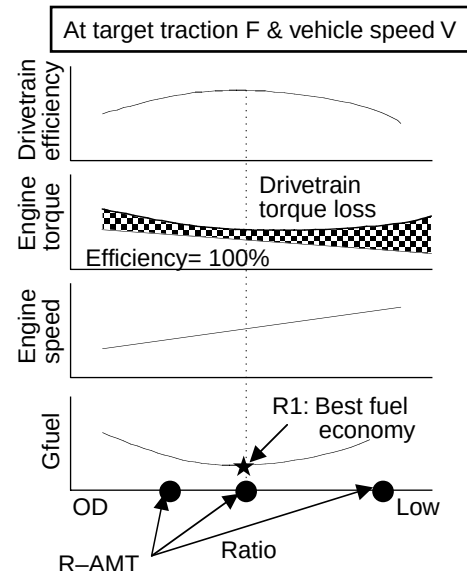


Fig.1 Gf-Bottom calculation for AMT

"Shift Schedule," to calculate, for each mode, the transmission ratio combination that produced the best mode fuel economy.

- The low ratio was defined so that the low overall ratio would be the same as that of the CVT vehicle.

The reason for doing this is to assure that the AMT vehicle has sufficient driving force at the low transmission ratio. Automatic transmission-equipped vehicles normally employ a torque converter as a starting component, while CVT vehicles employ, for example, a wet multiple disc clutch, torque converter, or electromagnetic clutch. When used as a starting component, a torque converter uses its torque amplification effect to boost starting torque, and so allows the low overall transmission ratio to be relatively small. However, when using a starting clutch or such, the absence of any torque amplification effect requires the low overall transmission ratio to be relatively high in order to assure adequate starting driving force. In a manual transmission (MT)-equipped car, which uses a dry clutch as a starting component, the driving force when starting depends on how the driver operates the clutch, i.e., a large driving force can be generated abruptly by engaging the clutch rapidly while throttling the engine. However, an AMT vehicle, because clutch operations are automatic, may not be able to deliver adequate driving force when starting with the same low transmission ratio as an ordinary MT vehicle. For this reason, the authors have assured adequate starting driving force by giving the AMT a low transmission ratio similar to a CVT's low overall transmission ratio, which was achieved by adjusting the final transmission ratio.

2.3. Fuel Economy Simulation Model

This model determines fuel consumption by first calculating the necessary driving force based on mode driving speed; then, adjusting for transmission efficiency, transmission ratio, engine accessory driving work and such, and finally determining engine speed and engine power/

torque. Fig.2 gives a block diagram of the model.

2.4. Verifying the Accuracy of the Fuel Economy Simulation Model

The model's accuracy was verified by comparing the results of simulation with data obtained during the mode driving of an actual vehicle.

Used in verification along with a CVT-equipped vehicle were the vehicle specifications listed in Tables 1 and 2. Accuracy of the model was verified by comparing the results of simulation with data obtained by measuring while driving an actual car having the same specifications as this CVT model.

Tables 1 and 2 summarize the CVT and AMT specifications used in this investigation. Both vehicles were the same model. Other than the actuator that performs shifting automatically in the AMT, there were no significant mechanical differences between the two. The conditions not

listed in the tables are as follows.

AMT conditions:

- Transmission friction was assumed to be the same as that of a 5-speed manual transmission as there is no significant mechanical difference between the AMT's power transmission system and that of a conventional MT.
- The energy needed for one shift was added to the engine work as actuator driving work.

Fig.3 shows a portion of the results for the LA4-CITY mode as an example of comparison and verification. From the top down, these are engine speed, engine torque, transmission ratio, and fuel consumption. The red and blue solid lines represent observed data and simulation data, respectively.

The difference between the observed data and simulation data for engine torque during deceleration is due to the low accuracy of the data used in simulation for negative-side

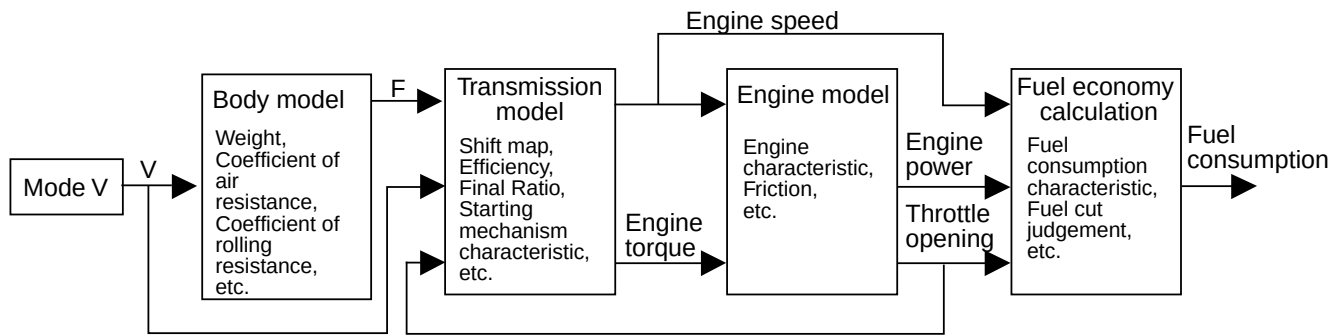


Fig.2 Simulation model structure

Table 1 Conditions of verification

Mode		US		J	
Engine	Type	1.6L 1CAM-VTEC		1.5L 1CAM-VTEC-3	
	Displacement (cc)	1590		1493	
	Horse power (kW/rpm)	85/6300		96/7000	
	Torque (Nm/rpm)	140/5400		139/5300	
Transmission	Type	Metal V-belt type CVT	Automated manual transmission	Metal V-belt type CVT	Automated manual transmission
	Starting mechanism	Wet-type multiple disc clutch	Automated clutch	Wet-type multiple disc clutch	Automated clutch
Body	Weight (kg)	1247.4	1190.7	1250	
	Tires	185/65R14		185/65R14	

Table 2 Conditions of simulation (Overall ratio)

Mode		US		J	
Transmission type		Metal V-belt type CVT	Automated manual transmission	Metal V-belt type CVT	Automated manual transmission
Overall ratio	Low	14.323	14.323	14.323	14.323
	2nd		8.382		10.145
	3rd		5.297		6.178
	4th		3.975		3.975
	5th		3.094		3.094
	Final	5.808	4.407	5.808	4.407

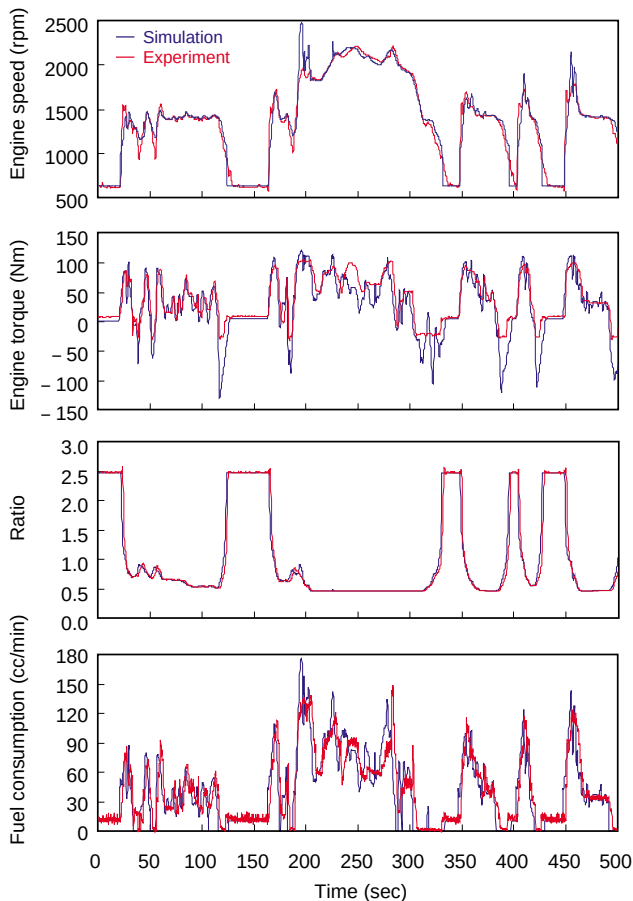


Fig.3 Result of verification
(Difference between simulation and experiment)

engine torque. However, as this is a fuel cut (F/C) range, the low accuracy of the engine data has almost no impact on fuel economy calculations. Aside from this, sufficient correlation does exist between the observed data and simulation data for the mode.

Fig.4, which gives the mode fuel economy at this time, shows a difference of less than 2% between the observed data and simulation data. The authors therefore judged this simulation model to offer sufficient accuracy for use in the further comparison and investigation of fuel economy.

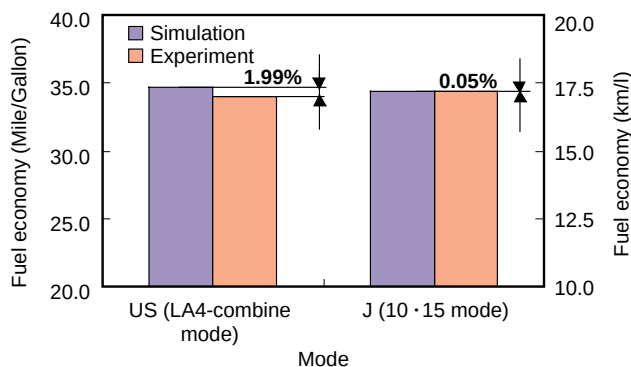


Fig.4 Mode fuel economy
(Difference between simulation and experiment)

3. Comparing CVT and AMT Fuel Economy

This section describes the fuel economy performance of CVT- and AMT-equipped vehicles by comparing the fuel economy of the two. CVT is widely thought to permit excellent engine efficiency while the AMT provides superior transmission efficiency. In this section, the authors consider which factors promote greater fuel economy.

3.1. Specifications of the Compared Vehicles

Table 1 lists all specifications other than transmission ratio, which is given in Table 2.

3.2. Comparison of Mode Fuel Economy

The mode fuel economy for both of a CVT and an AMT was compared by simulation for the 10·15 mode and the LA4-Combine mode ("LA4-COMB").

The results of simulation indicate that AMT yields superior fuel economy over CVT under current specifications (Fig.5).

Fig.6 gives average Pse ("Ave. Pse" in the illustration) and average BSFC ("Ave. BSFC") during acceleration and constant-speed driving in the 10·15 mode. AMT exhibited a lower average Pse, CVT a lower average BSFC, meaning that the low-Pse AMT permits a low P_{sf} and high transmission efficiency during driving, the low-BSFC CVT permits high engine thermal efficiency. Further, another possible factor in the superior fuel economy of the AMT is that the difference in average Pse for the CVT and the AMT is larger than the difference in average BSFC.

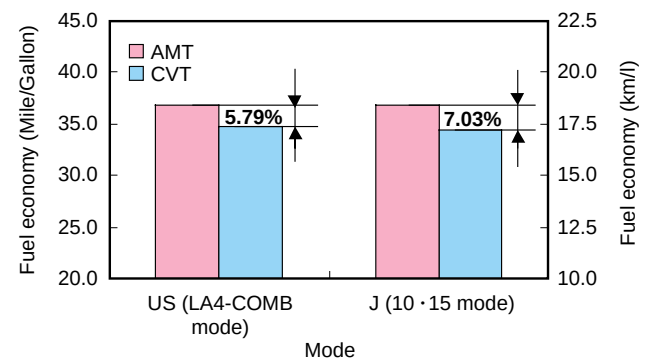


Fig.5 Fuel economy
(Difference between AMT and CVT)

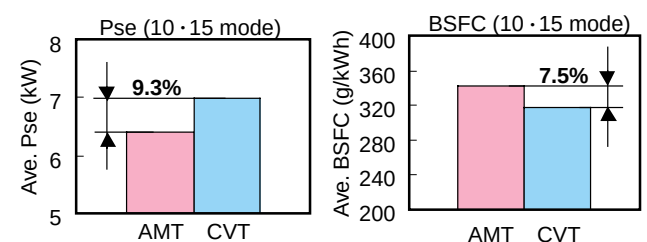


Fig.6 Average of Pse & average of BSFC
(Difference between AMT and CVT)

The graph in Fig.7 represents the relationship of engine speed to vehicle speed during acceleration and constant-speed driving in the 10·15 mode.

In the vehicle operating points there exists a range within the low-speed range that cannot be used as it used for fuel-cut and in surge prevention. Fig.7 shows how the CVT maintains a low engine speed whenever possible while avoiding this range, and how the AMT selects transmission ratios that result in low engine speeds whenever possible during constant-speed driving at 20, 40, and 50 km/h. A major difference between the CVT and the AMT is that the CVT, because of its continuously variable transmission ratio, permits complete low-engine-speed, high-load driving. In contrast, with AMT, a stepped-ratio transmission, the lowest engine speed possible at a given transmission ratio is limited to a specific vehicle speed, so that when vehicle speed rises, engine speed must also be increased until the transmission shifts to the next ratio. The result is high engine speeds coupled with low load.

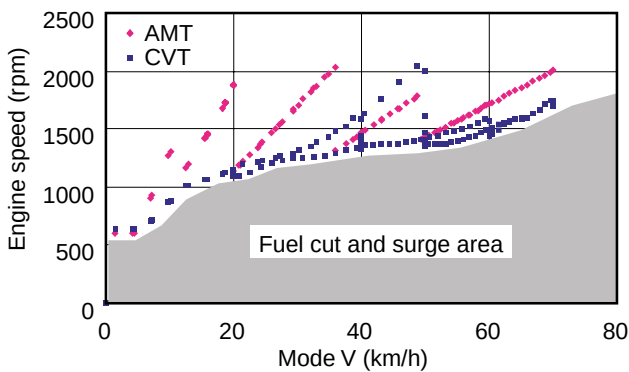


Fig.7 Comparison of engine speed
(Difference between AMT and CVT)

3.3. The Impact of Low Transmission Ratios and the Effect of Optimized Transmission Ratio Combinations

For this research, in order to have identical conditions for comparison, the AMT's final ratio was adjusted so that its low overall transmission ratio would be the same as that of CVT, the result being a large overall transmission ratio in comparison to an ordinary manual transmission. As a result, the AMT transmission ratios were less favorable with respect to fuel economy than those of the original manual transmission. For reference, the effects on fuel economy of different final ratio and optimized ratio combinations are discussed below.

These effects were investigated under the following sets of conditions.

- Mass production manual transmission ratios
- Different final ratio
- Different final ratio + optimized ratio combinations (same transmission ratios as those in Table 2)

Table 3 lists the overall ratios for each set of conditions. While Fig.8 shows the results of the investigation. The inclusion of the LA4-Highway mode ("LA4-HWY") in the LA4-COMB mode obscures the effects of transmission ratio

Table 3 Overall ratio
(Difference of final ratio & ratio combination)

Mode Conditions	US			J		
	a	b	c	a	b	c
Low	12.097	14.323	14.323	13.189	14.323	14.323
2nd	6.633	7.853	8.382	7.747	8.413	10.145
3rd	4.362	5.165	5.297	5.073	5.509	6.178
4th	3.383	4.006	3.975	3.689	4.006	3.975
5th	2.613	3.094	3.094	2.849	3.094	3.094
Final	3.722	4.407	4.407	4.058	4.407	4.407

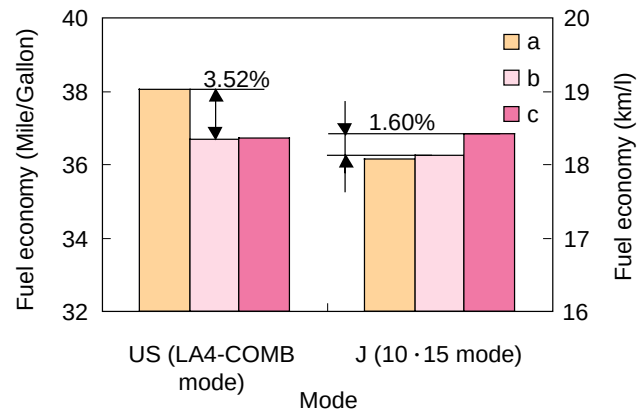


Fig.8 Fuel economy
(Difference of final ratio & ratio combination)

combination on fuel economy, and so the effect that is apparent is that of the large OD-side transmission ratio resulting from the change in final ratio.

4. Analysis of the Impact of Transmission Elements on Fuel Economy

This section considers the impact on mode fuel economy of the transmission elements of friction, transmission ratio range, and number of shift steps. By comparing these in the same modes, the authors investigated how each element contributes to mode fuel economy in CVT and AMT vehicles. The authors selected mode fuel economy as the basis of this comparison as they concluded that driving patterns with different driving loads and average vehicle speeds could be analyzed by using the 10·15 mode and LA4-CITY mode to represent driving at relatively low speeds with frequent stop-and-go situations, and the LA4-HWY mode to represent driving at relatively fast speeds with infrequent stop-and-go situations.

4.1. CVT

4.1.1. Transmission Ratio Range Impact on Fuel Economy

Here, the authors investigate the effect on fuel economy improvement of a wider transmission ratio range with smaller transmission ratios for all ratios except low, thereby achieving low-speed, high-load engine operating points and, therefore, higher engine thermal efficiency and reduced

friction.

Below are the conditions under which this investigation was performed.

Transmission ratio range: 5.0, 5.5, 6.0, 6.5

(The transmission ratio range was changed by expanding the OD-side ratio while leaving the low ratio fixed.)

Fig.9 shows the percentage of change in fuel economy for the transmission ratio range relative to the base transmission ratio range of 5.5 (the transmission ratio range of mass production CVT-equipped cars).

A 1.0 increase in the transmission ratio range resulted in a large 5.0% increase in fuel economy in the LA4-HWY mode, but a less than 1.0% improvement in the other modes. (The somewhat large 2.5% increase in the LA4-COMB mode was due to the effects of the LA4-HWY mode.)

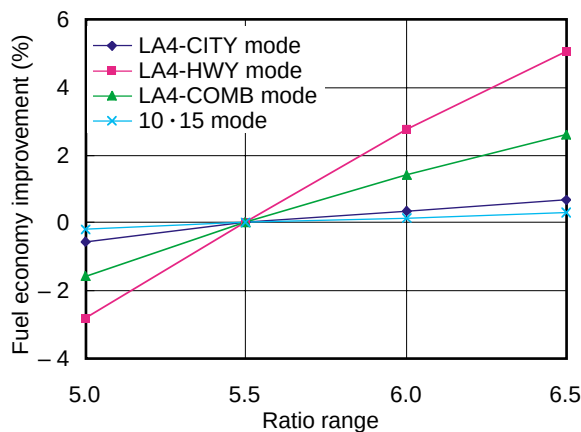


Fig.9 Fuel economy improvement of CVT (Difference of ratio range)

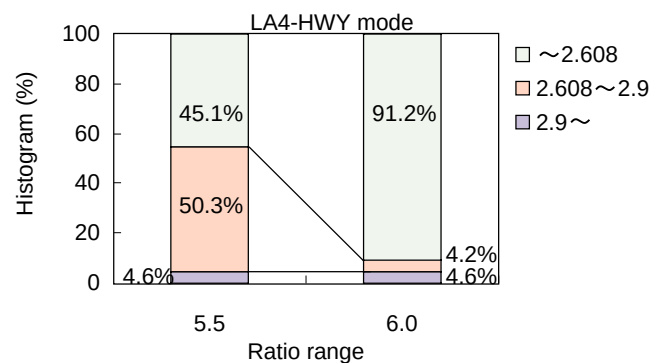
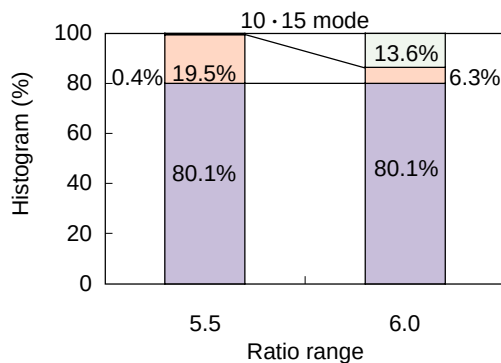


Fig.10 Ratio histogram

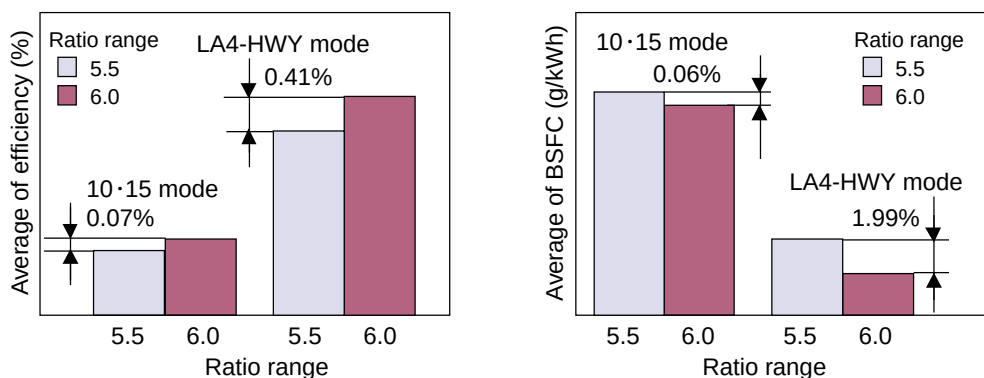


Fig.11 Average transmission efficiency & average BSFC

The results also demonstrate that in the case of the CVT, the wider transmission ratio range improved fuel economy in every mode.

Next, to determine the extent of the effect on fuel economy of low-speed, high-load driving, achieved with a wider transmission ratio range, in modes using differing driving patterns, the authors examined the 10-15 and LA4-HWY modes to compare frequency of ratio use during mode driving, average transmission efficiency, and average BSFC. The data used was collected during acceleration and constant-speed driving in each mode.

In the case of the 5.5 transmission ratio range, the graph in Fig.10 divides the range of ratio use during mode driving into 2.608 and below (the OD transmission ratio), 2.608 to 2.9, and over 2.9, and compares the frequency with which each range was used. The graph for the 10-15 mode (the left half of Fig. 10) shows that the over-2.9 range is used far more than any other, and that the 2.608-and-below range, i.e., the wider OD ratio range that accompanies the wider transmission ratio range, accounts for 13.6% of driving in that mode. In contrast, the transmission ratio range that is used far more frequently than any other in the LA4-HWY mode (the right half of Fig.10) is the under-2.608 ratio, which is used 45.1% of the time in the 5.5 transmission ratio range but 91.2% of the time in the case of the 6.0 transmission ratio range. In short, the expanded OD-side transmission ratio range is used more frequently in the LA4-HWY mode than in the 10-15 mode, meaning that in the former mode, the expanded transmission ratio range increases the frequency of low-speed, high-load engine operation.

Fig.11 compares average transmission efficiency and average BSFC in the 10-15 and LA4-HWY modes, and

shows the difference between the 5.5 and 6.0 transmission ratio ranges.

The reason for the improvement, albeit slight, in average transmission efficiency is believed to be this: With the wider transmission ratio range, the lower ratio ranges are used more frequently, and so the increase in transmission efficiency due to the lower input engine speed exceeded the drop in efficiency due to the metal belts' greater friction on the OD and low sides. The reason for the lower average BSFC is that the wider transmission ratio range resulted in a greater frequency of low-speed, high-load operation.

To summarize: In the 10·15 mode, the low frequency of low-speed, high-load engine operation translates into limited improvement of average transmission efficiency during mode driving and limited improvement in fuel economy due to the lower average BSFC, whereas in the LA4-HWY mode, the high frequency of low-speed, high-load engine operation increases improvement in fuel economy due to the lower average BSFC. This is what is responsible for the difference in fuel economy improvement in the two modes.

4.1.2. Impact on Fuel Economy of Friction Differences

This section considers the effect on fuel economy improvement of reducing friction to lower engine work.

Below are the conditions of this investigation:

Scenarios: Belt, oil pump, belt + oil pump

Rate of decrease: 0 → 100%

The reason for selecting the belt and oil pump as friction elements is that these two components together account for the majority of friction in a CVT (Fig.12).

Fig.13 shows the percentage improvement in fuel economy for different rates of friction reduction, starting from a friction reduction rate of zero.

Although there is some difference according to mode, a 0.5% improvement in fuel economy requires a roughly 5% decrease in friction in the belt-and-oil-pump scenario. In comparison to the aforementioned need for a 0.5 to 1.0 increase in transmission ratio range (0.1 in the LA4-HWY mode) to effect a 0.5% improvement in fuel economy, reducing friction is obviously an effective means of fuel economy improvement in CVT-equipped vehicles, while a wider transmission ratio range is effective when driving at relatively high speeds with less frequent stop-and-go situations.

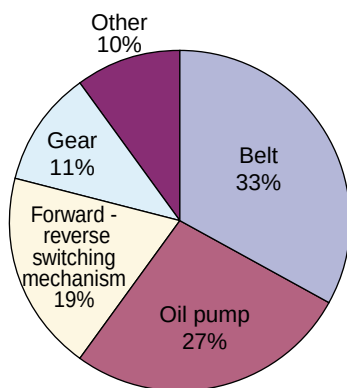


Fig.12 Power loss rate of CVT (at 40 km/h cruise)

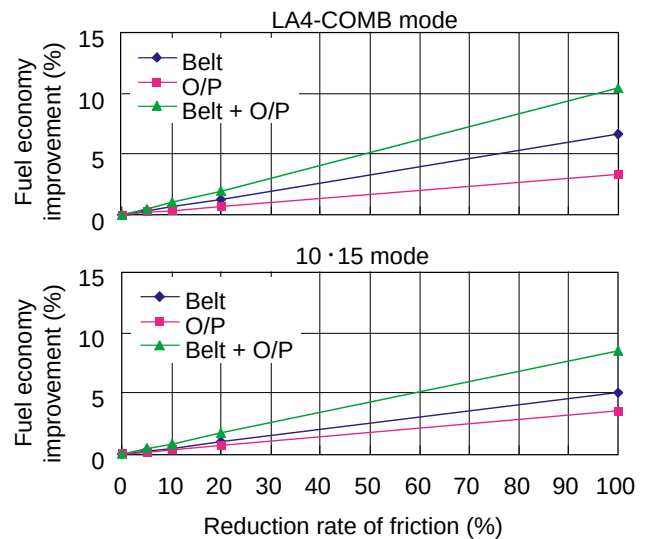


Fig.13 Fuel economy improvement of CVT (Difference of friction)

4.2. AMT

4.2.1. Transmission Ratio Range Impact on Fuel Economy

In this section, as in the case of the CVT, the authors investigate the effect on fuel economy improvement of a wider transmission ratio range with smaller transmission ratios for all ratios except low, thereby achieving low-speed, high-load engine operating points and, therefore, higher engine thermal efficiency. The effects of transmission and various friction reductions are also considered.

Whereas the CVT, being continuously variable, permits a simple widening of the transmission ratio range, the AMT requires that the proper distribution of transmission ratios be determined for the new transmission ratio range, and so the authors first sought to determine the proper distribution of transmission ratios.

The conditions used in this investigation are as follows.

Transmission ratio range: 4.6, 5.0, 5.5, 6.0

(The transmission ratio range was changed by expanding the OD-side ratio while leaving the low ratio fixed.)

Number of shift steps: 5

Fig.14 shows the percentage of changes in fuel economy for the different transmission ratio ranges relative to the base transmission ratio range of 4.6 (the transmission ratio range of mass production manual transmission-equipped cars).

As can be seen, an AMT with a wider transmission ratio range gets better fuel economy in the LA4-HWY mode but not necessarily in the other modes.

Fig.15 shows the relationship of a mode's average BSFC and average Pse to mode fuel economy at different transmission ratio ranges in the LA4-CITY mode with a low ratio that is the same as the low overall ratio of the CVT (left-hand graph) and with a low ratio that is the same as the low overall ratio of a mass production manual transmission-equipped automobile (right-hand graph).

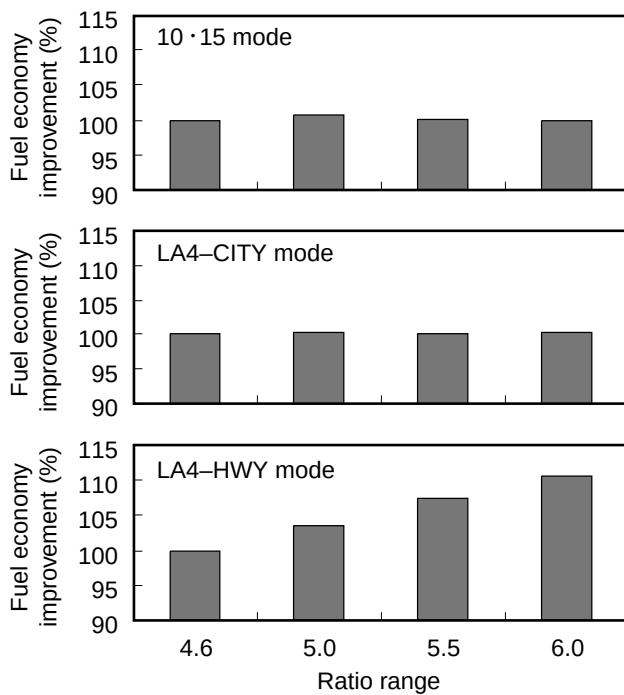


Fig.14 Fuel economy improvement of AMT (Difference of ratio range)

In Fig.15 (left-hand graph):

A transmission ratio range of 4.6 yielded the lowest average Pse, which represents engine work, but a large average BSFC, which represents engine thermal efficiency, for a resultantly low fuel economy.

There was no significant difference in average Pse among transmission ratio ranges 5.0, 5.5 and 6.0. However, the 5.5 transmission ratio range resulted in the lowest average BSFC and so the best fuel economy.

In Fig.15 (right-hand graph):

A transmission ratio range of 5.5 resulted in the lowest average BSFC but a large average Pse, and so fuel economy was poor. The 5.0 transmission ratio range yielded a large

average BSFC but the lowest average Pse and so the best fuel economy.

The preceding illustrates how exclusive focus on operating points with high engine thermal efficiency does not always improve fuel economy.

Engine work comprises the work needed for mode driving and various lost work, the latter of which depends on the transmission efficiency of the power transmission system.

This shows just how important it is to select transmission ratios and a shifting schedule that result in operating points that maximize the improvement in fuel economy when considering how to improve both engine thermal efficiency and transmission efficiency.

4.2.2. The Impact on Fuel Economy of Friction Differences

As with CVT, this section considers the effect on fuel economy improvement of reducing friction to lower engine work.

Below are the conditions of this investigation:

Rate of decrease: 0 → 100%

Transmission ratio range: 5.5

Fig.16 shows the rate of fuel economy change as related to rates of friction reduction with 0% friction reduction being base.

In descending order, the largest fuel economy change ratio for different friction reduction ratio were achieved in the modes LA4-HWY, LA4-COMB, LA4-CITY, and 10-15. Improving fuel economy by 0.5% requires a 20% reduction in friction in each mode.

4.2.3. Impact on Fuel Economy of the Number of Shift Steps

This section considers the effect on fuel economy improvement of increasing the number of shift steps to thereby increase the degree of freedom and achieve operating points with greater engine thermal efficiency.

The conditions used in this investigation are as follows.

Number of shift steps: 4, 5, 6

Transmission ratio range: 4.6, 5.0, 5.5, 6.0

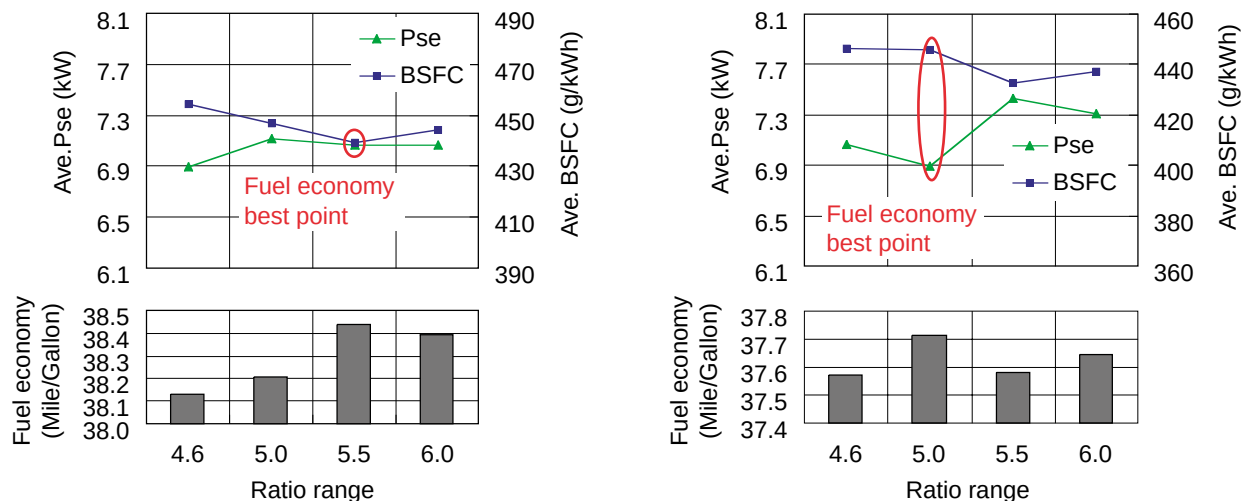


Fig.15 Relation of Pse, BSFC & fuel economy

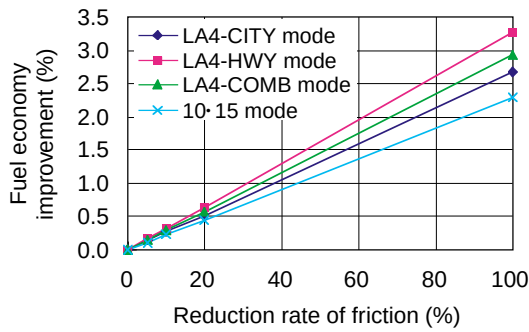


Fig.16 Fuel economy improvement of AMT (Difference of friction)

Fig.17 shows the percentage change in fuel economy for different numbers of shift steps relative to a base number of 5 steps.

Increasing the numbers of shift steps from 5 to 6 improves fuel economy by 0.5 to 1.0%. The percentage improvement in fuel economy tends to level off beyond a transmission ratio range of 6.0.

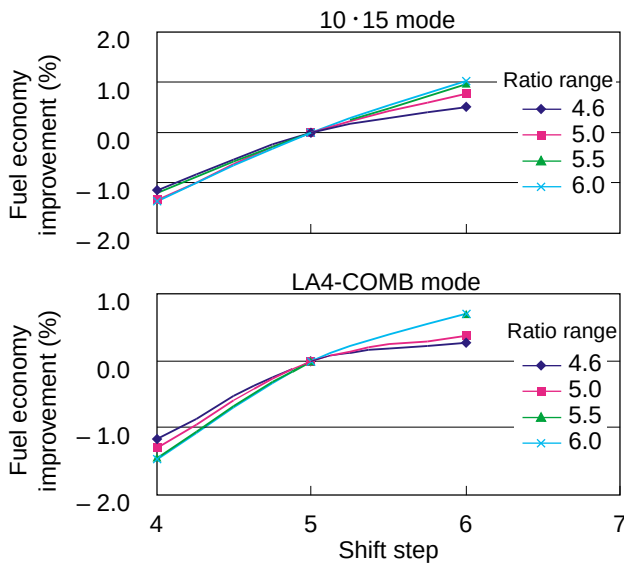


Fig.17 Fuel economy improvement of AMT (Difference between 4-speed and 6-speed)

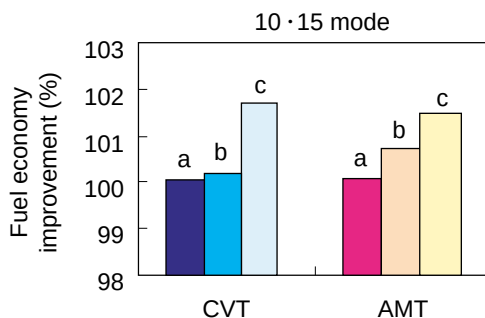


Fig.18 Simulation results under conditions of table 4

4.3. Summary of the Investigation of Fuel Economy

To summarize the preceding, friction reduction is an effective means of fuel economy improvement in the case of a CVT, as is a wider transmission ratio range when driving at high speeds, such as the LA4-HWY mode. For an AMT, friction reduction is less effective because of the already-high transmission efficiency of the AMT, for which a wider transmission ratio range is roughly as effective for high-speed driving as it is for the CVT.

In continuation of the preceding, the authors next predicted fuel economy improvements by determining what the percentage of improvement in fuel economy would be under conditions thought to be viable for both a CVT and an AMT.

Below are the sets of conditions investigated.

- Current specifications
- Wider transmission ratio range
- Wider transmission ratio range and fuel economy improvement techniques

As for the fuel economy improvement techniques, a wider transmission ratio range was used for both transmission types. In addition, friction reduction was used for the CVT, but for the AMT, a larger number of shift steps was used, as friction reduction would not significantly contribute to fuel economy improvements. The conditions are listed in Table 4.

The graph in Fig.18 shows the percentage changes in fuel economy achieved with the conditions in Table 4 relative to conditions a, which were the base conditions.

The percentage improvement in fuel economy was nearly the same for conditions a, b, and c with both the CVT and the AMT, verifying no change in the tendency of the AMT to result in better fuel economy than the CVT.

Table 4 Conditions of simulation

	Transmission type	
	CVT	AMT
a	Ratio range 5.5	Ratio range 4.6
b	Ratio range 6.0	Ratio range 5.0
c	b + Friction -20%	b + 6Speed

5. Conclusion

A simulation model permitting comparison of a CVT and an AMT according to the shift schedules that result in the best fuel economy for each was established to determine the fuel economy potential of each transmission type. By using the same technique to determine the shifting schedules for both CVT and AMT, the authors were able to compare the fuel economy of these different transmissions under more similar conditions than before.

The findings of this research are as follows:

- (1) The AMT is more advantageous than the CVT with respect to fuel economy.

Regarding further increases in fuel economy:

- (2) Reducing friction in the oil pump and belts was verified as an effective means for the CVT, as was a wider transmission ratio range for driving at high speeds, as in the LA4-HWY mode.
- (3) Friction reduction is minimally effective for the AMT, which already has a high transmission efficiency. However, a wider transmission ratio range was verified to be as effective for the type of high-speed driving typified by the LA4-HWY mode as it is for the CVT.
- (4) Essentially the same fuel economy ratio changes were achieved for both the CVT and the AMT when transmission elements were changed under conditions felt to be viable, verifying no change in the tendency of the AMT to be superior to the CVT with respect to fuel economy.

Acknowledgments

The authors wish to express their gratitude to the persons who provided various data and invaluable advice during the course of this research.

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Takashi AOKI

Evaluation of the Durability and Reliability of the Power Semiconductor Device for a Hybrid Car's Motor Driver

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ABSTRACT

The reliability of the power semiconductor device for a hybrid car's motor driver was verified by clarifying the relationship between the unit life characteristics and vehicle usage environment in terms of electric current and heat generation temperature. A reliability of 99.915% or more and a shape parameter (m) of 5.5 or more were obtained. Further, an evaluation method for durability and reliability was established.

1. Introduction

Amidst the growing demand for lower fuel consumption and reduced exhaust emissions from automobiles, the Integrated Motor Assist (IMA) system has been developed as one means of satisfying these demands. This system reduces the amount of fuel consumed by applying an electric motor located between a conventional gasoline engine and the transmission to collect energy dispersed by the vehicle during deceleration, by storing this energy in a battery, and by assisting the engine with this stored energy to drive the motor during acceleration.

A power semiconductor device that can perform rapid on-off switching of a large electric current is indispensable for achieving optimum control of the drive and regeneration energy conversion that takes place in this system. It is important to accurately verify the durability and reliability of this power semiconductor device in order to guarantee

the quality of the IMA system in the market. During the development of the IMA system, a method for evaluating the durability and reliability of power semiconductor devices for motor driving was established, a description of which is reported here.

2. Overview of the IMA System

Fig.1 shows the configuration of the IMA system. This system incorporates a lean-burning variable valve timing and lift electronic control system (VTEC) engine having a 1 liter displacement as the main power source, an ultra-thin brushless DC motor for the auxiliary power source, a power control unit (PCU) for converting the electrical power, and an Ni-MH battery for storing the electrical power. This system is characterized by continuously using the engine as the main power source, and supplementing it with the electric motor as an auxiliary power source according to the driving

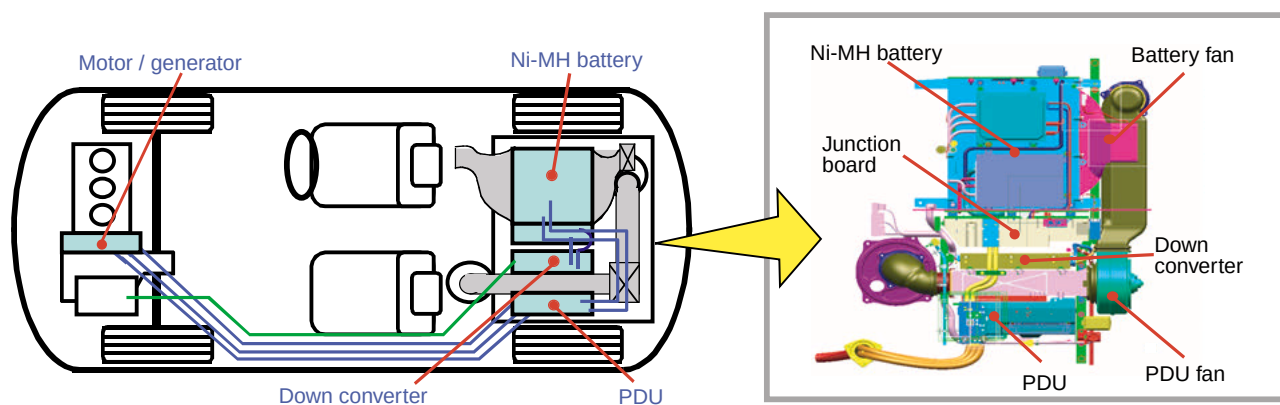


Fig.1 IMA system

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conditions. The overall system is simple, and the motor, battery and PCU can all be made compact and lightweight. The relationship between the driving conditions and system status is described next.

- During Acceleration
When a large load is applied to the engine during acceleration, the electric motor assists the engine by generating a maximum torque of approximately 5 kgf·m.
- When Cruising
The vehicle is powered by the gasoline engine alone when cruising for stable operation with low fuel consumption.
- During Deceleration
The motor fulfills the role of a generator to regenerate the deceleration energy and charge the battery which would otherwise be dispersed as heat from the brakes.
- When Stopped
Idling is stopped automatically to inhibit fuel consumption and reduce exhaust emissions.

3. Functional Overview of the Motor Drive Unit

A functional overview of the power drive unit (PDU) is shown in Fig.2. The major functional components of the PDU consist of a power semiconductor device in the form of an insulated gate bipolar transistor (IGBT), a diode, an intelligent power module (IPM) containing a built-in protective function, a pre-driver unit equipped with the gate drive power supply for the IGBT, an input power voltage sensor and an interface with the ECU, and an electrolytic capacitor that smooths the power supply current.

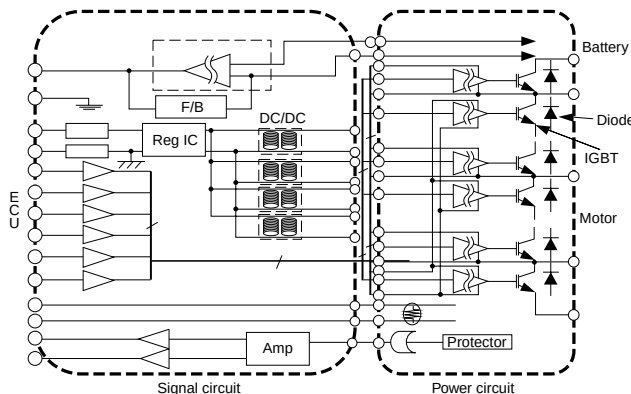


Fig.2 PDU block diagram

The motor is a permanent magnet, three-phase synchronous type. The relative positions of the stator and rotor are detected by three commutation sensors. A single cycle that is completed once for every 60° of rotation is detected by dividing into six parts, and as a result of switching six IGBTs on and off, the current and its direction of flow to the three-phase terminals U, V and W connected to the motor are switched, thereby converting electrical power from the battery to the motor or from the motor to the battery.

Drive power and regenerated electrical power are

controlled by advancing and delaying or by reducing and extending the switching timing, and by duty control.

The current that flows to the IGBT and diode relative to the current flowing to the 3-phase output can be approximated by the following expressions:

$$\begin{aligned} I(\text{IGBT}) &= 1/2 \times I_m \times (\text{Duty}/100) \\ I(\text{diode}) &= 1/2 \times I_m \times (1 - \text{Duty}/100) \end{aligned} \quad (1)$$

where I_m is the output current, and duty is the PWM duty of the input signal.

Consequently, the current flowing to the power semiconductor device can be deduced from the output current, which can be measured in an actual vehicle (motor input current), thereby making it possible to obtain the loss (calorific value) per device.

The loss characteristics of the power semiconductor device relative to the output current are shown in Fig.3.

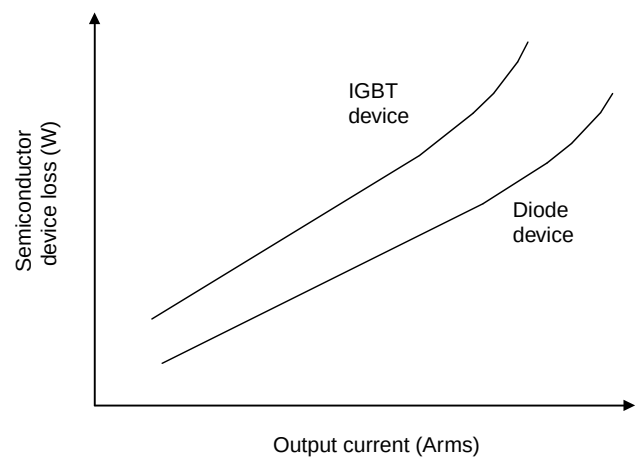


Fig.3 Power semiconductor device loss curve

4. Method for Evaluating the Durability and Reliability of the Power Semiconductor Device

The power semiconductor device, which has an important function among the constituent components of the PDU, operates at a high voltage, high current and high speed, and is exposed to harsh environmental conditions. The environment in which the components are used involves a combination of high temperature, humidity and vibration that subjects the components to stress. Since the power semiconductor device in particular generates heat, there are strict requirements for its temperature environment. Evaluations of durability and reliability were performed for the three environmental factors independently and in combination. In general, modeling these set conditions is difficult, and the establishment of a modeling technique is an important requirement. Since the power semiconductor device is a heat-generating component and is subjected to particularly severe conditions in use, load testing was conducted at various temperatures and currents to evaluate durability and reliability, the results of which are reported next.

The internal construction of the IPM containing the power semiconductor device is shown in Fig.4. The locations that reach a high temperature are especially susceptible to deterioration and malfunction,⁽¹⁾ and a summary of the operating conditions and component malfunction factors is shown in Table 1. It can be seen from this table that the environmental factors and malfunction sites are clearly defined, and that the junction temperature and case temperature should be measured as control points. Changes in the control temperature for various operating temperatures and currents when used in an actual vehicle are shown in Fig.5. Typical durability characteristics relative to stress in the form of temperature, current, etc. are shown in Fig.6 with the accompanying test data, while a Weibull chart is shown in Fig.7 for determining the reliability from the time to breakdown.

The principle verification concept was to determine the life span characteristics relative to the stress of the individual component being tested, and then to determine point A in Fig.6 based on the frequency of the vehicle use and vehicle life span. Details of the procedures used to evaluate durability and reliability are given next.

- (1) Durability data were obtained for the power semiconductor device alone (refer to the durability curve of Fig.6).
- (2) Current and temperature data were measured while

driving an actual vehicle.

- (3) The data were converted to the vehicle lifetime from the frequency data obtained in (2).
- (4) The actual vehicle data were converted to a single point temperature using the acceleration coefficient of (1), above this single point is the semiconductor life span point when applied to an actual vehicle (point A in Fig.6) (here, the t_a component is the excess time relative to actual capabilities).
- (5) Life span testing is performed under the same temperature conditions as those at point A, before plotting a Weibull chart (see Fig.7).
- (6) Since point A is the number of cycles for the vehicle life span at the acceleration temperature, the reliability can be determined by plotting the Weibull chart (see Fig.7).

As shown in Table 1 for semiconductor power devices, the IGBT chip, which is a heat-generating component, and the aluminum wire and solder that extend the electrode from the diode chip are important control points for determining life span. The critical parameter that determines the life span of the aluminum wire is the current. By measuring the PDU output current from data obtained when driving the vehicle, the current per power semiconductor device can be divided as shown in equation (1). The internal junction temperature difference (DT_j) can then be determined from equation (2) shown next.

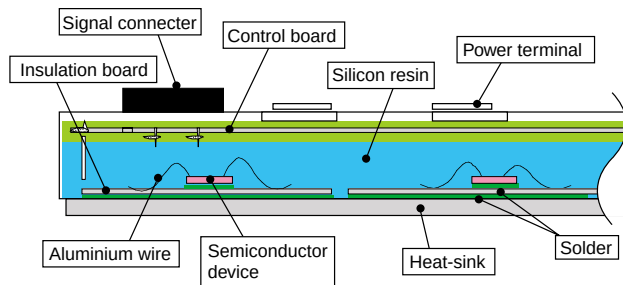


Fig.4 IPM internal diagram

Table 1 Malfunction factors

IMA driving mode	Control point	Malfunctioning point
Engine-start mode	Junction temperature	Aluminium-wire
Adjustable-speed mode	Case temperature	Chip-solder

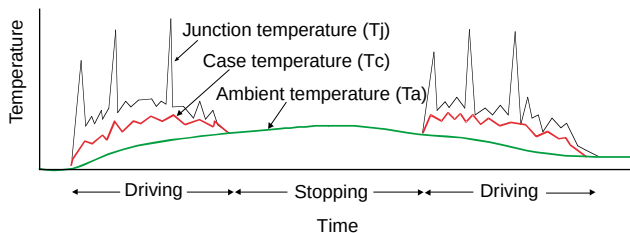


Fig.5 Vehicle temperatures transition

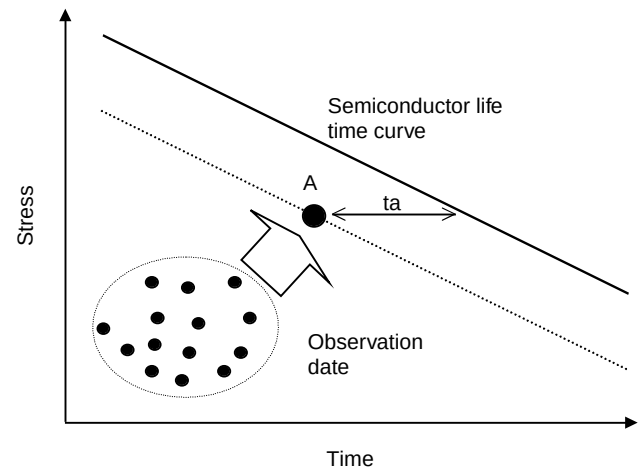


Fig.6 Durability curve

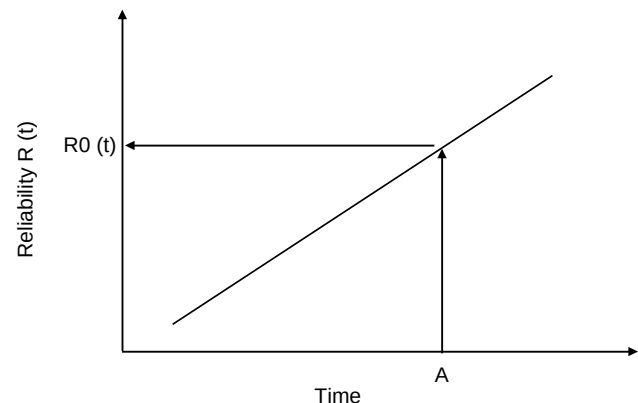


Fig.7 Weibull chart

$$\Delta T_j = R_{th} \times \{ (I_m \times V_{sat}) + P_{sw} \} \quad (2)$$

where T_j is the internal junction temperature, R_{th} is the thermal resistance of the IPM package, I_m is the PDU output current, V_{sat} is the power semiconductor ON voltage, and P_{sw} is the switching loss.

The temperature stress of the solder was determined by measuring and using the temperature directly below the IGBT chip. On the other hand, the current and temperature that determine the individual life span of the power semiconductor device were verified from the life span data obtained from a power cycle test (PCT) and thermal fatigue test (TFT).

PCT is a life span test that accelerates the deterioration of an aluminum wire carrying a heavy current, and TFT is a life span test that accelerates the solder deterioration caused by temperature changes.

5. Durability and Reliability Evaluation Results

5.1. Evaluation of the Reliability Relative to Current

Start-up current test conditions were determined by taking into consideration the engine oil type and viscosity characteristics at low temperatures as environmental effects during engine start-up. Since the run-on torque changes according to the position of the engine crank angle and the current duration also simultaneously changes, the current duration was set according to the characteristics of the various elements and vehicle data. The vehicle start-up test results are shown in Fig.8.

The longest current duration during start-up is t_1 in Fig.8. The number of times the power semiconductor device is used over the course of the vehicle life span refers to the number of times the ignition key is turned on and the number of times the engine is started after it has been stopped after idling. This number is larger than the number of times the

vehicle is started in conventional gasoline-powered vehicles. When the reliable vehicle lifetime has been reached and determined from the Weibull chart that had been prepared by plotting actual vehicle data for PCT durability under the same conditions as those for vehicle lifetime point A, the value was found to be 99.915%. In addition, although the IPM prototype installed in the IMA was a trial product made on a line that was nearly identical to the mass production line, the form of breakdown at this time was such that the shape parameter (m) was sufficiently larger than 1 ($m = 6.2$) according to the Weibull chart, thereby confirming durability deterioration.

5.2. Evaluation of the Durability and Reliability Relative to Temperature

The environmental temperature during vehicle use can be classified into radiant heat from the outside air, vehicle chassis or road surface, heat from other parts, and heat effects from the heat generation of the part itself. In consideration of market use, an environmental dynamo bench was used in vehicle tests settings, using that assumed a high-temperature environment with respect to the amount of solar radiation and so forth. Although the particular environment varies according to the location where the vehicle is used in terms of air temperature, amount of solar radiation and other parameters, a particularly harsh environment was defined in consideration of global specifications. The life span was also verified by testing vehicles in extremely hot and cold regions and gathering the test data from those vehicles.

Fig.9 shows the current, temperature and vehicle speed during general driving simulated on a dynamo bench. The case of a similar test performed while changing only the driving conditions to those for hard driving is shown in Fig.10. For both general driving and hard driving, the case temperature (base plate temperature) was 10-12°C higher than the outside air temperature. Next, driving tests were performed in the U.S., the data gathered during general

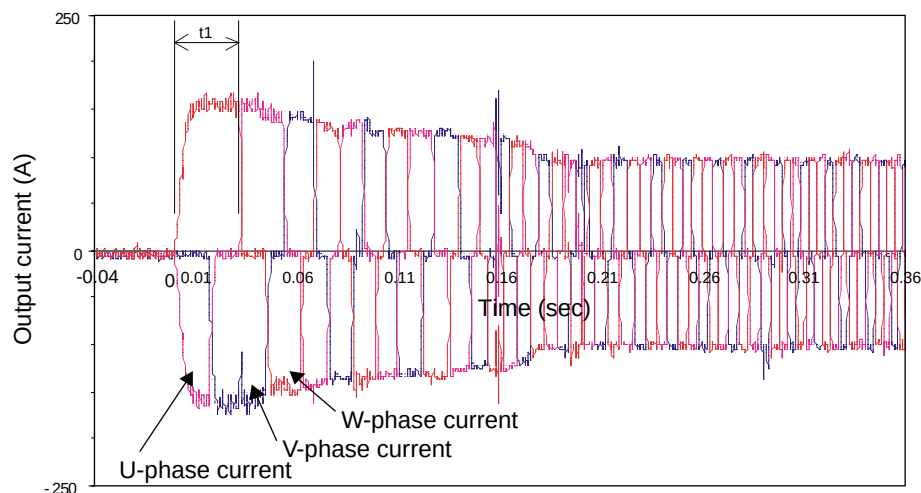


Fig.8 Starter current waveform

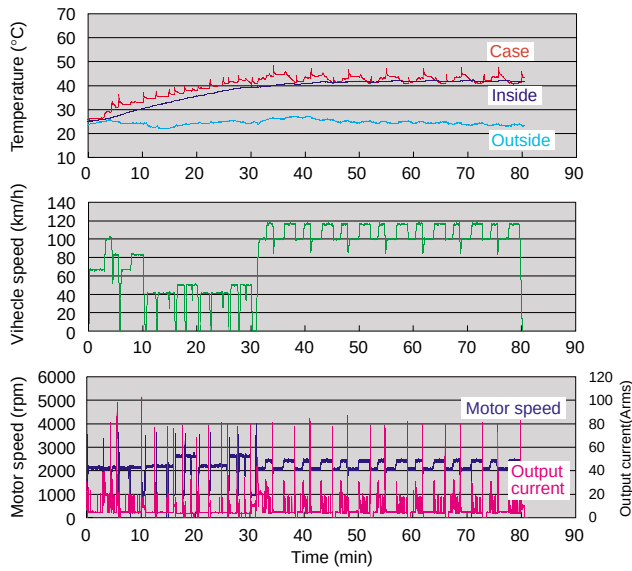


Fig.9 General driving test on dynamo bench

driving are shown in Fig.11, while the data taken during hard driving are shown in Fig.12. A temperature rise of 30–32°C was obtained, which is slightly higher than the data obtained on the dynamo bench. In addition, test data from cold climate region driving are shown in Fig.13. Since air cooling is used in cold- climate regions, the temperature difference until the fan operating temperature is reached is larger, with the temperature rising to about 45°C. On the basis of the above results, when the reliability for the vehicle lifetime had been determined from a Weibull chart according to point A by plotting various types of driving test data for TFT durability, values could be obtained ranging from 99.97% to 99.99%. Moreover, when the shape parameter (m) was determined from a Weibull chart in the same way as that used for evaluating the reliability relative to current, the m value was found to be 5.5, thereby confirming durability against deterioration.

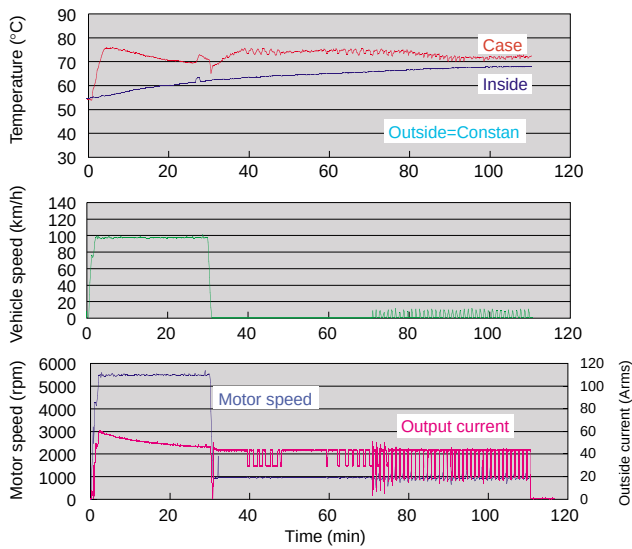


Fig.10 Hard driving test on dynamo bench

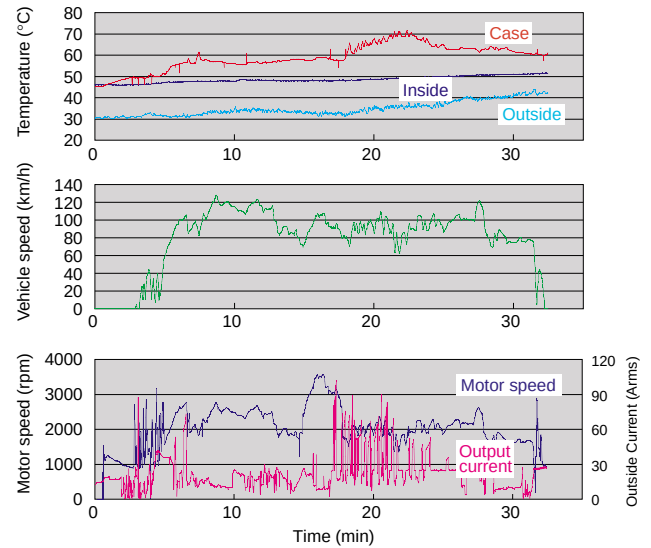


Fig.12 Hard driving test in U.S.

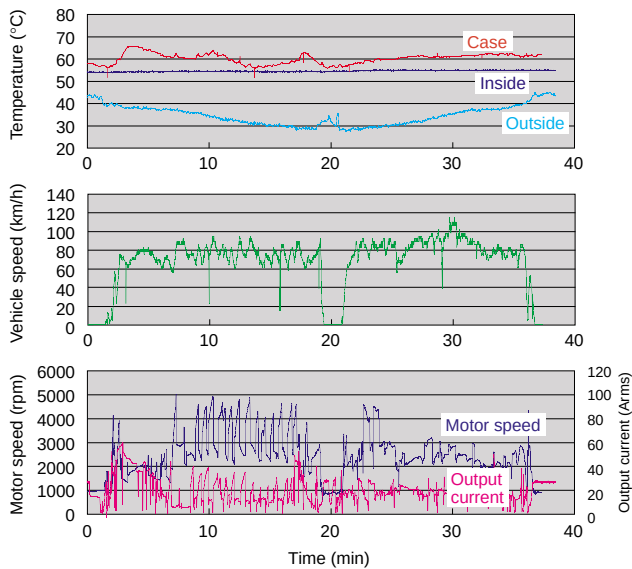


Fig.11 General driving test in U.S.

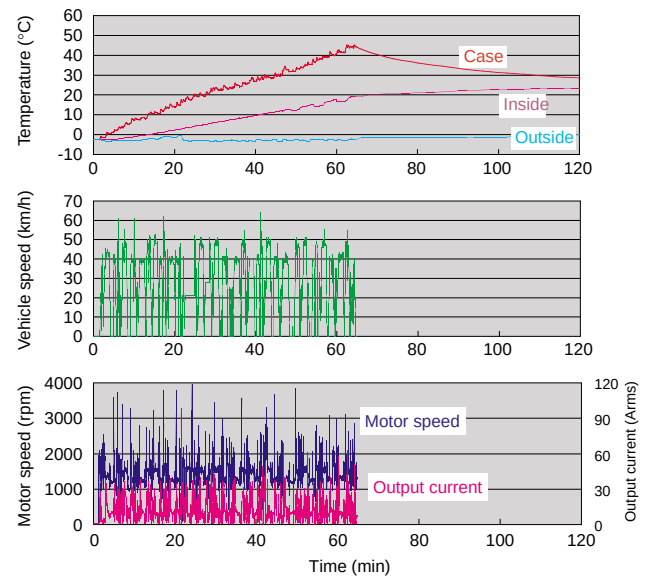


Fig.13 Cold-climate region driving test

6. Conclusion

Durability and reliability were evaluated by focusing on the currents and temperatures that affect the lifetime of power semiconductor devices. Reliability was determined using a Weibull chart drawn from various vehicle driving data and environmental test data, as well as from individual life span test data for the power semiconductor device. As a result, the reliability was determined to be 99.9% or better, and the shape parameter (m) ranged from 5.5 to 6.2, thereby confirming durability against deterioration. In addition, a method for evaluating durability and reliability under a large current load and high temperature applied to the motor driver installed in a hybrid car was also established.

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Energy Efficiency Improvements for a Series Hybrid Vehicle

Yusuke HASEGAWA* Satoshi AOYAGI*
Takahiro YONEKURA* Hiroyuki ABE*

ABSTRACT

A system was developed for a series hybrid electric vehicle that is characterized by mechanically separating the engine and axle to thereby enable operation in the optimum electric generation range. This system was developed to vastly improve fuel economy without sacrificing vehicle dynamic performance.

The study focused on real-time generating without surplus or deficit in accordance with motor power to improve energy transfer efficiency and decrease the size and weight of the energy storage unit. At the same time, generating efficiency was improved by operating the engine with a minimum Brake Specific Fuel Consumption (BSFC). The system was tested on a 1 500 kg equivalent inertia weight class vehicle whereby it was verified that a 94% fuel economy was achieved in comparison to gasoline-powered vehicles in the same class, while no sacrifices were made to dynamic performance.

1. Introduction

With increased interest in global environmental issues, demand has further increased for technology to reduce automobile fuel consumption. Attention is currently focused on reducing vehicle body size and weight, and efforts are also underway to develop engine technologies to improve fuel efficiency such as direct injection gasoline and diesel engines. In addition, hybrid systems are attracting attention as a total vehicle technology, with some hybrid vehicles having already been released to the market.⁽¹⁾⁽²⁾

This study focused in particular on the series hybrid electric vehicle (hereafter “SHEV”) characterized by a mechanically separate engine and axle enabling operation in the optimum electric generation range. Further, we attempted to vastly improve fuel efficiency for a SHEV without sacrificing vehicle dynamic performance compared to a gasoline fueled vehicle of the same equivalent inertia weight class.

2. Aim of This Study

Most conventional SHEV use an energy storage (hereafter “E/S”) system in which a compact generator is mounted on an electric vehicle.⁽³⁾ In these systems, the motor power is provided by the E/S, and the discharged energy is replenished by the generator. However, the following problems make it difficult to improve fuel efficiency.

- (1) The E/S weight and size increase in order to guarantee motor power with the E/S.
- (2) The E/S charge/discharge loss degrades energy transfer efficiency.

In this study, the authors thought that if real-time generating could be performed without surplus or deficit in accordance with the motor power, the E/S size and weight

could be reduced and at the same time the charge/discharge loss could also be greatly reduced and the energy transfer efficiency improved, thus overcoming these problems. It was thought that by utilizing the above-mentioned feature of the SHEV, the engine could be constantly operated in the high efficiency range even if the generation level was varied, thus yielding vast fuel efficiency improvement effects compared to the same class of gasoline fueled vehicles.

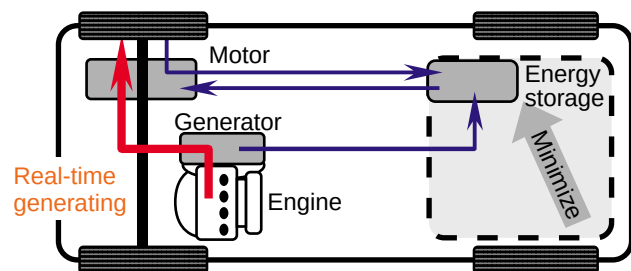


Fig.1 Real-time generating and minimized energy storage

Next, the effect of real-time generating on transfer efficiency within the energy transfer system shown in Fig.2 is explained. Here, η_{GEN} is the generator efficiency, $\eta_{\text{E/S}}$ is the E/S charge/discharge efficiency, and η_{MOT} is the motor efficiency.

The energy transfer efficiency when the conventional generating method and the real-time generating method are applied to this energy transfer system can be expressed by the following equations.

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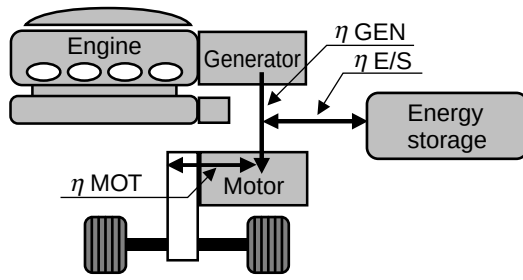


Fig.2 Energy transfer efficiency

Conventional generating method:

$$\text{Energy transfer efficiency} = \eta_{GEN} \times \eta_{E/S} (\text{Charge}) \times \eta_{E/S} (\text{Discharge}) \times \eta_{MOT} \quad (1)$$

Real-time generating method:

$$\text{Energy transfer efficiency} = \eta_{GEN} \times \eta_{MOT} \quad (2)$$

When calculated assuming each of the efficiencies in equations (1) and (2) above are 90%, the energy transfer efficiency for real-time generating is 81% while it is 65% for the conventional method.

3. Control System

3.1. Control System Configuration

Fig.3 shows the overall control system configuration.

The system can be broadly divided into the following three control units.

- (1) Generating control block
- (2) Brake Specific Fuel Consumption (BSFC) bottom trace controller
- (3) Engine, generator and motor controllers

The generating control block is comprised of the real-time generating controller, which forms the base, and the interval power controller, which turns the generator on and off according to the E/S state of charge (hereafter "SOC") when the generator power requirement is low. This block determines the target motor torque and generator power

requirement based on the acceleration pedal angle and the vehicle speed.

The BSFC bottom trace controller determines the target generator power and the throttle angle to minimize the BSFC based on the generator power requirement determined by the generating control block.

The motor, generator and engine controllers control their respective control subjects in accordance with the target torque, target generator power and throttle angle determined above.

The generating control block and BSFC bottom trace controller characteristic of this control system are described in the following sections.

3.2. Real-time Generating Control

Fig.4 shows the real-time generating controller structure outline.

This controller determines the basic generator power by estimating the electric power required to drive the motor from the target motor torque and the vehicle speed, and then adding other power consumption. Generating action can be started before actual power consumption starts by estimating the power requirement with a feed-forward system. This improves generating responsiveness, and makes it possible to reduce temporary discharge from the E/S even when the motor power changes suddenly, thereby achieving energy management to minimize charge/discharge loss.

In addition, the generator power requirement is

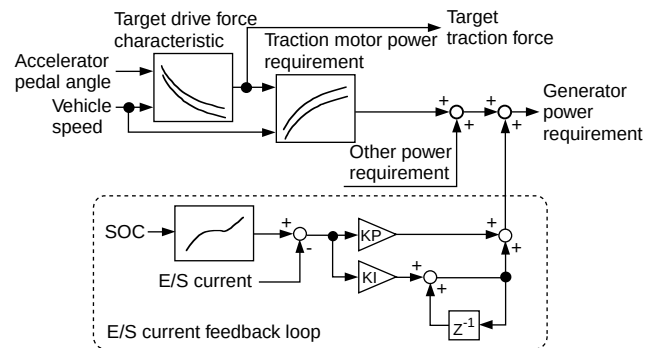


Fig.4 Real time generating control structure

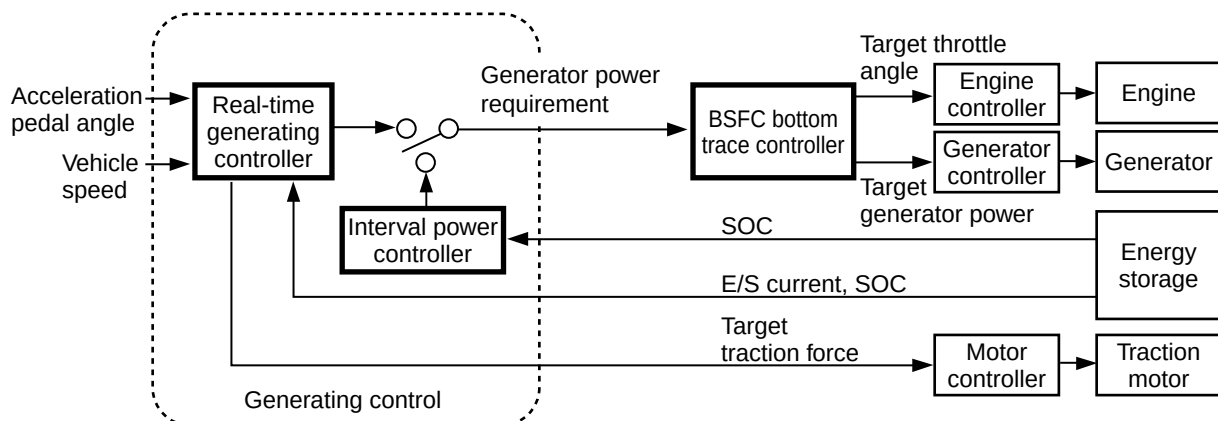


Fig.3 System block diagram

determined by adding a SOC feedback system to the above-mentioned feed-forward system and correcting the basic generator power according to the SOC. This prevents E/S over-charge and discharge caused by the discrepancy between the actual power consumption and estimated power due to temporary discharge from the E/S generated during over-response, recovery of deceleration energy, changes in motor efficiency and other factors. This makes it possible to maintain the SOC at a fairly constant level.

3.3. BSFC Bottom Trace Control

The generating efficiency is the product of generator efficiency and engine efficiency. However, since engine efficiency is dominant over almost the entire operating range, the degree to which engine efficiency can be increased during generation is critical. Utilizing the SHEV feature whereby the engine revolutions are not dependent on the vehicle speed, the required generator power can be generated above the BSFC bottom line shown in Fig.5.⁽⁴⁾⁽⁵⁾

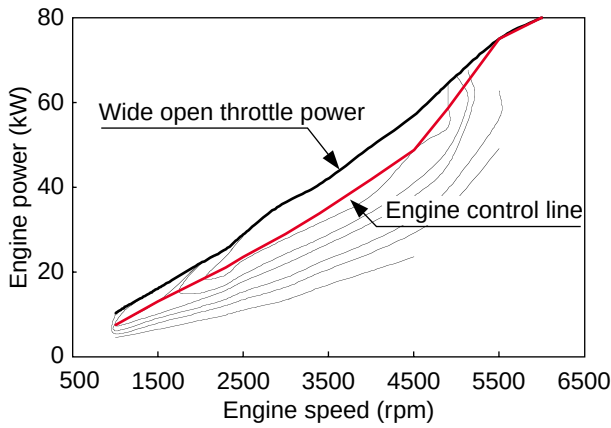


Fig.5 BSFC characteristic

Fig.6 shows the BSFC bottom trace control structure outline. The combination of the engine revolutions and the throttle angle that minimizes the BSFC with respect to the generator power requirement determined by the generating control block is calculated as the target. The actual engine revolutions are fed back to control the generation of a generator linked directly to the engine so that the engine follows the target revolutions, and the engine output is adjusted by controlling the throttle angle. This makes it possible to realize superior response characteristics, even when a particularly rapid generating response is required,

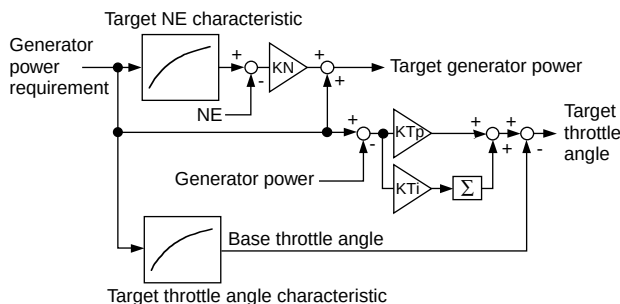


Fig.6 BSFC bottom trace control structure

by feeding back the actual revolutions and actively reducing the engine load to increase the engine revolutions.

3.4. Interval Power Control

As noted thus far, high efficiency generating which follows the motor power can essentially be performed by combining real-time generating control and BSFC bottom trace control. However, Fig.5 shows that the BSFC degrades in the low engine output range. Therefore, when the required motor power is small, the generator is controlled at intervals so that the low output range with poor BSFC is not used. As shown in Fig.7, during interval power control, generating is stopped and power is supplied with only the E/S when the SOC is at the upper limit or higher, and generating is performed at a level that exceeds the motor power when the SOC falls below the lower limit. In addition, the engine is stopped when not generating in order to reduce wasteful fuel consumption due to idling.

In this manner, interval power control works to prevent drops in generating efficiency. However, it also produces E/S charge/discharge loss, so the interval power control range must be determined according to the balance of these two factors.

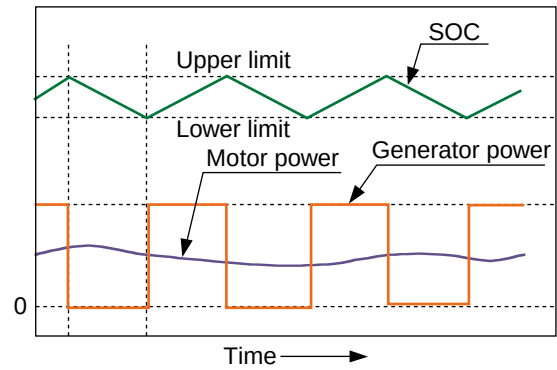


Fig.7 Interval power control

3.5. Recovery of Deceleration Energy

The motor regeneration level was set according to vehicle speed as shown in Fig.8 in order to avoid giving an unnatural feel to the driver when decelerating. In addition, the motor regeneration level is increased during braking to further increase recovery of deceleration energy.

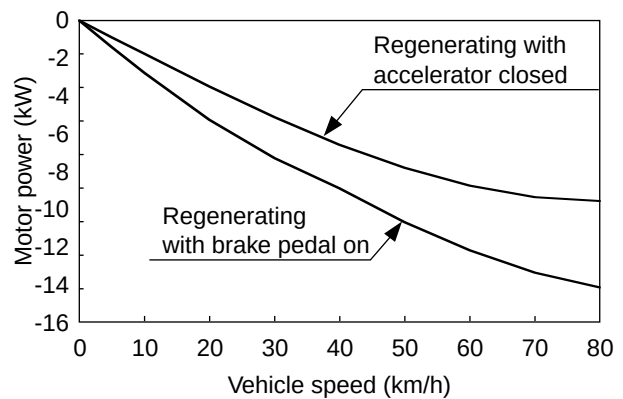


Fig.8 Regenerative braking power curve

4. Verification with an Actual Vehicle

4.1. Prototype Vehicle Specifications

The prototype vehicle was a 1500 kg equivalent inertia weight class, and Table 1 shows the major specifications.

Table 1 Major specification

Model		
Vehicle	Inertia weight (kg)	1500
	Maximum velocity (km/h)	130
Motor	Type	DC brushless
	Maximum power (kW)	73.5
	Maximum torque (N·m)	275
Engine	Type	Water-cooled gasoline engine
	Displacement (cm ³)	1493
	Valve train	SOHC VTEC
	Maximum power (kW/rpm)	95.6/7000
	Maximum torque (N·m/rpm)	139.3/5300
Generator	Type	DC brushless
	Maximum power (kW)	80
	Maximum torque (N·m)	130
Energy storage	Maximum power (kW)	30
	Energy capacity (Ah)	5
	Weight (kg)	60

In addition, Fig.9 shows the maximum motor power line for the prototype vehicle and the maximum traction line for a gasoline-fueled 4AT vehicle of the same inertia weight class. In order to achieve the same dynamic performance as the comparison vehicle, the motor maximum power was set at 73.5 kW in order to satisfy the traction characteristics. The maximum power generation was set at 80 kW in consideration of efficiency so that the entire motor output could be provided during real-time generation. Regarding the E/S weight, an 80 kW/5 Ah E/S is required to guarantee maximum motor power for a conventional SHEV. Therefore, assuming an output density of 500 W/kg, a 160 kg E/S must be mounted. However, with this system the E/S need only

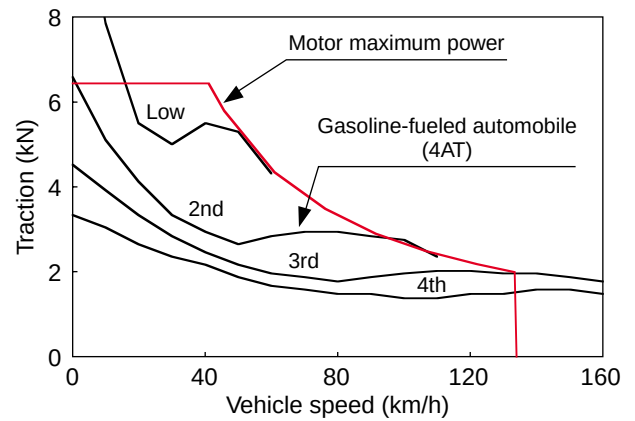


Fig.9 Drive force curve

guarantee the generating response lag during acceleration and the recovery of deceleration energy. Thus a 30 kW/5 Ah and 60 kg E/S can be mounted, thus realizing a weight reduction of 62.5%.

4.2. Control Algorithm

Fig.10 shows the transitions in motor power, generator power and SOC in the 10-15 mode utilized in order to verify the behavior of the entire control system. Power is generated during acceleration when the motor power increases, and power generation is stopped when the generator power requirement is low such as during cruising or idling, or when recovering deceleration energy. The 10-15 mode requires little motor power overall, so generating is frequently stopped by interval power control and the SOC decreases when generating is stopped. However, the consumed energy is replenished by the recovery of deceleration energy and by SOC feedback during real-time generating control, so the SOC is maintained at a fairly constant value.

Fig.11 shows the transition from interval power control to real-time generating control in Fig.10, particularly during acceleration. When the motor power reaches the prescribed value of 10 kW, the engine starts and generating starts. When generating starts, generating is performed at the motor power or higher in order to replenish the energy consumed by interval driving and the generating response lag, enabling the recovery of the SOC. In addition, the generator power

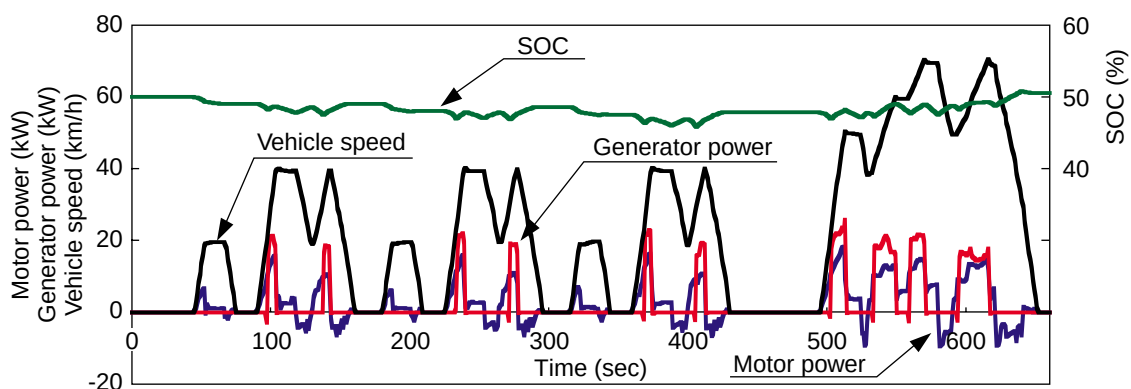


Fig.10 System behavior (10-15 mode)

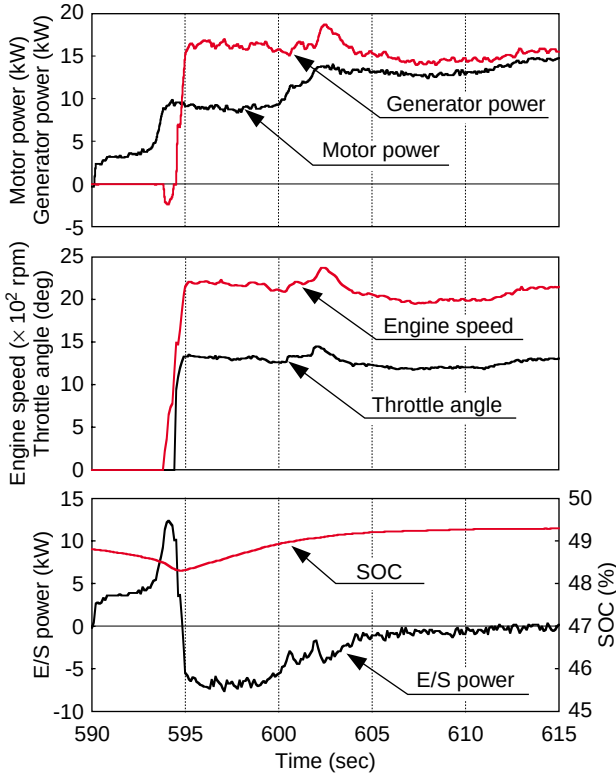


Fig.11 Generating behavior

and the engine power are about the same. Therefore, the engine driving point at this time is approximately 15 kW, 2000 rpm, which matches with the engine minimum BSFC point shown in Fig.5.

4.3. Efficiency

Figs.12 and 13 respectively show the energy transfer efficiency and engine driving range in the 10-15 mode. These graphs show that the energy transfer efficiency reaches approximately 80% of the initial target by performing real-time generating control, and the engine driving range is also on the BSFC bottom line due to the performance of BSFC bottom trace control.

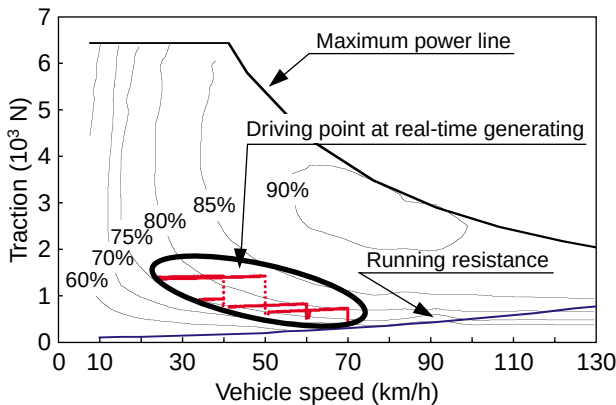
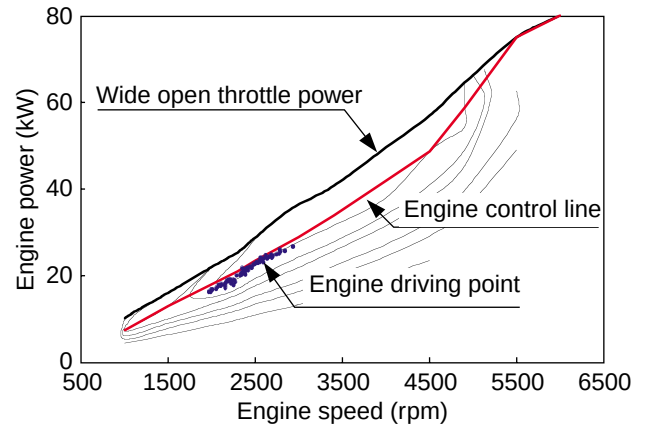

 Fig.12 Transfer efficiency characteristic ($\eta_{MOT} \times \eta_{GEN}$)


Fig.13 BSFC characteristic and engine driving point

4.4. Fuel Efficiency

Fig.14 shows the fuel efficiency improvement effects of each control method in the 10-15 mode. The mode fuel efficiency ultimately reached 23.3 km/L, thus confirming fuel efficiency improvement effects of 94% with respect to the comparison vehicle.

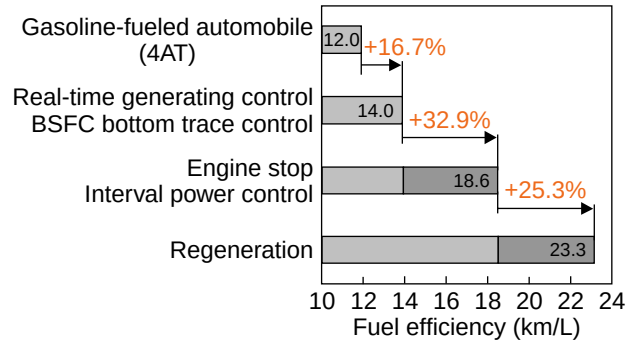


Fig.14 10-15 mode fuel efficiency

5. Conclusions

Focusing on the SHEV characteristic whereby the engine and axle are mechanically separated to facilitate operation in the optimum range of electric generation, the authors created a system that vastly improves fuel economy without sacrificing vehicle dynamic performance, and achieved the following results:

- (1) The E/S power requirement was reduced and at the same time the energy transfer efficiency was improved by performing real-time generating without surplus or deficit in accordance with the motor power. This resulted in an E/S weight reduction of 62.5% and achieved an energy transfer efficiency of 80% in the 10-15 mode.
- (2) The energy transfer efficiency was raised and generating was performed to constantly minimize the engine BSFC by real-time generating control and BSFC bottom trace control. This resulted in fuel efficiency improvement effects of 16.7% in the 10-15 mode with respect to the comparison vehicle.

- (3) Additional fuel efficiency improvement effects of 32.9% over (2) above were obtained by performing interval power control and stopping the engine in the low power range where engine efficiency is poor.
- (4) Fuel efficiency was further improved by 25.3% over (3) above by recovering deceleration energy to ultimately reach 23.3 km/l. This represents fuel efficiency improvement effects of 94.2% with respect to the comparison vehicle.

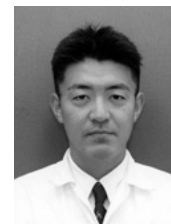
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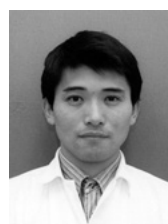
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Development of CFRP Rotor for Flywheel Batteries

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ABSTRACT

Carbon fiber reinforced plastic (CFRP) filament winding was selected as the forming method for the flywheel battery rotor due to its high specific energy. In order to prevent the delamination problem that often occurs with this forming process, a press fit multi-ring construction was employed. This multi-ring construction consists of a CFRP ring with a high elastic modulus, a CFRP ring with medium elastic modulus and an aluminum ring. A safety factor of 2.0 was set and each rotor ring was designed and manufactured with the optimal material and dimensions. Rotational tests were performed through 70 000 rpm whereby it was verified that the 2.0 safety factor is maintained. Further, by taking stress measurements up to 50 000 rpm, we were able to demonstrate the validity of the design.

1. Introduction

The effects of global warming on the environment is an important concern, and one of the main causes of global warming is thought to be greenhouse effects due to the emission of large amounts of CO₂. Auto manufacturers are conducting R&D on a wide range of systems such as reducing fuel consumption rates, alternate fuel vehicles, electric vehicles (EV) and hybrid electric vehicles (HEV), and a number of these efforts have already reached the practical application stage.

Based on the results of much of this research, current development trends are focusing on the development of gasoline/diesel HEV or alternative fuel HEV for the mid-term, and fuel cell vehicles (FCEV) for the long-term.

This paper reports the results of research on a flywheel battery rotor which is part of a concept for exploring the possibility of vast reductions in CO₂ emissions by combining an HEV system utilizing a natural gas engine which has lower CO₂ emissions than a gasoline engine of the same output ratio with the new technologies of a flywheel battery and an adsorbed natural gas (ANG) fuel tank. (Fig.1)

In order to increase the specific energy of the flywheel battery, fiber reinforced plastic (FRP) is often selected as the rotor material. FRP is strong in the tensile direction of the fibers, but extremely weak in the direction 90° to that. Various forming techniques have been devised to overcome this problem, but the performance anticipated from material reinforcement has not been obtained.⁽¹⁾⁻⁽⁴⁾ In addition, while strain and stress during rotation have been theoretically calculated, these results have not yet been verified through actual measurement. This study achieved the target performance by adopting a press fit multi-ring construction and designing and manufacturing a rotor with the optimal material and dimensions for each ring. In addition, the validity of the design was verified by measuring the strain and comparing the calculated and actual measured results.

This study is a part of the fiscal 1998 research for the “High Efficiency Clean Energy Vehicle R&D” project commissioned by the New Energy and Industrial Technology Development Organization (NEDO). This research aims to develop a vehicle with a running energy consumption and CO₂ emissions of no more than half that of existing vehicles through a combination of petroleum alternative clean energy sources and HEV systems.

2. Setting Performance Targets

Flywheel batteries are a type of mechanical battery that stores kinetic energy in a rotating disk. This energy is input and output in the form of electricity via a motor/generator (Mot/Gen) which acts as a motor during charge and a generator during discharge. Fig.2 shows a concept drawing of a flywheel battery.

Table 1 shows the performance targets for a flywheel battery which satisfy both the fuel and motive power performance targets of an ANG-HEV vehicle.

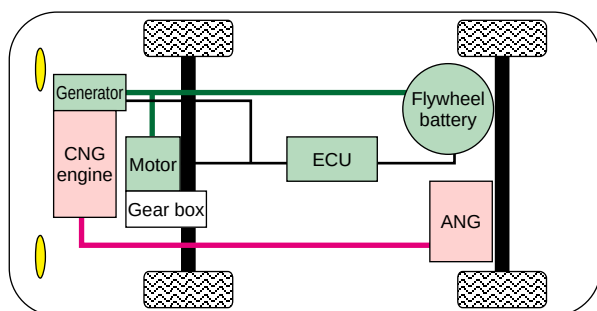


Fig.1 Outline of ANG-HEV system

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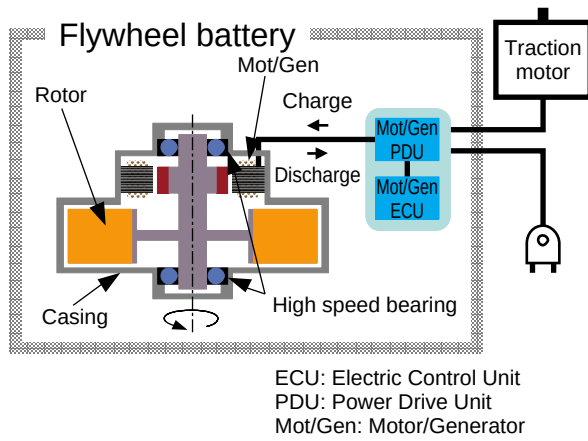


Fig.2 Outline of flywheel battery system

Table 1 Specifications of flywheel battery

Max power	30kW
Usable energy stored	500Wh
Total weight	100kg

3. Design

3.1. Investigation of Materials

Like a chemical battery, in addition to the total amount of energy that can be stored, high specific energy, high specific output and high efficiency are also required of the flywheel when applied as a battery. Of these, the requirement for the rotor, which is the main unit for storing energy, is "high specific energy". Therefore, the author verified the type of rotor material that would meet the requirement for "obtaining high specific energy".

The energy that can be stored was considered assuming the use of a rotating disk such as that shown in Fig.3. The disk was considered to be a solid disk made of isotropic materials in order to simplify the model.

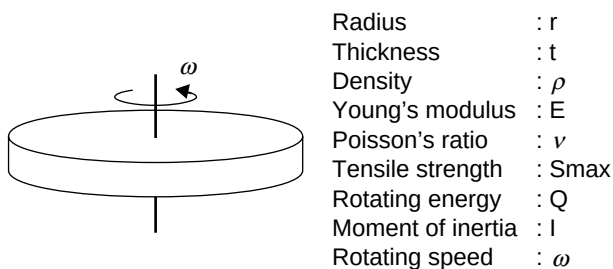


Fig.3 Rotating disk

The amount of kinetic energy that can be stored in a rotating disk is expressed by the following equation.

$$Q = \frac{1}{2} I \omega^2 = \frac{1}{4} \pi \rho r^4 t \omega^2$$

Next, in consideration of the rotational stress produced

on this disk, the maximum attainable revolutions are the revolutions at which the maximum stress produced equals the tensile strength of the materials.

$$S_{max} = \frac{1}{8} \rho r^2 (3 + \nu) \omega_{max}^2$$

Therefore, the maximum specific energy that can be achieved with a rotating disk made from a certain material can be defined by the following equation.

$$(Q/Wt)_{max} = \frac{1}{4} (r \omega_{max})^2 = \frac{2S_{max}}{\rho (3 + \nu)}$$

This equation shows that the maximum achievable specific energy of a rotating disk is determined by the specific strength, which is the value obtained by dividing the tensile strength of the material by the density.

Based on the above, the results of comparing the maximum specific energy of various materials are shown in Table 2.

Table 2 Comparison of available specific energy

	Density ρ (kg/m ³)	Poisson's ratio ν	Tensile strength S_{max} (MPa)	Specific energy Q/Wt (Wh/kg)
Maraging steel (MAS2)	8000	0.31	2942	62
Carbon steel (SCM435)	7830	0.28	1373	30
Titanium (Ti-6Al-4V)	4430	0.19	1177	46
Aluminum (A7075-T6)	2800	0.29	569	34
Thermoplastic (Nylon66)	1140	0.3	78	12
Synthetic fiber (Nylon66)	1140	0.3	665	98
Aramido fiber (Kevlar129)	1368	0.3	2334	287
Glass fiber (S-Glass)	2103	0.3	3158	253
Carbon fiber (T1000G)	1620	0.3	4459	463

Table 2 shows that carbon fiber reinforced plastic (CFRP) rotors have the highest specific energy, so CFRP was selected as the rotor material.

3.2. Investigation of Forming Methods

Carbon fibers are an anisotropic material that exhibit high tensile strength only in the tensile direction of the filaments. Therefore, it was necessary to design a construction which makes the fullest use of this high tensile strength, that is to say a construction which receives the high centrifugal stress produced by rotation in the tensile direction of the filaments.

There are two possible filament arrangements which receive the centrifugal stress in the tensile direction of the filament as follows.

- (I) Arranging the filaments so that they radiate axially outward in a radial direction from the center.
- (II) Arranging the filaments in ring shapes in the hoop direction.

Of these, evenly arranging the filaments from the smaller center areas to the larger peripheral areas is not possible with arrangement (I), making it difficult to actually form a

high strength rotor with this arrangement. Therefore, when forming a flywheel rotor using composite fiber materials, filament winding (II) is selected. (Fig.4)

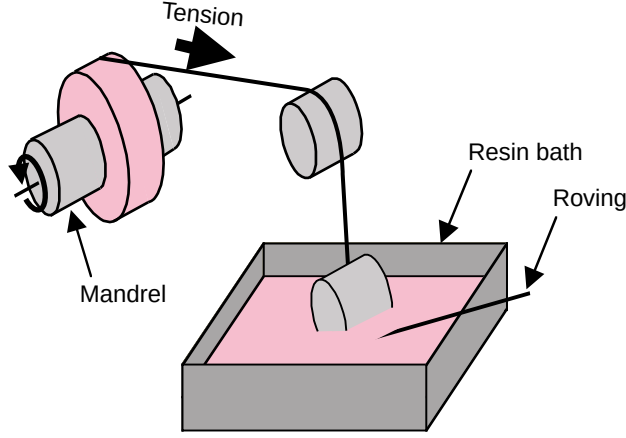


Fig.4 Filament winding

3.3. Calculating Rotational Stress

Filament winding formed rotors exhibit strong anisotropy with an extremely different material constant and strength in the hoop and radial directions. Therefore, stress calculation using a two-dimensional polar coordinate system was selected.⁽⁵⁾ The equations for calculating the rotational stress are shown below.

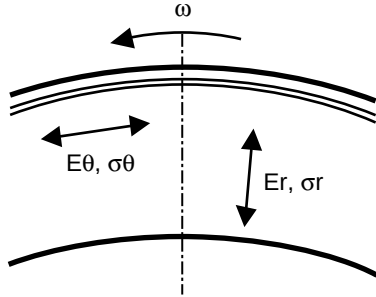


Fig.5 Stress and strain

The symbols used for calculation are defined as follows.

- r: Radius
- ϵ_θ : Hoop strain
- σ_θ : Hoop stress
- E_θ : Hoop Young's modulus
- ν_θ : Hoop Poisson's ratio
- η_θ : Hoop coefficient of thermal expansion
- ΔT : Temperature difference
- E_f : Fiber Young's modulus
- ν_f : Fiber Poisson's ratio
- ρ : Density
- u: Radial displacement
- ϵ_r : Radial strain
- σ_r : Radial stress
- E_r : Radial Young's modulus
- ν_r : Radial Poisson's ratio

- η_r : Radial coefficient of thermal expansion
- V_f : Fiber volume content
- E_m : Resin Young's modulus
- ν_m : Resin Poisson's ratio
- ω : Rotational angular velocity

The relations between the stress and strain are as follows.

$$\epsilon_r = \sigma_r / E_r - \nu_{\theta r} \cdot \sigma_\theta / E_\theta$$

$$\epsilon_\theta = \sigma_\theta / E_\theta - \nu_{r\theta} \cdot \sigma_r / E_r$$

Which can be simplified as follows.

$$F = E_r \cdot \nu_{\theta r} = E_\theta \cdot \nu_{r\theta}$$

$$E_r' = E_r / (1 - \nu_{\theta r} \cdot \nu_{r\theta})$$

$$E_\theta' = E_\theta / (1 - \nu_{\theta r} \cdot \nu_{r\theta})$$

$$F' = F / (1 - \nu_{\theta r} \cdot \nu_{r\theta})$$

The balance between the stress and centrifugal force in the radial direction is expressed by the following equation.

$$d(r \cdot \sigma_r) / dr - \sigma_\theta = -\rho \omega^2 r^2$$

The relations between displacement and strain are as follows.

$$\epsilon_r = du / dr$$

$$\epsilon_\theta = u / r$$

Thus, assuming E_r' , E_θ' and F' are constants the following equation is obtained.

$$E_r' \cdot r^2 \cdot d^2u / dr^2 + E_r' \cdot r \cdot du / dr - E_\theta' \cdot u = -\rho \omega^2 r^3$$

By solving this equation and taking into account temperature dependence, the displacement, strain and stress are expressed by the following equations.

$$u = X \cdot r^\alpha + Y \cdot r^{-\alpha} + A \cdot r^3 + F \cdot r$$

$$\epsilon_r = \alpha \cdot X \cdot r^{\alpha-1} - \alpha \cdot Y \cdot r^{-(\alpha+1)} + 3A \cdot r^2 + F$$

$$\epsilon_\theta = X \cdot r^{\alpha-1} + Y \cdot r^{-(\alpha+1)} + A \cdot r^2 + F$$

$$\sigma_r = B \cdot X \cdot r^{\alpha-1} - C \cdot Y \cdot r^{-(\alpha+1)} + D \cdot r^2 + G$$

$$\sigma_\theta = B \cdot \alpha \cdot X \cdot r^{\alpha-1} + C \cdot \alpha \cdot Y \cdot r^{-(\alpha+1)} + E \cdot r^2 + G$$

$$A = \rho \cdot \omega^2 \cdot (\alpha^2 - \nu_{\theta r}^2) / (E_\theta \cdot (\alpha^2 - 9))$$

$$B = E_\theta / (\alpha - \nu_{\theta r})$$

$$C = E_\theta / (\alpha + \nu_{\theta r})$$

$$D = \rho \cdot \omega^2 \cdot (3 + \nu_{\theta r}) / (\alpha^2 - 9)$$

$$E = \rho \cdot \omega^2 \cdot (\alpha^2 + 3\nu_{\theta r}) / (\alpha^2 - 9)$$

$$F = (\eta_\theta \cdot (\alpha^2 - \nu_{\theta r}) + \eta_r \cdot (\nu_{\theta r} - 1)) \cdot \Delta T / (\alpha^2 - 1)$$

$$G = E_\theta \cdot (\eta_\theta - \eta_r) \cdot \Delta T / (\alpha^2 - 1)$$

3.4. Determining Rotor Specifications and Investigating Strength

Flywheels are characteristic in that energy is stored as rotational kinetic energy, so more energy can be stored at higher revolutions. However, the design requirements were determined as shown in Table 3 according to the following required performances and limitations.

- Specific energy requirements as a system
- Canceling out the gyro-moment using two rotors rotating in opposite directions
- Mot/Gen control capability limit
- Achieving a safety factor of 2.0 based on results for compressed natural gas (CNG) tanks, etc.

CFRP was selected as the rotor material. In order to

Table 3 Rotor design requirements

Usable energy stored	250Wh*1
Total energy stored	333Wh
Available speed range	0 to 5000rpm
Operating speed range	25000 to 50000rpm
Burst speed	over 70000rpm
Rotor radius	$\approx$ Rotor Thickness
Inside diameter/outside diameter	$\approx$ 0.7

*1 Use two rotors for canceling out the gyro-moment

satisfy the performance targets noted in Table 3, the outside diameter of the rotor was changed and the amount of energy that can be stored in those rotors is indicated in Fig.6. This graph shows that a rotor outside diameter of $\phi 270$ is needed to store an effective energy of 250 Wh.

Therefore, the rotor dimensions were determined as follows.

Outside diameter: $\phi 270$ (mm)

Inside diameter: $\phi 190$ (mm)

Thickness: 130 (mm)

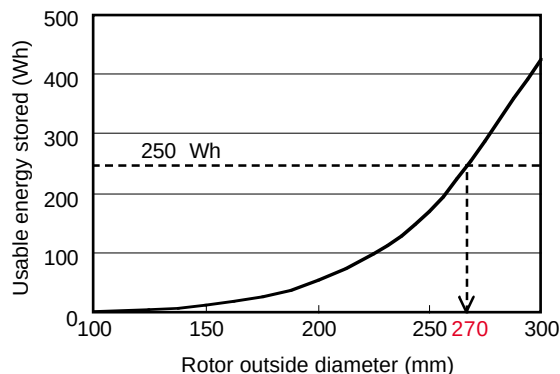


Fig.6 Relation between rotor outside diameter and stored energy

Fig.7 shows the stress calculation results at 70000 rpm for a filament wound rotor having an outside diameter of 270 mm and an inside diameter of 190 mm formed with Toray T1000G, which has the highest tensile strength of currently available carbon fibers. Table 4 shows the Toray T1000G ($V_f = 60\%$ CFRP) allowable stress limit and material constants used for the calculation.

Rotation produces large hoop stress in the tensile direction of the fibers. Further the elongation at the inner and outer hoops differs. Therefore stress is also produced in the radial direction. Filament winding shaped rotors exhibit

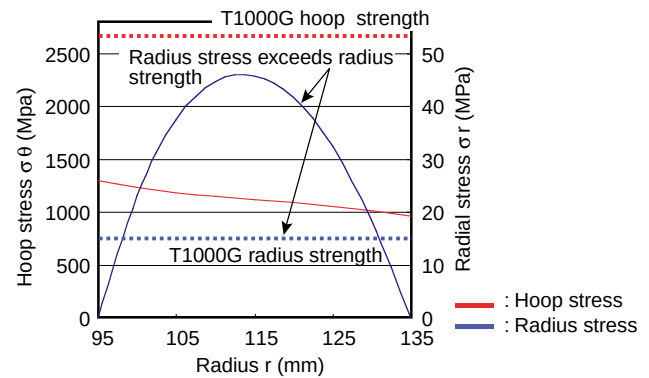


Fig.7 Stress of T1000G rotor at 70000rpm

extreme anisotropy and have a construction where in fibers distributed in layers in the radial direction are fixed by resin. Therefore, the tensile strength is particularly low in the radial direction. Fig.7 shows that the radial stress produced by rotation greatly exceeds the radial tensile strength for a CFRP single layer simple ring winding rotor, leading to delamination in the layers between the fibers. Therefore, suppressing radial stress due to rotation and improving the rotor construction to prevent delamination are important issues.

3.5. Investigation of Construction

The following approaches can be considered to prevent inter-layer delamination of the rotor:

- (I) Adopting a construction which produces compressive force in the radial direction due to rotation.
- (II) Applying an initial compressive force to the rotor which exceeds the produced radial tensile stress.
- (III) Improving the matrix resin strength.
- (IV) Adding reinforcing materials in the radial direction.

Possible forming methods for each of these are as follows:

- (I): Providing a density slope according to radius so that inner rings have a higher density, or providing an elastic modulus slope according to radius so that outer rings have a higher elastic modulus.
- (II): Adopting an integrated construction where in multiple rings are press fit with tightening margins.
- (III): Using a thermoplastic resin matrix.
- (IV): Adopting a three-dimensional fiber construction, or a structure with ring winding disks and radial arrangement disks adhered together.

Of these methods, forming technologies which satisfy the required reliability and strength do not exist for (III) thermoplastic resin matrixes or (IV) three-dimensional fiber constructions. The adhered construction (IV) is commonly used for FRP, but is excluded here because there have been

Table 4 T1000G material specification

Hoop strength σ_{θ} max (MPa)	Radial strength σ_r max (MPa)	Density ρ (kg/m ³)	Hoop modulus E_{θ} (GPa)	Radial modulus E_r (GPa)	Poisson's ratio ν
2675	15	1568	167.6	10	0.33

reports that cracking occurs along the reinforcing fibers rendering this method unusable in forming high strength disks.⁽³⁾ As a result of investigating (I) and (II), a multi-ring press fit construction (multi-ring construction) with an internal metal ring was selected. The densities and elastic moduli for each ring were varied by changing the fibers in each ring to create a construction that prevents delamination. The fibers used were Toray M40J high elasticity carbon fibers for the outer ring, Toray T300 general-purpose carbon fibers for the middle ring, and A7075-T6 for the inner metal ring.

Some filaments are cut when forming and machining the rings. Fig.8 shows the stress calculation results assuming an elastic modulus of 95%.

The stress calculation judgment criteria are that the stress in each part does not exceed the strength limit, and that the stress in the press fit portions does not become tensile. In other words, the press fit portions must not separate.

Fig.8 shows that the stress of the innermost hoop where the hoop stress is the highest for each ring does not exceed hoop strength, radial stress does not exceed radial strength, and the stress on the press fit portions is compressive. Therefore separation does not occur.

Based on the above, the rotor specifications were determined to be as follows: outside diameter, $\phi 270$ mm, maximum operating revolution, 50000 rpm, and limit revolution, 70000 rpm.

Table 5 shows these rotor specifications.

Table 5 Specification of flywheel rotor

Specification of flywheel rotor				
Item	Specification		Remarks	
Outside diameter	$\phi 270$ (mm)			
Inside diameter	$\phi 180$ (mm)			
Thickness	$26 \times 5 = 130$ (mm)			
Weight	6.87 (kg)			
Maximum energy stored	336 (Wh)		50000rpm	
	Outer ring	Middle ring	Inner ring	
Material	M40J	T300	A7075-T6	
Outside dia. (mm)	$\phi 270$	$\phi 235$	$\phi 190$	
Inside dia. (mm)	$\phi 235$	$\phi 190$	$\phi 180$	
Interference	400 (μm)		800 (μm)	
Vf of CFRP	60 (%)		60 (%)	

4. Rotor Fabrication

Fig.9 shows a single ring (thickness 26 mm) of the CFRP portion fabricated based on the rotor specifications shown in Table 5. The inner portion with a lighter color is the T300 ring, and the outer portion which appears blacker is the M40J ring. The M40J and T300 rings are mechanically press fit.

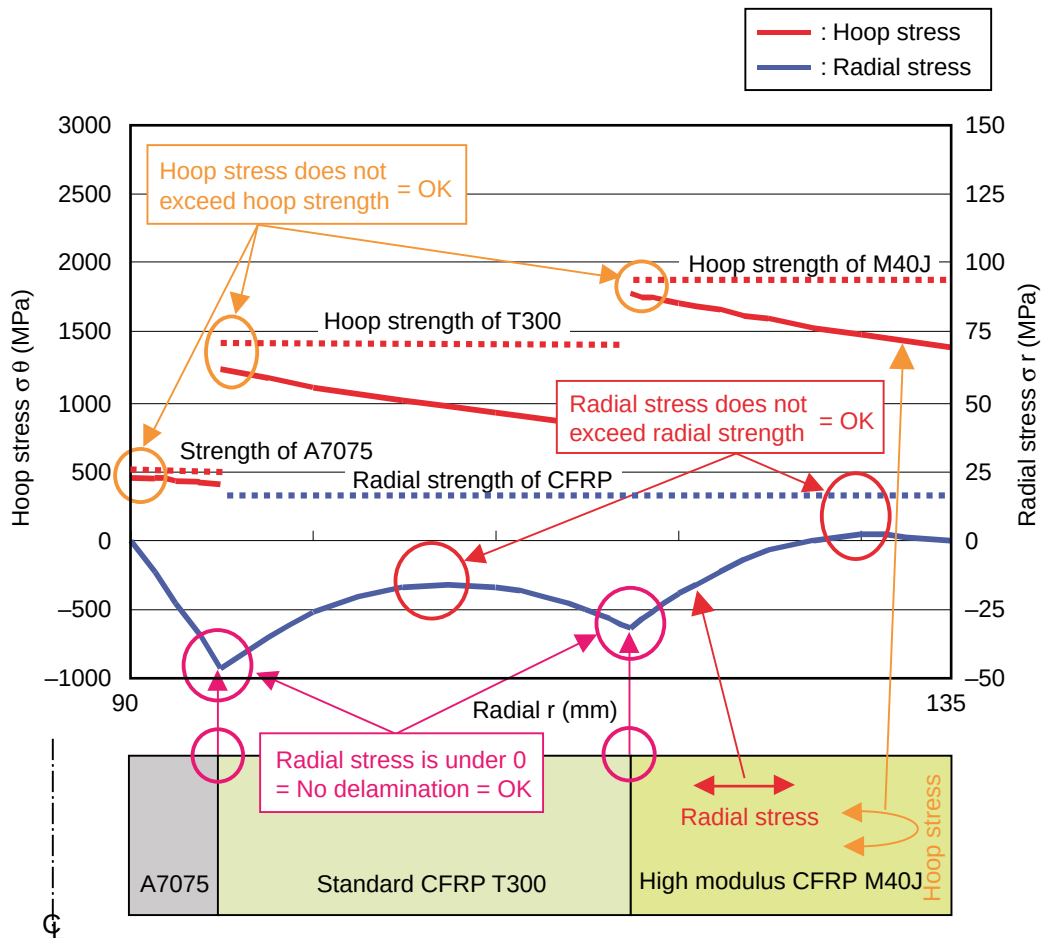


Fig.8 Stress calculation results

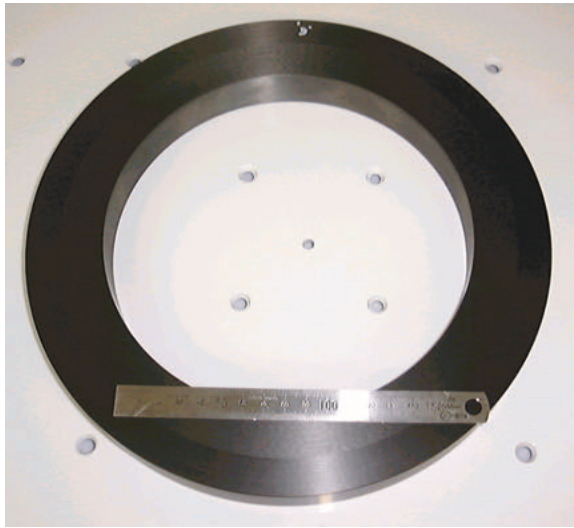


Fig.9 CFRP ring

5. Tests

5.1. Measuring the Change in Inside and Outside Diameter Due to Cooling

Carbon fibers have a negative coefficient of thermal expansion that is near zero. Thus, when carbon fibers are formed into a ring by filament winding and the ring is then cooled, only the resin contracts, and it does not contract in the direction of the fibers. Therefore, the outside diameter contracts and the inside diameter expands. The appropriateness of the dimensional change characteristics with respect to temperature was verified for the various rotor stress calculations. This was carried out by measuring the inside and outside diameters of the prototype rings at a normal temperature (27°C) and at -40°C and comparing these results with the calculated results of expansion and contraction.

Measurement was performed at four points on each of the three individual rings for a total of twelve points.

Fig.10 shows a comparison of the measured and calculated results, and shows that the measured and

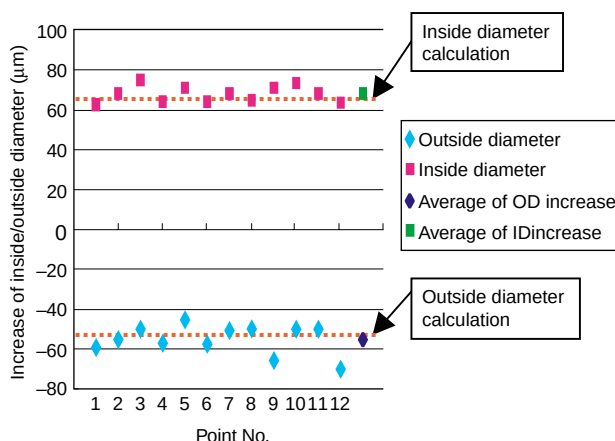


Fig.10 Increase of inside/outside diameter of CFRP ring at -40°C

calculated results match closely. Thus the calculated values for temperature change in inside and outside diameters were verified to be appropriate. Further cold fitting was judged to be possible with the press fitting margin setting determined assuming the expansion of the CFRP ring inner diameter.

Fig.11 shows the rotor constructed by cold fitting the innermost aluminum ring at the temperature of liquid nitrogen.



Fig.11 Complete rotor

5.2. Rotation Test up to 70000 rpm

A rotation test was conducted up to 70000 rpm to check whether the prototype rotors could achieve revolutions of 70000 rpm. Fig.12 shows the rotor in the set condition. The test was conducted in a vacuum using a spin tester with the rotor suspended and rotated by an air turbine.

Axial oscillation was monitored during the test, and the apparatus was set so that oscillation in excess of a certain value due to damage, delamination or any other reason would trigger a reverse in the air supply to the turbine thereby causing it to decelerate.

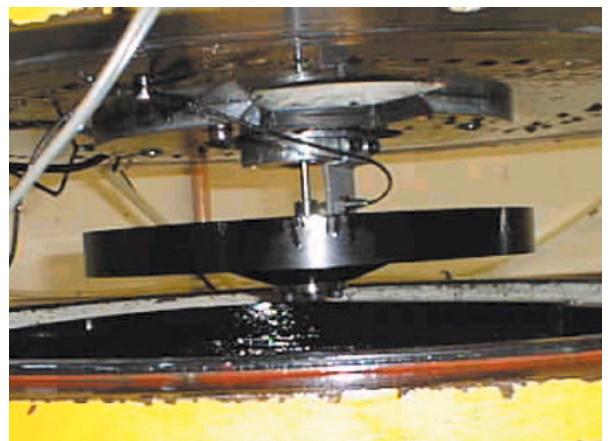


Fig.12 Rotor setting

The test conditions were set as follows, and the results of tests conducted using three rotors are shown in Fig.13.

- Degree of vacuum inside the spin tester: 133 Pa (1 Torr) or less
- Maintain 70000 rpm or more for 3 minutes

The horizontal axis shows the duration, and the vertical axis shows the revolutions. All rotors were able to reach and maintain 70000 rpm for 3 minutes without activating the trigger.

Based on these results, it was determined that a rotor could be obtained with a safety factor of 2.0 with respect to maximum operating revolutions of 50000 rpm.

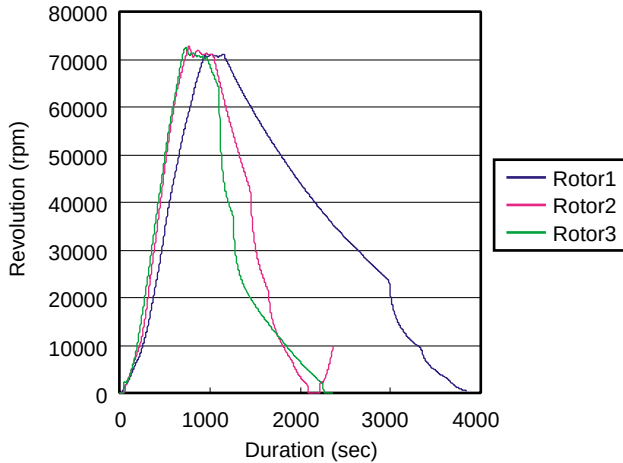


Fig.13 Results of 70000rpm testing

5.3. Rotational Strain Measurement

Aluminum rings were press fit into the individual prototype rings and strain gauges were attached to measure the rotational strain. These tests were conducted using a spin tester as in the 70000 rpm test. Fig.14 shows the rotor with the strain gauge attached, and Fig.15 shows a model drawing of the measurement points.

Hoop strain was measured at the three points of the center of the outer hoop and the center side and inner hoop of the CFRP portion. Radial strain was measured at one point at the center side of the CFRP portion.



Fig.14 Rotor with strain gauge

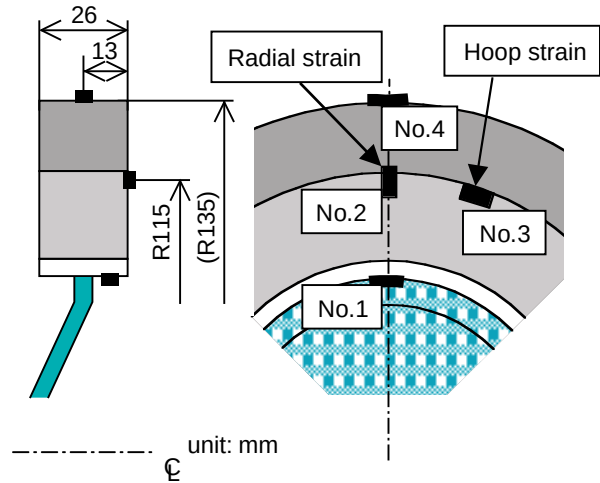


Fig.15 Outline of strain measurement points

Fig.16 shows a comparison of the strain measurement results and calculations up to 50 000 rpm. The strain was measured from two to four times at each point. This graph shows that extremely good measurement values were obtained, and that the measured strain closely matched the calculated values, thus verifying the validity of the filament winding formed anisotropic rotor design method.

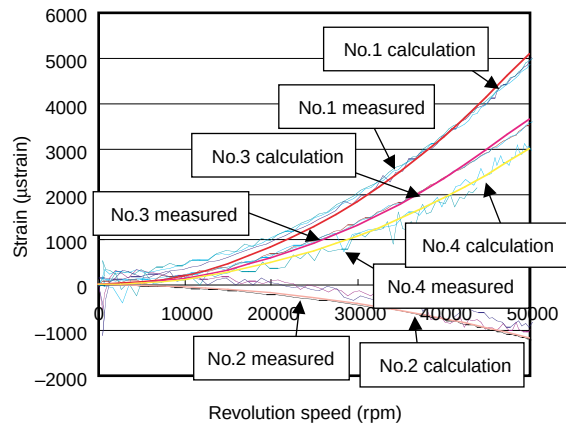


Fig.16 Comparison of calculation and measured strain

6. Conclusions

Regarding a flywheel battery rotor,

- (1) Materials and a construction that allow the specific energy to be increased were determined,
- (2) Rotor dimensions and shapes that satisfy the performance requirements were determined,
- (3) Rotation tests were conducted up to 70000 rpm, and
- (4) Strain was measured up to 50000 rpm.

These results proved that a rotor can be obtained which secures a safety factor of 2.0 over operating ranges up to 50000 rpm. Further, the calculated and measured values matched closely, thus verifying the validity of the design method.

Acknowledgment

In closing the author wishes to express his deep thanks to Ishikawajima-Harima Heavy Industries for its immense cooperation in promoting this research.

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Hiroyuki KOBAYASHI

Thermal Behavior Mechanism of Lithium-ion Batteries for Electric and Hybrid Vehicles

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Noboru SATO*

ABSTRACT

Thermodynamics experiment and study were carried out for the Lithium-ion batteries that are expected as the power sources for electric and hybrid vehicles. It is confirmed that the heat coming in and going out depend on charging and discharging, respectively. And the heat generation factors will be decomposed to three elements: reaction heat value Q_r , polarization heat value Q_p , and joule heat value Q_j . Furthermore, the contribution degree of each factor was able to be expressed quantitatively, by dividing these heat generation factors for charging and discharging.

The accuracy of thermodynamics logic was verified since the thermodynamics calculation coincided with the experimental data of heat generation for the practical Li-ion batteries for EV. It is possible to utilize the calculation and simulation for the development of Li-ion battery and the establishment of thermal control technology not only for EV but also for HEV.

1. Introduction

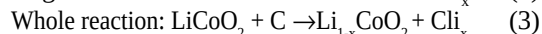
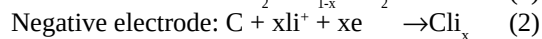
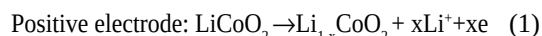
The active development of electric vehicles (EV), hybrid electric vehicles (HEV), and fuel cell vehicles (FCVs) has been accompanied by leaps in battery technology as applied to energy storage. Particularly high expectations are being placed on advances in high-performance nickel-metallic hydrogen (Ni-MH) batteries and lithium (Li)-ion batteries. Ni-MH batteries offer superior performance, reliability, and safety, and are already on the market in the Honda EV Plus released 3 years ago and the HEV Insight released six months ago.

Heat control is an important issue in Ni-MH, lithium-ion, and other cold batteries, which are susceptible to the effects of heat. Temperatures above 50°C can lower the charging efficiency or lower the longevity properties of these batteries. Heat management must also take battery reactions and heat generation factors into account, as exothermic behavior can vary greatly depending on the battery system.

However, there are almost no papers that discuss the thermal behavior of electric vehicle batteries from a thermodynamic perspective⁽¹⁾. Only one existing report quantitatively analyzes the thermal behavior of Ni-MH batteries⁽²⁾. Working from this report, the authors focus on lithium-ion batteries, which are expected to have expanded application in the future, with the goal of clarifying the exothermic mechanism through analysis of heat generation factors, and determining the proper approach to heat management for lithium-ion batteries.

2. The Operating Principle and Characteristics of Lithium-Ion Batteries

A lithium-ion battery generally comprises a positive electrode of lithium cobalt oxide (LiCoO_2), lithium manganate (LiMnO_2 or LiMn_2O_4), or lithium nickel oxide (LiNiO_2), or a hybrid material comprised thereof; a negative electrode of hard carbon or graphite; an electrolyte of a nonaqueous solvent comprising ethylene carbonate (EC) or propylene carbonate; and, LiPF_6 as a supporting electrolyte. This charging and discharging reactions of this type of battery are expressed with equations (1) through (3) below.



In short, these are solid-state reactions in an intercalation mechanism in which (Fig.1) the negative electrode occludes Lithium ions during charging while the positive electrode occludes Lithium ions during discharging.

The use of electrode active materials that are lighter than those in lead batteries or Ni-MH batteries results in the characteristically large energy density per weight of lithium-ion batteries. This, combined with their large volume energy density, is why lithium-ion batteries have replaced Ni-MH batteries in almost all computers and mobile phone applications. The next logical step is to develop technology for large lithium-ion batteries, with the development of such batteries for EVs and HEVs is gaining momentum.

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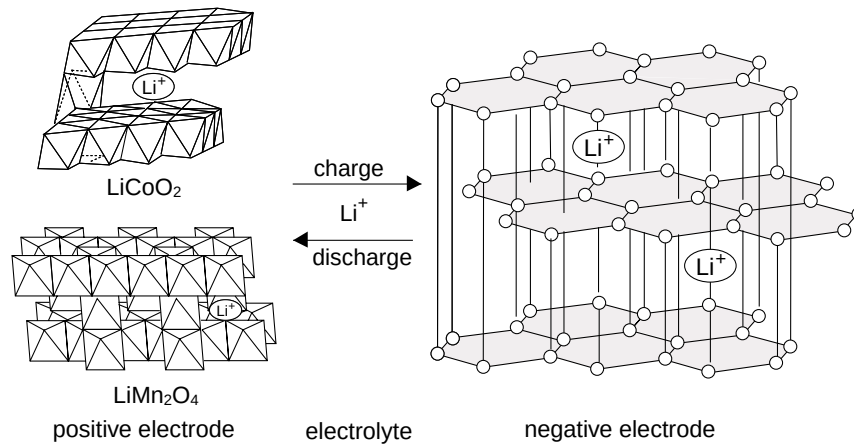


Fig.1 Crystal structures of lithium cobalt oxide and lithium manganese oxide as the positive active material and carbon material as the negative material

3. Impact of the Thermal Action of Lithium-Ion Batteries

The performance of nearly all types of secondary cells is normally degraded by the action of heat. At low temperatures, output falls primarily because of the electrolyte's decrease in electric conductivity, while high temperatures often lower the battery's longevity. As an example, data from the heat-induced capacity degradation of a lithium-ion battery for a 1.25 Ah appliance are given.

The experiment conditions were defined as an ambient temperature of 25°C, 2.5 total hours of charge time at a 1A constant current and a 4.2V constant voltage, discharging at a constant current of 1A, 2A, 3A and 4A, and a final voltage of 2.5V. There was a 10 minute rest time after charging and discharging, and the entire process from charge to discharge was defined as one cycle. Fig.2 gives the relationship between cycle count and capacity ratio at each current value in this experiment. Fig.3 shows the relationship between current value and average battery temperature during

discharge.

Because of the safety issues that still preclude experimentation that would intentionally induce temperature increases in a large-sized lithium-ion battery for electric vehicle application, data from appliance batteries were used instead. However, exothermic mechanisms and effects are essentially the same.

In Fig.3, the lower battery temperature during 4A discharge than at 3A discharge is caused by the shorter discharge time at 4A. Therefore, the minimum voltage of 2.5V was reached and the next cycle was begun before the battery temperature could rise keeping battery temperature relatively low.

Let us next consider degradable rate. The primary causes of degradation are most likely electrode bulk degradation, growth of electrode surface film and changes in composition, and the accompanying electrolyte decomposition. As equations (4) through (12) show, degradation progresses due to the complex interaction of multiple reactions⁽³⁾⁻⁽⁹⁾.

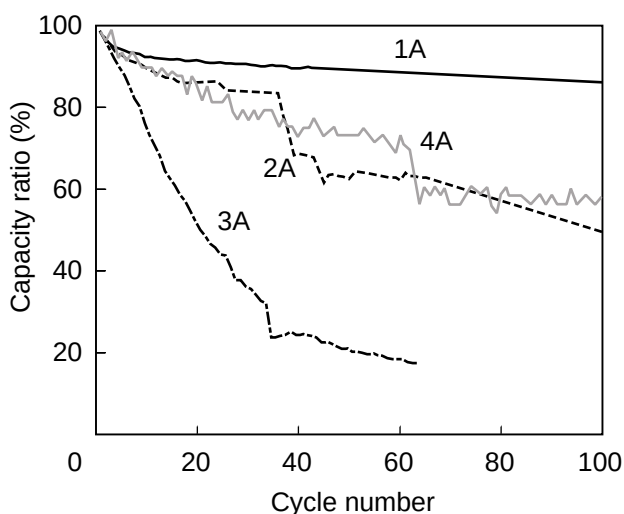


Fig.2 Change of the capacity ratio for each discharge current

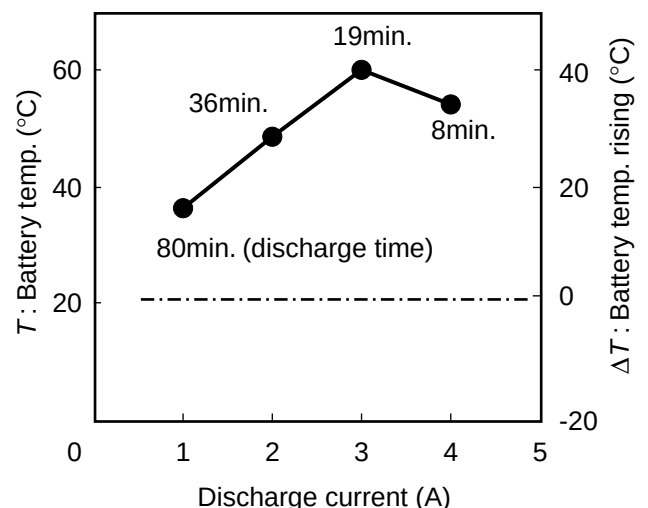
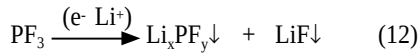
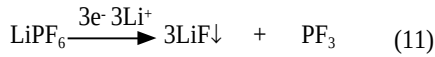
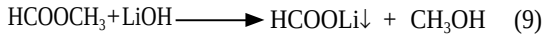
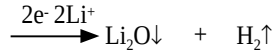
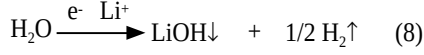
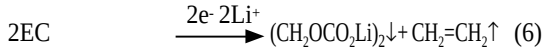
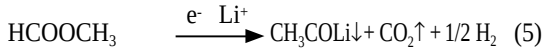
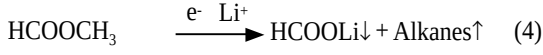


Fig.3 Relationship between the discharge current and the battery temperature rising



At the same time, the electrode degradable reaction rate can be expressed with equation (13).

$$K = I/nF \quad (13)$$

K : Electrode degradable reaction rate (mol/s)

I : Current value consumed by degradation (A)

n : Reaction order

F : Faraday constant (9.64846×10^4 C/equiv.)

To calculate the electrode degradable reaction rate using equation (13), capacity at the point where initial capacity has dropped by 20% (this point is defined as cycle life) is divided by the total amount of time elapsed throughout the cycle life, thus arriving at the current value that is consumed

by battery degradation. As reaction order (n) is either 1 or 2, as in equations (4) through (12), equation (13) is used to calculate the degradable reaction rate at each temperature for each reaction order. The results are in Table 1.

Defining activation energy as E_a (kJ/mol), gas constant as R (8.31451 J/K·mol), and frequency factor as A (A/mol·s), Arrhenius equation can be expressed as in equation (14).

$$\ln K = -E_a/RT + \ln A \quad (14)$$

Equation (14) can then be used to logarithmically convert degradable reaction rate K from the inverse of battery absolute temperature. These results are plotted in Fig.4.

This approximation equation verifies that correlation coefficient R is 0.987, suggesting good correlation. Thus, the mechanism by which degradation is effected by temperature change can be approximated using degradable reaction rate K .

Equation (14) yields the following values: $E_a = 150.6$ (kJ/mol), $A_{n=1} = 3.643$ (mol/s), $A_{n=2} = 3.500$ (mol/s).

Here, activation energy refers to E_a in Fig.5, and is the potential energy barrier that must normally be overcome for the reaction to proceed. In other words, it is the energy needed for the degradation reaction to occur in the lithium-ion battery.

At low temperatures, the electrode surface in a lithium-ion battery is relatively stable, as indicated by the energy level in Fig.5 (the broken line). At higher temperatures, however, the potential energy level rises to the energy level indicated by the solid line. In this state, the smaller difference relative to the potential energy in the activated state makes it easier for the peak in the illustration to be exceeded, at which point the degradable reaction rate speeds up.

Reaction energy when $n=1$ is roughly the same as when $n=2$, but frequency factor A exhibits different values. This

Table 1 Degradable reaction rate

I (A)	1	2	3	4
ΔT (°C)	11.3	23.7	35.0	29.2
Cycle number (∞)	500	38	10	27
Degradable rate: $K_{n=1}$ (mol/s)	1.25E-09	1.99E-08	8.24E-08	3.23E-08
Degradable rate: $K_{n=2}$ (mol/s)	6.25E-10	9.93E-09	4.12E-08	1.61E-08

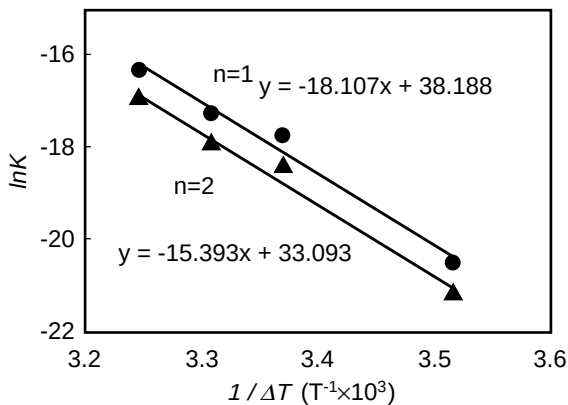


Fig.4 Arrhenius plot for the practical Li-ion battery

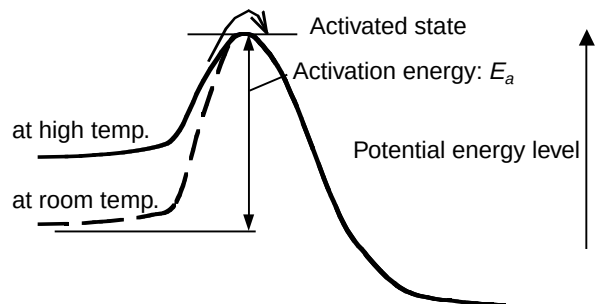


Fig.5 Energy barrier for chemical reactions

represents the differing frequency of the various reactions in equations (4) through (12), and is also seen as a result of complex degradation reactions.

The preceding shows how battery temperature is a major determinant of degradation. Next is an in-depth analysis of the exothermic mechanism operating during charge and discharge of the battery.

4. Analysis of the Battery's Exothermic Mechanism

A battery generally releases and absorbs heat during charge and discharge reactions. Because a reversible battery maintains a thermodynamic relationship at a constant temperature and pressure, the release of heat in battery reactions can be expressed in terms of ΔG , the change in Gibbs free energy equivalent to electric work, as in equation (15)

$$\Delta G = \Delta H - T\Delta S \quad (15)$$

where ΔH is the battery reaction enthalpy change, T is absolute temperature, and ΔS is the battery reaction entropy change. If the battery reaction progresses in reverse, then the discharge and charge reactions are the reverse of one another, and so the intake and release of heat is reversed as well. $T\Delta S$ in equation (15) corresponds to thermal energy in the battery reaction, and is expressed as reaction heat value Q_r in equation (16).

$$Q_r = T\Delta S = T(-\Delta G/\delta T) \quad (16)$$

On the other hand, as for ΔG , if equation (17) holds true when the battery's electromotive force is E_e , Faraday constant is F , and charge number involved in the battery reaction is n , equation (16) can be converted to equation (18).

$$\Delta G = -nFE_e \quad (17)$$

$$Q_r = nFT(\delta E_e/\delta T) \quad (18)$$

However, as the battery's actual terminal voltage (E_p) deviates from equilibrium electromotive force (i.e., electrode potential) due to electrochemical polarization of the battery, it is accompanied by heat generation Q_p , which is the energy loss due to this polarization. Q_p is heat generation during charge and discharge. The battery reaction also results in side reactions, self-discharge and such due to electrolyte decomposition, which are manifested as thermal factor Q_s . Also added is Joule heat Q_j , which is caused by the battery's electrical resistance component. Total heat generation in the battery reaction, Q_t , is therefore expressed with equation (19).

$$\begin{aligned} Q_t &= Q_r + Q_p + Q_s + Q_j \\ &= nFT(\delta E_e/\delta T) + Q_p + Q_s + Q_j \end{aligned} \quad (19)$$

5. Analysis of the Thermal Behavior of Lithium-Ion Batteries

5.1. Thermal Behavior During Charge and Discharge

The side reaction accompanying electrolyte decomposition at the end of the charge period is an irreversible gas generation reaction, which results in the

generation of flammable gas. This is an issue that necessitates control in the stage before this side reaction occurs. Therefore, unlike a Ni-MH battery, the term Q_s is necessarily zero. In actuality, the side reaction that accompanies degradation most likely generates minute quantities of heat as stated in section 3, but this amount is small enough to disregard in comparison to other quantities of heat.

During discharge, the battery heat reaction component (Q_r) is exothermic, and so heat stress caused by the battery reaction is greater than during charging. The addition of Joule heat Q_j resulting from the battery's electrical resistance component further increases heat generation during discharge. Although this heat generation is too large to disregard in batteries having considerable self-discharge and such, the self-discharge of currently marketed lithium-ion batteries is low enough to disregard for the purposes of analyzing thermal behavior, and so is not addressed here. Thus, the term Q_s is zero during discharge as well.

In short, in the case of a lithium-ion battery, equation (19) can be replaced by equation (20) depending on use restrictions. It should be noted, however, that, as stated above, the dominant thermal factors are different during charge and discharge.

$$Q_t = Q_r + Q_p + Q_j = nFT(\delta E_e/\delta T) + Q_p + Q_j \quad (20)$$

5.2. Analysis of Thermal Behavior Factors

5.2.1. Reaction Heat Value: Q_r

The reaction heat during charge in a lithium-ion battery corresponds to the heat absorbed when Lithium ions from the positive electrode are intercalated by the negative electrode. In addition, a quantity of heat equivalent to this is generated during discharge. If total generated heat in equations (1) and (2) is Q_1 (kJ/mol) and battery charge current is I_c , then the corresponding reaction heat value per unit time (Q_r) is as expressed in equation (21). Thus, reaction heat value per unit time is expressible with a first-order function of charge current.

$$\begin{aligned} Q_r &= Q_1/(F/3600) \cdot I_c \\ &= -3.37 \times 10^{-2} \cdot Q_1 \cdot I_c \text{ (kJ/h)} \end{aligned} \quad (21)$$

5.2.2. Polarization Heat Value: Q_p

The reason that polarization results in a discrepancy with theoretical electromotive force in an actual battery reaction is that energy is required for the diffusion and movement of atoms in the battery reaction process. Lithium ions have a low chemical diffusion coefficient of $5 \times 10^{-9} \text{ cm}^2/\text{s}^{(10)}$, and so diffusion and movement inside the electrodes are the predominant factors. Also discussed in section 3, the electrode surface degradation accompanying the cycle also increases polarization. Thus, the key to controlling heat generation is a structural design that reduces polarization.

R_{pd} , the polarization resistance component that accompanies this polarization during discharge is manifested as the polarization resistance component corresponding to a portion of total battery resistance during discharge (R_{td}), and so can be expressed as shown in equation (22), where R_e is the electrical resistance component. Polarization heat value per unit time (Q_{pd}) is expressed with equation (23), where I_d is the discharge current. In the case of charging, resistance

component R_{pd} and current value I_d are replaced with R_{pc} and I_c , respectively, yielding equations (24) and (25).

$$R_{pd} = R_{td} - R_e \quad (22)$$

$$Q_{pd} = I_d^2 \cdot R_{pd} (W) = 3.60 \cdot I_d^2 \cdot R_{pd} (kJ/h) \quad (23)$$

$$R_{pc} = R_{tc} - R_e \quad (24)$$

$$Q_{pc} = I_c^2 \cdot R_{pc} (W) = 3.60 \cdot I_c^2 \cdot R_{pc} (kJ/h) \quad (25)$$

Because this heat generation is manifested as total Joule heat, polarization heat values (Q_{pd} and Q_{pc}) are not calculated directly, but rather must be calculated by separating the polarization resistance components out of the total resistance components in equations (22) and (24) and then calculating them as the heat generation corresponding to that resistance component.

5.2.3. Joule Heat of the Electrical Resistance Component: Q_j

Heat generation that is dependent on the battery's internal resistance is manifested as Joule heat. However, as shown in equations (22) and (24), internal resistance comprises electrical resistance and polarization-induced reaction resistance, with the heat generated being the total of the two. As for the latter, as stated in section 5.2.2, Joule heat generated per unit time due to the electrical resistance portion is expressible with equation (26), where I_d is the discharge current. Joule heat during charging can be expressed with equation (27) by replacing I_d with I_c and electrical resistance R_{ed} with R_{ec} .

$$Q_j = I_d^2 \cdot R_{ed} (W) = 3.60 \cdot I_d^2 \cdot R_{ed} (kJ/h) \quad (26)$$

$$Q_j = I_c^2 \cdot R_{ec} (W) = 3.60 \cdot I_c^2 \cdot R_{ec} (kJ/h) \quad (27)$$

Thus, reducing this Joule heat necessitates a material design and a structural design that minimize the battery's electrical resistance component (R_e), which is determined by the current network linking the collector and the electrode active material, by the separator (a nonconductive material), and the electrolyte, among others. As Fig.6 shows, R_{td}

changes depending on the discharge condition (DOD), increasing in the zone where the DOD value is large. Here, 50% DOD is defined as the baseline value of 1.0.

As Fig.7 shows, the PC/DMC/LiPF₆ electrolytes commonly used in lithium-ion batteries have an electric conductivity that varies depending on temperature. A shift in battery temperature to the low-temperature side therefore increases the battery's internal resistance.

5.2.4. Heat Balance During Charge and Discharge

(1) Heat balance during charge: Based on the above and using equation (20), charge heat balance per unit time (Q_c) is expressed with equation (28).

$$\begin{aligned} Q_c &= Q_r + Q_{pc} + Q_j = -3.73 \times 10^{-2} \cdot Q_i \cdot I_c \\ &\quad + 3.60 \cdot I_c^2 \cdot R_{pc} + 3.60 \cdot I_c^2 \cdot R_{ec} \\ &= -3.73 \times 10^{-2} \cdot Q_i \cdot I_c + 3.60 \cdot I_c^2 \cdot R_{tc} (kJ/h) \end{aligned} \quad (28)$$

(2) Heat balance during discharge: Similarly, discharge heat balance per unit time (Q_d) is expressed with equation (29).

$$\begin{aligned} Q_d &= Q_r + Q_{pd} + Q_j = 3.73 \times 10^{-2} \cdot Q_i \cdot I_d \\ &\quad + 3.60 \cdot I_d^2 \cdot R_{pd} + 3.60 \cdot I_d^2 \cdot R_{ed} \\ &= 3.73 \times 10^{-2} \cdot Q_i \cdot I_d + 3.60 \cdot I_d^2 \cdot R_{td} (kJ/h) \end{aligned} \quad (29)$$

5.2.5. Changes in Reaction Heat and Joule Heat According to Current Value

Here, equations (28) and (29) are used to analyze changes in reaction heat and Joule heat according to current value. In equation (28), the 1st term is the endothermic reaction heat; the 2nd term, exothermic Joule heat. The 1st term in equation (29) is the exothermic reaction heat. This relationship is shown in Fig.8.

The generated heat (Q_j) and internal resistance (R_i) contained in the 1st and 2nd terms change depending on DOD. Here, each term's heat generation is calculated for 50% DOD, a representative value. Q_i is 9.39 kJ/mol while internal resistance (R_i) depends on battery size: 1-2 mΩ per EV cell, 3-5 mΩ per HEV cell. As these are test calculations, changes in the reaction heat and endothermic Joule heat during discharge were calculated with a value of 1.2 mΩ for

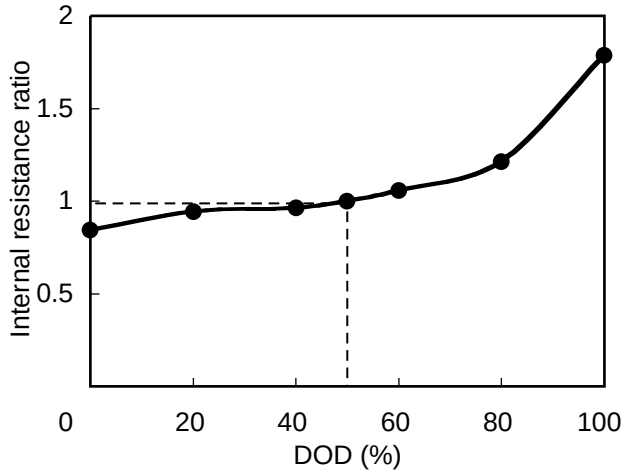


Fig.6 Relationship between DOD and internal resistance ratio

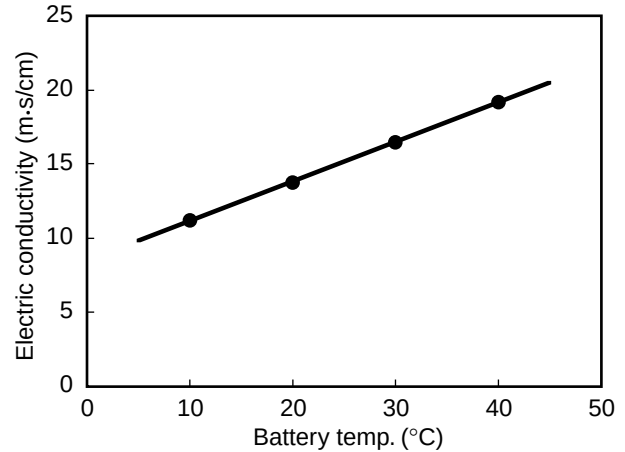


Fig.7 Battery temperature dependence on electric conductivity of 1.0M-PC/DMC/LiPF₆ electrolyte

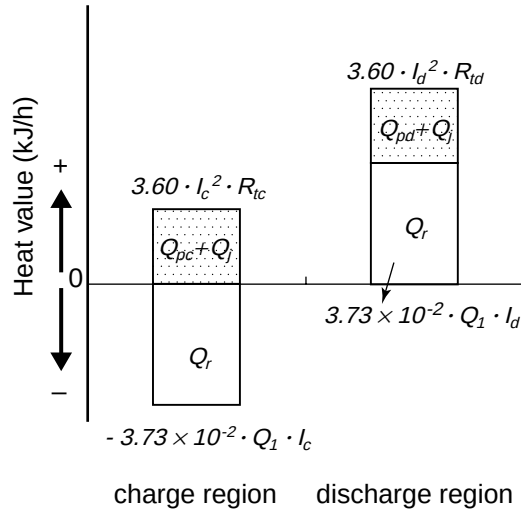


Fig.8 Heat intake and release model of Li-ion batteries during charge and discharge process

EV cells and 4 mΩ for HEV cells (Fig.9).

During charge, the heat of reaction is endothermic. In the case of the EV lithium-ion battery, the endothermic heat of reaction exceeds the exothermic Joule heat up to 70A or 80A. Thus, Q_c , the value of heat during charge, is negative, and the battery temperature drops. During discharge, however, both the heat of reaction and Joule heat are exothermic, and the temperature rises continuously. In an EV lithium-ion battery, the temperature increase due to the reaction heat makes a large contribution to total temperature increase up to 70A or 80A, but beyond that, the contribution of Joule heat is larger.

In contrast, a low-capacity lithium-ion battery, such as one designed for HEVs, has a larger internal resistance than an EV lithium-ion battery, and so the contribution of exothermic Joule heat exceeds that of the reaction heat at

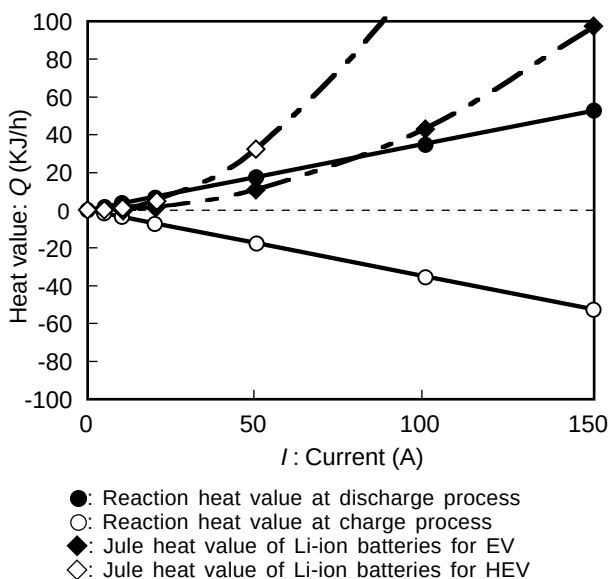


Fig.9 Relationship between current and heat value

about 20A to 30A. In short, an HEV lithium-ion battery has a greater frequency occurrence of large currents of 20A or higher, and so internal resistance must be reduced in order to prolong battery longevity.

6. Comparison of Simulation and Actual Measurements

This section compares actual measurements with the results of heat generation simulation performed using equations (28) and (29). From battery output (W) in the vehicle driving mode in Fig.10, mode current value was calculated with equation (30), after which heat generation was calculated with equation (28) for negative current values. Equation (29) was used for positive current values.

$$I = \{E_e - (E_e^2 - 4 \cdot R_i \cdot W)^{1/2}\} / 2 \cdot R_i \quad (30)$$

As in equation (31), this heat generation value was divided by C (kJ/K) (battery heat capacity) to arrive at ΔT , battery temperature change per unit time. In accordance with the battery component materials, a value of 3.42 (kJ/K) was used for battery heat capacity. As Fig.11 shows, the accuracy of simulation was improved by also including thermal radiation rate per unit time (T_r) at 25°C in an insulated state.

$$\Delta T = Q_i / C - T_r \text{ (K/h)} \quad (31)$$

The results of simulation are shown in Fig.12 along with the actual measurements. The actual measurements — which were taken with a large 80Ah EV lithium-ion battery wrapped in insulating material and so indicate a battery temperature in a state approaching an insulated one — correspond well with the results of simulation. The reason for the slight discrepancy between the simulation results and actual measurements starting at 70% DOD is that the value of internal resistance R_i at 50% DOD was used throughout the simulation process for the purpose of simplification. In actuality, internal resistance varies depending on DOD, showing a particularly large increase in the high DOD zone (Fig.6). In other words, the simulated values were slightly larger than the measured values in the high DOD zone because the internal resistance value used in the simulation

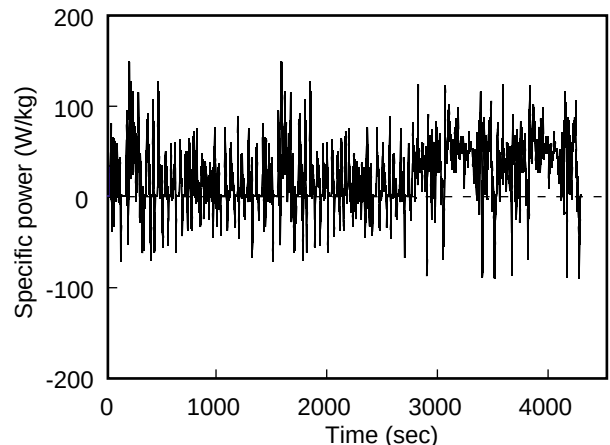


Fig.10 Vehicle test mode

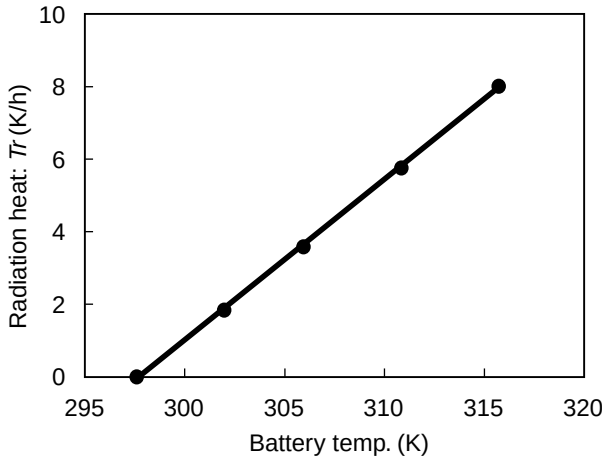


Fig.11 Correlation between battery temperature and radiation heat with an adiabatic state

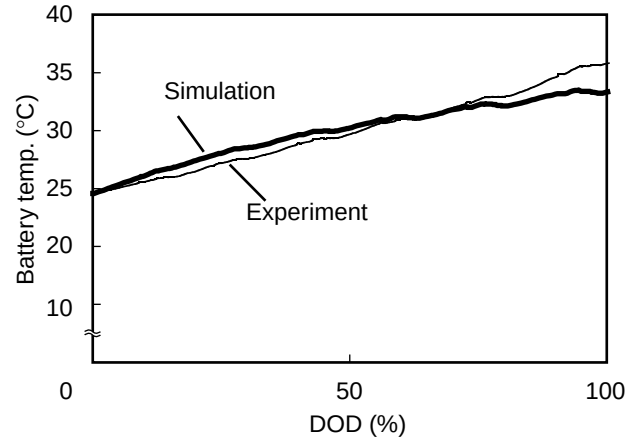


Fig.12 Comparison of measured heat value with simulation data

was lower than actual internal resistance.

The preceding verifies the theoretical correctness of the results of exothermic behavior analysis described in section 5. Although the simulation described in this paper assumed a lithium-ion battery for EV applications, similar trial calculations are, of course, also possible for HEV applications, as well.

7. Conclusion

Thermodynamic analysis of battery reactions permitted the fine categorization of the specific heat generation factors listed below, and the estimation of a battery's exothermic properties.

- (1) Heat generation factors can be divided into three types: reaction heat value (Q_r), polarization heat value (Q_p), and Joule heat (Q_j), expressible as $Q_r = 3.73 \times 10^{-2} \cdot Q_1 \cdot I$, $Q_p = 3.60 \cdot R_p \cdot I^2$, and $Q_j = 3.60 \cdot R_e \cdot I^2$.
- (2) When used as the positive electrode, LiCoO_2 results in an endothermic reaction heat value during charge. In a large, 80Ah EV lithium-ion battery, this endothermic reaction heat exceeds the exothermic Joule heat with a current value of up to 70A or 80A, resulting in a negative value for charge heat (Q_c).
- (3) Further categorizing the heat generation factors into charge and discharge factors and performing simulation using these values yield results that corresponded well with actual measurements, thereby verifying the correctness of this analytic method.
- (4) This method can quantitatively estimate the exothermic properties of lithium-ion batteries of differing capacities, such as those for EV and HEV applications.

8. Notes

The authors' analysis of exothermic behavior mechanism not only made it possible to quantitatively analyze heat generation by a lithium-ion battery under actual load conditions, but also provided guidelines for the design of lithium-ion batteries for electric vehicles. It is hoped that

this paper will help further efforts to develop batteries for HEV and EV applications.

List of Symbols

- ΔG : Change in Gibbs free energy
- ΔH : Enthalpy change of the battery reaction
- ΔS : Entropy change
- E_a : Activation energy
- T : Absolute temperature
- K : Electrode degradable reaction rate
- R : Gas constant
- A : Frequency factor
- F : Faraday constant
- C : Battery heat capacity
- W : Battery output
- T_r : Thermal radiation temperature
- Q : Quantity of heat
- Q_t : Total battery heat
- Q_c : Charge heat
- Q_d : Discharge heat
- Q_r : Reaction heat value
- Q_p : Polarization heat value
- Q_s : Side reaction heat
- Q_j : Joule heat
- I_c : Battery charge current
- I_d : Battery discharge current
- R_t : Battery total resistance
- R_{tc} : Battery resistance during charge
- R_{td} : Battery resistance during discharge
- R_p : Polarization resistance
- R_{pc} : Polarization resistance during charge
- R_{pd} : Polarization resistance during discharge
- R_e : Electrical resistance

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Development of a Lane mark Recognition System for a Lane Keeping Assist System

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ABSTRACT

A lane mark recognition system that processes images taken by a camera for a Lane Keeping Assist System (LKAS) has been developed. This system is a main component of the Honda Intelligent Driver Support System. This system uses a new algorithm that is a combination of Hough transformation and the process of extracting positive and negative edges which are features of lane marks. By employing this algorithm, we can remove noise and execute high speed image processing with a special chip. As a result, we can process images in real time even in such complex situation as a parting road or double broken line conditions.

In experiments on real images, we were able to verify a high rate of recognition and accuracy of output data sufficient for steering control.

1. Introduction

Recent advances in electronics technology have spurred considerable R&D on intelligent system for automobiles, and some technologies have already been commercialized. One advanced experimental system being developed, for instance, uses laser radar installed on the vehicle front to detect forward cars and automatically adjust distance to the forward car⁽¹⁾. The authors have developed a more evolved system: the Honda Intelligent Driver Support System. This system includes the Lane Keeping Assist System (LKAS), which recognizes lane marks from images taken by a camera and controls Electronic Power Steering (EPS) torque based on those results in order to assist the driver in staying in the current lane. The issues in developing this LKAS system were (1.) the inability of conventional line mark recognition systems to provide sufficient positional accuracy for lane keeping control and (2.) their inability to handle lane marks in complex situations such as turnoffs and double broken lane marks. To overcome these issues, we developed a new lane mark recognition system that supports LKAS. This paper gives an overview of this lane mark recognition system and describes the results from experimentation using actual road images.

2. Overview of the System

2.1. Overview of the Honda Intelligent Driver Support System

The Honda Intelligent Driver Support System is comprised of two main systems (Fig.1): the Adaptive Cruise Control (ACC) system, which uses laser radar to determine the distance to the forward car, then operates the accelerator

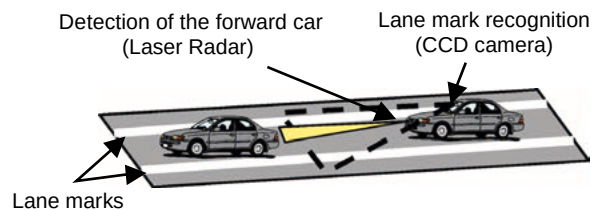


Fig.1 Main functions of Honda Intelligent Driver Support System

and brake to maintain proper distance to the forward car, and LKAS, which assists the driver in staying in the lane by using a CCD camera to take images of the road in front of the car, then controlling EPS torque according to the lane marks determined by processing those images. As the lane mark recognition system discussed in this paper was developed for LKAS, an overview of LKAS follows.

2.2 The Flow of Processing in LKAS

Fig.2 is a flowchart of LKAS processing, which controls EPS torque to help the driver to steer in order to keep the vehicle in the current lane. The data processing designed to achieve this goal begins by taking images of the road surface ahead of the car with a CCD camera mounted near the rearview mirror. Next, these images are processed by the image processing hardware and the CPU to extract the lane position and road curvature. This lane information is then used to calculate the EPS torque that is needed to assist the driver in staying in the lane. Lastly, this assist torque is sent

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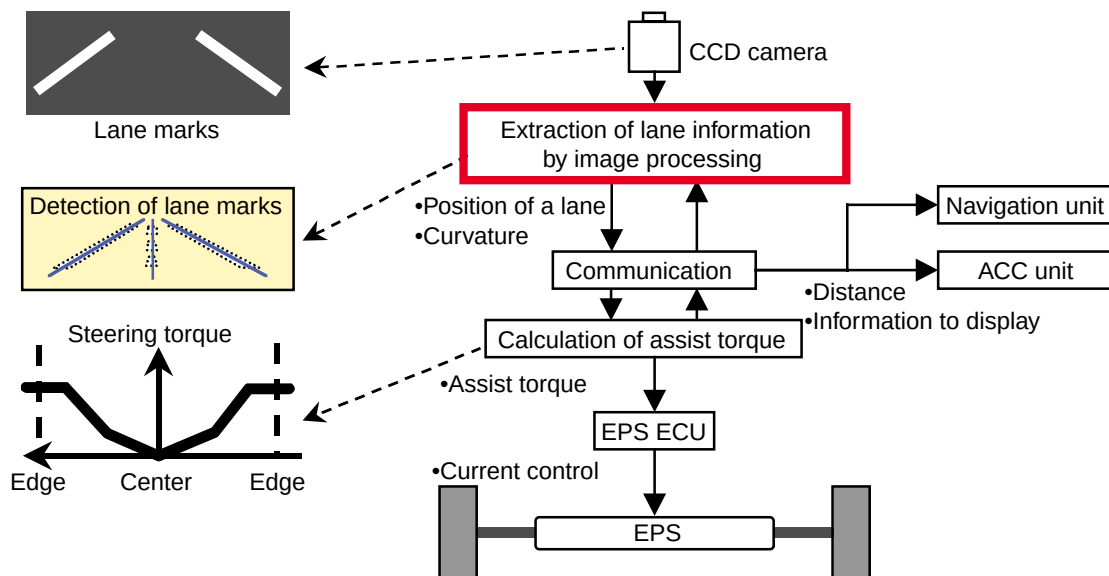


Fig.2 Flowchart of the LKAS process

to the ECU of the EPS for use in control of the EPS.

The flowchart in Fig.3 illustrates the image-processing portion of the process. As is indicated, the lane mark recognition process consists of two steps. In step 1, several sequences of points, each representing candidate lane marks, are identified. In this step, the goal is to extract as many sequences of points (i.e., lane mark candidates) as possible. Although these will include guard rails and other objects that are not lane marks, as well as lane marks that are unrelated to the car's current lane (as in the case of a turnoff), these extraneous sequences of points can be eliminated in the next step; the primary goal here is to avoiding missing the actual lane mark. In step 2, the candidate that is most suitable as the lane mark is selected, and lane information such as lane position and road curvature are calculated. Past

data is stored so that lane mark positions can be determined with greater accuracy in this step. Thus, lane mark position is estimated by selecting the most likely lane mark candidate based on consistency with past data, and then statistically processing the past data together with the selected lane-mark sequence of points. Steps 1 and 2 are discussed in detail later in this paper. First, section 3 describes the process of extracting sequences of points representing lane-mark candidates; section 4, the data evaluation process.

3. Method for Extracting Sequences of Points Representing Lane-Mark Candidates

3.1. Images Used

Fig.4 is an example of the type of images that must be processed in LKAS. In this system, the first stage of image processing is edge extraction to eliminate the effect of changes in density due to differences in brightness. Fig.4 is an edge-extracted image. Because the Honda Intelligent Driver Support System assumes use on expressways, the lane mark recognition system also targets lane marks not on ordinary roads but on expressways. Expressway characteristics include merges and turnoffs (b) at interchanges and service areas, along with double broken lines at relatively sharp curves (f). Here, "double broken line" refers to a broken line that is painted on the inside of an ordinary lane mark, and which is thicker and broken at shorter intervals, visually increasing the driver's sensation of speed so as to encourage him to decelerate. Other elements that must be addressed include road repair marks (d), guardrails, and the shadows of guardrails (e). An example of a complex image is one resulting from the shadows of a railway bridge (c).

3.2. The Basic Principles of Lane Mark Recognition

Let us consider the basic principles of how lane marks are recognized through image processing. In this system, lane marks are represented using pairs of positive and negative edges. A positive edge is an edge where density

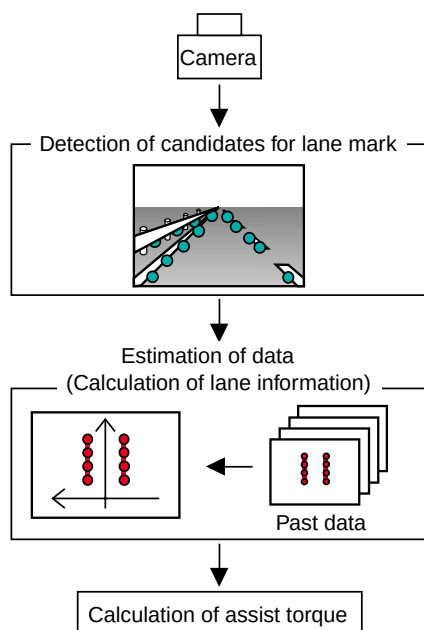


Fig.3 Flowchart of the lane mark recognition

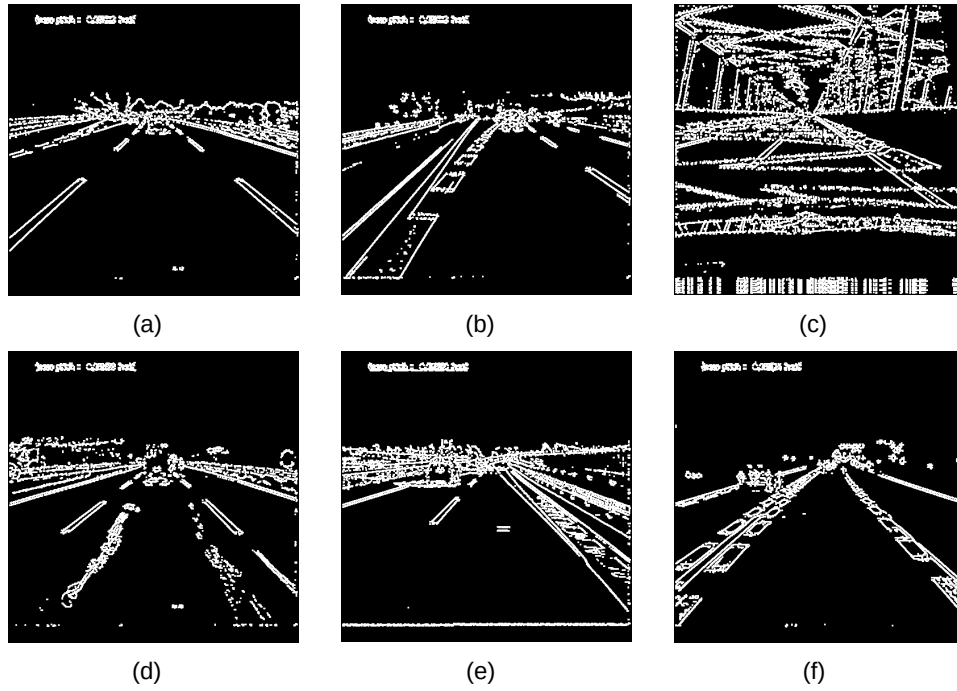


Fig.4 Sample of road images (after edge extracting process)

increases when the image is scanned from left to right. Similarly, a negative edge is where density decreases from left to right (Fig.5). A point is deemed a candidate lane mark when a positive and a negative edge are arranged in such a way that:

- (1) they are within a predetermined distance of each other and
- (2) they appear in the sequence positive $\rightarrow$ negative when

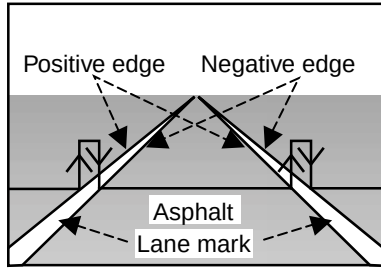


Fig.5 Definition of the positive/negative edge

the image is scanned from left to right.

Also employed in the recognition of a lane mark is the linear arrangement of a broken edge. On an expressway, the portion of a lane mark that is relatively close to the vehicle is, on the screen, nearly linear, even in places where the road curves. This portion is valid as it occupies the majority of the screen. Hough transformation⁽²⁾⁽³⁾ is widely used as an image processing technique for extracting sequences of points that are arranged linearly on screen, and research on the use of Hough transformation in lane mark recognition has, in fact, been carried out⁽⁴⁾⁽⁵⁾. As this system also utilizes this technique, explained next is the principle behind Hough transformation and how it is actualized on a computer. First, let us assume that a point (x, y) exists on plane X-Y, like that

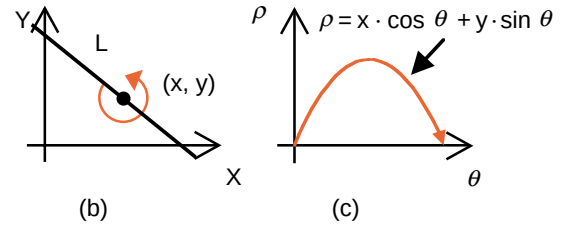
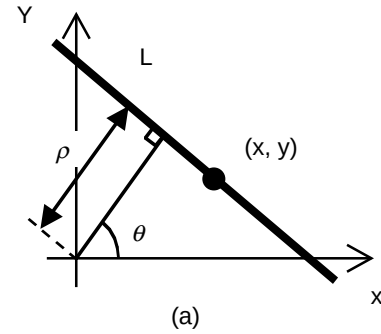


Fig.6 Definition of Hough transformation (1)

in Fig.6-a. As shown in the figure, ρ and θ are calculated for straight line L , which passes through this point, where ρ is the length of a perpendicular from the origin to L , and θ is the angle between this perpendicular and the X axis. The relationship between x , y , ρ , and θ is such that the following holds true.

$$\rho = x \cdot \cos \theta + y \cdot \sin \theta \quad (1)$$

When straight line L is rotated around point (x, y) (see Fig.6-b), ρ and θ change. This change, when drawn on plane ρ - θ , appears as in Fig.6-c. Next, Fig.7 shows this operation performed for multiple points (x_i, y_i) ($i = 1, 2, \dots$) arranged

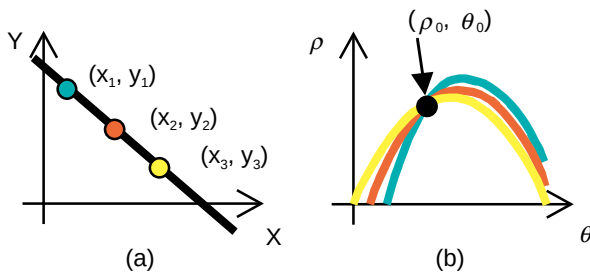


Fig.7 Definition of Hough transformation (2)

linearly on plane X-Y. Here, the intersection of the curves drawn on plane ρ - θ correspond to a common line passing through the points on plane X-Y. Thus, the straight line passing through the sequence of points is obtained by calculating the intersection of the curves on plane ρ - θ .

Defining the intersection on plane ρ - θ as (ρ_0, θ_0) , then the straight line can be represented thus:

$$x \cdot \cos \theta_0 + y \cdot \sin \theta_0 = \rho_0$$

Next, the steps for implementing Hough transformation on a computer are given.

(Step 1)

A two-dimensional array corresponding to plane ρ - θ is

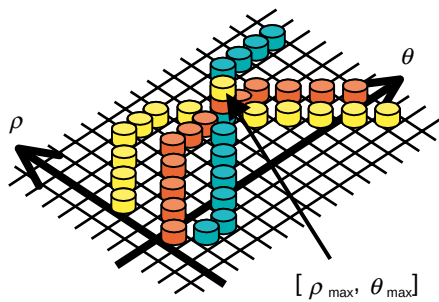


Fig.8 Implementation of Hough transformation

prepared (Fig.8). Each element functions as a counter.

(Step 2)

For each point on the image (x, y) , ρ is calculated for a value of θ from 0 [degrees] to 180 [degrees] so that the counter value of the array element referenced by $[\rho, \theta]$ is increased by 1 (i.e., voting).

(Step 3)

The above process (Step 2) is carried out for all edge points.

(Step 4)

The two-dimensional array with the maximum count value is selected. Defining the index of this array as $[\rho_{\max}, \theta_{\max}]$, then in the plane X-Y, this straight line is such that

$$\rho_{\max} = x \cdot \cos \theta_{\max} + y \cdot \sin \theta_{\max}$$

For instance, if an interval of 1 [degree] is used for θ , then equation (1) must be solved 180 times for each point in the image. This process, because it involves the repetition of simple calculations, is well suited to computerization.

The important requirements for Hough transformation can be summarized thus:

- (1) A linear sequence of points on image plane x-y correspond to a local peak in Hough space ρ - θ (Fig.9).
- (2) The height of a local peak in a Hough space corresponds to the number of points in the corresponding linear sequence of points (Fig.9).

Another characteristic advantage of Hough transformation is its ability to extract a straight line indicating an edge's general direction, without dealing excessively with the edge's intricate jaggedness. Conversely, its disadvantage is that when edge points are locally concentrated, as with a double broken line, numerous extraneous lines other than the one desired are also extracted.

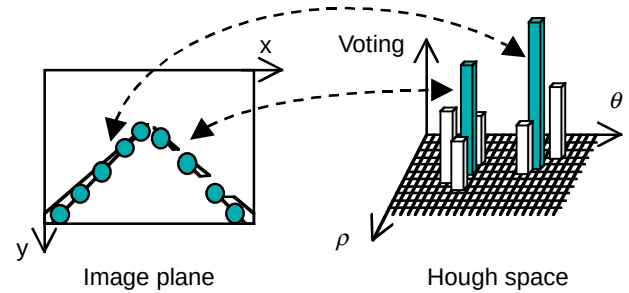


Fig.9 Hough transformation

3.3. Characteristics of the Lane Mark Recognition Technique of the Honda Intelligent Driver Support System

This system, by combining the basic principles explained in section 3.2, has achieved a lane mark recognition technique that can handle double broken lines and similarly complex images and which is resistant to the effects of noise. This section describes the characteristics of that technique.

The first technique is:

- (1) The ability to handle double broken lines and similarly complex images

As Fig.10 shows, edge extraction from a double broken line image results in a highly dense concentration of points

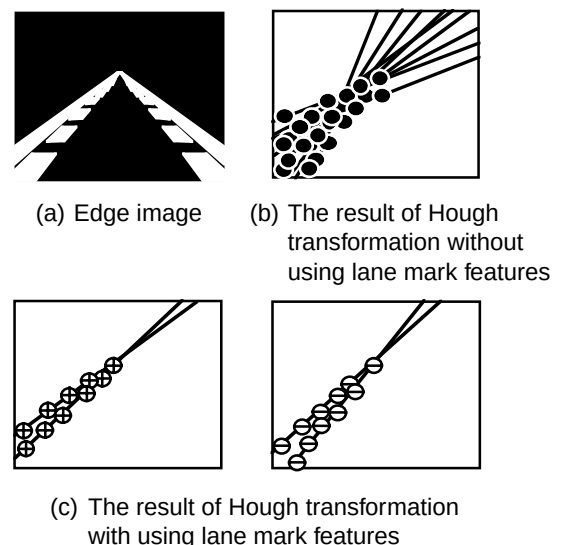


Fig.10 Features of the a lane mark recognition algorithm (1)

being extracted. In such cases, Hough transformation would normally result in line segments in many directions being extracted, obscuring the desired lane mark (Fig.10-b). Therefore, line segments in these extraneous directions are excluded during Hough transformation by utilizing the lane mark characteristics described in Fig.5 in section 3.2 (Fig.10-c). This makes it possible to recognize even a lane mark as complex as a double broken line with good probability.

The second characteristic of this technique is:

- (2) The ability to yield sequences of points, not linear parameters, as the results of recognition

Hough transformation normally yields linear parameters, which, when used as the results of recognition without modification in the processing of a gently curving lane mark like that shown in Fig.11-a, result in a considerable mismatch between the resultant line and the actual lane mark position, preventing sufficient accuracy in lane mark position (Fig.11-b). One way of avoiding this is to raise the angular resolution of Hough transformation. This approach, although permitting accurate representation of lane mark position by making it possible to represent curves with multiple line segments, is not practical because of the huge volume of Hough transformation calculations required. For this system, it was therefore decided to output not the Hough peak itself that is extracted as the candidate lane mark, but rather the sequences of points that comprise the Hough peak (Fig.11-c). In the next step, which is data assessment, these results are statistically processed along with past data, yielding a

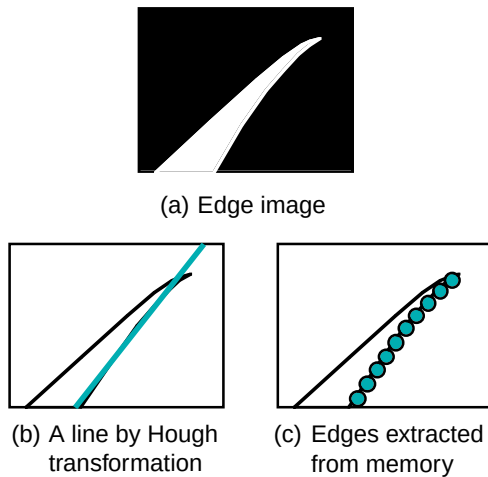


Fig.11 Features of the a lane mark recognition algorithm (2)

more-accurate lane mark position.

The third characteristic of this technique is:

- (3) Ease of hardware customization

Hough transformation, which in this technique accounts for the largest portion of the calculations, involves the repeated calculation of equation (1) in section 3.2 and so is conducive to hardware customization. In actuality, this system uses customized hardware for these calculations to achieve speeds nearly 100 times faster than software

processing with system CPUs. This makes it possible to process an entire image with limited hardware resources, and to detect the desired lane mark among multiple candidate sequences of lines, even in cases of turnouts and similarly confusing lane marks.

4. Data Assessment

The process up to section 3 yields segmented sequences of points as the candidate lane marks. These results are then used to:

- (1) identify the true lane mark and supplement lane marks that had not been extracted; and
- (2) estimate the true position of the lane mark

An overview of this process is given in Fig.12. First, the sequences of points for each lane mark segment are projectively transformed from the image coordinate system to a real world coordinate system. This projective transformation uses parameters identified in a stationary state, and so at this point noise caused by road gradient and vehicle pitch and roll changes, etc., enter. Next, the geometric interrelationship between the segments on the real world coordinate system is determined and represented with a graph and matched with a road shape model similarly expressed as a graph. This yields a candidate set for the current road shape while in motion. At this stage, the misextracted lines and omitted lines in the matched models are identified to calculate assessment values that indicate agreement between the models, as well as relative distance between the vehicle and the lane marks, among others. In the next stage, a map is drafted after collecting these candidates for a certain short period of time assuming that the lane marks flow smoothly, with no change in road structure while the vehicle is in motion during this short period. Next, candidate lists of current road shape and maps are generated by categorizing and arranging under the assumption that the lane marks and the vehicle's position in the lane continue smoothly. Lastly, each candidate list is compared with expressway design standards to determine the appropriateness of the road width and curvature change rate, etc. Once the optimum list is selected, least squares approximation is used to identify the true position of the lane mark.

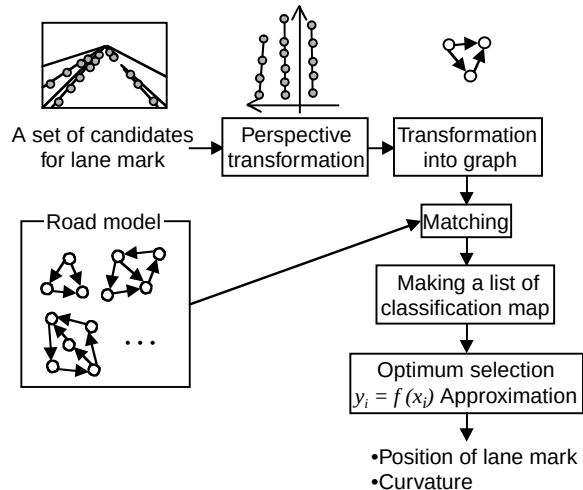


Fig.12 Flowchart of estimating data

4.1. Geometric Representation of Road Shape

The set of candidate lane marks is matched with the road shape model in order to select the true lane mark. Normally, the graph is represented thus, using node V and arc E , which represents the properties between nodes:

$$G = (V, E)$$

V has attributes and attribute value P , which in this system is defined thus:

$$P = (p, \text{value}), p \in \{\text{solid line, broken line, unclear}\}$$

“value” is the value of similitude to p

$$E = (e, \text{value}), e \in \{\text{parallel, serial, unrelated}\}$$

“value” is the value of similitude to e

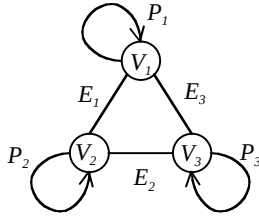


Fig.13 Example of graph

If interrelationship E is such that $E_{ij} = E_{ji}$, then graph G is a non-oriented graph. If $E_{ij} \neq E_{ji}$, then it is an oriented graph. In graph matching problems, ordinarily there is no significant difference between oriented and non-oriented graphs, which can be used according to the type of problem in question. This system symbolizes and defines E so that an absence of any relationship (“unrelated”) is represented with a non-oriented graph, a “parallel” relationship with an oriented graph whose orientation depends on the vehicle’s direction of travel, i.e., with the two in a mirror-image relationship, preventing a match with geometrically similar structures and making it possible to interpret solutions uniformly.

An example road shape model is given in Fig.14. This model’s definition is designed to be comprised of link information from map data of the current area provided by the navigation system.

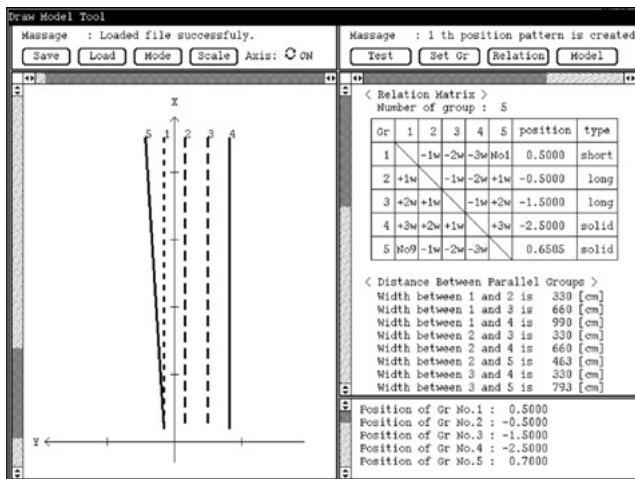


Fig.14 Sample of definition of road model

4.2. Matching Principle and Algorithm

Matching of candidate lane mark set graph G_1 and model graph G_2 can be treated as a problem of finding the portions of the two graphs that both have the same shape. Thus, one has only to create a combined graph from the combinations of nodes of G_1 and G_2 and then use it to solve the problem of extracting clique N . Here, no detailed description is given of the combined graph creation procedure, the N clique extraction algorithm⁽³⁾, or the method for calculating the assessment values. However, during creation of the combined graph, the matching process is sped up by omitting V_1 , V_2 and other combinations that lie outside of the lane on the assumption that the vehicle is located on the inside of the road division.

4.3. Estimating Lane Mark Position and Curvature

N number of lane marks on the real world coordinate system are expressed with the quadratic function $f = ax_i^2 + bx_i + c_i$ ($i=1 \sim N$), and coefficients a , b , c_i are calculated with the method of least squares. In other words, given that e_i is the error between each function and the original sequence of points (x_{ij}, y_{ij}) , then a , b , c_j are solved to minimize $e = \sum e_i$. This is done by solving simultaneous linear equation with $N+2$ variables.

$$\begin{pmatrix} \sum_{i=1}^N \sum_{j=1}^{n_i} x_{ij}^4 & \sum_{i=1}^N \sum_{j=1}^{n_i} x_{ij}^3 & \sum_{i=1}^N \sum_{j=1}^{n_i} x_{ij}^2 & \sum_{i=1}^N \sum_{j=1}^{n_i} x_{ij}^2 & \dots & \sum_{i=1}^N \sum_{j=1}^{n_i} x_{ij}^2 \\ \sum_{i=1}^N \sum_{j=1}^{n_i} x_{ij}^3 & \sum_{i=1}^N \sum_{j=1}^{n_i} x_{ij}^2 & \sum_{i=1}^N \sum_{j=1}^{n_i} x_{ij} & \sum_{i=1}^N \sum_{j=1}^{n_i} x_{ij} & \dots & \sum_{i=1}^N \sum_{j=1}^{n_i} x_{ij} \\ \sum_{i=1}^N \sum_{j=1}^{n_i} x_{ij}^2 & \sum_{i=1}^N \sum_{j=1}^{n_i} x_{ij} & \sum_{i=1}^N \sum_{j=1}^{n_i} 1 & 0 & \dots & 0 \\ \sum_{i=1}^N \sum_{j=1}^{n_i} x_{ij}^2 & \sum_{i=1}^N \sum_{j=1}^{n_i} x_{ij} & \sum_{i=1}^N \sum_{j=1}^{n_i} 1 & 0 & \dots & 0 \\ \vdots & \vdots & \vdots & \vdots & \ddots & \vdots \\ \sum_{i=1}^N \sum_{j=1}^{n_i} x_{ij}^2 & \sum_{i=1}^N \sum_{j=1}^{n_i} x_{ij} & \sum_{i=1}^N \sum_{j=1}^{n_i} 1 & 0 & \dots & \sum_{i=1}^N \sum_{j=1}^{n_i} 1 \end{pmatrix} \begin{pmatrix} a \\ b \\ c_1 \\ c_2 \\ \vdots \\ c_N \end{pmatrix} = \begin{pmatrix} \sum_{i=1}^N \sum_{j=1}^{n_i} x_{ij}^2 y_{ij} \\ \sum_{i=1}^N \sum_{j=1}^{n_i} x_{ij} y_{ij} \\ \sum_{i=1}^N \sum_{j=1}^{n_i} y_{ij} \\ \sum_{i=1}^N \sum_{j=1}^{n_i} y_{2j} \\ \vdots \\ \sum_{i=1}^N \sum_{j=1}^{n_i} y_{Nj} \end{pmatrix}$$

Here, each sequence of points (x_{ij}, y_{ij}) must be transformed to a simultaneous coordinate system using information about the vehicle’s path of motion (x, y, θ) . To reduce the amount of memory required and the cost of coordinate transformation calculations, this system compresses the point sequence coordinates into the shape of the coefficient matrix of normal matrix $A_i X=B_i$ for each lane mark, then performs time correction for each of these coefficient matrices. Fig.15, which shows the sequences of points on the map list and the estimated approximation curves (the coordinate origin in the illustration is the vehicle), demonstrates how this method of estimation has accurately approximated the lane marks, although lane marks far from the vehicle are subject to the effects of vehicle movement and other noise components.

5. Hardware Configuration

Fig.16 shows the LKAS ECU configuration. This ECU comprises three major parts: the image preprocessing unit, image processing CPU, and control CPU. Customized hardware in the image preprocessing unit performs positive and negative edge extraction and Hough transformation of the road images taken by the camera. This is the processing described in section 3.2. In the image processing CPU, using

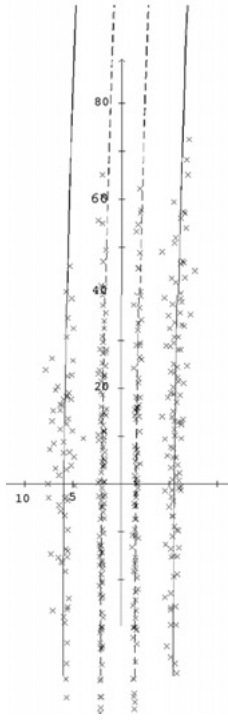


Fig.15 Points in map list and approximation functions

the results of processing by the image preprocessing unit, software narrows down the candidate lane marks, reconstructs the lane mark point sequences, and assesses the data as described in section 4. Lastly, the image processing CPU calculates the road position information and road curvature and sends them to the control CPU.

6. Results of Experimentation

6.1. Land Mark Recognition Rate

This system was used with actual road images to determine its lane mark recognition rate. Using a set of road images taken in various conditions on expressways, the experimenters judged the results of calculations as successful if they met the following criteria:

[Criteria] Edges of the target lane mark were detected over a distance of at least 50 m ahead of the vehicle, with a quantity and distribution that are sufficient for control purposes.

This set did not contain images in which lane mark recognition was completely impossible, e.g., due to road construction or backlight. The results are given in Table 1. The results for some of the images in the set are shown in Fig.17.

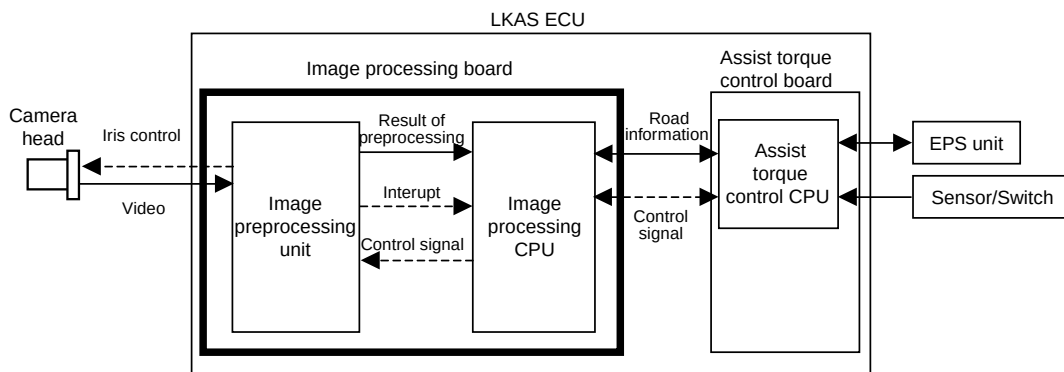


Fig.16 Architecture of LKAS ECU

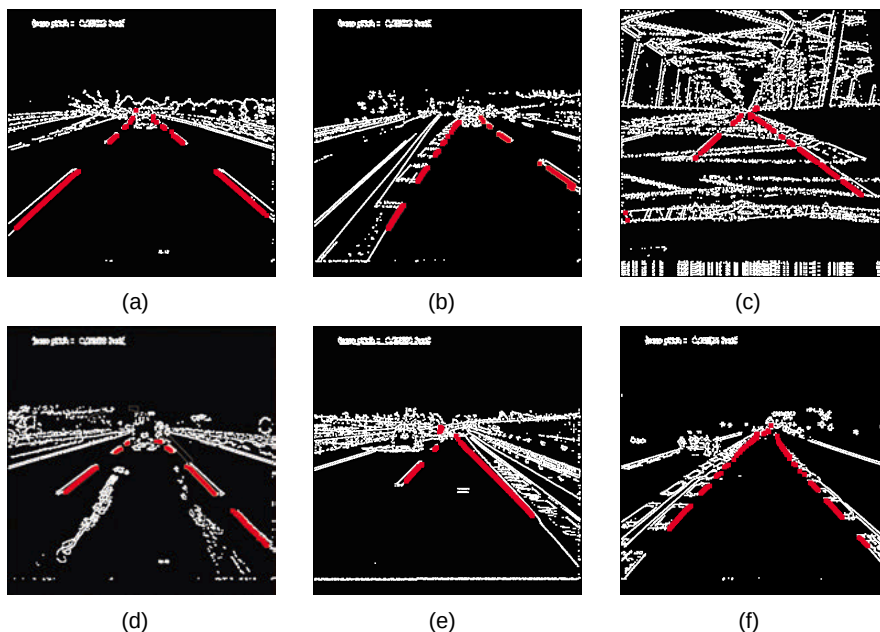


Fig.17 The result of lane mark recognition

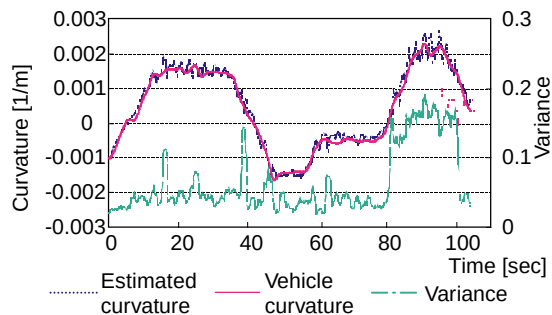
Table 1 Result of lane mark recognition

Number of image samples	Number of success	Recognition rate
293	274	94%

LKAS, which assumes steering by the driver, requires decision-making by the driver in all circumstances and gives priority to operations by the driver, and so is capable of assuring proper performance in this image processing.

6.2. Lane Mark Position Accuracy

Road images videotaped on an actual expressway along with vehicle motion data were computer-processed using this system; the results are in Fig.18. The data are the road curvatures and their distribution as determined by image processing, along with road curvatures calculated offline from vehicle path information. The section traveled was a winding section of the Tohoku Expressway, between the Shirakawa and Nasu Kogen rest areas. In the illustration, a double broken line section starting at the 80-second mark has resulted in an abrupt worsening of distribution, but experimentation that recreated these conditions on a test course verified assist control to be possible at this level without causing the driver to experience any incongruity.



(Tohoku expressway • Nasukougen PA to Shirakawa PA)

Fig.18 Estimated curvature and variance

7. Conclusions

The following conclusions were reached concerning the development of a new lane mark recognition algorithm that combines Hough transformation with extraction of positive and negative edges, which are used as characteristic of lane marks:

- (1) Eliminating extraneous linear components (noise) and using customized computer chips to repeat simple calculations speeds up processing and made possible the development of a lane mark recognition system that can recognize, in real time, double broken line lane marks and lane marks in turnouts and other complex situations on actual expressways.
- (2) Testing of the system on images of an actual expressway yielded a high rate of recognition and showed position accuracy to also be of a level sufficient for steering control.

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Algorithm for Crash Judgment by Frequency Analysis

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ABSTRACT

A crash judgment algorithm that uses frequency analysis has been developed. This algorithm judges whether or not a crash has occurred by calculating both energy elements and frequency elements. Frequency analysis is implemented in hardware at the current microprocessor level using a Hadamard transform that does not require multiplication or division in the calculation of frequency elements. This approach improves the accuracy of crash judgments and reduces development time.

1. Introduction

Heightened awareness in recent years of the importance of passive safety has led to the inclusion of crash tests in domestic safety standards. Overseas, no longer are crash tests limited to the conventional full frontal barrier tests, but offset crashes, as typified by Offset Deformable Barrier (ODB) tests are also being implemented. Fig.1 gives examples of the most common crash tests and the corresponding

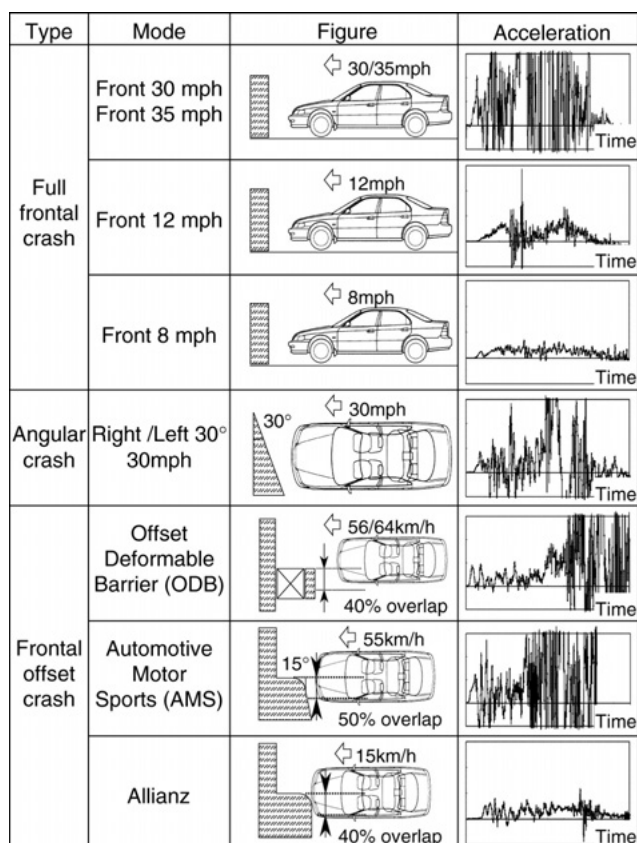


Fig.1 Common crash modes and acceleration wave forms

acceleration waveforms.

There are two important requirements for protecting occupants in the event of a crash including the following:

- The body crash performance needs to absorb the energy of the crash;
- The body crash performance needs to maintain survival space for occupants; and
- The occupant restraint system performance needs to minimize injury by helping prevent secondary impacts between occupants and interior components.

Crash judgment algorithms control the operation timing of certain components of such occupant restraint systems, namely, airbags and seatbelt pretensioners, including the turning on and off of these components.

The authors have developed a new crash judgment algorithm that incorporates a new frequency analysis technique. This paper discusses this algorithm and how it makes crash judgment easier than before.

2. Crash Judgment Algorithm

2.1. Overview of Electronic SRS

Early SRS placed a mechanical sensor in the vehicle front and a separate mechanical sensor in a unit located in the

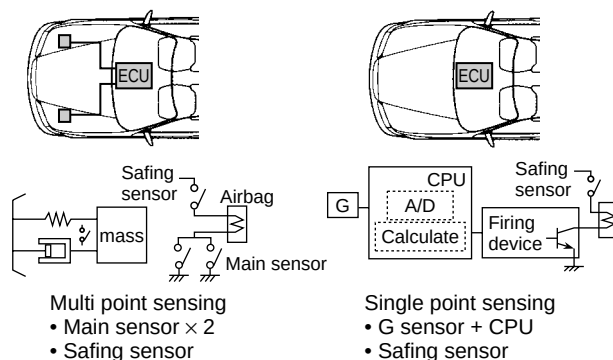


Fig.2 Mechanical/Electrical SRS

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vehicle center (Fig.2, left side). With these sensors serving as electrical contacts, the system was designed so that when both sensors turned on simultaneously, electricity would be delivered to the airbag inflators, inflating the airbags with gas. This type of system is called a mechanical multipoint sensing system.

In the summer of 1993, a different type of system was released: electronic single point sensing (Fig.2, right side). In this system, the output from an acceleration sensor inside the unit (again located in the vehicle center) is processed by a CPU. When the results of CPU calculation indicate that a crash has occurred, the unit turns on the firing circuit.

It is the crash judgment algorithm that processes acceleration sensor output and determines whether a crash has occurred. Because acceleration differs from one car model to another, the threshold value used to judge whether a crash has occurred also differs from one car model to another. "Setting" is the process of determining the right threshold value for a given car model.

2.2. The Difficulty of Crash Judgment

There are two major difficulties in crash judgment:

- There are ON requirements and OFF requirements, each of which demands high reliability.
- ON requirements include a time requirement called TTF, discussed below.

An airbag must not deploy during normal driving (even when driving on rough roads) or during light crashes at speeds of approximately 8 mph BEV in which occupant injury would be minimal even without airbag deployment. Here BEV means Barrier Equivalent Velocity. On the other hand, an airbag can not fail to deploy in a crash in which deployment is imperative. The performance targets of deployment ("ON") and non-deployment ("OFF") require a

high level of reliability.

The performance target of deployment uses a predetermined time duration from crash start until the deployment signal is sent to the airbags; this is called Time to Fire (TTF). Ordinarily, TTF is set so that an airbag deploys at the point where the occupant has been displaced 5 inches. Since an airbag usually requires 30 ms to deploy, TTF is equal to [time required for the occupant to be displaced 5 inches] - 30 ms. In the case of a full frontal high-speed crash, this value is between 8 and 15 ms, which is the time before acceleration rises to the high levels that characterize high-speed crashes. Fig.3 gives the acceleration, velocity, and displacement of a high-speed crash. As this example illustrates, in judging a high-speed crash, it is necessary to predict whether this will be a major crash requiring airbag deployment, i.e., whether high-speed acceleration will occur, before sufficient crash acceleration has been generated. This is the difficulty of crash judgment.

2.3. Overview of Conventional Crash Judgment Algorithms

In Fig.4, an example of a conventional crash judgment algorithm, calculations are based on the acceleration differential, i.e., the rate of change in acceleration. Normally, other data are also used in the calculations, such as the acceleration integral, which indicates change in velocity, and the double integral of acceleration, which indicates displacement. According to the technique employed in this example, table processing is used during acceleration waveform computations. This allows for calculations to assess only a predetermined range of data. In cases of such processing, information beyond this range is sometimes absent from the body of data, the result being that sufficient extraction of acceleration waveform features is not possible.

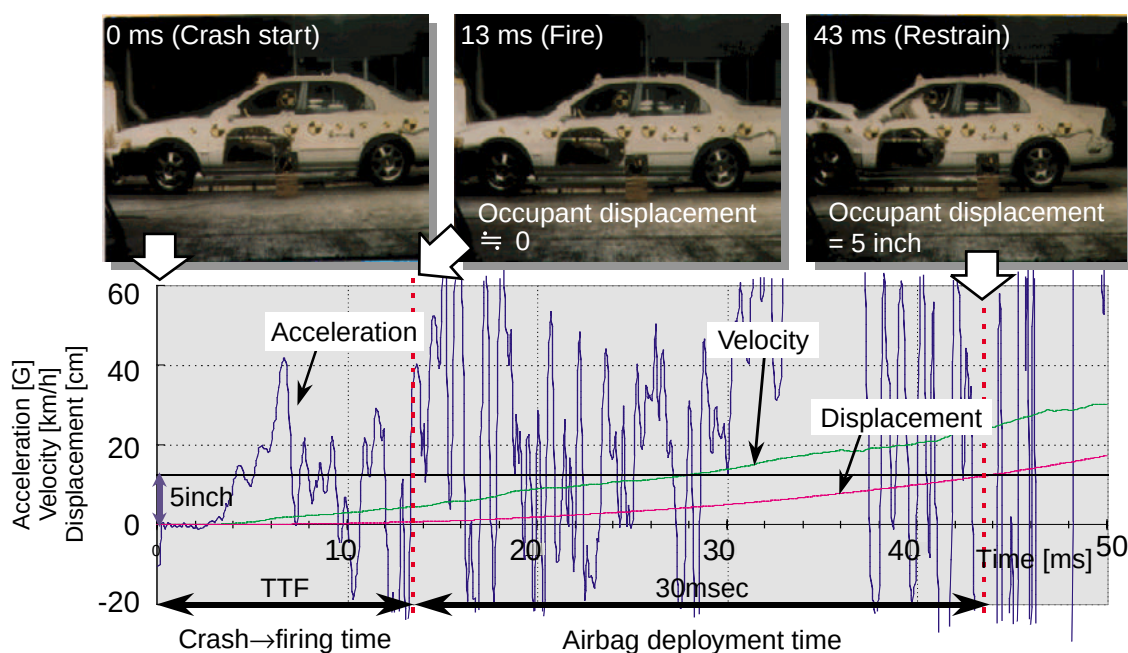


Fig.3 High speed crash

3. Overview of the New Crash Judgment Algorithm

3.1. The Validity of Crash Judgment by Frequency Analysis

The authors have employed frequency analysis to extract acceleration waveform characteristics. This method is already widely used for feature extraction for vibration, as in voice recognition and mode analysis. Fig.5 gives an example of the result of a type of analysis termed Short Time Fourier Transform (STFT)^{(1),(2)}, which yields a time variation showing fluctuation in the frequency element of the

acceleration waveform.

The results of this analysis show that in order to make a crash judgment based on frequency analysis, a mere frequency element is insufficient. Rather, as indicated in Fig.6, it is necessary to calculate the product of the frequency element data and the energy at that point. This can also be surmised from the mechanism in which, during a crash, energy accumulation (whereby stress first accumulates in a member) is followed by generation of high frequency elements, whereby stress is discharged through collapse. A simplified model of this process is shown in Fig.7.

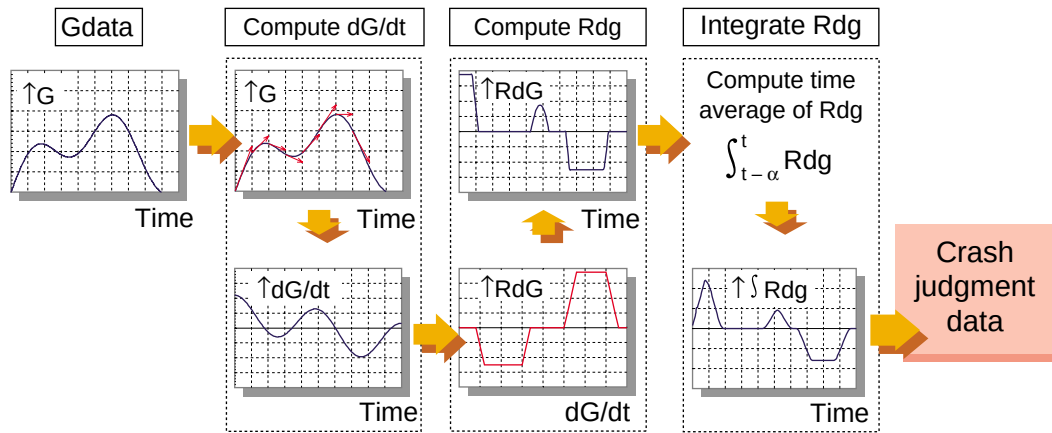


Fig.4 One of previous crash algorithm

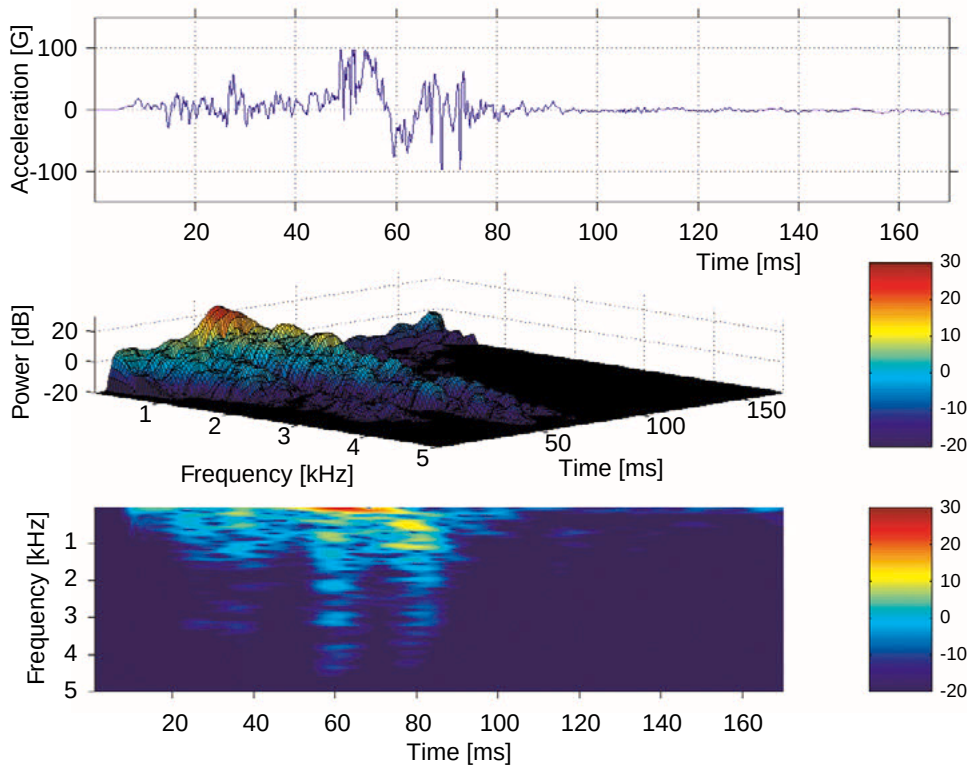


Fig.5 STFT analysis

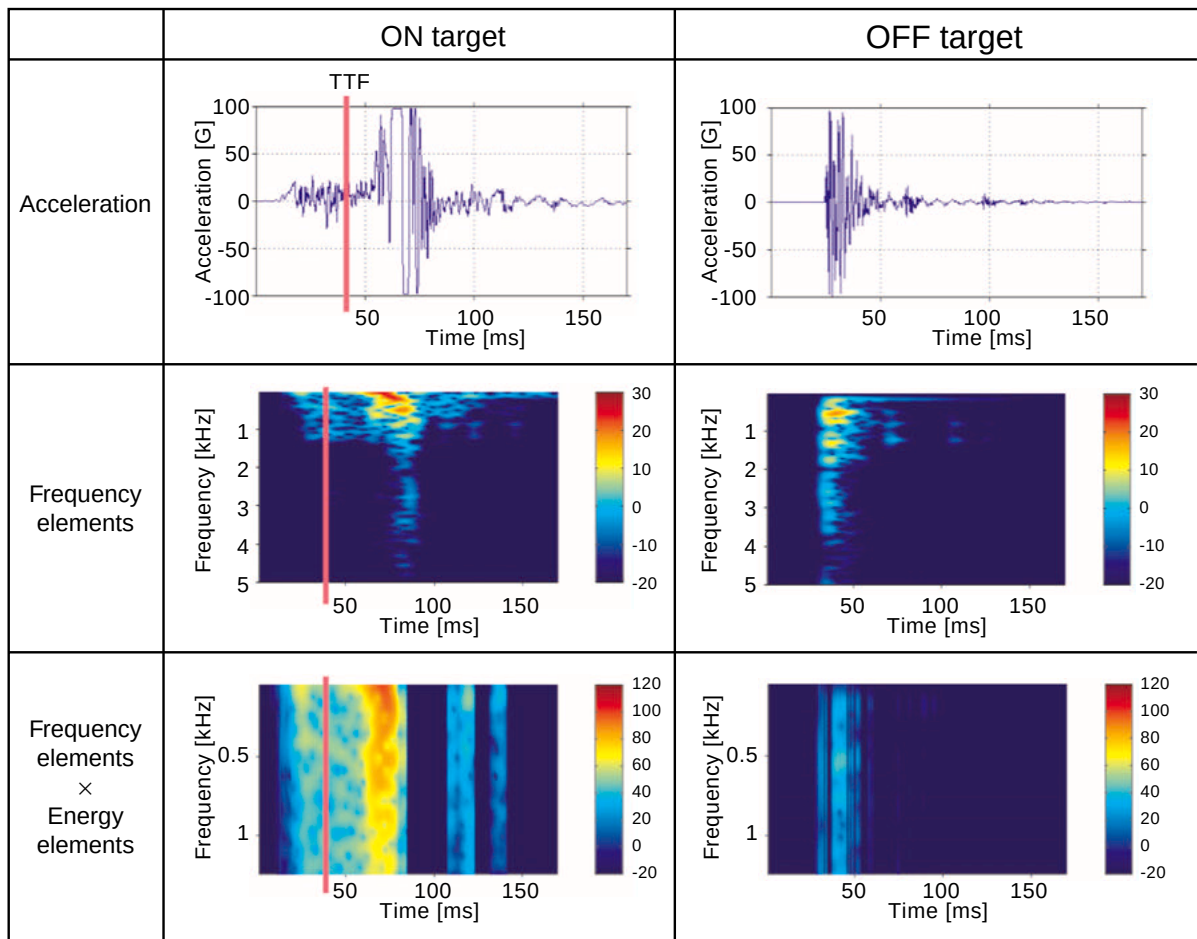


Fig.6 Frequency elements and Energy elements

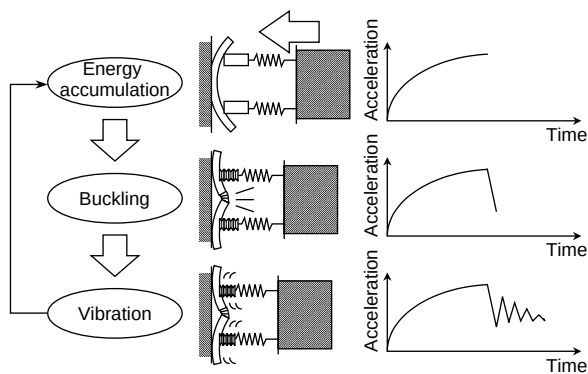


Fig.7 Model of generating a crash acceleration

3.2. Realization in ECU: Hadamard Transform

Fourier transform, used in the analysis thus far, provides a high accuracy of frequency decomposition, but is difficult to apply to current sensor ECU hardware because of the large volume of real-number multiplication and division operations. The authors have therefore selected Hadamard transform⁽³⁾, which although having a lower accuracy of

frequency decomposition, makes it possible to perform frequency decomposition with addition and subtraction alone. This reduces the volume of calculations to 1.9% in terms of time ratio.

Hadamard transform is a method of frequency analysis that is characterized by the following.

- Use of a matrix comprising only the (1, -1) transformation matrix (the only computations are addition and subtraction)
- The original waveform is expressed as an orthogonal, square wavegroup (Fourier transform in the broad sense)

This linear first order transformation can be expressed with the following equation.

$$\begin{pmatrix} Y_0 \\ Y_1 \\ \vdots \\ Y_{n-1} \end{pmatrix} = \begin{pmatrix} H \\ n \times n \end{pmatrix} \begin{pmatrix} X_0 \\ X_1 \\ \vdots \\ X_{n-1} \end{pmatrix}$$

Transformed data
Transformation matrix
Data before transformation

As shown below, a second-order basic matrix is used to generate the other transformation matrices recursively, thus: 2nd order $\rightarrow$ 4th order, 4th order $\rightarrow$ 8th order, ..., nth order $\rightarrow$ 2nth order.

$$\text{2nd order transformation matrix: } H_2 = \begin{pmatrix} 1 & 1 \\ 1 & -1 \end{pmatrix}$$

$$\text{4th order transformation matrix: } H_4 = \begin{pmatrix} H_2 & H_2 \\ H_2 & -H_2 \end{pmatrix} = \begin{pmatrix} 1 & 1 & 1 & 1 \\ 1 & -1 & 1 & -1 \\ 1 & 1 & -1 & -1 \\ 1 & -1 & -1 & 1 \end{pmatrix}$$

$$\text{8th order transformation matrix: } H_8 = \begin{pmatrix} H_4 & H_4 \\ H_4 & -H_4 \end{pmatrix} = \begin{pmatrix} 1 & 1 & 1 & 1 & 1 & 1 & 1 & 1 \\ 1 & -1 & 1 & -1 & 1 & -1 & 1 & -1 \\ 1 & 1 & -1 & -1 & 1 & 1 & -1 & -1 \\ 1 & -1 & -1 & 1 & 1 & -1 & -1 & 1 \\ 1 & 1 & 1 & 1 & -1 & -1 & -1 & -1 \\ 1 & -1 & 1 & -1 & -1 & 1 & -1 & 1 \\ 1 & 1 & -1 & -1 & -1 & 1 & 1 & 1 \\ 1 & -1 & -1 & 1 & -1 & 1 & 1 & -1 \end{pmatrix}$$

$$\text{nth order transformation matrix: } H_n = \begin{pmatrix} H_{n/2} & H_{n/2} \\ H_{n/2} & -H_{n/2} \end{pmatrix}$$

Fig.8 gives the fundamental function for Hadamard transform and the resultant frequency characteristics for 16th order, 1 ms sampling. Although the effects of error due to square wave decomposition do exist, the basic frequency spectrum has been decomposed in keeping with theory. Here, SQ refers to sequency, an index in which the order of

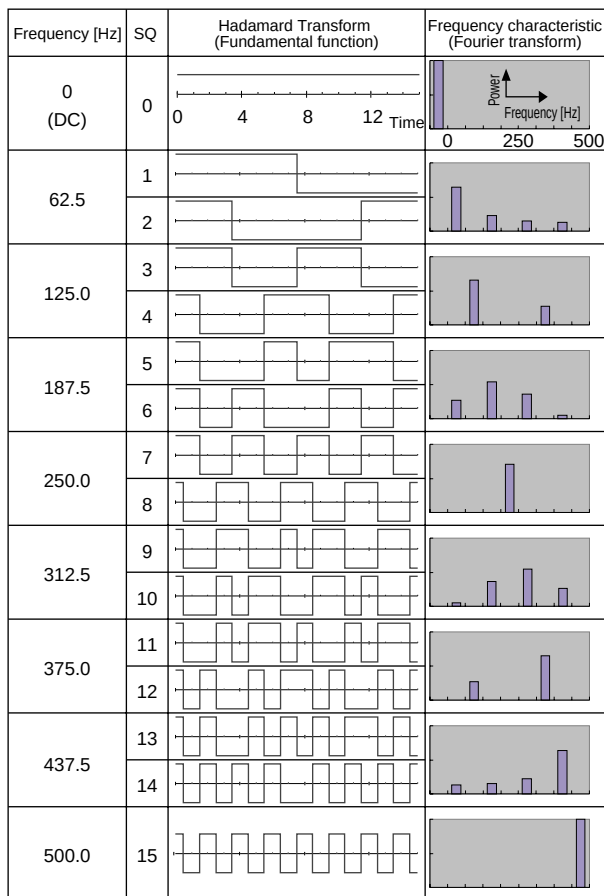


Fig.8 Frequency characteristic of Hadamard transform

Hadamard transform corresponds to frequency.

In the example application in an actual crash waveform in Fig.9, 16th-order Hadamard transform is used, generating 16 types of data for each frequency from one piece of acceleration data. The fact that the peak is at SQ4 indicates that that is where the correlation between the original acceleration waveform and the shape of the SQ4 fundamental function is the closest. In short, each of the 16 types of data that have been extracted from the original acceleration data for each frequency element represents the degree of correlation between that frequency and the fundamental function.

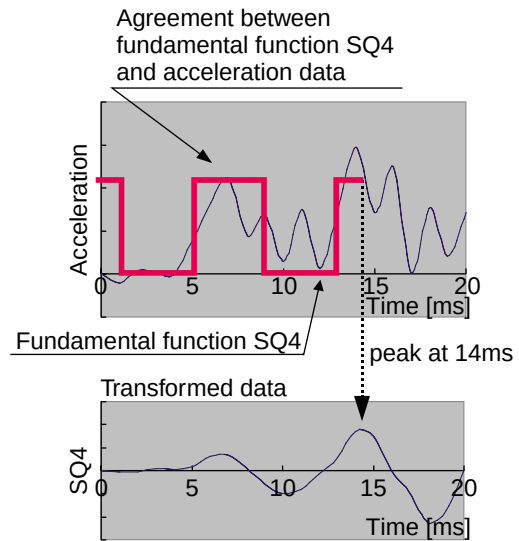
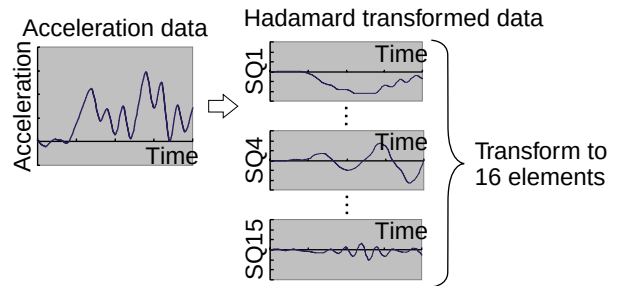


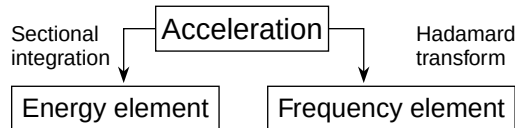
Fig.9 General idea of frequency analysis

3.3. Overview of the Algorithm

Using the results of analysis up to this point, the new algorithm decomposes the acceleration signal into energy elements and frequency elements. If specific energy elements and frequency elements (i.e., the Hadamard transform data) exceed the predetermined threshold value, it then judges that a crash has occurred.

The simplified block diagram and judgment flowchart in Fig.10 demonstrate how the threshold value for the frequency element is variable, changing in steps according to energy element output.

Block diagram



Judgment flow chart

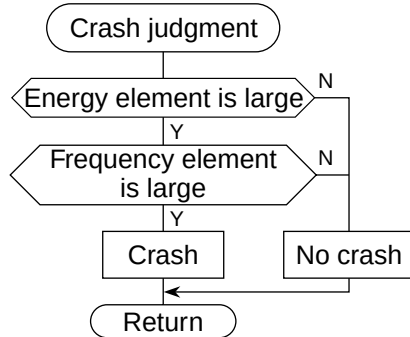


Fig.10 Concept of the crash algorithm

3.4. The Verification Process

Because this algorithm uses frequency elements as the basis of crash judgment, it is crucial to verify toughness (i.e., robustness) with respect to variation in frequency elements. This is because, in a real-world crash outside of the crash mode for which the algorithm was set, the frequency elements generated during the crash could vary and prevent the predetermined requirements from being met. Conventional algorithms, which base their calculations essentially on velocity change, have been empirically shown to maintain a relatively proportional relationship even in such situations. For this reason, the authors tested the suitability of this algorithm with respect to the following parameters.

- The effects of resonance
- Crash velocity and mode
- Frequency variation

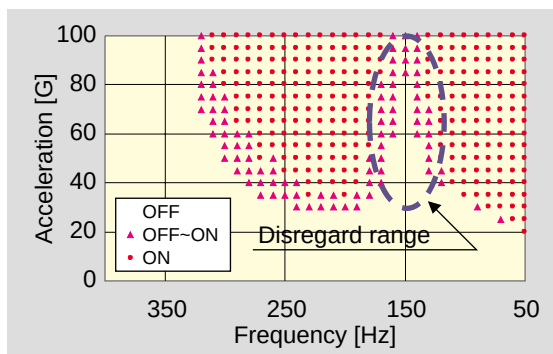
As for validation of robustness against resonance, the authors used crash waveforms from past measurements to verify that this algorithm permits easy setting even for crash waveforms where the resonance of the ECU mounting bracket was sufficiently large to warrant investigation. Fig.11 shows the ON characteristics by frequency for a conventional algorithm and the new algorithm. This shows how the new algorithm makes it possible to avoid bracket resonance by using a setting that disregards a specific frequency element, which would be difficult to distinguish with a conventional algorithm.

For validation of robustness with respect to crash velocity and mode, barrier testing simulating real-world crashes at non-regulated velocities was implemented. This was done with two crash modes, in which the suitability of airbag deployment was verified. The results are summarized in Table 1.

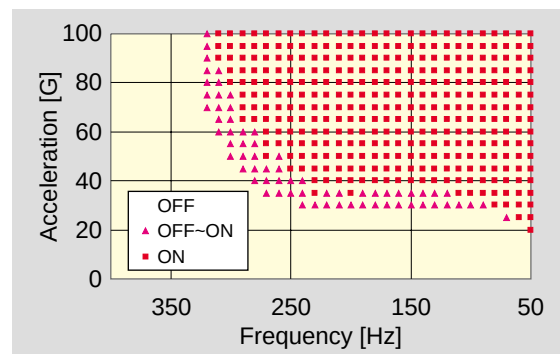
To verify the algorithm's robustness against the effects of frequency variation, the frequency elements of actual crash waveforms were varied using a random number generator program. Fig.12 shows the method of generation: The original acceleration waveform is decomposed into frequency elements, the amplitude of which is then changed with random numbers to produce variations that are inversely transformed to determine the waveform's usability under certain restraint conditions.

Table 1 Non-regulation crash modes

Mode	Figure	Target	Result	Judgement
Full frontal crash 24mph		15.8	14	○
Frontal offset pole crash 24mph		43.3	34	○



New algorithm



Previous algorithm

Fig.11 Disregarding a frequency range

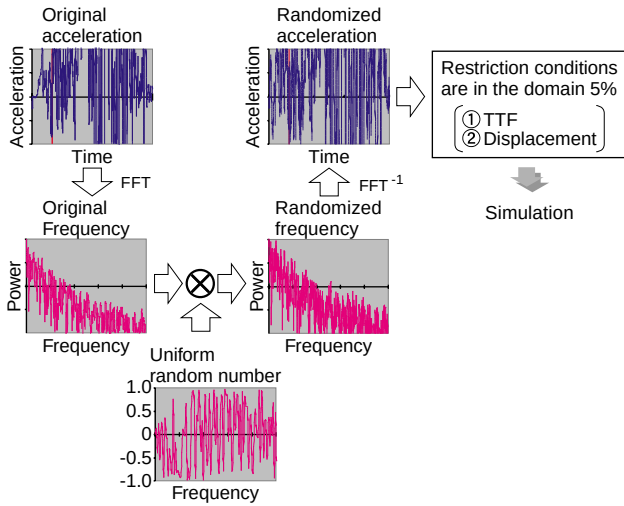


Fig.12 Changing frequency element

4. Validation of Effectiveness

Let us consider an example of the improved effectiveness of this algorithm in comparison with that of conventional algorithms.

In Fig.13, the feature with which the conventional algorithm meets only the OFF target are shown as “property 1.” Here, the issue is maintaining the ON target. Conventional algorithm property 2 is the opposite of 1 in that it meets only the ON target, the issue being the maintenance of the OFF target. Each line represents the velocity change of each crash waveform.

In Fig.14, the new algorithm was used for these waveforms. Note how the new algorithm easily meets both the ON and OFF targets.

In addition, the algorithm’s simplified structure has made it possible to create a tool for automatically searching for

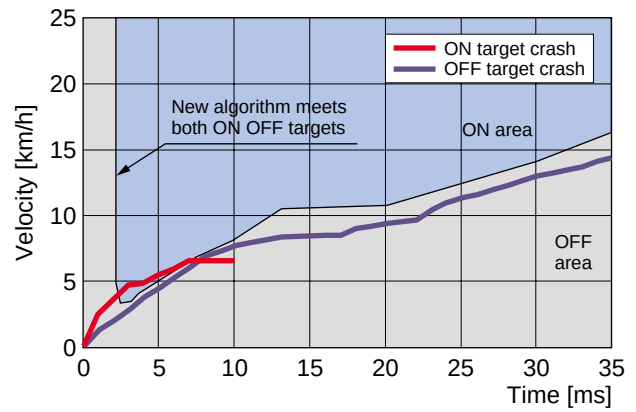


Fig.14 Characteristic of new algorithm

correlation between frequency elements and energy elements. This tool has greatly reduced the duration of setting by making it possible to approximate the setting semi-automatically.

5. Conclusion

By applying the Hadamard transform, the authors have created a crash judgment algorithm that employs frequency analysis at the level of microprocessors that are mountable in current sensor ECU hardware. Moreover, the validity of this algorithm was demonstrated.

This algorithm improves the accuracy of crash judgment and greatly reduces the amount of time required for setting.

The authors plan to make additional improvements to further enhance the performance and setting efficiency of the algorithm, thereby expanding the range of applicable vehicle models.

Acknowledgement

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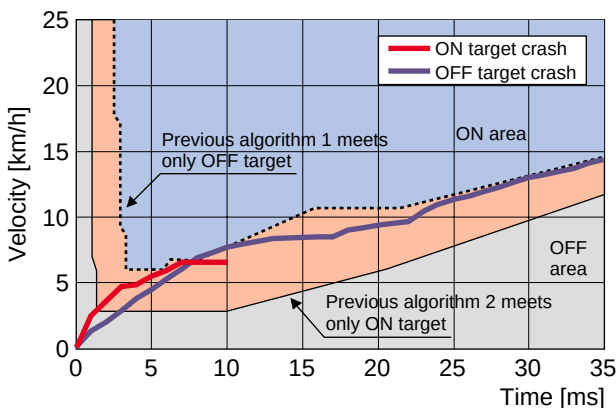


Fig.13 Characteristic of previous algorithm

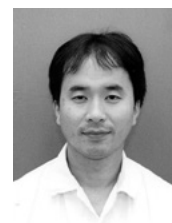
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An Application of Occupant Deceleration Control by the Crash Load Control Structure to an Actual Vehicle

Yasuki MOTOZAWA* Takahiro KAMEI*

ABSTRACT

In order to reduce the maximum value of occupant deceleration in a vehicle crash, a body structure was devised to approximate the optimal vehicle deceleration wave form by utilizing the difference of the deformation mode of the body. This concept of structure was successfully applied to an actual vehicle.

1. Introduction

When considering vehicle crash safety performance, optimizing the change in vehicle deceleration over time in a crash (hereafter referred to as the “vehicle deceleration wave form”) is an important issue for simultaneously securing survival space and reducing injury severity. Previous research mainly performed optimization using the sensitivity analysis method⁽¹⁾⁽²⁾, but the authors have derived and generalized mathematical solutions for the optimal vehicle deceleration wave forms for chest deceleration of the occupant through mathematical studies using a two-mass model⁽³⁾. These mathematical solutions have in common a three-part structure consisting of high initial deceleration, low or negative middle deceleration, and then constant deceleration later on.

However, vehicle deceleration is generally determined by the vehicle structure deformation load in a crash which is applied to the vehicle mass. This deformation load increases with the amount of vehicle deformation, which makes it difficult to reduce the deceleration during the middle period of a crash compared to that at the start of a crash. Therefore, the authors devised a fundamental model for approximating the optimal vehicle deceleration wave form by reducing the crash load during the middle period of a crash utilizing the difference in load characteristics of the deformation mode. In addition, application of this model to an actual vehicle structure was studied.

2. Mathematical Solutions for the Optimum Vehicle Deceleration Wave Form

Previously the authors had mathematically studied the vehicle deceleration wave form that minimizes the maximum occupant deceleration value using a two-mass model like that shown in Fig.1. Here, the authors introduced a transient function concept and obtained mathematical solutions for the optimal vehicle deceleration wave form⁽³⁾.

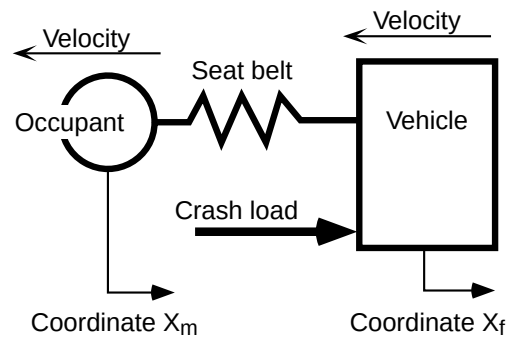


Fig.1 One-dimensional two-mass model

The transient function is the optimal occupant deceleration wave form that results in a complete ride-down condition during a certain short control time after the crash starts. Solutions of the function are not unique. A corresponding basic optimal vehicle deceleration wave form type (shown by equations (4), (5) and (6)) exists for each of the three basic types of transient functions shown by equations (1), (2) and (3). Here, X_m indicates the occupant coordinate, and X_f the vehicle coordinate. In these equations the time t_c when control ends is expressed using the transient velocity $\gamma = \pi/t_c$. Also, the deceleration wave form after $t = t_c$ is the constant value C for all cases.

$$\ddot{X}_m = -C \left[\frac{\gamma}{\pi} \right]^3 \left[2t - 3 \frac{\pi}{\gamma} \right] t^2$$

[Power polynomial type transient function] (1)

$$\ddot{X}_m = \frac{C}{2} (1 - \cos \gamma t)$$

[Cosine type transient function] (2)

$$\ddot{X}_m = C \frac{\gamma}{\pi} \left[t - \frac{1}{2\gamma} \sin 2\gamma t \right]$$

[Sine type transient function] (3)

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$$\ddot{x}_f = -C \left[\frac{\gamma}{\pi} \right]^3 \left\{ \left[2t - 3 \frac{\pi}{\gamma} \right] t^2 + \frac{6}{\omega^2} \left[2t - \frac{\pi}{\gamma} \right] \right\}$$

[Power polynomial type] (4)

$$\ddot{x}_f = \frac{C}{2} \left\{ 1 + \left[\frac{\gamma^2}{\omega^2} - 1 \right] \cos \gamma t \right\}$$

[Cosine type] (5)

$$\ddot{x}_f = C \frac{\gamma}{\pi} \left\{ t + \left[\frac{4\gamma^2}{\omega^2} - 1 \right] \frac{1}{2\gamma} \sin 2\gamma t \right\}$$

[Sine type] (6)

These wave forms are shown in Fig.2. The optimal vehicle deceleration wave form exhibits a curved form which differs for each type of function. However, what each wave form has in common is a three-part structure consisting of high initial deceleration, negative middle deceleration, and then constant deceleration later on.

3. Study of Crash Load Control Structures

3.1. Study of Target Vehicle Deceleration Wave Forms

When considering the vehicle deceleration wave forms shown in Fig.2 as actual target vehicle deceleration wave forms, creating a negative deceleration wave form for the middle period of a crash becomes an issue. Vehicle deceleration in a crash is generally produced by the vehicle deformation load (crash load) applied to the vehicle mass, so the crash load is limited to the direction opposite that of the velocity, and the deceleration tends to increase as deformation progresses. For this reason, it is difficult to produce stable low deceleration with conventional vehicle structures during the middle period of a crash. Furthermore, the crash load must be in the opposite direction of the vehicle mass to produce negative deceleration, making it even more difficult to achieve with this vehicle structure.

However, when for example the realistic values of $\omega \approx 701/s$ and $C \approx 400m/s^2$ are set in the mathematical solution for the cosine type vehicle deceleration wave form shown in equation (5), if the control time t_c is set to 35 msec or more, the negative region of the vehicle deceleration wave form disappears as shown in Fig.3.

When considering this type of vehicle deceleration wave form as the target wave form, it is possible for the actual vehicle deceleration wave form to approach the target by

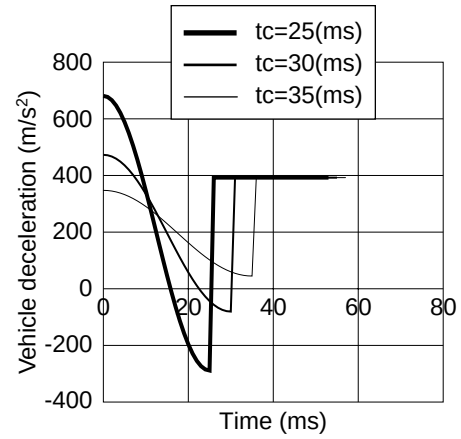


Fig.3 Deceleration curves for various t_c values

applying control so that the crash load is reduced during the middle period of a crash.

In addition, the linear combinations of the mathematical solutions for each different function type are also mathematical solutions, so various curve shapes can be considered as long as the optimal deceleration wave form has the common feature of the three-part structure. For this reason, even deceleration wave forms which change in a stepped manner can be approximate optimal deceleration wave forms. Therefore, the crash load does not necessarily have to be continuously controlled, and any means of reducing the crash load during the middle period of a crash can be considered effective for optimizing the vehicle deceleration wave form.

Thus, the target vehicle deceleration wave form was based on the cosine type and a realistic approximate optimal vehicle deceleration wave form was selected consisting of an initial high constant deceleration, a constant middle deceleration of approximately zero, and a constant later deceleration of value C . Next, methods for controlling the crash load with the vehicle structure were studied as a means of creating this wave form.

3.2. Crash Load Control by Deformation Mode

A number of methods can be considered for controlling the crash load. Fig.4 shows the basic concepts of load control structures. In consideration of application to existing vehicle

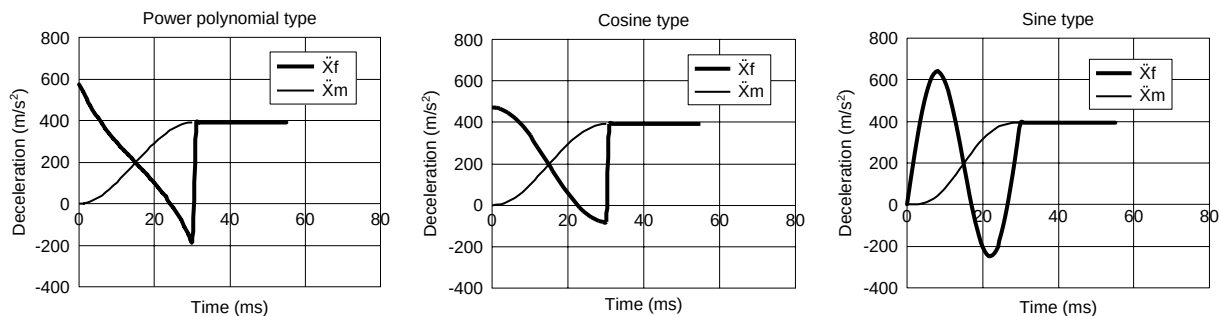


Fig.2 Wave forms of the mathematical solutions to the transient function forms

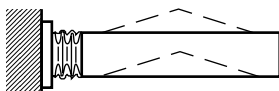
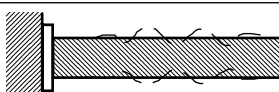
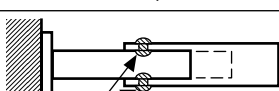
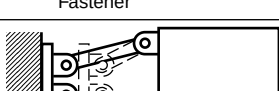
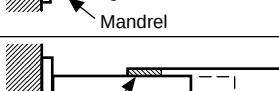
Control method	Basic construction
Combination of axial collapse and bending	
Controllable axial collapse	 Tailored composite material
Breakable fastener	 Fastener
Variation of the cosine component (geometry)	
Variation of the cross-sectional area	 Mandrel
Variation of the friction factor (from static to dynamic)	 Friction element

Fig.4 Methods for controlling the crash load

structures, a method which does not require mechanical elements is best. Therefore, the authors focused on methods utilizing the difference in load properties related to the deformation mode.

The vehicle structures which generate crash loads in general vehicles are hollow metal beam components such as front side members. When a load is generated on these structures compression collapse (hereafter “axial collapse”) or bending deformation is produced. Axial collapse is stable and generates a fairly constant load, whereas if bending deformation exceeds the maximum load the load decreases rapidly. Therefore, if the transition to bending deformation can be controlled, the load can be decreased at an optional time during the crash. This can be accomplished by a number

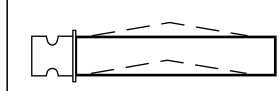
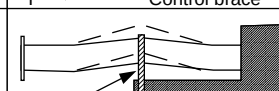
Control method	Basic construction
Difference in the maximum load between axial collapse and bending	
Control brace	 F Control brace
Breakable strap	 Breakable strap

Fig.5 Methods for controlling the transition to bending deformation

of methods as shown in Fig.5. Here the simplest structure is a combination of the axial collapse and bending zones in series. A structure was studied which maintains a large load at the start of the crash for a certain time and then rapidly decreases the load by providing a difference between the load generated in the axial collapse zone and the maximum bending zone load.

3.3. Fundamental Model of Crash Load Control Structure

Fig.6 shows the fundamental model of the devised crash load control structure. This fundamental model is a hollow beam component which hypothesizes a front side member, and is comprised of an axial collapse zone at the front and a bending zone in the middle. The axial collapse zone has a shape which concentrates stress to induce regular buckling deformation. The bending zone has a slightly angled bend shape in the center in order to direct the bending deformation in a stable manner. The cross section of each zone is set so that the axial collapse zone has a slightly smaller deformation load than the maximum load of the bending zone.

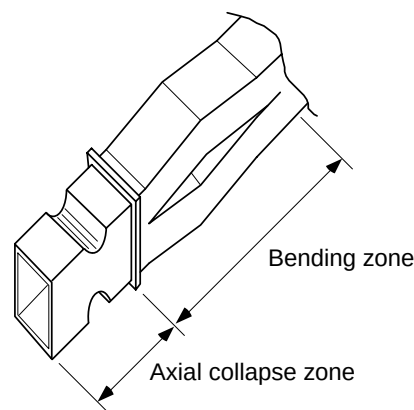


Fig.6 Fundamental model of a crash load control structure

Fig.7 shows the deformation process in the fundamental model. Fig.7-A shows the 1st stage. First, the stress centralizing shape of the axial collapse zone causes it to start deforming immediately after the start of the crash. A fairly constant load is produced after the axial collapse zone starts to deform, but the load increases as the regular buckling deformation advances over the entire axial collapse zone, and eventually reaches the maximum load of the bending zone. Fig.7-B shows the 2nd stage. When the bending zone exceeds the maximum load, it immediately bends significantly and the load drops to a fraction of that in the 1st stage. Fig.7-C shows the 3rd stage. As the bending deformation progresses, the beam bottoms out and the load increases again at the end of the crash.

Fig.8 shows the results of obtaining the change in crash load over time through a simulation using a finite element model created based on the fundamental model. This simulation attached a mass corresponding to a vehicle to the

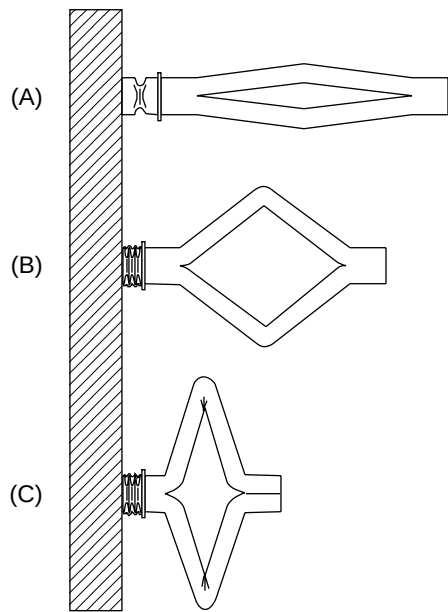


Fig.7 Deformation process in the fundamental model

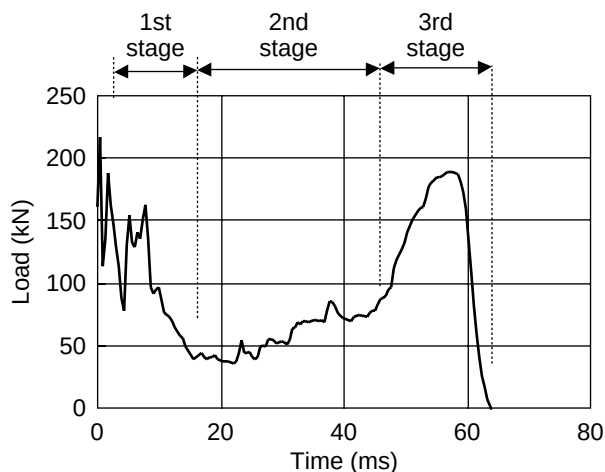


Fig.8 Change in crash load over time for the fundamental model

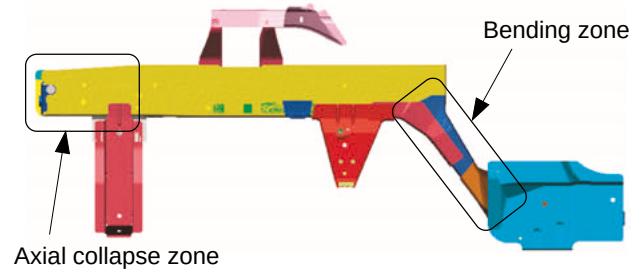
rear end of a beam, crashed the beam into a rigid wall with an initial velocity of 15.56 m/sec, and obtained the load generated on the rear end of the beam. The graph shows that a large load is maintained for a certain time at the start of the crash, the load decreases during the middle period, and then rises again toward the end of the crash, thus confirming crash load control based on the fundamental model. These results indicated that it was possible to achieve deceleration close to the target vehicle deceleration curve without significantly changing the conventional vehicle structure of actual vehicles.

4. Study of Crash Load Control in an Actual Vehicle Structure

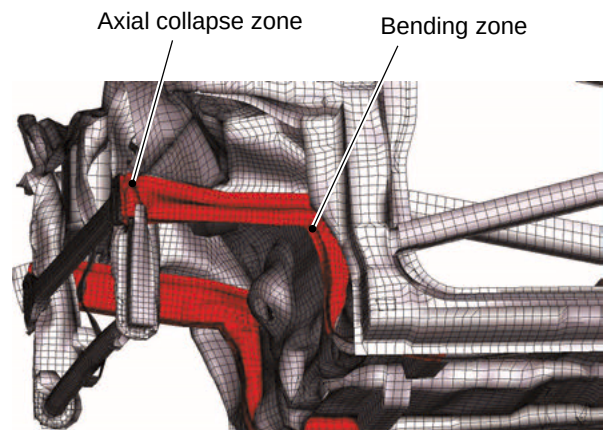
The fundamental model shown in Fig.6 in Section 3 has an extremely simple structure. However, in an actual vehicle

the engine mount is normally established near the midpoint of the horizontal part of the front side member, so the bending deformation of this part is controlled by the mass of the engine and the transmission. For this reason, it is difficult particularly in compact vehicles to adopt a design which provides a bending zone shaped like the fundamental model in the horizontal part of the front side member. Therefore, the existing bent shape in the connection between the cabin and the rear end of the front side member in the conventional vehicle structure was used as the bending zone, and an attempt was made to achieve the same functions as the fundamental model by making the maximum deformation load of this zone slightly larger than the axial collapse load of the front side member.

Fig.9 shows the structure of a production compact vehicle which performs vehicle deceleration control by introducing this approach. In this example there is almost no change to the vehicle structure, and reduction of the dummy chest deceleration was verified through barrier crash tests (Fig.10).



Front side member (before crash)



Front structure (after crash)

Fig.9 Application of the crash load control method to a production vehicle

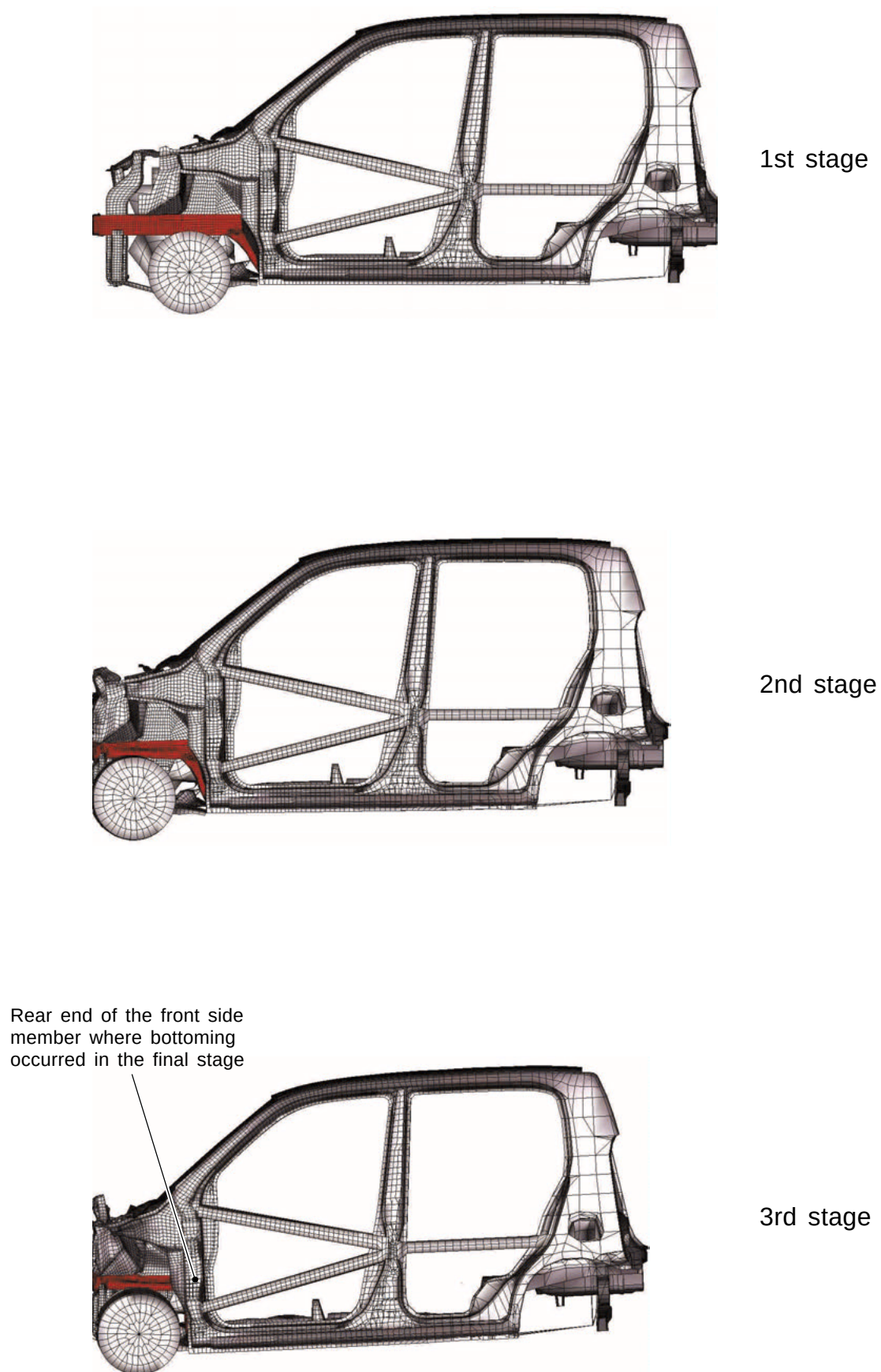


Fig.10 FEM simulation of barrier crash test

5. Conclusions

- (1) Focusing on the difference in load properties due to the hollow beam deformation mode, a fundamental model was devised for creating low deceleration during the middle period of a crash, which is a feature of the optimal vehicle deceleration wave form. This was achieved by controlling the change in crash load due to the transition from axial collapse to bending deformation by making the maximum deformation load of the axial collapse zone slightly smaller than that for the bending zone.
- (2) A finite element simulation model was created based on the fundamental model, and the change in crash load over time was obtained. The crash load was confirmed to decrease during the middle period and then rise again toward the end of the crash, thus verifying the functionality of the fundamental model.
- (3) Studies were carried out to apply the concept of the fundamental model to the bending deformation of the front side member of an actual vehicle. As a result, the dummy chest deceleration in barrier tests could be reduced without significantly changing the conventional vehicle structure.

Further studies will be carried out in the future on structures which optimize vehicle deceleration for various crash modes.

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Study of Occupant Deceleration Control During a Crash Due to Vehicle Mass Separation

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ABSTRACT

In order to reduce the maximum value of occupant deceleration in a vehicle crash, a physical model which utilizes the separated mass of a vehicle was devised to yield an optimal vehicle deceleration wave form. The validity of this model has been certified by a mathematical study and a three-dimensional dummy simulation.

1. Introduction

When considering vehicle crash safety performance, optimizing the change in vehicle deceleration over time in a crash (hereafter referred to as the “vehicle deceleration wave form”) is an important issue for simultaneously securing survival space and reducing injury severity. Previous research mainly performed optimization using the sensitivity analysis method, but the authors have derived and generalized mathematical solutions for the optimal vehicle deceleration wave forms for chest deceleration of the occupant through mathematical studies using a two-mass model. These mathematical solutions have in common a three-part structure consisting of high initial deceleration, low or negative middle deceleration, and then constant deceleration later on.⁽¹⁾

However, the vehicle deceleration in a general crash is determined by the vehicle structure deformation load applied to the vehicle mass. Therefore, the direction of the load relative to the vehicle mass is constant. This makes it difficult to generate negative deceleration. This paper reports on a physical model devised for generating negative deceleration on the vehicle, presents a mathematical study conducted based on this model, and examines the possibility of creating an optimal vehicle deceleration wave form. Furthermore, the validity of the vehicle deceleration wave forms obtained from this physical model are verified through a mathematical study and a three-dimensional dummy simulation.

2. Model for the Separation of Vehicle Mass

2.1. Optimal Vehicle Deceleration Wave Form

In the past the authors had mathematically studied the vehicle deceleration wave form that minimizes the maximum occupant deceleration value using a two-mass model like that shown in Fig.1. Here, the authors introduced a transient function concept and obtained mathematical solutions for

the optimal vehicle deceleration wave form⁽¹⁾. The transient function is the optimal occupant deceleration wave form that results in a complete ride-down condition during a certain short control time after the crash starts. Solutions of the function are not unique, and a corresponding basic optimal vehicle deceleration wave form type exists for each of the three basic types of transient functions.

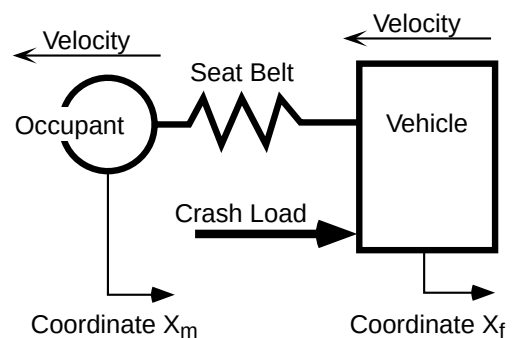


Fig.1 One-dimensional two-mass model

These wave forms are shown in Fig.2, and have in common a three-part structure consisting of high initial deceleration, low or negative middle deceleration, and then constant deceleration later on.

When considering these wave forms as the target vehicle deceleration wave forms for an actual vehicle, creating the deceleration wave form for the middle period of a crash becomes an issue. Vehicle deceleration in a crash is generally produced by the vehicle deformation load (crash load) applied to the vehicle mass. However, the direction of the crash load is limited to the direction opposite that of the velocity, and the deceleration tends to increase as deformation progresses. For this reason, it was impossible

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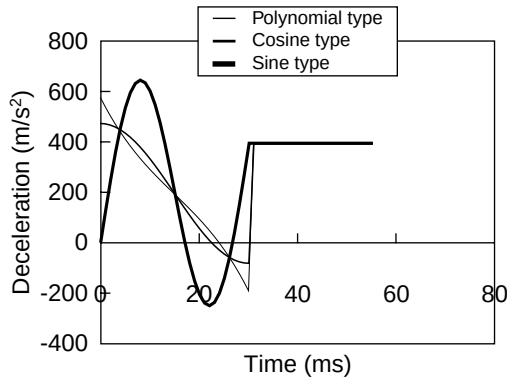


Fig.2 Wave forms of the mathematical solutions

to produce stable low or negative deceleration with conventional vehicle structures during the middle period of a crash. Therefore, the authors devised a new physical model as a means of solving this problem.

2.2 Model for the Separation of Vehicle Mass

Negative vehicle deceleration can be created by applying a crash load to the vehicle in the direction opposite the normal direction. This requires a new mass, but this new mass can be created inside the vehicle by separating the vehicle mass.

Fig.3 shows a model with the vehicle mass separated. Here, the occupant is treated as mass m and the seat belt as a spring with spring constant k . The vehicle is separated into two parts, with the part to which the seat belt is linked expressed by mass M_B and called the binding. Similarly, the remaining vehicle portion is called the frame and expressed by M_f . All masses have an initial velocity of V_0 .

The initial high deceleration and subsequent negative deceleration are required only with the binding. The coordinate system consists of the occupant coordinate X_m , and binding coordinate X_B , and the frame coordinate X_f , and the crash start time is $t = 0$. At this point the binding and frame are relatively separated only by the control gap S_0 . After the crash starts, the respective crash loads F_f and F_B act separately on the frame and binding, and these parts move independently of each other. If the vehicle is designed so that the binding deceleration is larger than the frame deceleration at this time, the gap decreases from S_0 and the

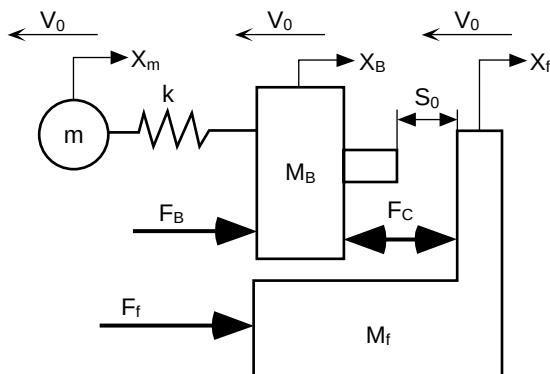


Fig.3 Model for the separation of vehicle mass

two parts eventually contact each other. This contact time is labeled t_{c1} , and the time from 0 to t_{c1} is called the first control time. After contact, a load F_C is produced between the binding and the frame. Here, F_C acts in the negative deceleration (acceleration) direction relative to the binding, so if F_C is sufficiently large, negative deceleration is created on the binding. The model assumes F_f , F_B and F_C to be constant values, and assumes a completely plastic crash. Therefore, the load F_C is eliminated when the binding and the frame reach the same velocity. This time is labeled t_{c2} , and the time from t_{c1} to t_{c2} is called the second control time. After t_{c2} , the binding and frame decelerate as a single unit.

However, while it is clear that negative deceleration can be obtained from this model, it is not known whether the optimal vehicle deceleration wave form can be obtained. Therefore, a mathematical study is conducted next to determine the possibility of realizing the optimal vehicle deceleration wave form.

3. Mathematical Study of Model for the Separation of Vehicle Mass

When a fixed load is applied to the binding as an input, the output occupant deceleration wave form clearly shows the same form as a cosine type transient function, and this also appears in the solution for equation (19) noted hereafter. In addition, the motion equations for this model differ for the three time regions of the first control time, the second control time, and thereafter. Therefore, individual cosine type occupant deceleration wave forms appear in the first and second control times, but when these are linked if the occupant deceleration wave form for the entire control time satisfies the transient function conditions, namely starting from the origin, a tangential gradient of 0 at the origin and a complete ride-down condition at t_{c2} , then this wave form is a single transient function which combines cosine type deceleration wave forms.

Based on this thinking, the time regions are related by solving the equation for each control time with the crash load and control gap as unknown constants and t_{c1} and t_{c2} as independent variables, and using the occupant and binding conditions at the end of the first control time t_{c1} as the initial conditions for the second control time equation. Then, the optimal crash load and optimal control gap are calculated by applying the transient function conditions to the obtained occupant deceleration equation.

When t_{c1} and t_{c2} are set arbitrarily, if these exist as finite values, then they also determine the vehicle deceleration. This is the vehicle deceleration that corresponds to the transient function, or in other words the optimal vehicle deceleration wave form.

3.1. Equations of Motion and Condition

First, the motion equations for the first control time are as follows, with the time differential expressed as $(\cdot)$.

$$\ddot{X}_m = -\omega^2(X_m - X_B) \quad (1)$$

$$\ddot{X}_B = \alpha\omega^2(X_m - X_B) + g_B \quad (2)$$

$$\ddot{X}_f = g_f \quad (3)$$

Provided that,

$$\omega = \sqrt{\frac{k}{m}}, \alpha = \frac{m}{M_B}, g_B = \frac{F_B}{M_B}, g_f = \frac{F_f}{M_f} \quad (4)$$

The initial conditions for these are as follows.

$$\begin{aligned} X_m(0) &= X_B(0) = 0, X_f(0) = S_0, \\ \dot{X}_m(0) &= \dot{X}_B(0) = \dot{X}_f(0) = -V_0 \end{aligned} \quad (5)$$

Also, t_{c1} is the time at which the binding and frame contact each other, so the condition for t_{c1} is as follows.

$$X_B(t_{c1}) = X_f(t_{c1}) \quad (6)$$

Next, the motion equations for the second control time are as follows.

$$\ddot{X}_m = -\omega^2(X_m - X_B) \quad (7)$$

$$\ddot{X}_B = \alpha\omega^2(X_m - X_B) + g_B - g_C \quad (8)$$

$$\ddot{X}_f = g_f + \alpha_B g_C \quad (9)$$

Provided that,

$$g_C = \frac{F_C}{M_B}, \quad \alpha_B = \frac{M_B}{M_f} \quad (10)$$

t_{c2} is the time at which the binding and frame reach the same velocity, so the condition for t_{c2} is:

$$\dot{X}_B(t_{c2}) = \dot{X}_f(t_{c2}) \quad (11)$$

In addition, the transient function conditions at the origin are automatically satisfied by the initial conditions. The remaining conditions, that is to say the conditions for the occupant to reach a complete ride-down condition at t_{c2} , are that the occupant deceleration is C, with C being the target occupant deceleration setting value, and that the occupant and the binding are traveling at the same velocity. These conditions are expressed as follows.

$$\ddot{X}_m(t_{c2}) = C \quad (12)$$

$$\dot{X}_m(t_{c2}) = \dot{X}_B(t_{c2}) \quad (13)$$

Then, the condition equation to maintain the complete ride-down condition after the control time is that the deceleration for the entire vehicle and the occupant deceleration are the same value, or in other words:

$$\frac{F_f + F_B}{M_f + M_B + m} = C \quad (14)$$

3.2. Solutions to the Equations

These equations can be solved by various methods, but this study solves the equations by introducing the relative coordinate as follows.

$$X_m - X_B = X_r \quad (15)$$

This equation was then used to convert the simultaneous equations (1)(2) and (7)(8) into the following single-variable equations.

$$\ddot{X}_r + (1 + \alpha)\omega^2 X_r = -g_B \quad (16)$$

$$\ddot{X}_r + (1 + \alpha)\omega^2 X_r = -g_B + g_C \quad (17)$$

Based on the initial conditions (5), the initial conditions for X_r are as follows.

$$X_r(0) = 0, \quad \dot{X}_r(0) = 0 \quad (18)$$

Solving equation (16) based on equation (18), the solution during the first control time regarding to X_r as follows.

$$X_r = -\frac{g_B}{\beta^2}(1 - \cos\beta t) \quad (19)$$

Provided that,

$$\beta = \omega\sqrt{1 + \alpha} \quad (20)$$

In addition, solving equation (17) with the $X_r(t_{c1})$ and $\dot{X}_r(t_{c1})$ obtained from equation (19), the solution for the second control time is as follows.

$$X_r = -\frac{g_B - g_C}{\beta^2} - \frac{g_C}{\beta^2}\cos\beta(t - t_{c1}) + \frac{g_B}{\beta^2}\cos\beta t \quad (21)$$

On the other hand, the ride-down condition equations (12) and (13) are converted as follows by equation (15).

$$X_r(t_{c2}) = -\frac{C}{\omega^2}, \quad \dot{X}_r(t_{c2}) = 0 \quad (22)$$

Applying these to equation (21) yields the following.

$$g_B = (1 + \alpha)C \frac{\sin\beta(t_{c2} - t_{c1})}{\sin\beta(t_{c2} - t_{c1}) - \sin\beta t_{c2} + \sin\beta t_{c1}} \quad (23)$$

$$g_C = (1 + \alpha)C \frac{\sin\beta t_{c2}}{\sin\beta(t_{c2} - t_{c1}) - \sin\beta t_{c2} + \sin\beta t_{c1}} \quad (24)$$

Then, applying these to equations (6) and (11),

$$g_f = C \frac{\sin\beta(t_{c2} - t_{c1}) - K(1 - \frac{t_{c1}}{t_{c2}})\sin\beta t_{c2}}{\sin\beta(t_{c2} - t_{c1}) - \sin\beta t_{c2} + \sin\beta t_{c1}} \quad (25)$$

Provided that,

$$K = 1 + \alpha_B + \alpha \alpha_B \quad (26)$$

Based on the above equations (23) to (25), the appropriate crash load is calculated by the following equations.

These equations clearly show that the values are finite. Therefore, it is possible to obtain the optimal vehicle deceleration wave form from the vehicle mass separation model presented in this paper.

$$F_f = C \frac{M_f \sin \beta (t_{c2} - t_{c1}) - (M_f + M_B + m)(1 - t_{c1}/t_{c2}) \sin \beta t_{c2}}{\sin \beta (t_{c2} - t_{c1}) - \sin \beta t_{c2} + \sin \beta t_{c1}} \quad (27)$$

$$F_B = C \frac{(M_B + m) \sin \beta (t_{c2} - t_{c1})}{\sin \beta (t_{c2} - t_{c1}) - \sin \beta t_{c2} + \sin \beta t_{c1}} \quad (28)$$

$$F_C = C \frac{(M_B + m) \sin \beta t_{c2}}{\sin \beta (t_{c2} - t_{c1}) - \sin \beta t_{c2} + \sin \beta t_{c1}} \quad (29)$$

Note that as a condition for the F_f obtained from equation (27) here to be a positive value, the limitation on the relationship between t_{c1} and t_{c2} is derived approximately as follows.

$$t_{c2} > \frac{-t_{c1} + \sqrt{K t_{c1}^2 + \frac{3(K-1)^2}{\beta^2}}}{K-1} \quad (30)$$

However, this limitation need to be considered only when t_{c1} is set extremely small within the range of $t_{c1} < \frac{1}{\beta} \sqrt{\frac{3(K-1)}{K}}$. In addition, the setting range for t_{c1} and t_{c2} is $0 < t_{c1} < t_{c2} < \pi/\beta$ so that F_B and F_C are positive values.

The optimal control gap value is obtained as follows by calculating $X_B(t_{c1})$ and $X_f(t_{c1})$, and substituting these into equation (6).

$$S_0 = C \frac{\frac{\alpha}{\beta^2} (1 - \cos \beta t_{c1}) \sin \beta (t_{c2} - t_{c1}) + \frac{t_{c1}^2}{2} K (1 - \frac{t_{c1}}{t_{c2}}) \sin \beta t_{c2}}{\sin \beta (t_{c2} - t_{c1}) - \sin \beta t_{c2} + \sin \beta t_{c1}} \quad (31)$$

Finally, substituting equation (27) and (28) into condition equation (14) shows that this equation can not be satisfied for the period after the control time except the case of $t_{c1} = t_{c2}$, which has no physical meaning. This means that the appropriate crash load within the control time is always insufficient for the required crash load after the control time. Therefore, one or both of F_f and F_B must be changed from the values in equations (27) and (28) at t_{c2} to values that satisfy equation 14. In this study, F_f was changed.

Based on the above analysis results, the deceleration wave forms given by equations (1) to (3) and (7) to (9) for the occupant m , binding M_B , and frame M_f are shown in Fig.4. This example sets $m = 33$ kg, $M_B = 40$ kg, $M_f = 500$ kg, $k \approx 16$ kgf/mm, $C = 40G$, $t_{c1} = 12$ ms, and $t_{c2} = 24$ ms.

Note that the crash load values obtained by equations (27) to (29) at this time are $F_f \approx 8.8$ tons (until t_{c2}) and 20.3 tons (after t_{c2}), $F_B \approx 2.7$ tons, and $F_C \approx 2.4$ tons, and the control gap is $S_0 \approx 32$ mm.

4. Verification through Simulation

The binding deceleration wave form shown in Fig.4 was input to a dummy simulation model to mathematically confirm the validity of the wave form. The model used the ATB program, and applied a three-point seat belt as the standard binding condition. Load limiters and air bags were not applied.

Fig.5 shows the input binding deceleration wave form, the deceleration response of the dummy's chest (dummy

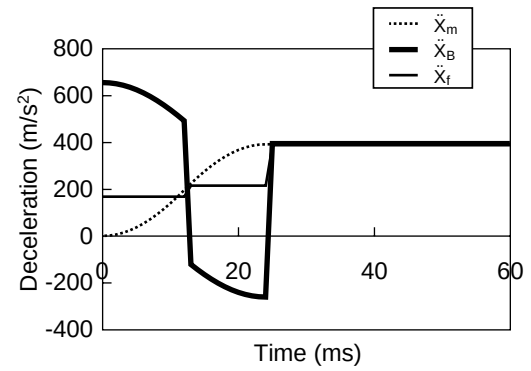


Fig.4 Theoretical vehicle and occupant deceleration wave form from the vehicle mass separation model

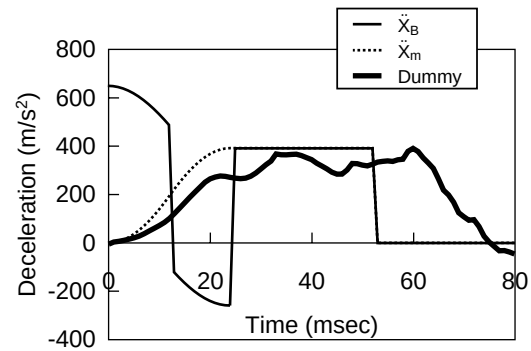


Fig.5 Vehicle and dummy chest deceleration in a three dimensional simulation

coordinates), and the corresponding mathematically obtained occupant deceleration wave form.

This figure shows that although slightly delayed, the graph of dummy chest deceleration matches closely with the calculated graph, and forms a fairly flat portion between 30 and 60 ms.

The slight delay in the dummy graph relative to the calculated graph is thought mainly to occur because the mass corresponding to m increases slightly during the course of the crash due to the effects of the arm and head portions, and because the spring constant of the seat belt in the simulation is adjusted so that elongation is nonlinear in small elongation regions to match with reality. In addition, the fluctuation appearing after 20 ms is presumed to be because the dummy forms a complex multi-mass system.

5. Observations

Creating negative acceleration is impossible with a model that treats the vehicle as a single mass as shown in Fig.1. However, this is possible with a model that separates the vehicle mass. Further, the feasibility of creating an optimal vehicle deceleration wave form using this model was verified both numerically and through simulation.

The studies presented in this paper use a simplified physical model. The applicability of this model to an actual vehicle structure has not yet been studied, and various

technical problems can be expected. Furthermore, verification not only through simulation but also using actual dummies must also be investigated in the future. However, it is significant that this paper has shown the possibility of creating vehicle deceleration wave forms which include a negative deceleration portion using only the crash load due to vehicle mass separation.

6. Conclusions

- (1) A vehicle mass separation model which had not been studied before was devised as a means of creating negative vehicle deceleration.
- (2) It was verified through a mathematical study that the optimal vehicle deceleration wave form can be obtained from this vehicle mass separation model, and a mathematical solution for the optimal vehicle deceleration wave form was obtained using this model.
- (3) The obtained mathematical solution was input to a three-dimensional dummy simulation and the chest deceleration response was obtained. These results verified the validity of the mathematically solved optimal vehicle deceleration wave form with respect to reducing chest deceleration.

Future issues include verifying the validity of this model using actual dummies and investigating the applicability to an actual vehicle structure.

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Takahiro KAMEI



Yasuki MOTOZAWA

Low Order Modeling of an Acoustic Path for Active Noise Control and Design Method of Feedback Control

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Shuichi ADACHI**

ABSTRACT

In order to reduce the cost of active noise reduction (ANC), this research focused on feedback (FB) control, which characteristically does by not require reference signal sensors. The purpose of this paper is low order modeling of acoustic transfer characteristic (plant) between a control noise source and an error sensor arranged at the control point; and, the designing of a controller based on the results. Specifically, this entails constructing as simple an FB-ANC system as possible (a 1 input/1 output system), then investigating the noise reduction performance of that system. First, the authors employed physical modeling in order to model the plant in the lowest possible order. In addition, Smith predictor-FB-ANC is presented, and FB-ANC based on PID (Proportional Integral Derivative) control is designed for the low-order plant model obtained. Finally, the validity of these results is demonstrated with computer simulation using actual data.

1. Introduction

A major issue in active noise control (ANC) is its high cost. For instance, active control of interior road noise employing feed forward (FF) control⁽¹⁾⁻⁽³⁾ requires numerous references signals (i.e., input signals) because of the complexity of input paths. Consequently, installing this system at low-cost has been considered difficult to achieve. Another issue is determining the proper placement of reference signal sensors.

As a means of resolving these issues, this research project focuses on feedback (FB) control, which characteristically does not require reference signal sensors. In FB-ANC research, the Tight Coupled Monopole (TCM) method^{(4), (5)}, whereby error sensors (i.e., microphones) are placed near the control noise source (i.e., speakers), is well-known. However, few cases have addressed FB-ANC from the control engineering perspective of using modeling and design of a controller based on a model. In recent years, there has been some research from a control engineering perspective⁽⁶⁾⁻⁽¹⁰⁾, but FB-ANC research is still less extensive than FF-ANC research. FB-ANC offers the advantage of a simplified installation system, yet there have been few investigations of its strengths and weaknesses with respect to noise reduction. Hence, clarifying the performance of FB-ANC is an important prerequisite for further advancements in ANC technology.

Against this background, this paper has as its objectives the low-order modeling of acoustic transfer characteristics from a control sound source to an error sensor located at the control point (i.e., the “plant”), and the designing of a

controller based on those results⁽¹¹⁾. Specifically, this entails constructing as simple an FB-ANC system as possible (single input/single output), then investigating the noise reduction performance of that system. First, physical modeling is employed in order to model the plant in the lowest possible order. Next, to create a high-precision model, gray-box modeling⁽¹²⁾, which uses physical modeling information and experimental data is applied. In addition, Smith predictor type FB-ANC^{(13), (14)} is proposed, and FB-ANC based on Proportional Integral Derivative (PID) control is designed for the low-order plant model obtained. Finally, the validity of these results is demonstrated with computer simulation using actual data.

Throughout this paper, the terms “speaker” and “microphone” respectively refer to the control sound source and error sensor used in this research.

2. Physical Modeling of the Acoustic Path

2.1. Conventional Modeling Techniques

In the field of ANC, system identification is frequently used as a means of modeling. Least Mean Square (LMS) algorithm is widely employed as the identification method because of low calculations volume, Finite Impulse Response (FIR) as the identification model because of its stability⁽¹⁵⁾. Nevertheless, precise identification requires numerous FIR model parameters in order for impulse responses to converge. Designing an FB controller with an FIR model in accordance with control theory is also difficult because of the need to express the plant with an Infinite Impulse Response (IIR) model.

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On the other hand, there have been cases reported in which the plant was expressed as an IIR model to reduce the parameters in the mathematical model, and in which the Simplified Hyperstable Adaptive Recursive Filter (SHARF) method was used as the identification method⁽¹⁶⁾. The actual number of parameters, however, was determined by trial and error. Therefore, as a means of determining the number of parameters with quantitative indicators, we proposed an IIR model identification method that employs a subspace method⁽¹⁰⁾. Using this model we have expressed the plant with an IIR model having twelve parameters (i.e., a sixth-order IIR filter).

Nevertheless, it is not clear to what extent the number of parameters can be reduced while still maintaining the requisite precision. Therefore, determining this will not only simplify the designing of FB controllers, but it is also an important factor in achieving low-cost ANC. We therefore use physical modeling of the plant to obtain a high-precision, low-order model^{(7), (17), (18)}.

2.2. Physical Modeling

Below, physical modeling is performed separately for the three components of the plant: speaker, sound field, and microphone⁽¹⁸⁾.

2.2.1. Physical Modeling of the Speaker

When the speaker's mechanical system is equivalently converted into electrical circuits, the transfer function $P_s(s)$ from voltage e_0 [V], which is impressed on the circuit terminal, to vibration velocity v_0 [m/s] of the diaphragm is expressed as shown below:

$$P_s(s) = \frac{As}{a_1s^2 + a_2s + C_Mr} \quad (1)$$

where

$$a_1 = \frac{8}{3} a^3 \rho r + mr + Lr_M$$

$$a_2 = r \cdot r_M + C_M L + A^2$$

where

r [kg/m³] = air density,
 a [m] = cone radius,
 m [kg] = diaphragm and coil mass,
 A = electrodynamic converter's force factor,
 r [W] = coil resistance,
 L [H] = coil inductance,
 r_M [Ns/m] = mechanical resistance,
 C_M [N/m] = stiffness of the diaphragm support and
 s = a complex number in Laplacian transformation.

2.2.2. Physical Modeling of the Sound Field

We assume that the distance from speaker to microphone can be made sufficiently short relative to the wavelength of the sound to be controlled. Because the effect of direct sounds, rather than reflected sounds, are predominant even in a closed space such as an indoor chamber, the effect of the natural acoustic mode is believed to be minimal. Therefore, a free sound field is assumed for simplicity's sake.

The speaker's radiative surface is assumed to approximate a planar piston because vibration velocity of the diaphragm is equivalent to that of air particles on the diaphragm. The result is an equation of transfer function

$P_A(s)$ from diaphragm air particle velocity v_0 [m/s] to the sound pressure at a point on the central axis located distance l [m] from the planar piston. That equation is as shown below:

$$P_A(s) = \rho c \left\{ \exp\left(-\frac{l}{c}s\right) - \exp\left(-\frac{\sqrt{l^2 + a^2}}{c}s\right) \right\} \quad (2)$$

where c [m/s] is the speed of sound⁽¹⁸⁾. When the distance to the measurement point is extremely long relative to speaker radius, or in the case of a low-range frequency, the approximation

$$P_A(s) = \rho \frac{a^2}{2l} s \cdot \exp\left(-\frac{l}{c}s\right) \quad (3)$$

holds true. Equation (3) shows that the sound radiation characteristics comprise derivative element $(\rho a^2/2l)s$ and time delay element $\exp\{-(l/c)s\}$. As the target of noise reduction was the 40-500 Hz frequency range, we used equations (2) and (3) to calculate changes in sound pressure over distance when 40 Hz and 500 Hz sine waves were excited. The radius is 0.0475 which is the same as the actual speaker used in the experiment. The results, shown in Fig.1, show equation (3) to be fully capable of expressing the transfer function of a sound field when the distance is 0.1 m or greater.

In addition, when the effects of the natural acoustic mode are such that they cannot be disregarded, they, too, must be incorporated into modeling.

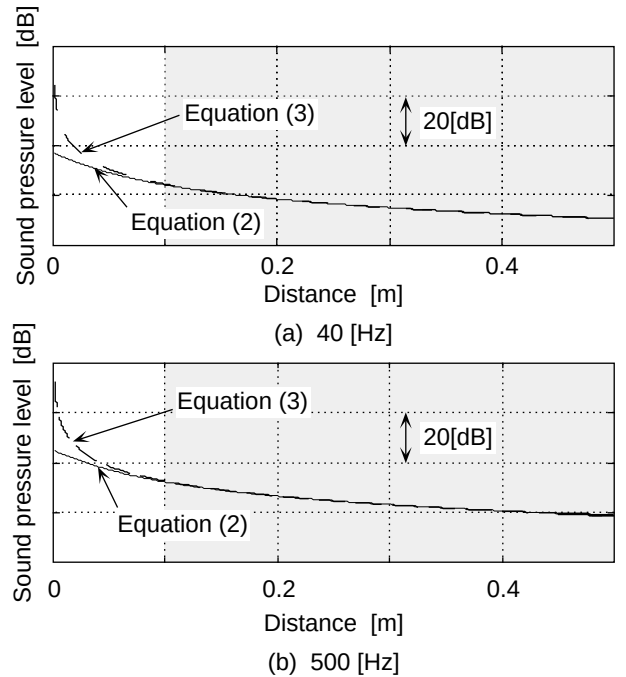


Fig.1 Distance vs sound pressure

2.2.3. Physical Modeling of the Microphone

This research project assumes the use of a condenser microphone, using stiffness control. As the frequency range to be controlled has flat characteristics, with no phase differences, the microphone's characteristics can be expressed in terms of proportional gain. Here, for simplicity's sake, the microphone's transfer function $P_M(s)$ is expressed

below:

$$P_M(s) = 1 \quad (4)$$

2.3. Physical Model of the Acoustic Path

A physical model of the acoustic path can be expressed with the three aforementioned components. Table 1 gives the physical parameters of the speaker used. Using the physical parameters in Table 1 to calculate the transfer function of the plant for a speaker-to-microphone distance of 0.1 m, the calculations are as shown below:

$$\begin{aligned} P(s) &= P_S(s)P_A(s)P_M(s) \\ &= \frac{(\rho A a^2 / 2l) s^2}{a_1 s^2 + a_2 s + C_{MR}} \cdot \exp\left(-\frac{l}{c} s\right) \\ &= \frac{2.03 \times 10^{-2} s^2}{9.03 \times 10^{-3} s^2 + 12.43 s + 3512.6} \\ &\quad \cdot \exp(-2.94 \times 10^{-4} s) \\ &= P_C(s) \exp(-\tau s) \end{aligned} \quad (5)$$

where

$$\begin{aligned} P_C(s) &= \frac{2.03 \times 10^{-2} s^2}{9.03 \times 10^{-3} s^2 + 12.43 s + 3512.6} \\ \tau &= 2.94 \times 10^{-4} \end{aligned}$$

Thus, the plant can be expressed as the second-order lumped-parameter part $P_C(s)$ and time delay element $\exp(-\tau s)$.

The next step is to validate the physical model obtained. To do this, system identification was performed by using input/output data of the plant measured in the semi-anechoic chamber. A total of 20000 samples were taken at a sampling frequency of 1378 Hz. An FIR model with 84 parameters was used for identification. Bode plots of the physical model and the results of identification are shown in Fig.2. Although the models generally agree in the gain diagram, the physical model and FIR model in the phase diagram do not agree. Let us consider the reason for this.

2.4. A Consideration of Phase Difference

The input/output coherence function obtained in the system identification experiment is shown in Fig.3. These

Table 1 Parameter of the loudspeaker (Panasonic WS-A10-K)

Parameter		Value
Mass	m	3.1×10^{-3} [kg]
Mechanical resistance	r_m	2.6 [Ns/m]
Mechanical stiffness	C_m	1351 [N/m]
Coil resistance	r	2.6 [Ω]
Coil inductance	L	5.4×10^{-5} [H]
Cone radius	a	0.0475 [m]
Force factor	A	2.98 [Tm, N/A]
Air density	ρ	1.21 [kg/m ³]
Wave speed	c	340 [m/s]

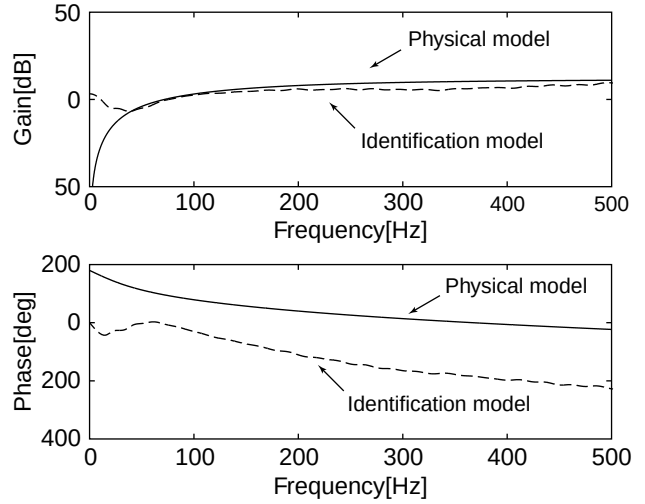


Fig.2 Comparison of Bode plot between a physical model and an identification model

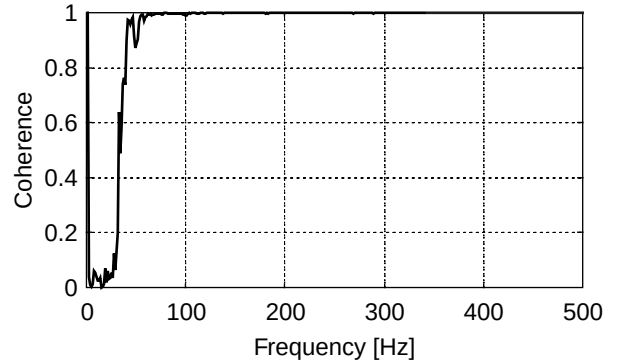


Fig.3 Coherence function

results suggest that because of the difficulty of producing noise from the speakers in the low-frequency band (below 40 Hz), coherence was extremely low. In Fig.2, phase is expressed continuously, rather than from -180 to 180 degrees because of the control system design. This means that phase has been calculated based on the phase of a band having low coherence and low identification results reliability. This results in the incorrect calculation of the absolute value for phase in the high-coherence range. This makes it impossible to determine whether the phase determined by identification or the phase calculated with the physical models is correct.

In the next section, therefore, phase is accurately estimated and gray-box modeling using the estimated phase are performed.

3. Gray-box Modeling of Acoustic Path

In this section, we construct a highly accurate model using gray-box modeling⁽¹²⁾. Here, physical model information are used for the order of the plant (structural information about the plant), and system identification data are used for adjusting parameters.

Physical modeling, a method for directly describing the dynamic characteristics of the subject in accordance with

the laws of physics, is referred to as white box modeling. System identification is referred to as black box modeling because the subject is considered a black box. Gray-box modeling is so called because it uses both of these modeling techniques to achieve greater modeling accuracy.

In estimating the phase in the coherent frequency range, the cross-correlation function is expressed as

$$\phi_{uy}(\tau) = E[u(t)y(t+\tau)] \quad (6)$$

where $u(t)$ is input, $y(t)$ output, and τ —time delay. The cross-correlation function determines the time delay between input and output and so can be used to estimate the phase. Fig.4, which shows the phase estimated from the cross-correlation function, demonstrates how the estimated phase very closely approximates not the physical model, but the identification model.

Next, we performed gray-box modeling using the structural information for the physical model along with the phase estimated with the cross-correlation function, as in Fig.4. The difference between the phases determined with the physical model and with the cross-correlation function was determined and added to the physical model as a time delay. However, because simply adding to the physical model does not assure proper expression of phase characteristics, the parameters of the lumped parameter part were also adjusted.

The transfer function of the model obtained by this gray-box modeling is expressed in the equation (7):

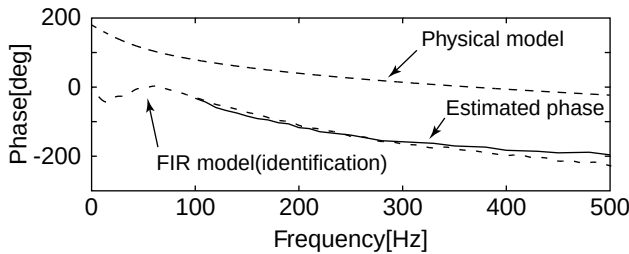


Fig.4 Estimated phase using cross-correlation function

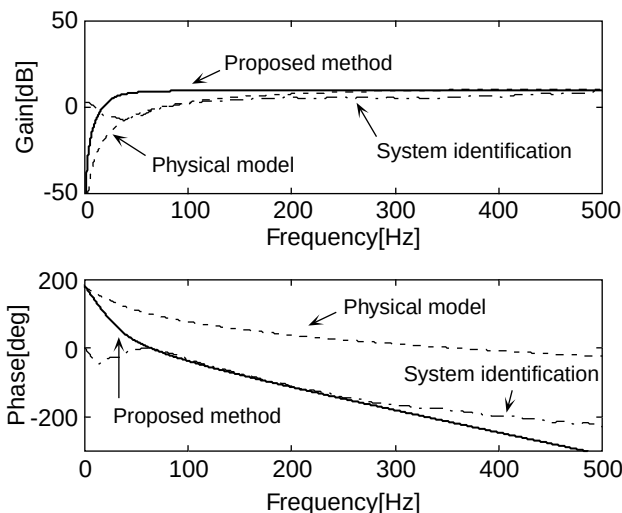


Fig.5 Comparison of Bode plot

$$P(s) = \frac{0.485s^2}{0.155s^2 + 45.1s + 4550} e^{-1.75 \times 10^{-3}s} \quad (7)$$

The Bode plots are in Fig.5.

In Fig.5, the phase corresponds up to approximately 250 Hz, beyond which it lags relative to the phase suggested by the system identification model. This is because the phases were compared primarily in the low range -- which is effective for ANC -- due to the difficulty of getting the phases to correspond throughout the 0-500 Hz range when expressing the physical model with a second-order system and time delay.

4. The Proposed Smith Predictor-type FB-ANC

With equation (5), the target plant was expressed in terms of the lumped parameter part and the time delay element that occurs because of the distance delay. The fact that the plant is a time delay series entails three problems:

- 1.) The inability to express the plant with a rational polynomial expression makes it difficult to design the controller in accordance with control theory.
- 2.) Even when using Padé approximation⁽¹⁹⁾ for the time delay element, a higher degree of approximation accuracy results in a higher plant order, which in turn makes for a high-order controller.
- 3.) The existence of the time delay element results in a non-minimum phase system, and the waterbed effect⁽²⁰⁾ makes broadband sound reduction difficult⁽⁹⁾.

Overcoming these problems requires elimination of the time delay element. To do this, we propose a type of FB-ANC using the Smith predictor^{(13), (14)}, an effective control technique for chemical and other plants where time delay elements are predominant.

As shown in Fig.6, the Smith predictor takes the approach of designing a control system by canceling the plant's time delay element to create a closed loop system free of any time delay. In contrast, FB-ANC treats noise as a disturbance, which is added after the plant, and is considered a disturbance-driven approach because of the absence of command input. Thus, the Smith predictor configuration shown in Fig.6 cannot be directly applied to FB-ANC.

However, we have made it possible to apply the Smith predictor to FB-ANC by doing the following. The design specifications of FB-ANC are intended to stabilize a closed loop system and reduce sensitivity function $S(s)$ in the frequency band to control (i.e., the transfer function from

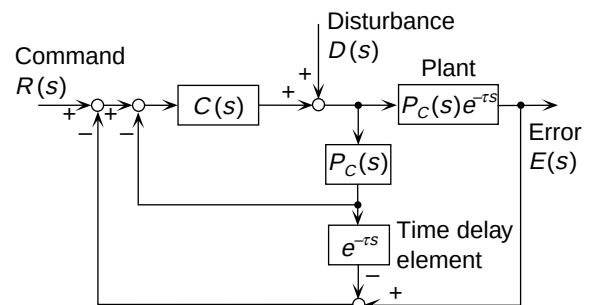


Fig.6 The Smith predictor

noise $D(s)$ to error $E(s)$). With the configuration in Fig.6, the sensitivity function when disturbance is added after the plant is determined with this equation:

$$S(s) = \frac{1 + P_C(s)C(s)\{1 - \exp(-\tau s)\}}{1 + P_C(s)C(s)} \quad (8)$$

Thus, the plant's time delay element is reflected in the sensitivity function. To prevent the time delay element from being reflected in the sensitivity function, the authors have proposed a Smith predictor-type FB-ANC configuration, shown in Fig.7, in which the sensitivity function is:

$$S(s) = \frac{1}{1 + P_C(s)C(s)} \quad (9)$$

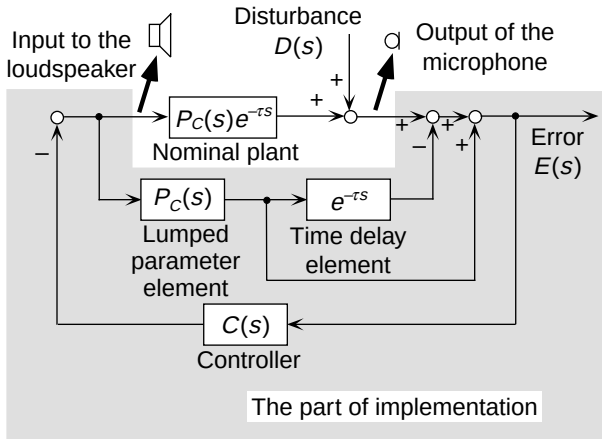


Fig.7 The proposed Smith predictor type FB-ANC for a nominal plant

With this configuration, design of the controller to the plant's lumped parameter part $P_C(s)$ is possible with equation (9).

The Smith predictor is interpretable within the Internal Model Control (IMC) framework whose application in FB-ANC we have proposed previously^{(9),(21)}. The Smith predictor is characterized by its facilitating of control system design using a $P_C(s)$ from which the plant's time delay element has been removed.

5. The PID Controller Design Method

This section discusses the structure of, and design method for, controllers. With respect to controller structure, focus is placed on PID control⁽²²⁾ as a way of achieving low-cost ANC. Also known as three-term control, PID control is characterized by a simple structure comprising proportional control, integral control, and derivative control. PID control is used quite extensively by the industry, but no cases can be found of its application in ANC. Therefore, this section discusses the design of FB-ANC that uses a PID controller.

The design method used by the authors is a PID controller design method that was proposed by Saeki and which enables the systematic determination of the three design parameters of PID control⁽²³⁾.

Stated simply, the actual design process begins by using the following equation to determine one of the PID control parameters k_i from the control constraints of the sensitivity function in the low frequency band.

$$C(s) = k_p + k_d \frac{s}{1 + T_s} + k_i \frac{1}{s} \quad (10)$$

Next, the following parameter sets concerning k_p, k_d , which satisfy the nominal performance conditions (i.e., noise reduction as nominal performance) and robust stability conditions (i.e., stability conditions that account for uncertainty in the plant model), are solved for each frequency.

$$K_S(\omega) \equiv \{[k_p, k_d] \mid |S(j\omega)| < \alpha(\omega)/|W_S(j\omega)|\} \quad (11)$$

$$K_T(\omega) \equiv \{[k_p, k_d] \mid |T(j\omega)| < \beta(\omega)/|W_T(j\omega)|\} \quad (12)$$

The elements of the set of the intersection of the sets in equations (11) and (12)

$$K(\omega) \equiv K_S(\omega) \cap K_T(\omega) \quad (13)$$

are now the PID control parameters that satisfy the design specifications. From the elements of the set in equation (13), the parameters are selected and the robust PID controller is determined.

6. Verification of Control Performance

6.1. Validity of the Proposed Smith Predictor-type FB-ANC

To determine the validity of the proposed Smith predictor-type FB-ANC, we used computer simulation to predict controlled sound pressure. The experiment shown in Fig.8 simulated road noise. Noise inside the vehicle was measured and used as noise data for simulation, which was performed with a discrete time series and a sampling frequency of 10 240 Hz.

The plant used was the physical model discussed in section 2. The PID controller designed for the plant's lumped parameter portion $P_C(s)$ was used as controller $C(s)$, shown in Fig.7. Given $a=0.58$ and $b=0.75$, the parameters were selected from parameter set K , which satisfies the nominal performance conditions and robust stability conditions, and the PID controller was determined with the following equation.

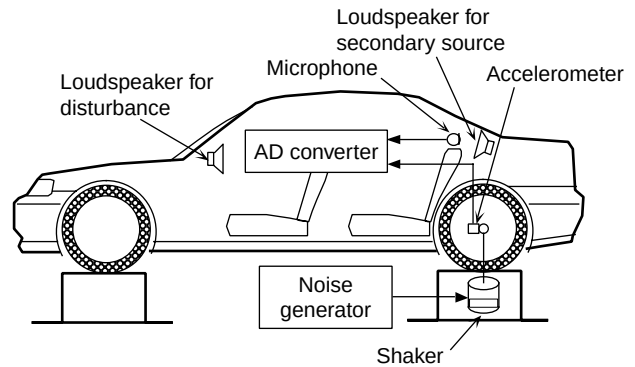


Fig.8 Data acquisition experiment simulating road noise in the cabin

$$C(s) = 0.2125 + \frac{s}{1 + s/(2\pi \cdot 500)} (1 \times 10^{-6}) + \frac{900}{s} \quad (14)$$

Fig.9 shows the sensitivity function, which was used as an indicator for predicting noise reduction performance during control; Fig.10, the results of sound pressure prediction simulation. A noise reduction of 4 to 7 dB is predicted from Fig.9. In actuality, the results of simulation indicate a 4 to 7 dB noise reduction in the broad band from 40 to 500 Hz, as Fig.10 shows. There was a marked reduction of 7 dB in the 100-200 Hz band. An equivalent noise reduction effect was also verified when the FB-ANC composition was applied to lumped parameter portion $P_c(s)$ of the nominal plant. This verified the validity of the PID controller designed using the physical model, and the validity of the proposed Smith predictor-type FB-ANC configuration.

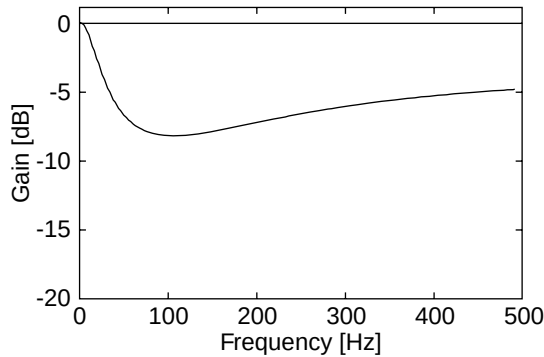


Fig.9 Sensitivity function for a physical model

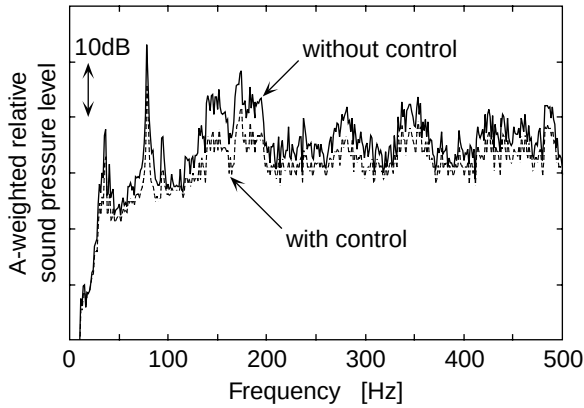


Fig.10 Sound pressure level for a physical model

6.2. Controller Based on a Gray-box Model

Next, the control performance of the controller designed with the gray-box model is compared with that of the controller designed with a physical model as described in the previous section. The transfer function of the PID controller designed with the gray-box model is represented by the following equation.

$$C(s) = 0.15 + \frac{s}{1 + s/(2\pi \cdot 500)} (1 \times 10^{-5}) + \frac{700}{s} \quad (15)$$

The sensitivity function is shown in Fig.11; the results of the simulation of sound pressure prediction, in Fig.12. These results demonstrate that noise reduction in the 100-200 Hz band is superior to that of the controller designed with the physical model. This confirms the validity of plant models obtained with gray-box modeling with respect to noise reduction performance.

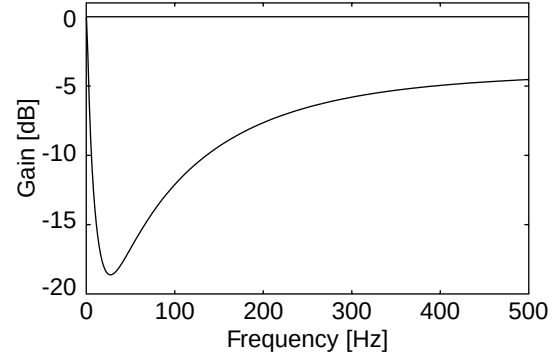


Fig.11 Sensitivity function for a gray-box model

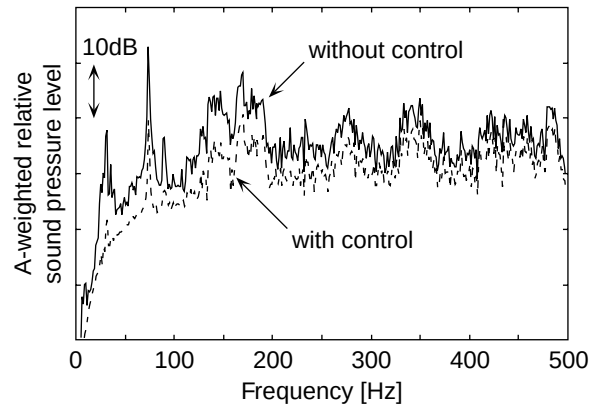


Fig.12 Sound pressure level for a gray-box model

7. Conclusion

Focusing on FB-ANC as a means of constructing low-cost ANC systems, this paper yields the following conclusions through the low-order modeling of a plant, and by designing a simple-structure controller with the results of said modeling:

- 1.) By using gray-box modeling as a means of high-precision low-order modeling, the plant is expressed with a second-order series and a time delay element.
- 2.) Smith predictor-type FB-ANC system eliminating time delay is proposed for controller design.
- 3.) By applying PID control to ANC, a simple, robust PID controller taking into account the uncertainty of time delay is designed. The validity of the plant model obtained and the controller designed is verified by predictive simulation of control performance using signals obtained from an actual vehicle.

We next plan to verify the proposed method's validity by implementing in an actual vehicle.

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Shuichi ADACHI

Reliability of Pb-free Soldered Joints for Automotive Applications

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ABSTRACT

Tests were conducted to evaluate the grain growth rate of the structure and the intermetallic compound layer fracture rate. These tests included the physical properties, as well as peeling, which is a mechanical property of Pb free soldered joints. In addition, thermal shock tests (acceleration tests) were performed to evaluate the life span as related to thermal fatigue.

First, the grain growth rate and intermetallic compound layer fracture rate of the Pb-free soldered structure were confirmed to follow the rule of parabolic curves. Next, the life span of Pb-free soldered joints under market conditions was predicted using Miner's rule of reliability engineering based on the effects of heat stress obtained through acceleration tests and the heat stress frequency in an automotive application.

1. Introduction

There are environmental concerns regarding the use of Sn-Pb solder. This has spurred much research toward a Pb-free solder. Although these global efforts have been led by America and Europe, such research has gained momentum in Japan, as well, with the recent passage of the Electrical Appliance Recycling Law (due to take effect in 2001). For instance, the Industrial Structure Council, of the Ministry of International Trade and Industry has also designated Pb reduction as a goal for automotive electronics and electrical equipment.

A Pb-free solder selected for automotive electronics and electrical equipment must maintain component functions in its respective utilization environment and for the duration of use. This, in turn, makes it necessary to fully understand the environment of the solder joints of automotive electronics and electrical equipment. The grain growth and intermetallic compound layer fractures that have been detected in the solder joints of automotive electronics and electrical equipment recovered after market use are said to have a significant impact on reliability, and so determination of mechanical strength alone (e.g., bulk material strength and joint tensile strength) is not sufficient for the reliability assessment of solder joints. There are few reports on reliability assurance against thermal fatigue for different Pb-free solder compositions from both the "micro" perspective of grain growth and intermetallic compound layer fracture as well as the "macro" perspective of mechanical properties.

The authors therefore studied the grain growth, intermetallic compound layer fractures, and mechanical strength of Pb-free solder, establishing testing conditions

that simulate the actual environments in which automotive electronic components are used, and using a failure model to calculate joint life.

2. Approach to the Reliability Assessment of Solder Joints

Automotive electronics and electrical equipment are used in extremely severe environments. The use of increasingly compact components has translated into narrower joint width. Automotive interior components, in particular, are simultaneously subjected to heat stress, vibration, humidity and other conditions. The most important aspect of solder joint reliability is thermal fatigue countermeasures. Solder has a low melting point, and it has a difference in coefficient of linear expansion between a substrate and a component that leads to considerable thermal stress, which, after repeated occurrence, can cause thermal fatigue failure. Pb-free solder is no exception even though it differs in composition from Sn-Pb solder and therefore also differs in melting point, grain growth rate, intermetallic compound layer fracture rate, and mechanical properties and similar, all of which have a large effect on solder joint life. The methods and standards for assessing components made with Pb-free solder are therefore particularly important. Assessment standards must focus on wear failure and degradation failure by providing testing conditions and testing times that simulate actual market environments, while the assessment method that must be constructed is one for indoor acceleration testing. As an example, the authors assess the reliability of bare chip die bonding.

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3. EXPERIMENT SAMPLES AND METHOD

3.1. Experiment Samples

Elements having low toxicity that are already used in actual solder applications were selected for the solder used in this experimentation. With a Sn base, eutectic Pb, Bi, Ag, Sb, and Zn, as well as individual elements in combination, were used to create the following solders: Sn-37Pb, Sn-3.5Ag, Sn-3.5Ag-0.75Cu, Sn-58Bi, Sn-2Ag-0.5Cu-7.5Bi, Sn-5Sb, and Sn-9Zn. These are the solders from which the samples for each test were selected. Sn-9Zn was not subjected to intermetallic compound layer fracture rate testing or thermal shock testing because its poor wettability with Ni makes it difficult to establish sample conditions. Furthermore, Sn-5Sb was not subjected to grain growth testing because the similar atomic numbers of Sn and Sb make it difficult to distinguish their phases by microscope.

3.2. Conditions for Intermetallic Compound Layer Fracture Rate Testing

Experimentation was carried out to determine how the fracture rate of the intermetallic compound layer fractures that form during solder melting and joining differs according to solder composition. Solder was exposed to a high temperature (120°C) to accelerate fracture growth rates. To identify the growth process, the solder was measured at the times shown in Table 1. Test pieces were made by soldering solder paste to Ni plates, after which the thickness of the intermetallic compound layer fracture formed on the joint interface was measured. As for the method of measurement, after high temperature exposure was applied for the predetermined period of time, a test piece was cut so that the cross-section of the solder would be perpendicular. The cross-section was then polished and thickness of the intermetallic compound layer fractures was measured by observing composition with element mapping and Backscattered Electron Image (BEI) made with an Energy Dispersive Spectrometer (EDS).

Table 1 Measurement time

Time (hr)	0	250	500	1000
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3.3. Conditions for Grain Growth Testing

Solder with JISA-grade impurity levels was used in all experimentation, the goal of which was to observe aging of the structure by accelerating the rate of aging by increasing temperature. Test pieces were made by dripping approximately 0.12 cc of flux (activated rosin) onto Ni-plated Cu sheets (17.5 × 17.5 × 1.75 t), then depositing 0.2 g of solder. The test piece was covered with paraffin paper and placed in a solder bath that was set to a temperature of 50°C over the solder's melting point and heated for approximately 30 seconds after the solder melted, thus completing the soldering process. After being air-cooled, the test pieces were washed with isopropyl alcohol to remove the remaining flux, followed by drying.

For testing, samples were placed in a thermostatic chamber maintained at 120°C and removed at different times

for observation. As Sn-58Bi has a low melting point (138°C) and its phase growth is faster than that of the other solders, samples were removed for microscopic observation (with a JEOL model JSM-5 310LV microscope) at ×1000 and ×3500 resolution at 0, 10, 20, 60, 120, 240, 1200, 3000, 12000, 24000, and 30000 minutes. Photographs taken during this observation were processed by linear analysis. An example is shown in Fig.1.

For instance, in the case of Sn-58Bi, a straight line of length L and parallel to one side of the observation results was drawn, and the size of the Bi phase was determined by dividing the ratio of Bi phases on that line by the number of Bi phases intersected.

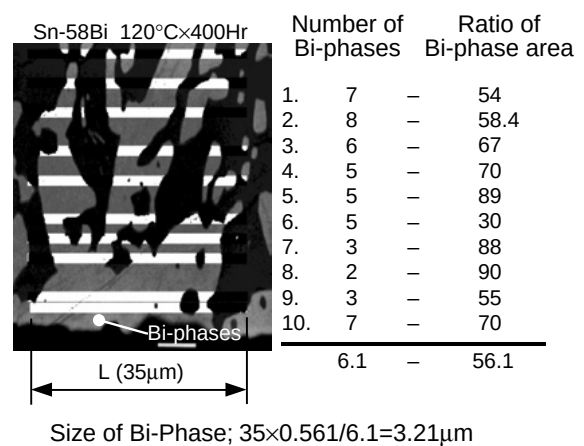


Fig.1 Linear analysis data

3.4. Conditions for Peeling Testing

Experimentation was also carried out to determine how the heating of solder joints would have different effects on joint strength according to differences in solder composition. Test pieces were made by placing solder cut from solder tape into 5 × 15 × 0.1 mm strips between sheet copper (Fig.2) and placing it into a solder bath for reflow. Reflow time was set at 20 seconds after solder melting, and solder bath temperature was set according to each solder's melting point: Sn-58Bi = 200°C, Sn-5Sb = 270°C, other solders = 250°C. The heating temperature was 120°C, and six heat exposure times were used to simulate aging (Table 2).

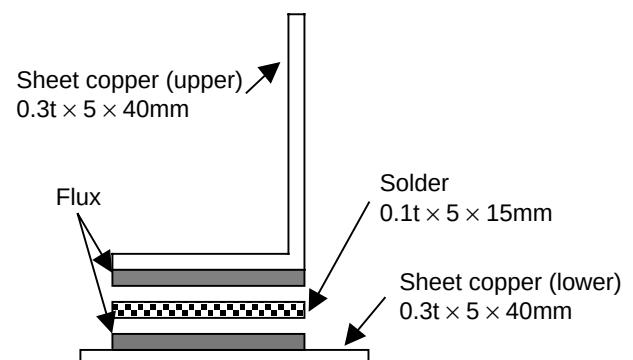


Fig.2 Test piece

Table 2 Heat exposure time

Time (min)	0	10	20	60	6000	12000	60000
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For testing, a tensile tester (Shimadzu AG-2000B) was used to perform peeling testing by placing the bent portion of each test piece into the tester's gripping device to subject the test piece to a tensile load. The pull rate was 10 mm/min⁽¹⁾. A constant load was used in this peeling testing (Fig.3). The point at which cracking occurred was defined as the end of life of the solder joint. Although peak load may seem more similar in actual conditions, a relatively stable constant load was used because of the effect on output of fillet shapes and other conditions.

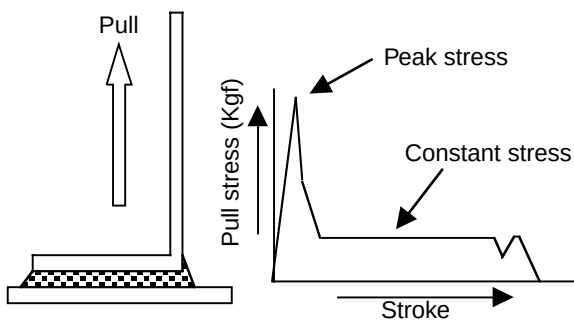


Fig.3 Pull test method

3.5. Conditions of Thermal Shock Testing

Experimentation was carried out to determine how the effect of thermal shock on solder joints differs according to solder composition. Bare chip die bonded test pieces were selected and an ultrasonic survey instrument was used to confirm sample quality (Fig.4). The three testing conditions applied simulated the conditions under which various automotive electronics and electrical equipment are actually used (Fig.5). In thermal shock testing, a thermal shock tester was used to determine and compare life of the joint. The ultrasonic survey instrument was used to observe the solder joints and determine the state of peeling at different times (Table 3).

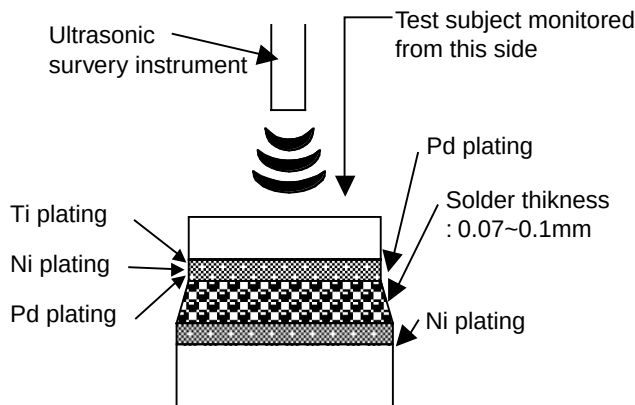


Fig.4 Specification of test subject

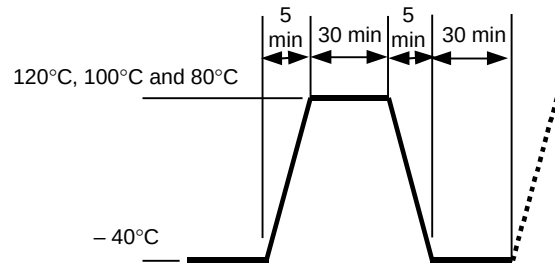


Fig.5 Conditions of thermal shock test

Table 3 Measurement time

Temperature	Time (hour)
$\Delta T 120^{\circ}\text{C}$	0, 505, 1000, 1500, 2000, 3770, 5000
$\Delta T 140^{\circ}\text{C}$	0, 406, 1000, 1732, 3000, 4000, 5000
$\Delta T 160^{\circ}\text{C}$	0, 100, 240, 537, 1060, 1533, 2000, 3000

4. Results of Intermetallic Compound Layer Fracture Rate Testing

Intermetallic compound layer fracturing occurs because of the formation of intermetallic compounds by Sn and other solder components due to mutual diffusion between these metals. This intermetallic compound layer fracturing occurs both during solder melting and joining and during post-joining high temperature exposure, making the joint brittle and creating Kirkendall void and cracks. Automotive electronics and electrical equipment, in particular, are subjected to repeated heating/cooling cycles, and require adequate consideration for the lowered reliability that can result when the thermal stress resulting from the materials' different coefficients of expansion causes joint cracks, thus having a major negative impact on mechanical strength. Fig.6 is an example of intermetallic compound layer fracturing.

All of the solder used in this testing underwent Sn-Ni intermetallic compound layer fracturing, the thickness of which varied depending on the type of solder. Fig.7 also suggests the occurrence of Sn-Cu intermetallic compound layer fracturing in Sn-3.5Ag-0.75Cu and Sn-2Ag-0.5Cu-7.5Bi. The rate of Sn-Ni intermetallic compound layer fracture is shown in Fig.8. This shows that in Pb-free solders, as in Sn-Pb solders, the intermetallic compound layer fracture rate is proportional to the square root of time, thus obeying the parabolic law⁽²⁾.

$$X = k\sqrt{D \cdot t}$$

where layer thickness = X, holding time = t, constant = k, diffusion constant = D.

The compositions exhibiting the slowest intermetallic compound layer fracture rate were Sn-3.5Ag-0.75Cu and Sn-2Ag-0.5Cu-7.5Bi, in both of which, as Fig.7 also shows, Cu is concentrated at the interface, suggesting that the addition of Cu tends to retard the growth of Sn-Ni intermetallic compounds⁽³⁾.

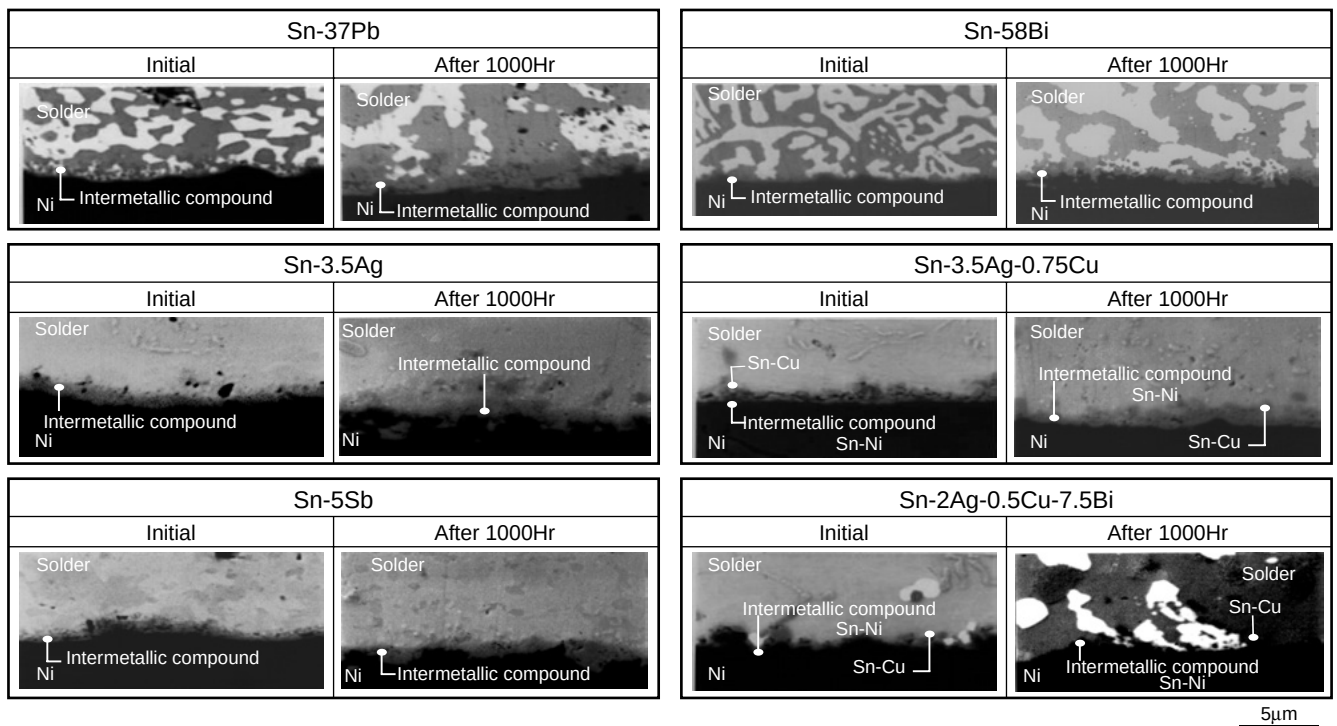


Fig.6 SEM image of intermetallic compound layer

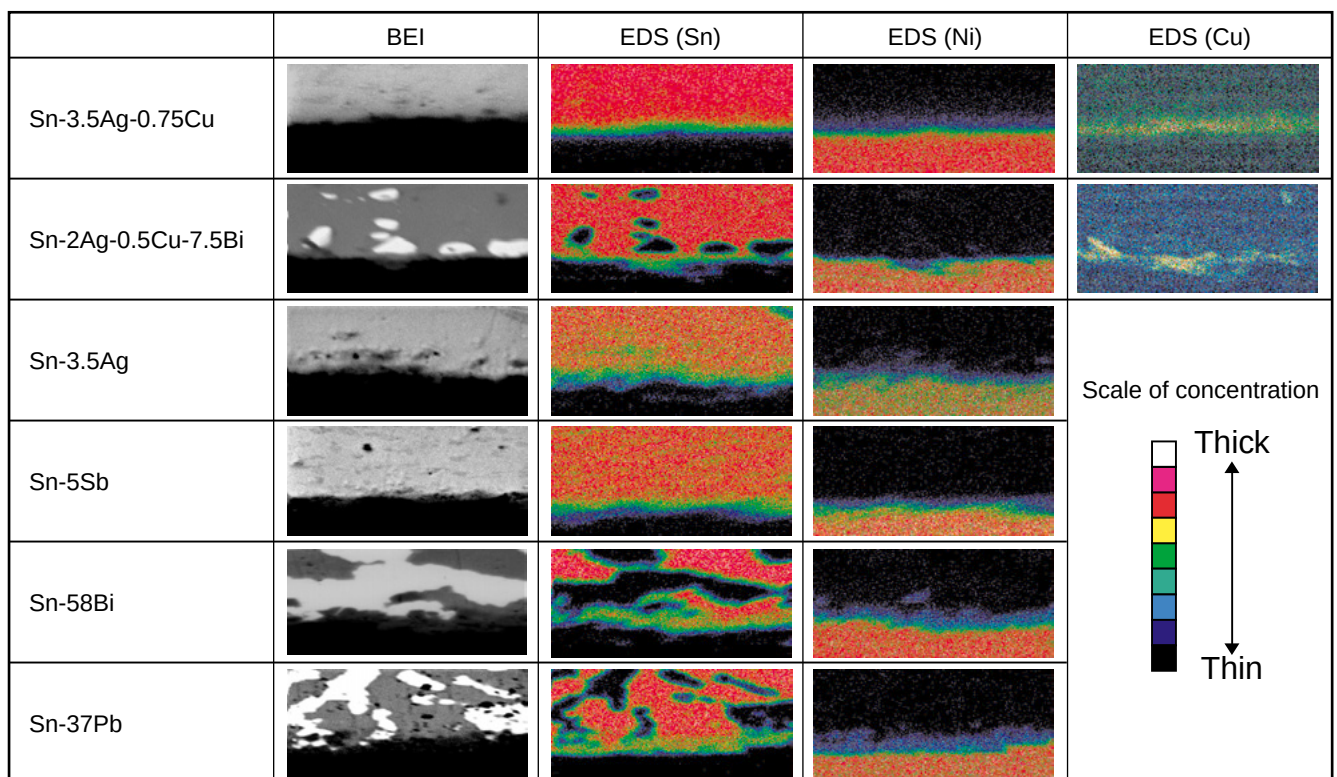


Fig.7 Distribution maps of the intermetallic compound layer of each element

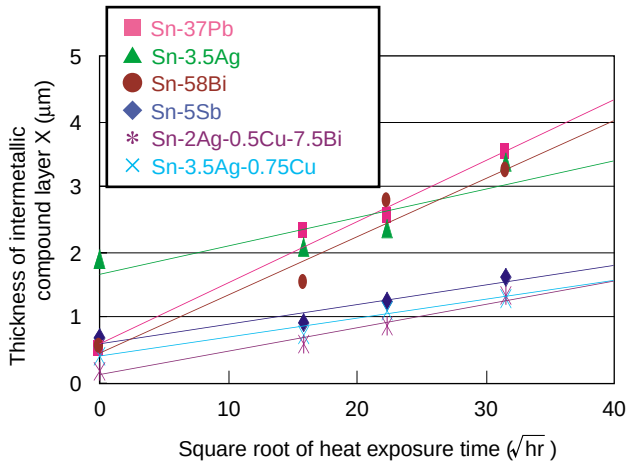


Fig.8 Growth pattern of the intermetallic compound layer

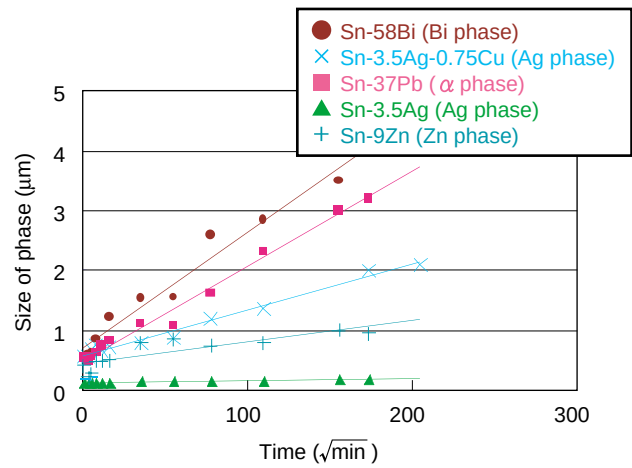


Fig.9 Change in phase size due to aging of each solder

5. Results of Grain Growth Testing

As in Sn-Pb solder, grain growth in Pb-free solder is due to a change in each phase's solid solubility limit density and surface energy, and has been shown to progress so as to decrease the free energy of phase equilibrium⁽⁴⁾ at a rate that obeys the parabolic law (Fig.9). The fastest grain growth rate was exhibited by a Sn-Bi solder, Sn-58Bi; the slowest, by a Sn-Ag solder, Sn-3.5Ag. As in the phase diagrams for these Pb-free solders, a greater change in solid solubility

limit density at a given temperature tends to be accompanied by a faster rate of grain growth. In addition, Sn-2Ag-0.5Cu-7.5Bi exhibited peculiar structural change (Fig.10). After 10 minutes of high temperature exposure at 120°C, the Bi diffused throughout the Sn matrix, resulting in voids where the Bi had been. Further high temperature exposure was shown to enlarge or eliminate existing voids and create new voids. Sn-2Ag-0.5Cu-0.5Bi was omitted from the plot in Fig.9 as its Bi phase has been eliminated.

As for the effects of grain growth on solder joints, because

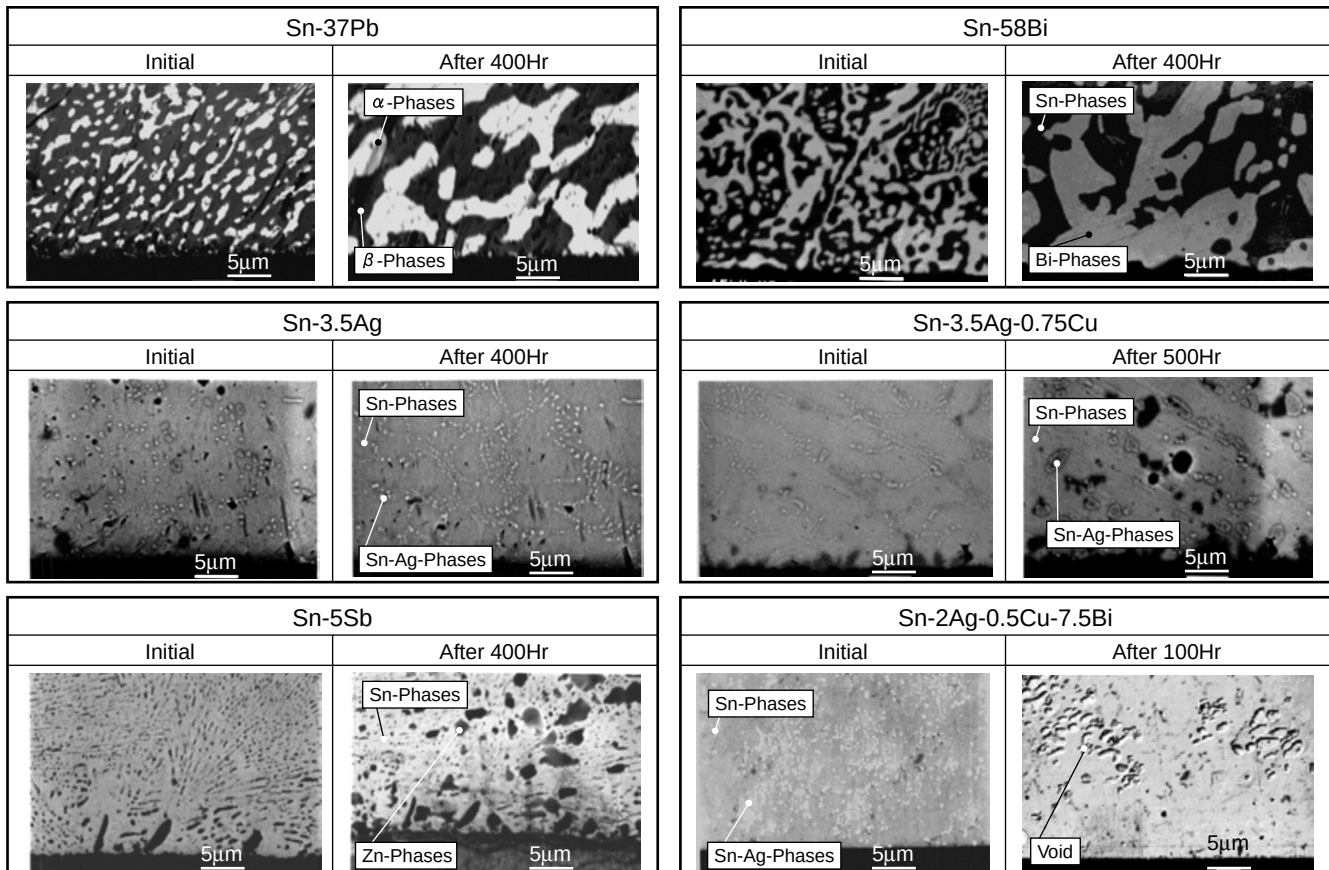


Fig.10 Aging of Pb-free soldered area relative to temperature

solder joint peeling due to thermal shock is the result of the formation and progression of microcracks caused by local plastic deformation (i.e., sliding deformation of dislocation), which in turn is caused by repeated stress at the site where maximum stress is concentrated, peeling in the solder interior is also most likely due to grain growth. Degradation of the solder joint strength due to grain growth is generally thought to obey the law of Hall-Petch⁽⁵⁾.

The relationship between phase (i.e., crystal particle) size and strength is expressed in terms of strength ρ thus, where D is average phase diameter:

$$\rho = k \sqrt{D}$$

This shows how greater phase growth through high temperature exposure decreases strength and makes peeling more likely.

6. Results of Peeling Testing

Fig.11 shows that high temperature exposure of 500 hours at 120°C caused Sn-9Zn, a Sn-Bi solder, to undergo a marked drop in solder load as determined by measurement of peeling load. As for the drop in load of Sn-2Ag-0.5Cu-7.5Bi, the high temperature exposure caused the Sn-Cu intermetallic compound layer to increase in thickness, causing Bi in the solder to remain on the surface between the intermetallic compound and the solder. Eventually, Bi density on the surface increases to the point where brittle peeling occurs. One factor in the decreased load of Sn-58Bi is believed to be the marked effect of Bi cleavage, a result of grain growth. The sharp decline in the peeling load of Sn-9Zn is believed to be due to the formation of Kirkendall void because of the high diffusion rates of Cu and Zn⁽³⁾. In contrast, Sn-Ag solder remained stable, showing little change in load. This is believed to be because of the slow growth of the Sn-Cu intermetallic compound, and the resultant few factors degrading the intermetallic compound/solder interface.

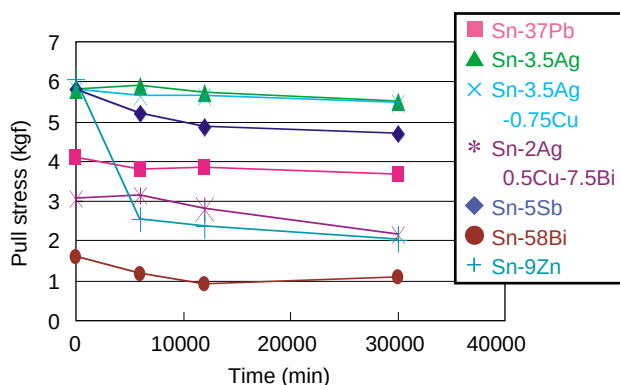


Fig.11 Pull strength of connection point after high temperature exposure

7. Results of Peeling Testing

The results of thermal shock testing (Fig.12) and peeling testing (Fig.13) verified that peeling starts with intermetallic compounds in the case of Sn-58Bi and Sn-2Ag-0.5Cu-7.5Bi, which are Bi solders, but in the solder interior in the case of all other solders. This is believed to be because of the extreme hardness and brittleness of Sn-58Bi and Sn-2Ag-0.5Cu-7.5Bi, which does not elongate and which, consequently, is

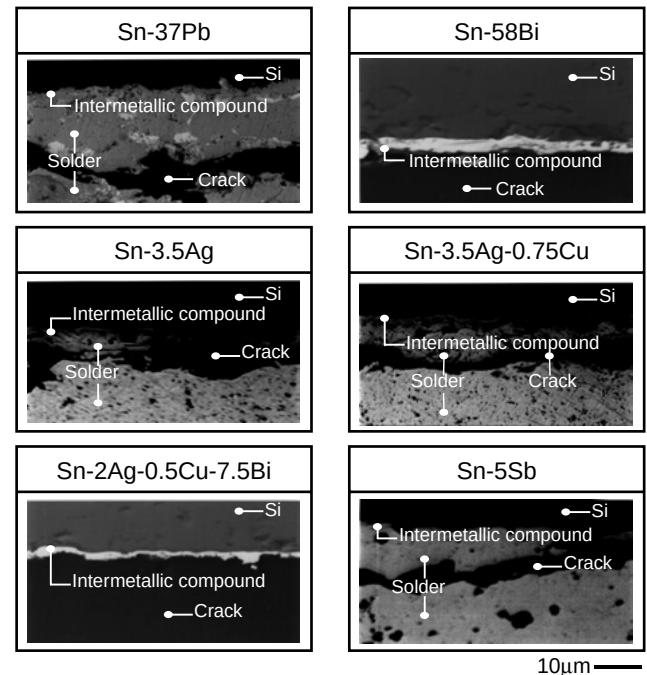


Fig.12 SEM image of cracks after thermal shock (under Si chip)

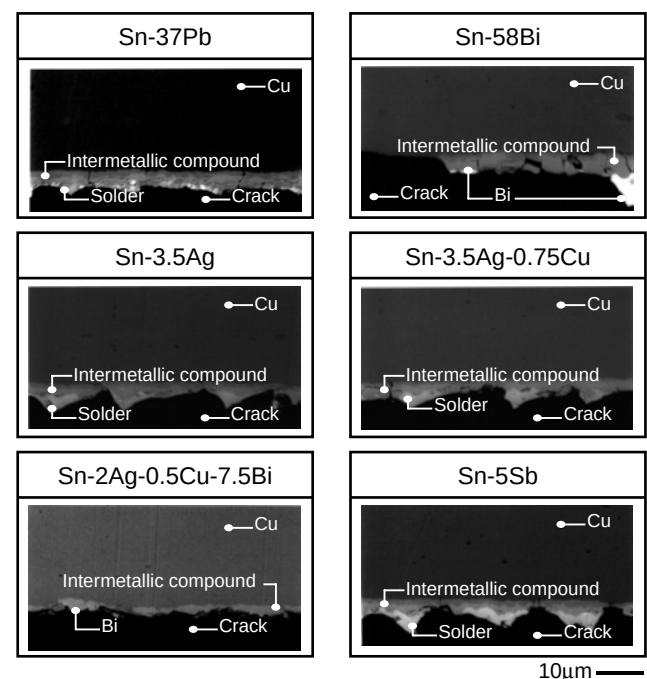


Fig.13 Peel strength of soldered area due to aging

unable to withstand distortion at the intermetallic compound/solder surface during thermal expansion, and so begins to peel at the surface where strength is lowest. In addition, intermetallic compound growth rates (i.e., change rates) are high in the case of both solders. Therefore peeling most likely began at the intermetallic compound surface. Other solders, in contrast, have a low intermetallic compound growth rate (i.e., rate of change) and are softer than Sn-Bi solders with respect to elongation. Thus no peeling occurs at the intermetallic compound/solder surface. Instead, it begins in the solder interior, where strength is lowest.

8. Prediction of Solder Joint Lifespan

8.1. Heat History in Automotive Applications

Thermal fatigue is the most important issue in the reliability of the solder joints of automotive electronics and electrical equipment. This is because these solder joints can ultimately undergo thermal fatigue failure due to the thermal stress and thermal distortion caused by repeated self-heat generation by the elements themselves, cooling, the manner and frequency of use, and seasonal changes in environmental temperature.

8.2. Thermal Fatigue Life Diagram

The thermal fatigue failure of a solder is believed to be compound fatigue/creep failure in which fatigue failure, the primary condition, causes creep to accelerate. Generally, the relationship expression for the thermal fatigue life (N_f) of solder joints is expressed with the Coffin-Manson empirical equation:

$$N_f \propto C \cdot f^m \cdot (\Delta \epsilon)^{-n} \cdot \exp(Q / k \cdot T)$$

where C = material constant; m , n = exponents, f = cycle frequency, $\Delta \epsilon$ = plastic deformation, Q = activation energy, k = Boltzmann constant.

If C and f are constant and the creep term is coefficient K , then

$$N_f = a \cdot K \cdot (\Delta \epsilon)^{-n}$$

Taking the logarithm of both sides results in the following equation

$$\begin{aligned} \log N_f &= -n \log \Delta \epsilon + \log a \cdot K \\ y &= -n \cdot X + a \cdot K \end{aligned}$$

which results in a straight line on the log-log graph.

8.3. Calculation of Expected Life

In the mechanical design of a solder joint, not only the material strength but also the S-N curve, which indicates the relationship between stress and the number of cycles, is also important. A typical method for predicting joint life with an S-N curve is Miner's rule, which states that when cyclic stress S_n (a varying stress) is applied n_t times (i.e., $S_1 = 1$ time, $S_2 = 2$ times,...), and N_t is the life cycle when stress S_t is applied, then

$$\sum_{t=1}^k n_t / N_t = 1$$

is the cumulative fatigue resulting from this varying stress⁽⁶⁾.

Thermal shock testing with different values of ΔT was carried out to determine N_t , the life being defined as the cycle number at which peeling equal to a certain surface area or beyond occurred. Fig.15 is the life diagram for different values of ΔT . A typical example of the thermal shock testing results for $\Delta T = 140^\circ\text{C}$ is given in Fig.14. The stress frequency to which components are subjected in market conditions is also listed in Fig.15 to account for utilization frequency and season changes and such⁽⁷⁾. Table 4 gives the results of marketed Pb-free solder joint life predictions made with these results and by applying Miner's law.

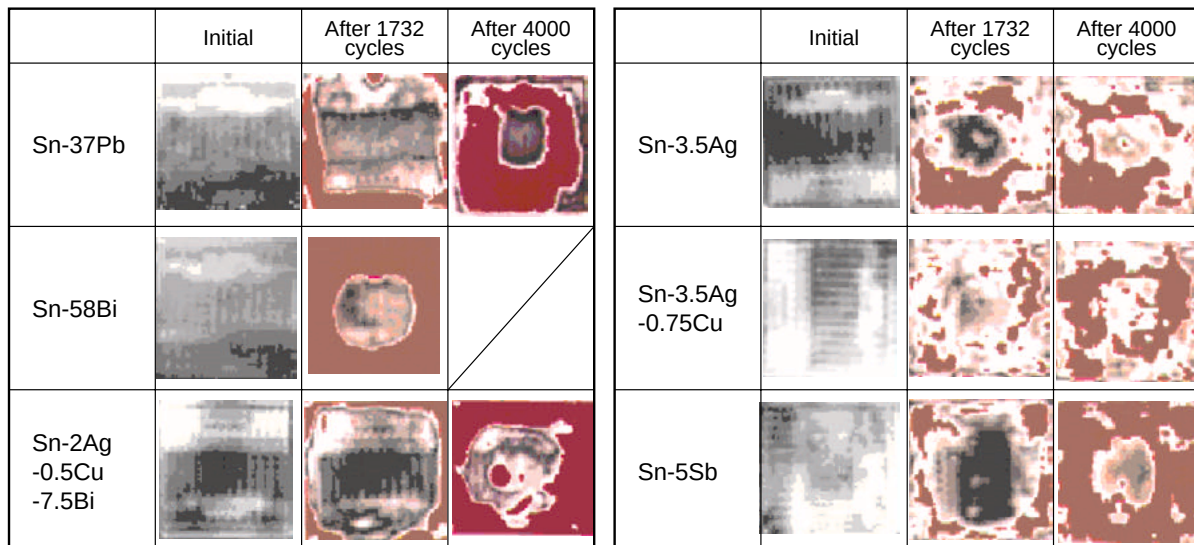


Fig.14 Cracks in soldered area due to heat impact (Red areas represent cracks)

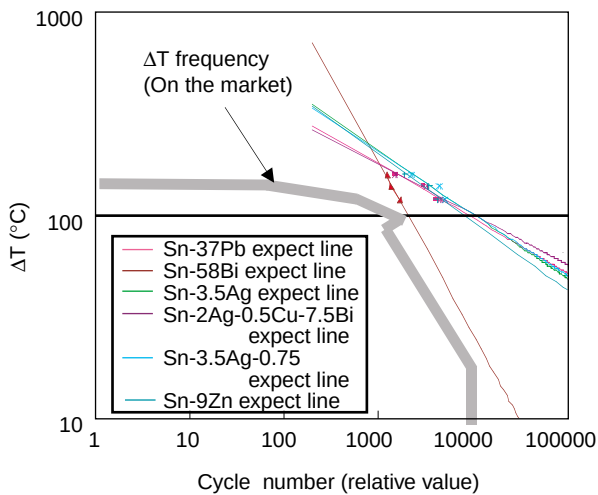


Fig.15 Requirement line for thermal fatigue

Table 4 Predicted life of solder joint

	Sn-37Pb	Sn-58Bi	Sn-3.5Ag	Sn-2Ag-0.5 Cu-7.5Bi	Sn-3.5Ag -0.75Cu	Sn-5Sb
Time to failure (N)	0.93	4.57	0.79	0.95	0.85	0.90

The results show all solders other than Sn-58Bi to have a life equal to or greater than that of Sn-37Pb.

9. Conclusion

The findings of this research are as follows:

- (1) The intermetallic compound layer fracture rate and grain growth rate of Pb-free solder was shown to obey the parabolic law, being proportional to the square root of time.
- (2) The results of the thermal fatigue life prediction of Pb-free solder joints in automotive applications showed all solders other than Sn-58Bi to have a life comparable to that of Sn-37Pb.
- (3) Miner's law, which is concerned with reliability engineering, can be used in the life prediction of Pb-free solder joints.

The authors plan to perform similar life prediction for different solder compositions, different methods of soldering, and different environments (e.g., heat, thermal shock, and vibration) as a next activity.

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Application of the Magic Formula Tire Model to Motorcycle Maneuverability Analysis

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ABSTRACT

The Magic Formula Tire Model, recently utilized in maneuverability analysis for automobiles, was applied to a motorcycle simulation model. The correlation between the simulated and measured characteristics for straight running stability and turning performance was compared with those of the current Carpet Plotted Tire Model. Further, the ease of use of the Magic Formula was investigated. The results show that correlation with actual tire characteristics is high for the Magic Formula Tire Model and, further, that the changing of tire properties can be easily accomplished with this model.

1. Introduction

Tire properties are an important determinant of motorcycle maneuverability, and tire models are crucial to behavior simulations employing a mechanical analysis language. The Carpet Plotted Tire Model (CP Tire Model) has traditionally been used as a tire model for expressing properties that include the nonlinear effects of the large camber angle characteristic of motorcycles.

For the CP Tire Model, Linear interpolation methods are used to smoothly express tire properties from discrete measurement data. However, using small increments in interpolation to assure accuracy not only requires processing huge amounts of data, but also makes it difficult to change tire properties.

In this paper, the application of the Magic Formula Tire Model (MF Tire Model) to motorcycles is discussed. Specifically, the correlation between simulation and actual measurement in straight running stability and turning performance, and the efficiency of changing tire properties are discussed.

2. A Tire Model for Motorcycles

Motorcycles have different tire slip angles and camber angles during running than automobiles (Fig.1) ⁽¹⁾⁻⁽³⁾.

They therefore require a tire model that is capable of assuring accurate representation even with large camber angles.

2.1. The MF Tire Model

Lateral force (F_{y0}) in the motorcycle version of the MF Tire Model, which consists of lateral force and camber thrust, is shown in equation (1) described below. The second term in equation (1) is the equation for camber angle. In equation (1), α represents tire slip angle, γ camber angle, and F_z

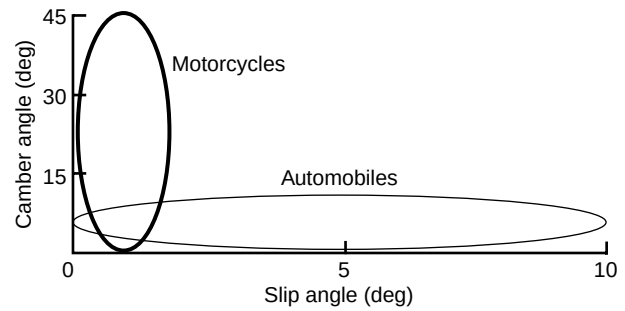


Fig.1 Comparison of motorcycle tires with automobile tires for tire slip angle and camber angle

vertical load. As with the MF equation used for automobiles, the tire model's lateral force F_{y0} is determined by the "MF coefficients" B_y , C_y , D_y , E_y , B_γ , C_γ , E_γ , and Sh_y , and by variables indicating a tire's condition: vertical load, tire slip angle, and camber angle. Values p_0 through p_{17} and γ_0 through γ_9 , which are determined by smoothing measurement data, are termed "MF parameters" and are used to represent differences in tire properties.

$$F_{y0} = D_y \sin[C_y \arctan\{B_y a_y - E_y (B_y a_y - \arctan(B_y a_y))\} + C_\gamma \arctan\{B_\gamma \gamma_y - E_\gamma (B_\gamma \gamma_y - \arctan(B_\gamma \gamma_y))\}] \quad (1)$$

$$\alpha_y = a + S_{hy} \quad (2)$$

$$C_y = p_0 \lambda_0 \quad (3)$$

$$D_y = p_1 \exp(p_2 df_z) (1 - p_3 \gamma_y^2) \lambda_1 F_z \quad (4)$$

$$E_y = \{p_4 + p_5 \gamma_y^2 + (p_6 + p_7 \gamma_y) \operatorname{sgn}(\alpha_y)\} \lambda_2 \quad (5)$$

$$K_y = p_8 F_{z0} \sin[p_9 \arctan\{F_z / (p_{10} + p_{11} \gamma_y^2) F_{z0} \lambda_3\} (1 - p_{12} \gamma_y^2) \lambda_3 \lambda_4] \quad (6)$$

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$$B_y = K_y / (C_y D_y) \quad (7)$$

$$S_{hy} = p_{13} \lambda_5 \quad (8)$$

$$C_g = p_{14} \lambda_6 \quad (9)$$

$$K_\gamma = (p_{15} + p_{16} df_z) F_z \lambda_7 \quad (10)$$

$$E_\gamma = p_{17} \lambda_8 \quad (11)$$

$$B_\gamma = K_\gamma / (C_\gamma D_\gamma) \quad (12)$$

where

F_{z0} : Nominal load

$$df_z = F_z - F'_{z0} / F'_{z0}$$

$$F'_{z0} = F_{z0} l_9$$

Total aligning moment (M_{z0}) is represented by equation (13). Again, the MF coefficients are B_t , C_t , D_t , E_t , B_r , D_r , S_{ht} , and S_{hr} ; the MF parameters are q_0 through q_{25} , γ_{10} , and γ_{11} .

$$M_{z0} = -D_t \cos[C_t \arctan\{B_t \alpha_t - E_t(B_t \alpha_t - \arctan(B_t \alpha_t))\}] \cos(\alpha) F_{y0} (\gamma=0) + D_r \cos[\arctan(B_r \alpha_r)] \cos(\alpha) \quad (13)$$

$$\alpha_t = \alpha + S_{ht} (S_{ht} = 0) \quad (14)$$

$$\alpha_r = \alpha + S_{hr} \quad (15)$$

$$\gamma_z = \gamma \lambda_{10} \quad (16)$$

$$B_t = (q_0 + q_1 df_z + q_2 df_z^2)(1 + q_3 \gamma_z + q_4 |\gamma_z|) \lambda_4 / \lambda_1 \quad (17)$$

$$C_t = q_5 \quad (18)$$

$$D_t = F_z (q_6 + q_7 df_z)(1 + q_8 |\gamma_z| + q_9 \gamma_z^2) (R_0 / F_{z0}) \lambda_1 \quad (19)$$

$$E_t = (q_{10} + q_{11} df_z + q_{12} df_z^2) \{1 + (q_{13} + q_{14} \gamma_z) (2 / \pi) \arctan(B_t C_t \alpha_t)\} \quad (20)$$

$$B_r = q_{15} \lambda_4 / \lambda_1 \quad (21)$$

$$D_r = F_z (q_{16} + q_{17} df_z + (q_{18} + q_{19} df_z) \gamma_z + (q_{20} + q_{21} df_z) \gamma_z |\gamma_z|) R_0 \lambda_{11} \lambda_1 \quad (22)$$

$$S_{hr} = q_{22} + q_{23} df_z + (q_{24} + q_{25} df_z) \gamma_z \quad (23)$$

where

R_0 : Unloaded tire radius

λ_i : Pneumatic trail

γ_z : Scaled camber angle

2.2. The CP Tire Model

Spline interpolation was performed with the data for tire lateral force and total aligning moment, which were obtained from different combinations of vertical load, tire slip angle, and camber angle using a tire tester. Fig.2 is an example of the resultant spline displayed on a curved surface. Tire slip angle and vertical load were used to calculate, for each camber angle, tire lateral force and total aligning moment

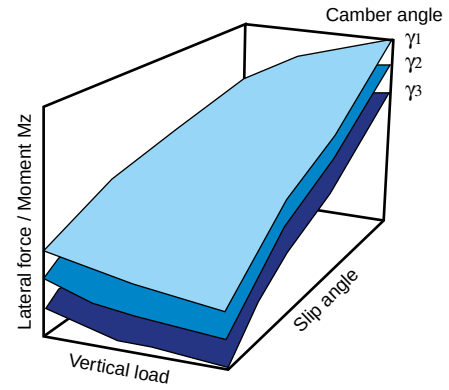


Fig.2 Surface planes of tire properties

on the data surface. In the event that no data surface exists for a corresponding camber angle, the program was designed to perform linear interpolation between the previous and next surfaces.

3. The Simulation Model

The motorcycle model used in the simulation is comprised of eight elements (Fig.3): front tire, rear tire, sprung steering components, unsprung steering components, engine, main body, rider, and rear swing arm. There are nine degrees of freedom of movement: six degrees in the main body, one degree in the steering, and two for the front and rear suspension. The front and rear suspension has nonlinear springs and speed-dependent damping units. The rider is secured to the main body, so only the rider's mass is taken into account. The rigidity of the frame, steering system, and rear swing arm are not taken into account. The running simulation assumes open loop running with no feedback control of the steering system by the rider.

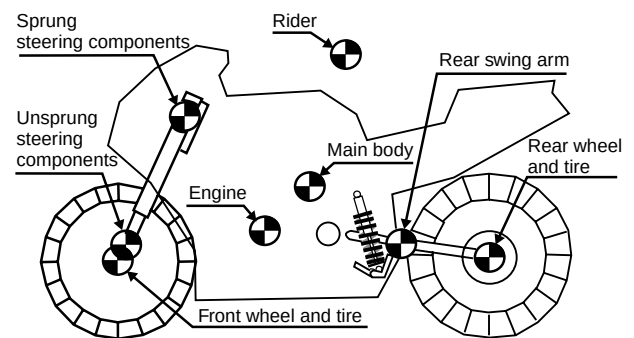


Fig.3 Motorcycle simulation model by multibody analysis language

4. Tire Properties

Tire properties were measured using the flat plank tire tester shown in Fig.4.

The basic data measurement for the tire model used in the simulation was three vertical loads, camber from -5 degrees to 45 degrees; and, tire slip angle, -10 degrees.



Fig.4 Flat plank tire tester

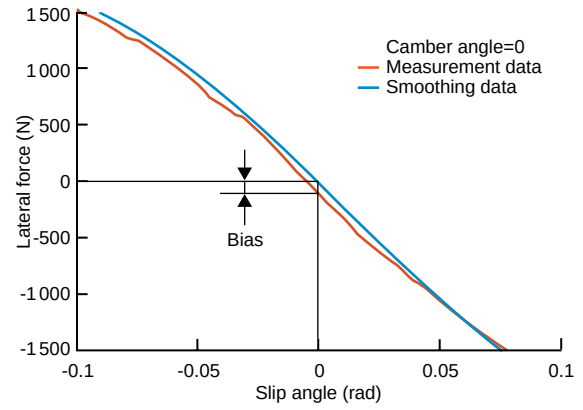


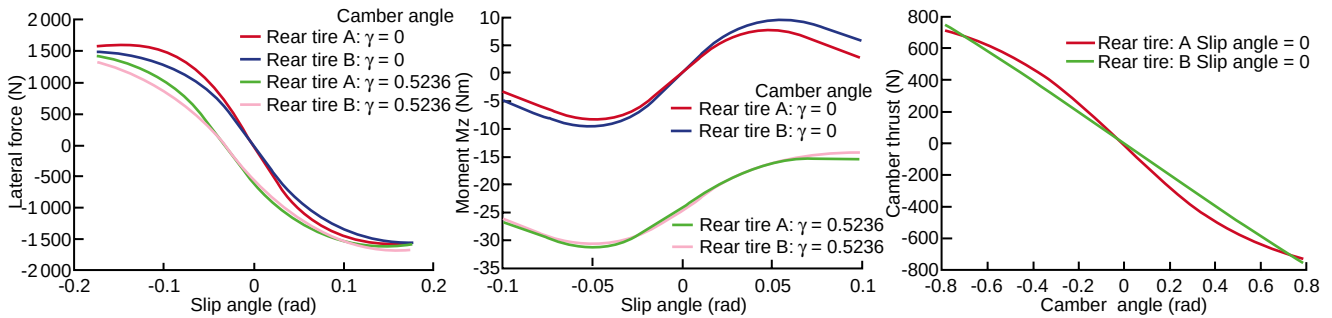
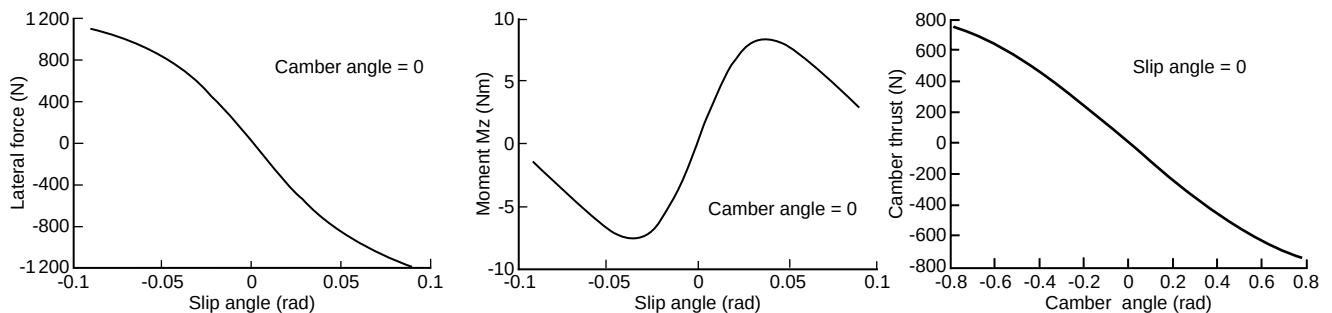
Fig.5 Comparison of tire measurement data before and after smoothing

The results of simulation might differ from prediction when a bias exists in the measurement data.

An example of this would be when tire lateral force or total aligning moment occurs in the straight running state (i.e., camber angle and tire slip angle are both 0 degrees), particularly when operating at lower motorcycle velocities in the simulation. In such cases, this bias is removed in the smoothing stage of the MF Tire Model, wherein the MF parameters are determined. Fig.5 shows measurement data and a post-smoothing curve. The horizontal axis is the tire slip angle; the vertical axis, tire lateral force. This shows that bias in the measurement data is no longer present after smoothing. In the CP Tire Model, measurement data is approximated directly with a spline function.

Then, the difference of properties between two types of

rear tires (A, B) obtained from the measurements is explained. In the graph, tire slip angle is shown on the horizontal axis while tire lateral force and total aligning moment are shown on the vertical axis. The curve representing the lateral force of Tire A has a sharp incline during straight running, i.e., both camber angle and tire slip angle are zero degrees (Fig.6). Near 30 degrees of camber angle there are differences in the absolute value of lateral force. Therefore, Tire A exhibits larger lateral force. During straight running, the curve indicating the total aligning moment of Tire B has a large slope, but there is no significant difference near 30 degrees of camber angle. This shows that the camber thrust stiffness of Tire A is larger than that of Tire B. Fig.7 indicates the front tire's lateral force, total aligning moment, and camber thrust.


 Fig.6 Characteristics of rear tires for lateral force, moment M_z (aligning torque and camber torque), and camber thrust

 Fig.7 Characteristics of front tire for lateral force, moment M_z (aligning torque and camber torque), and camber thrust

5. A Comparison of Running Simulation and an Actual Running Test

5.1. Test Method

Table 1 lists the specifications of the test motorcycle, which was equipped with a steering angle sensor on the steering column; a steering force sensor inside the handlebar pipe, 2-kg (mass) yaw and roll gyro-sensors on the fuel tank, and a 2-kg data recording system on the passenger seat.

A front tire having the properties shown in Fig.7 was used. Two different rear tires, one for each test mode, were used in the straight running stability test. The tire combinations used for this actual running test are shown in Table 2. The body behaviors recorded were steering torque, steering angle, yaw rate and roll angle. A pulsating steering torque was input at 165, 185 and 200 km/h. During the

circular turning test, the rider approached a constant 45m radius at constant velocity. The rider leaned with the motorcycle until steady state turning was achieved. The tire combinations are "Test 2" indicated in Table 2. The range of the test velocities was 40 to 60km/h by 5km/h increments.

5.2. Simulation Results

5.2.1. Straight Running Stability

A comparison between the actual results, the CP Tire Model, and the MF Tire Model has been undertaken under the test conditions of Test 1 and Test 2 illustrated in Table 2. In these tests, yaw rate peak gain relative to steering torque, damping resonance frequency, and yaw rate damping ratio are presented as changes relative to velocity in Figs.8 and 9 respectively.

From Test 1, as velocity is increased, both the CP Tire Model and the MF Tire Model have the tendency of low damping ratio, although there are only slight changes in the damping resonance frequency. This tendency corresponds with the actual measurements.

A comparison between Tire A and Tire B reveals corresponding trends. Tire A shows a higher damping ratio and damping resonance frequency than Tire B having lower lateral force, thereby providing high stability.

The absolute values for yaw rate peak gain relative to steering torque are slightly different from the measured values, but show a corresponding trend.

In the comparison between Tire A to Tire B, it is shown that Tire A with large lateral force capability has a smaller gain than Tire B.

The increase in the lateral force capability of the rear tire generally leads to the higher stability.

Therefore, these results are considered appropriate ⁽²⁾⁽⁴⁾.

Table 1 Test motorcycle specifications

Items	Specifications
Wheelbase (m)	1.401
Caster angle (deg)	23.7
Caster (m)	0.095
Height of center of gravity (m)	0.537
Curb Weight (kg)	204.0

Table 2 Tire specifications for actual running test

	Test 1	Test 2
Front tire	130/70ZR16	130/70ZR16
Rear tire	Tire A: 190/50ZR17	Tire B: 180/55ZR17

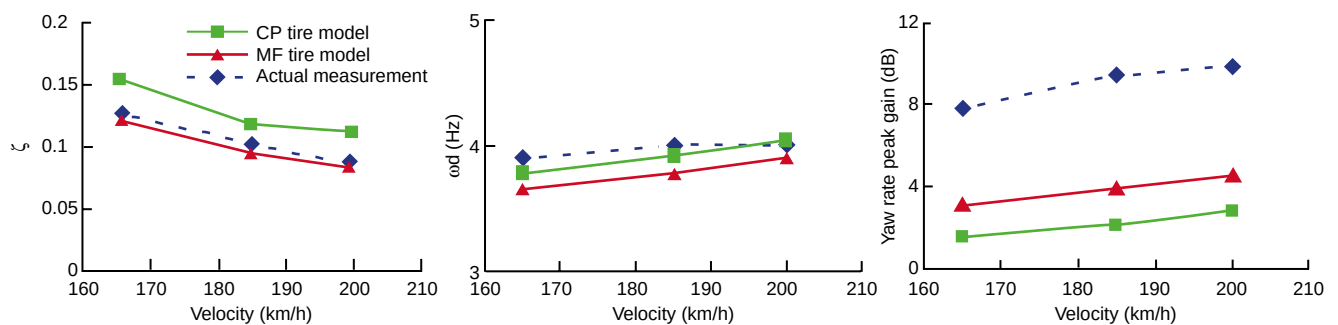


Fig.8 Comparison between simulation and experiment for damping properties in Test 1

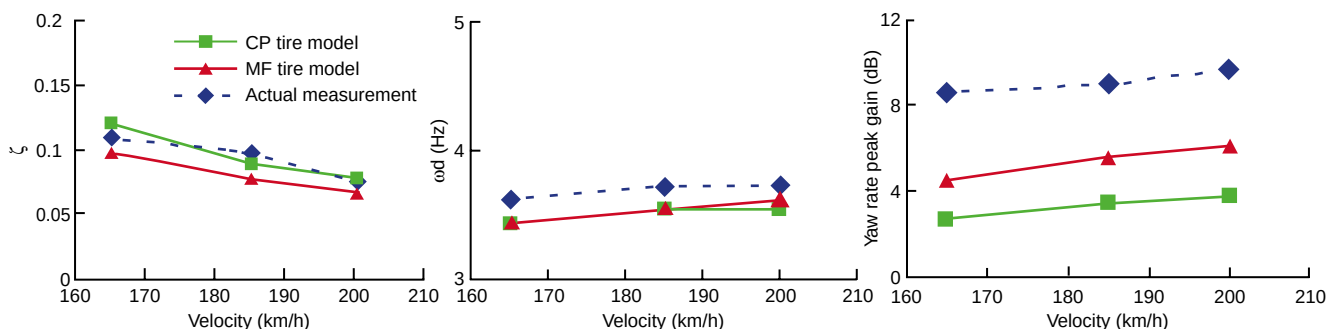


Fig.9 Comparison between simulation and experiment for damping properties in Test 2

The time domain data for steering torque, steering angle, and yaw rate in Fig.10 also indicates that the MF Tire Model is closer to the actual running results than the CP Tire Model.

5.2.2. Turning Performance

Fig.11 shows time domain data for steering torque, steering angle, yaw rate, and roll angle obtained in a circular turning simulation performed with the CP Tire Model and the MF Tire Model. Lateral acceleration was 0.5 G. The tendency and absolute values of steering angle, yaw rate, and roll angle indicated by both tire models correspond to the actual measurements. The steering torque and steering angle relative to lateral acceleration are shown in Fig.12. The tendency and absolute values of steering angle closely correspond to the actual measurement. Hence, the CP and MF Tire Models closely predict the actual measurements.

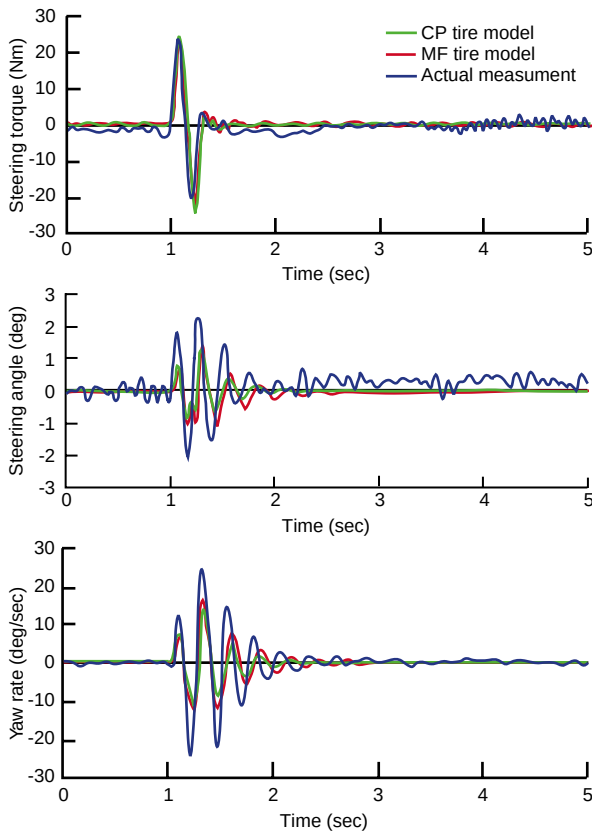


Fig.10 Comparison between simulation and experimental steering torque, steering angle and yaw rate

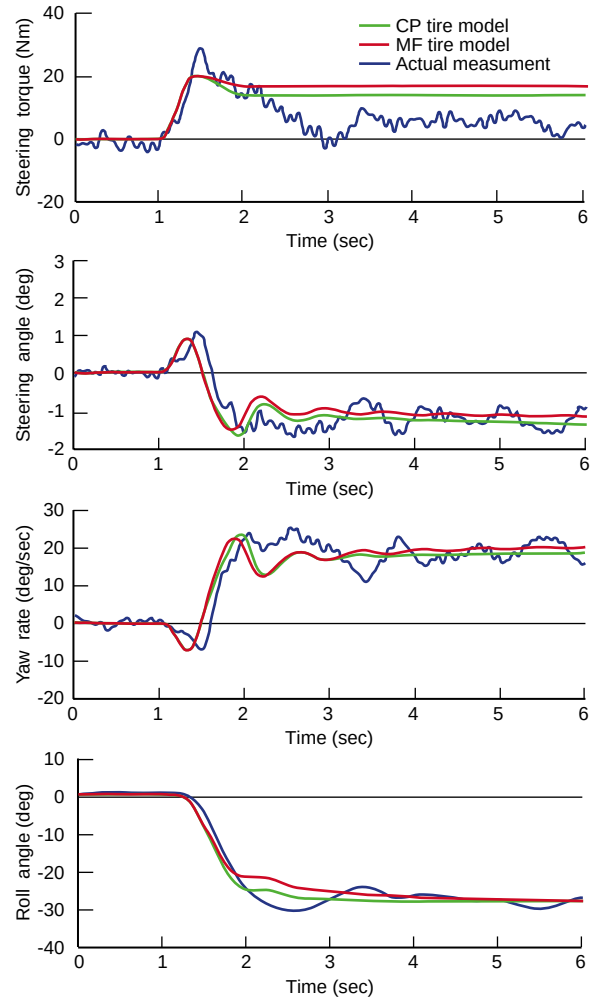


Fig.11 Comparison between simulation and experimental steering torque, steering angle, yaw rate and roll angle

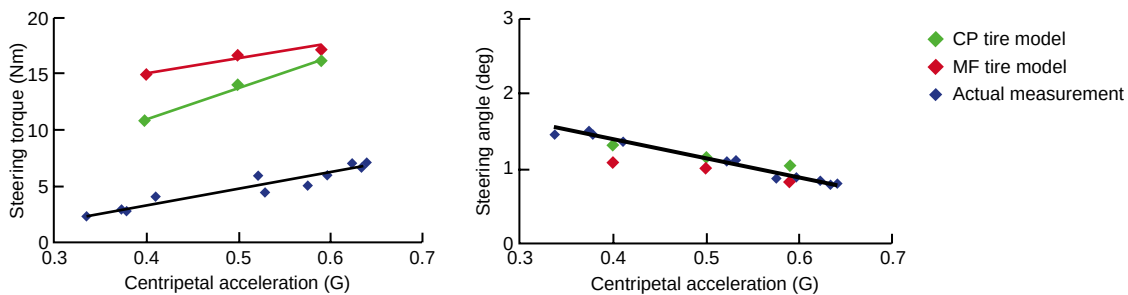


Fig.12 Comparison between simulation and experimental steady state turn

There is, however, a significant difference in steering torque properties. This is most likely due to the fact that the simulation model assumed open loop control, with no feedback control of the steering system. It is also necessary to consider a rider model that accounts for steering operations and individual differences among riders and so forth.

As for the applicability of the CP and the MF Tire Models to motorcycle simulation models, it takes almost the same period of time to convert the same amount of tire measurement data into the individual data format. However, looking at the ease of analysis, the MF Tire Model is more efficient in tire changes and parametric studies.

6. Conclusion

Motorcycle simulations performed with the CP Tire Model and the MF Tire Model verified the correlation of these models with actual measurements, and confirmed the following.

- (1) With respect to straight running stability, the MF Tire Model shows better correlation with actual measurements in comparison with the CP Tire Model.
- (2) With respect to turning performance, the MF Tire Model shows roughly the same correlation with the actual measurements as the CP Tire Model.
- (3) The MF Tire Model based on the tire measurement data, which is applied to the Motorcycle Simulation Model, facilitates changes in tire properties better in comparison with the CP Tire Model. In order to improve the accuracy of motorcycle simulation, it is necessary to adopt frame rigidity and the dynamic properties of the rider in the simulation model.

Furthermore, it is also required to develop a tool that is capable of calculating tire properties and expediting parameter studies.

7. Acknowledgements

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Development of High Fatigue Strength Ultra-Fine Grained Stainless Steel for Cylinder Head Gasket

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ABSTRACT

Fatigue testing of metallic cylinder head gaskets has shown that there is a performance discrepancy between the original sheet material and the manufactured gasket. It was determined that this discrepancy was caused by the grain boundary cracking that occurs during head forming. Therefore, an improved austenitic stainless spring steel possessing high fatigue strength was developed to improve gasket performance. This new steel features a grain-refined structure achieved by controlling recrystallization with reverse transformation.

The developed steel has a composite structure consisting of a rolling induced martensite structure and a recrystallized austenitic structure by reverse transformation having an average grain size kept to within $1 \sim 2 \mu\text{m}$. When this material is applied to a cylinder head gasket, fatigue strength is increased 30% and there is no loss of combustion gas sealing performance.

1. Introduction

As engine performance improves, conventionally common graphite or similar inorganic cylinder head gaskets (hereafter referred to as “head gaskets”) are being replaced by metallic gaskets, which offer superior durability and functionality. As shown in Fig.1, the head gaskets are made from thin sheet stainless steel (approx. 0.2mm), such as SUS301, and utilize the repulsive force of the beads formed thereon to form seals against combustion gas, cooling water and lubricating oil. The bead seal for the combustion chamber (hereafter referred to as “bore bead”) is subject to the harshest environment comprised of high temperatures and high pressure gas. Furthermore, in recent years the bore bead has tended to be used in increasingly even harsh environments, particularly due to the following three factors: (1) lower

engine rigidity due to thinner cylinder heads and cylinder blocks and the change to all aluminum parts with the aim of reducing engine weight, (2) reduced center distances due to expanding bore diameters with the aim of reducing the size of high output engines, and (3) higher compression ratios in order to reduce fuel consumption.

Under these conditions, there is an increase in cylinder head lift caused by combustion pressure (hereafter referred to as “gap”). This leads to repeated crushing of the bore head by the cylinder head with each explosion and the resultant concern about reduced fatigue life. However, fatigue test results for the thin material sheets before bore bead forming and for the gasket after bore bead forming do not always show the same tendencies, making it impossible to judge applicability of the stainless steel sheet materials as a gasket material based on fatigue strength.

Therefore, the causes of the discrepancy in fatigue characteristics between material and product were clarified by analyzing the state of bore bead fatigue cracking propagation, and an austenitic stainless steel sheet with greatly improved fatigue strength was developed. This paper reports these contents.

2. Causes of Bore Bead Cracking

2.1. Bore Bead Fatigue Strength

The head gasket is formed so that the bore bead encloses the combustion chamber, so that the gasket is arranged between the cylinder head and the cylinder block. During operation, a stress amplitude acts locally under high tensile stress at the foot and the underside of the bore bead peak as shown in Fig.2. The gap reacts to this stress amplitude, with a greater stress amplitude produced on gasket bore beads

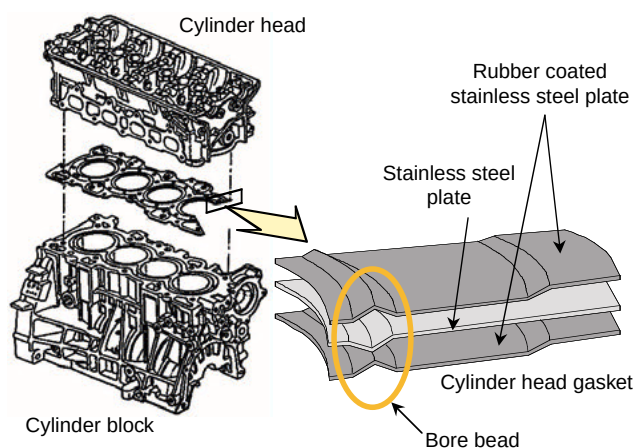


Fig.1 Illustration of cylinder head gasket

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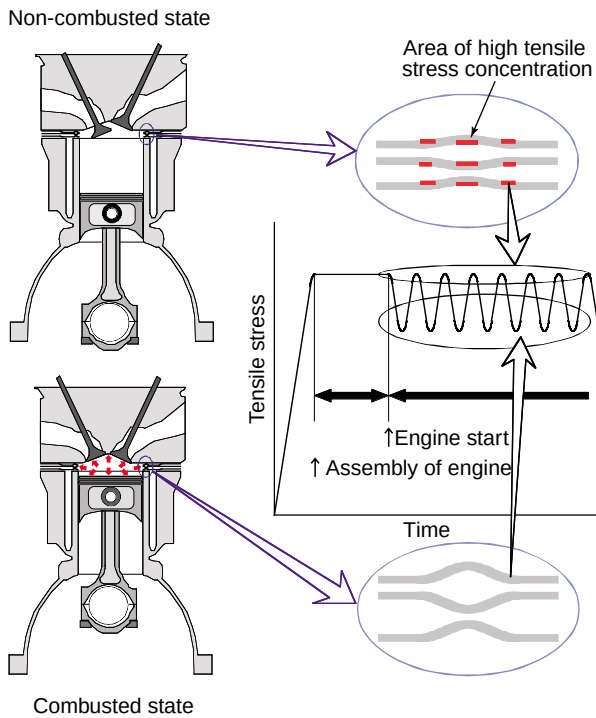


Fig.2 Tensile stress of gasket during engine running

for engines having larger gaps. For this reason, gasket fatigue strength is generally evaluated using the gap instead of a single stress event.

Therefore, crack limit displacement was set as an index for expressing gasket fatigue strength. Crack limit displacement is the displacement of the “no fatigue cracking” limit of the bore bead when a compressive load is repeatedly applied to the bore bead for up to 10^7 cycles during vibration fatigue tests.

2.2. Discrepancy between Material and Bore Bead Fatigue Characteristics

Fig.3 shows the results of planar bending fatigue strength measurements taken in the direction perpendicular to 0.2

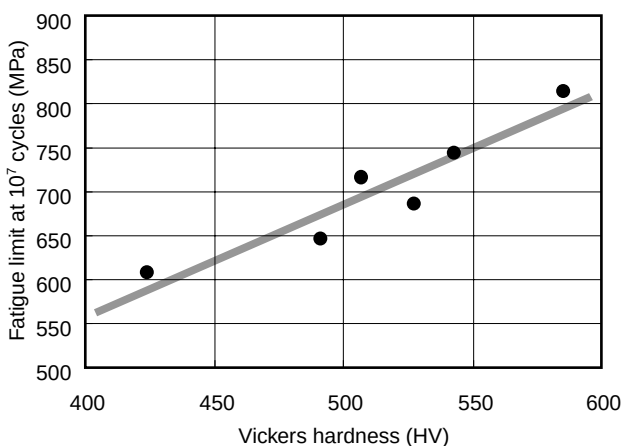


Fig.3 Relationship between fatigue limit stress and hardness for as-rolled sheet

mm thick SUS301 steel sheets with varying hardnesses. These results show that the fatigue limit increases linearly as the hardness increases. In these tests all cracking occurred from the surface, and the effects of non-ferrous inclusions⁽¹⁾⁻⁽³⁾ which cause the variations in the fatigue limit generally seen in high strength materials were not evident.

Next, Fig.4 shows the results of similar fatigue tests performed on bore beads manufactured using SUS301 materials of varying hardness. These results show a clear linear relationship between crack limit displacement and hardness. However, in contrast to the fatigue characteristics of the material, the crack limit displacement dropped as the hardness increased. In order to determine whether this was caused by the test conditions, minute strain gauges were affixed to the foot of cracked bore beads and the fatigue test results in Fig.4 were replaced with mean stress and stress amplitude. These results are shown in Fig.5, and indicate that although cracking is not affected by mean stress, it is highly dependent on stress amplitude. According to these tests, the failure limit stress amplitude did not vary with mean stress but rather was fairly constant at 90 MPa.

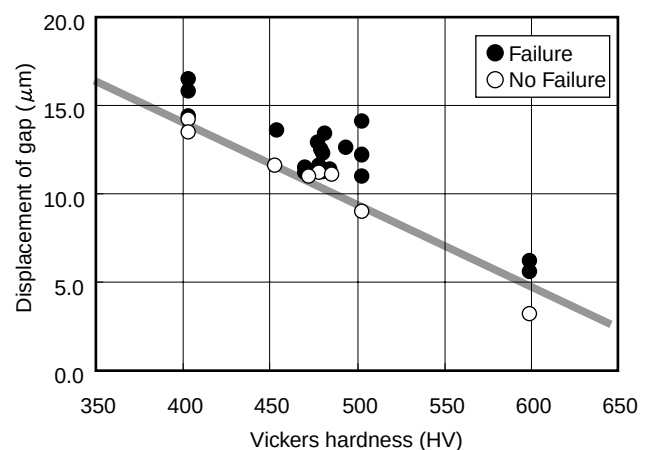


Fig.4 Relationship between displacement of gap and hardness
Solid line shows the displacement of gap against failure

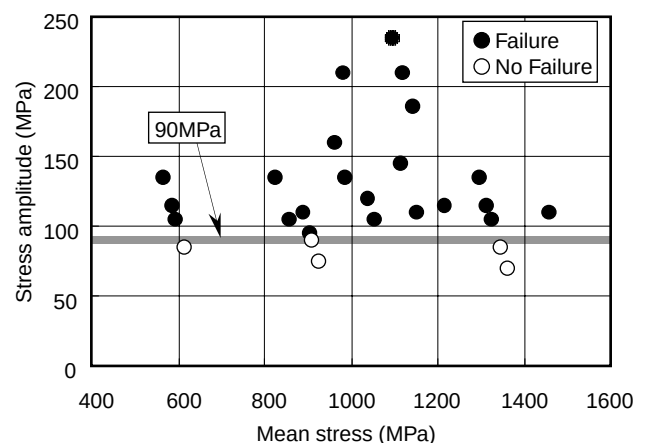


Fig.5 Effects of stress amplitude and mean stress on fatigue strength after 10^7 cycles

These results showed that the fatigue characteristics of the stainless steel sheet materials and the gasket products did not match, and that high material fatigue strength did not necessarily indicate superior fatigue characteristics when manufactured into a gasket. In other words, bore bead fatigue strength cannot be improved simply by increasing the material fatigue strength⁽⁴⁾⁽⁵⁾.

2.3. Bore Bead Cracking Condition

During bore bead forming, stress concentrates locally at the bead foot and peak as shown in Fig.6. In particular, high compressive stress is applied during bead forming at the bead foot, a starting point for fatigue failure. This is why the fatigue strength of the head gasket product exhibits reverse material fatigue strength tendencies. Further, this is thought to be the effect of local stress concentration during forming.

Fig.7 shows the results of observing the bead foot surface condition with a scanning electron microscope during bead forming and during a fatigue test. These results show the following fatigue conditions: a) although some grain boundary erosion was confirmed to have occurred in the acid-washing process after cold rolling and annealing, the surface was fairly clean before bead forming; b) bead forming caused cracking to occur at the grain boundaries, creating

undulations in the surface; c) in the early stages of the fatigue test, fatigue cracking advanced from the grain boundary cracks generated during bead forming; d) in the latter stages of the fatigue test, fatigue cracking advanced further and developed into large cracks.

Thus, the cause of the discrepancy in fatigue strength characteristics between gasket and material is thought to be due to the micro-cracks at the grain boundaries of the material surface generated during bead forming which easily lead to fatigue crack initiation⁽⁶⁾. Therefore, fatigue tests were carried out with the same load using test pieces whereon the surface of the bead foot was polished after bead forming to remove the grain boundary cracks produced by bead forming and unpolished test pieces. As shown in Fig.8, this was confirmed to extend the crack occurrence life by approximately 15 times.

Based on the above results, the discrepancy in fatigue strength characteristics between material and gasket is caused by the generation of micro-cracks in the grain boundaries at the bead foot surface due to bead forming. Conversely, it can be said that drops in fatigue strength produced when machining gaskets can be reduced by suppressing the formation of micro-cracks in the grain boundaries⁽⁷⁾.

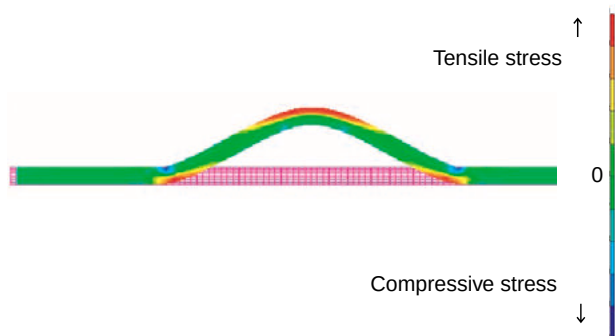


Fig.6 Distribution of stress in section of bead-formed sheet

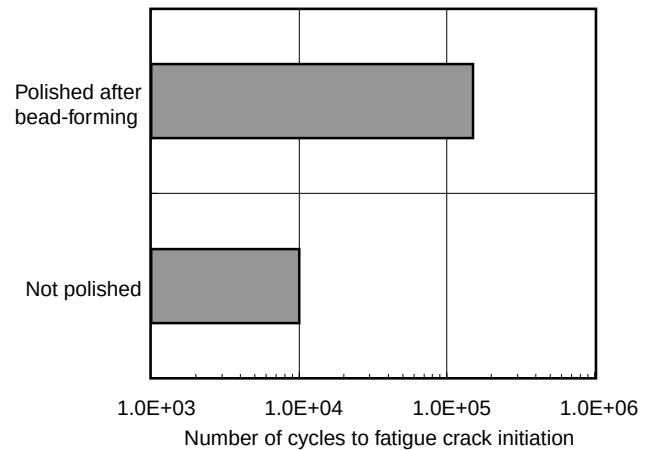


Fig.8 Effect of polish after bead-forming on fatigue crack initiation

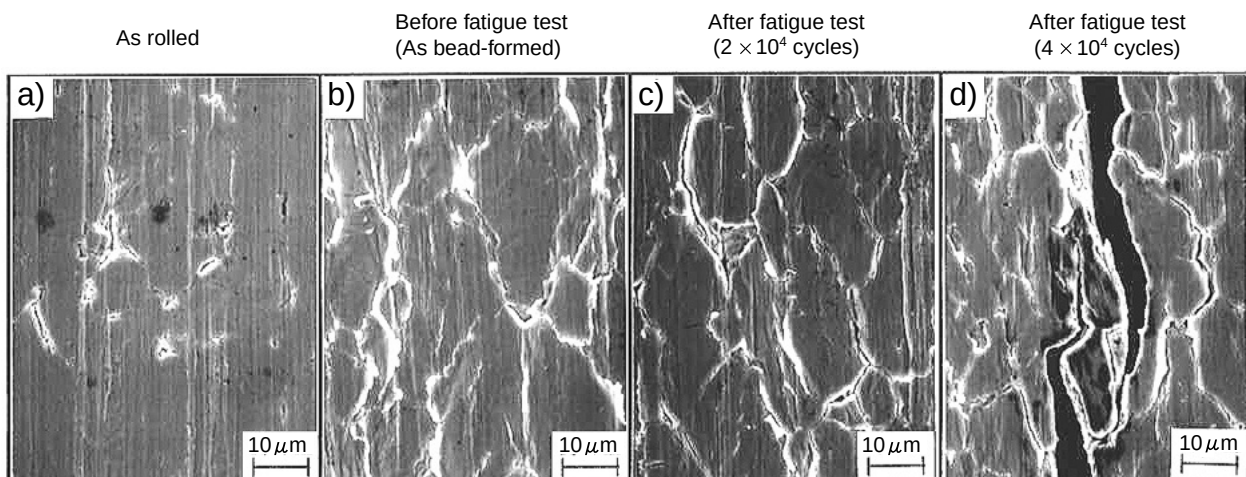


Fig.7 Surface of bead foot profile

3. Techniques for Suppressing Grain Boundary Cracking

3.1. Investigation of Grain Refinement Methods

Grain refinement which disperses deformation stress and improves grain boundary strength is thought to be the most effective means of suppressing grain boundary cracking. SUS301 is a metastable austenitic stainless steel which attains high strength by inducing a martensite structure through cold rolling. As shown in Fig.9, the sectional structure of the completed material after cold rolling has an induced martensite structure within former austenitic grains elongated in the rolling direction.

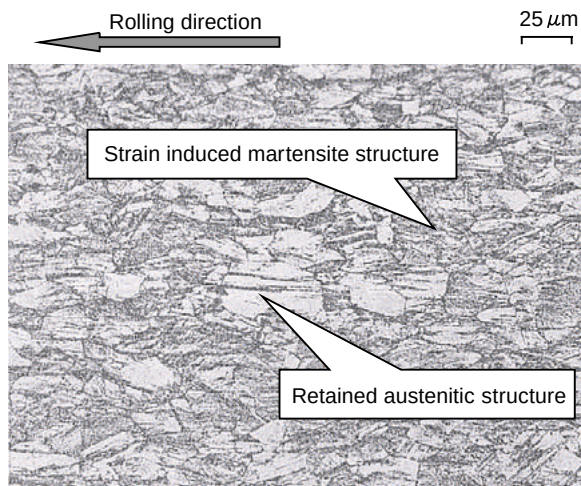


Fig.9 Microstructure of SUS301 after cold rolling

The grain refinement obtained in the completed material depends heavily on controlling the cold rolling and annealing steps in the final processing. In the final processing, the material structure undergoes a “reverse transformation sequence” in which the martensitic structure is annealed to form austenite and this is retransmitted to form the composite austenitic and martensitic structure by the final rolling. This final grain refinement is heavily dependent on the prior austenitic grain size. The control of this prior austenitic structure is accomplished by controlling the driving forces that govern recrystallization. Therefore, it is particularly important to control the strain energy of the cold rolling before annealing as well as the thermal energy of the annealing process.⁽⁸⁾⁻⁽¹¹⁾

3.2. Investigation of the Optimum Component System

The critical manufacturing processes for grain refinement are the cold rolling and annealing processes. However, the material components also have a great effect on grain growth behavior in the annealing process.

Fig.10 shows the change in austenitic recrystallized grain size with reverse transformation with respect to the annealing temperature for two types of steel, SUS301 with a carbon content of approximately 0.10% and SUS301L wherein the

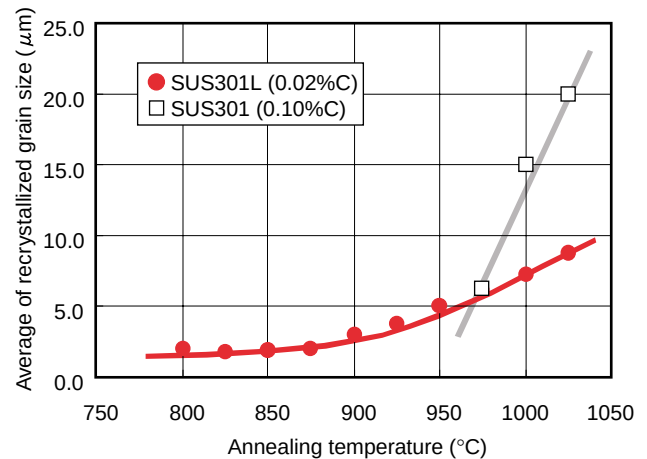


Fig.10 Effect of carbon on recrystallized grain size

carbon content was reduced to approximately 0.02%. When carbon-content is high, the recrystallization start temperature increases and, further grain growth also progresses rapidly, which facilitates roughness. In contrast, when there is little carbon, recrystallization starts from a relatively low temperature and the grain growth behavior is also slower, making this an effective means for refinement.

Fig.11 shows the half-value width of X-ray diffraction of austenitic grains in the annealed SUS301L. The half-value width caused by recrystallization drops sharply at around 800°C, indicating that the start temperature for recrystallization with reverse transformation exists in this temperature area. In addition, there is almost no change in the half-value width at temperatures above 800°C, suggesting that recrystallization is complete and grain growth is occurring.

In this manner, lowering the carbon content is an effective means of controlling grain refinement. However, since bore bead spring performance must also be ensured when applied as a head gasket material, an appropriate amount of nitrogen was added to the developed material to compensate for the drop in matrix hardness resulting from the lower carbon content.

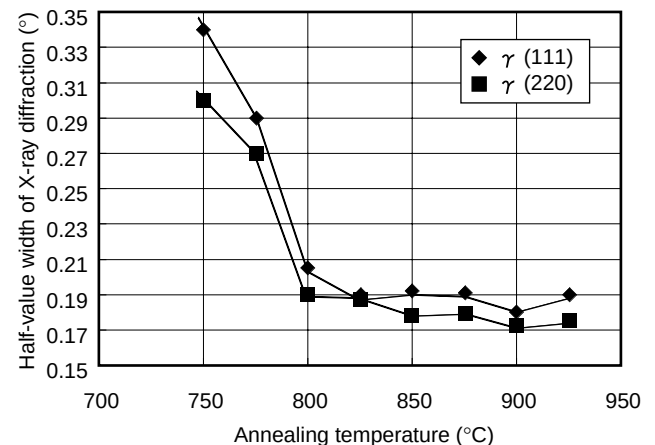


Fig.11 Relationship between half-value width of X-ray diffraction and annealing temperature

3.3. Grain Refinement Limit

Fig.12 shows the relationship between the average crystallized grain size and the hardness of the developed material after annealing. When the average crystallized grain size was 2.0 μm or less, the reverse transformation recrystallized austenitic structure was a composite structure consisting of rolling induced martensite structure enclosed by a reverse transformation recrystallized austenitic structure as shown in Fig.13. Like the full recrystallized single-phase austenitic structure, the relationship between the average crystallized grain size and hardness for this type of composite structure exhibits a clear linear relationship following the Hall-Petch formula⁽¹²⁾⁽¹³⁾. If grain refinement is further pursued to achieve an ultra-fine grained structure with a grain size of 1.0 μm or less, the ratio of rolling induced martensite structures increases and this structure is no longer enclosed

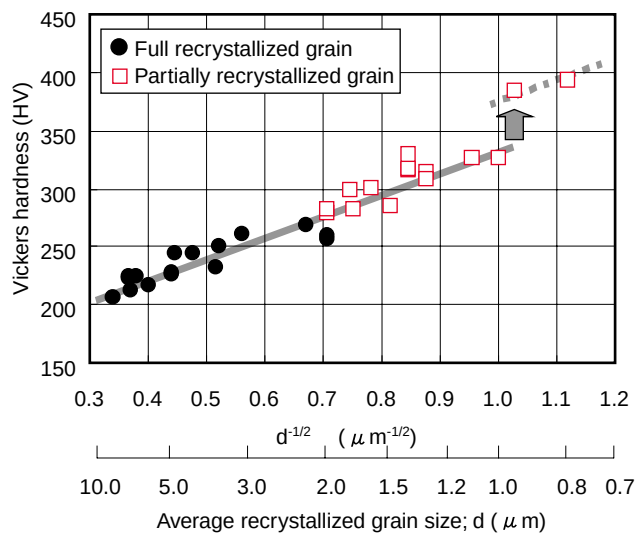


Fig.12 Relationship between hardness and average recrystallized grain size

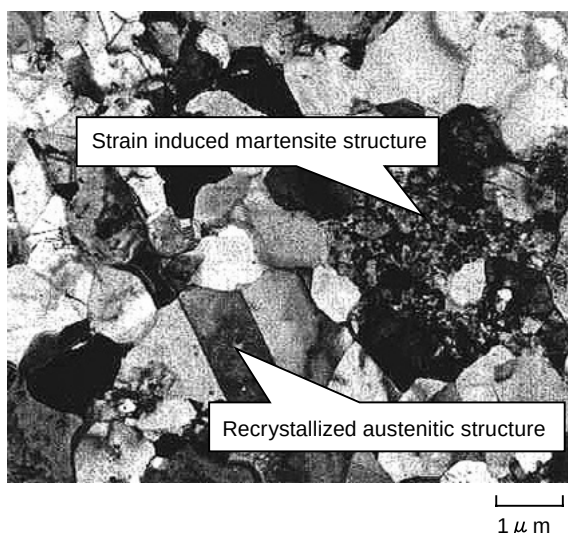


Fig.13 Transmission electron micrograph of post-bright annealing (Average of grain size: 1.0~2.0 μm)

by the reverse transformation recrystallized austenitic structure as shown in Fig.14. As a result, the relationship between grain size and hardness does not follow the Hall-Petch formula, so the limit of grain refinement effects for this material is thought to be up to approximately 1.0 μm .

3.4. Effectiveness of a Composite Structure

The developed material lowers the carbon content and adds nitrogen to SUS301 in order to control refinement of the reverse transformation recrystallized structure. Therefore the machining hardness behavior after cold rolling differs. Fig.15 shows a comparison of hardness after 48% cold rolling. First is the complete austenitic structure with an average full recrystallized grain size of approximately 2.0 μm . In this case, grain refinement strengthens the grain boundary, resulting in an increase in hardness of approximately 10 HV compared to SUS301L which is a low carbon material like the developed material. However, this is still approximately 30 HV below the target SUS301 material hardness (470 HV).

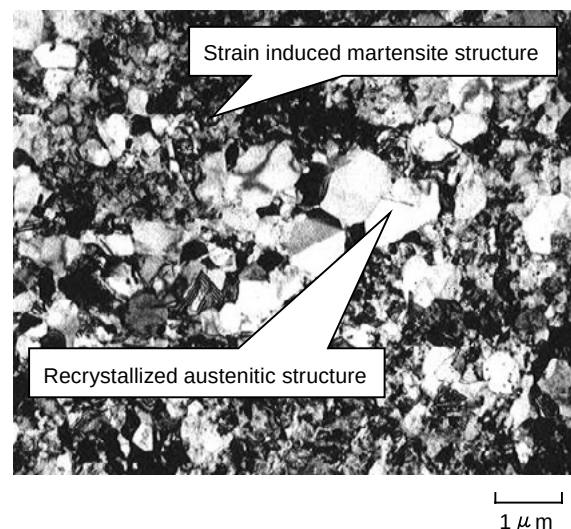


Fig.14 Transmission electron micrograph of post-bright annealing (Average of grain size: less than 1.0 μm)

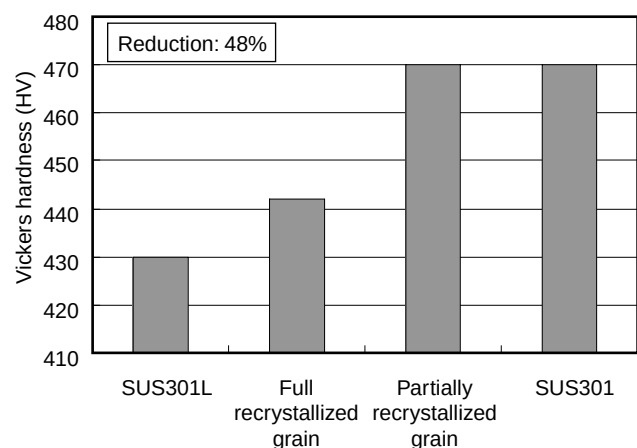


Fig.15 Comparison with hardness after 48% cold rolling

In order to raise the hardness, the first thing that comes to mind are methods of increasing the cold rolling reduction rate. However, increasing the reduction rate more than is necessary increases the amount of rolling induced martensite structure produced and greatly deteriorates the workability, and therefore is inappropriate.

Next, the amount of added nitrogen can be increased to raise the solid solution strengthening performance. However, the nitrogen partial pressure (pN_2) cannot be raised with the normal Argon Oxygen Decarburization (AOD) refining method, making it difficult to increase the nitrogen to the same level as carbon. Increasing the chromium also raises the nitrogen solid solidifying process, but this increases costs. Furthermore, if large amounts of nitrogen are added, the nitride solid solution treatment temperature rises sharply, making it difficult to establish grain refinement as a mass production process. In addition, methods of increasing hardness through aging also increase costs, and are therefore inappropriate.

Therefore, when an ultra-fine grained composite structure consisting of rolling induced martensite structure and reverse transformation recrystallized austenitic structure having an average grain size of 2.0 μm or less is adopted, the synergistic effects of a stronger matrix and further grain refinement make it possible to obtain hardness equivalent to SUS301 with the same rolling reduction rate.

3.5. Bore Bead Fatigue Strength Improvement Effects

Fig.16 shows the relationship between the average grain size of the reverse transformation recrystallized structure after bright annealing and the fatigue strength of this same material cold rolled to a hardness of HV470 and having a formed bore bead. These results show that the fatigue strength increases as the grain size becomes smaller not only for the full recrystallized structure material, but also for the partially recrystallized composite structure material containing martensite structure. In these tests, the bead crack limit displacement was increased by approximately 35% by refining the crystallized grains to achieve a composite

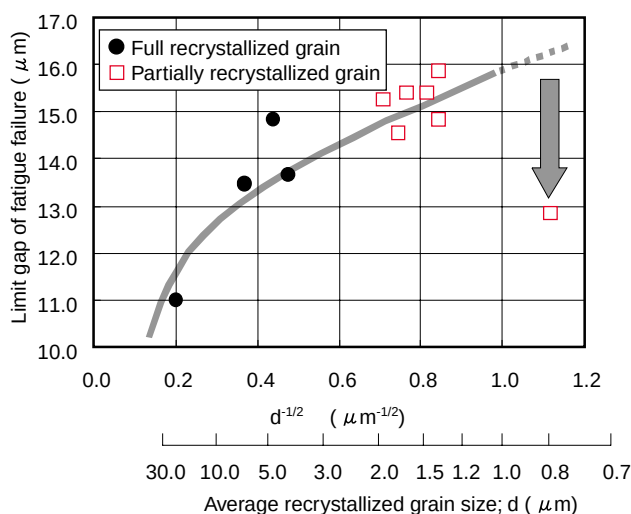


Fig.16 Relationship between limit gap of fatigue failure and average recrystallized grain size

structure state of approximately 2.0 μm , or about 1/15 the average SUS301 materials grain size of approximately 20 to 25 μm .

However, this tendency holds true only for an average grain size of 1.0 μm or more where the relationship between the crystallized grain size and hardness follows the Hall-Petch formula. Based on these results, even if a martensite structure remains after annealing, most of it is enclosed by a reverse transformation recrystallized austenitic structure. Therefore fatigue strength improvement effects can be obtained from grain refinement within if the material falls within a range that follows the Hall-Petch formula.

As described above, the authors developed a metastable austenitic stainless steel for application as a head gasket material having a composite structure consisting of a reverse transformation recrystallized austenitic structure mixed with a martensite structure, and an average grain size which is controlled to 1 to 2 μm . This developed material was named NAR301L-HS1, and ensures the same level of hardness needed to seal high pressure combustion gas as SUS301, while at the same time dramatically improving gasket fatigue strength.

4. Effects of Application to an Actual Vehicle

Fig.17 shows the results of bore bead fatigue tests performed up to 10^7 cycles using test pieces. These results show that even taking into account the variance in fatigue strength due to hardness and crystallized grain size, the composite structure grain-refined material has a crack limit displacement approximately 30% higher than SUS301.

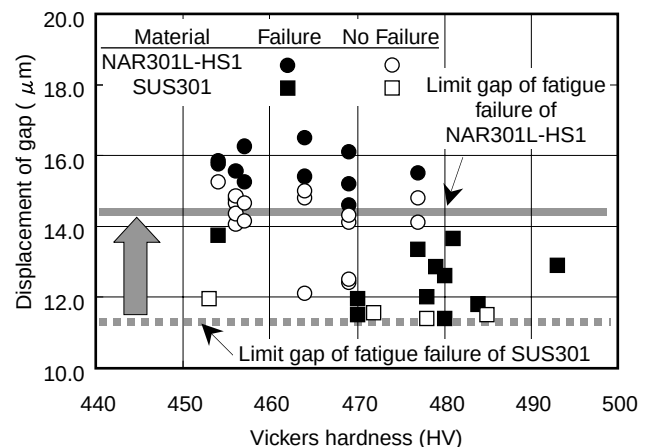


Fig.17 Relationship between displacement of gap and hardness after cold rolling

Fig.18 shows the results of applying the composite structure grain-refined material to the head gasket of a newly developed engine. This engine uses the basic structure of same-bore pitch engines (A to D) currently on the market, while aiming for even higher output. The maximum combustion gas pressure (P_{max}) is higher than conventional engines, and the maximum gap is also increased by approximately 50%.

For this reason the crack limit displacement of the

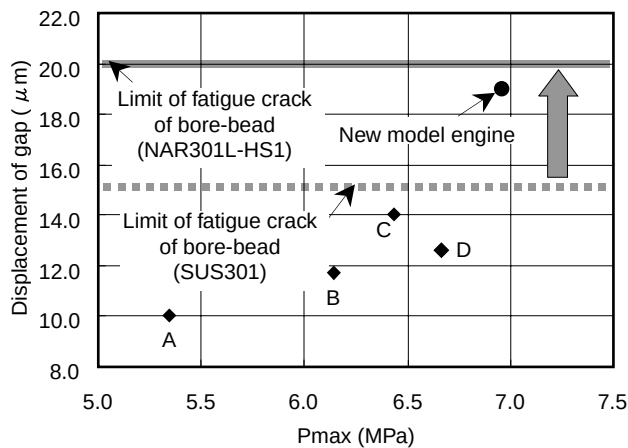


Fig.18 Effect of application of cylinder head gasket for new model engine

SUS301 material 3-layer structure head gasket used in conventional engines is exceeded by a wide margin. Therefore, a head gasket with superior fatigue characteristics was considered necessary. Generally, the crack limit displacement is increased by increasing the number of bore bead layers to reduce the gap load on each layer, or a stepped type gasket which concentrates the load on the bore is adopted to reduce the gap itself. However, all of these countermeasures involve exclusive parts and increased costs. On the other hand, the newly developed composite structure grain-refined material makes it possible to develop a low cost head gasket with superior durability.

5. Conclusion

The bore bead fatigue failure mechanism of the cylinder head gasket was analyzed, crystallized grain refinement with a composite structure was proposed as a means of improving fatigue strength, and a new material was developed. As a result, the following conclusions were obtained:

- (1) The discrepancy between the fatigue strength characteristics of the material and the head gasket was caused by cracking at grain boundaries due to bore bead forming, and fatigue failure initiated and advanced from these grain boundary cracks.
- (2) Refinement of the crystallized grains is an effective means for suppressing grain boundary cracking.
- (3) Refinement to an average grain size of approximately 1.0 μm is possible by controlling the reverse transformation recrystallization behavior. The material forms a composite structure with an average grain size of 2.0 μm or less consisting of rolling induced martensite structure enclosed by a reverse transformation recrystallized austenitic structure. In addition, the relationship between grain size and hardness follows the Hall-Petch formula.
- (4) Metastable austenitic stainless steel was developed for application as a head gasket material. This material has a composite structure consisting of martensite structure and a reverse transformation recrystallized structure.

Further, the average grain size is controlled to 1 to 2 μm . This ensures the level of hardness needed to seal high pressure combustion gas as possessed SUS301, while at the same time improving gasket fatigue strength by 30% or more.

Acknowledgment

In closing the authors wishes to express his deep thanks to all concerned parties at Sumitomo Metal Industries who participated in this joint development and at Nihon Leakless Kogyo for manufacturing the cylinder head gaskets.

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Takashi KATSURAI

Development of High-sealing Foamed Rubber Gasket Material

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Takuya AOKI* Atsuo YAMAGUCHI**

ABSTRACT

A high-performance durable formed rubber gasket material characterized by both superior compressive recovery and adaptability to rough surfaces was developed for automotive internal combustion engine sealing applications. Research was performed on gaskets exhibiting high sealing properties and a simplified fastening structure. As a result of the research, a gasket was produced with low surface pressure sealing properties that far exceed those of the conventional gasket materials due to the structure of the base rubber material, which is a foamed rubber material with a closed cell construction, and the optimization of the additive composition. The application of this gasket to an engine intake system facilitates the streamlining of the fastening structure.

1. Introduction

Internal-combustion engines that are lighter, more compact, and less expensive are a universal goal of the automobile industry. Recent progress toward achieving this goal includes the switch by automobile manufacturers from aluminum alloys to resin materials, and magnesium alloys as structural materials principally for air intake system parts, and the development of lightweight, compact engines through part modularization. Against this background, the field of fastening seal structures is facing the inability of rubber coated metal (RCM), joint seats, molded rubber rings and other conventionally used materials to provide the requisite sealability in some areas. This is due to lower seal surface flatness and increased roughness, lower flange stiffness, fewer joint bolts, and smaller bolt sizes. Thus, materials capable of sealing at lower joint surface pressures are needed⁽¹⁾⁻⁽³⁾.

In response, the authors have used a high-compressive-recovery foaming material to develop a gasket material that offers better low-surface-pressure sealability than existing gasket materials.

2. Determinants of Sealability

2.1. Analysis of Sealability Performance Factors

In a given fastening state, possible sealing pressure is generally determined with the experimental equation (1) below. Thus, sealability can be improved by raising the gasket coefficient, surface pressure, and surface roughness.

$$\text{Possible seal pressure} = k \times \rho \times \exp(p \times r) \quad (1)$$

k: Gasket coefficient
ρ: Seal medium viscosity
p: Surface pressure
r: Surface roughness

These sealability determinants can be further divided into sealing surface conditions, tightening force, flange stiffness, and gasket material properties (Fig.1). Improving any one of these factors will improve sealability and thereby achieve advanced performance. However, improvements made with existing technology have run counter to the goal of lightweight, low-cost engine components.

Thus, reconciling reduced weight and low cost on the one hand with sealability in a system requiring seals on the other calls for a technique whereby the compressive recovery of the gasket is improved and, at the same time, surface hardness (an indicator of conformability to the sealing surface) is lowered in order to improve the sealability of the gasket itself and to compensate for the lower sealability that results from reduced flange thickness (stiffness), fewer bolts (tightening force), and the elimination of surface processing, to thereby assure a lightweight and low cost gasket.

More specifically, a decreased weight, a streamlined fastening structure, low cost, and sealability are all possible with a high-performance gasket possessing sufficiently high compressive recovery, low surface hardness, and excellent durability and reliability to offset the reduced fastening seal functionality that has resulted from the switch to materials

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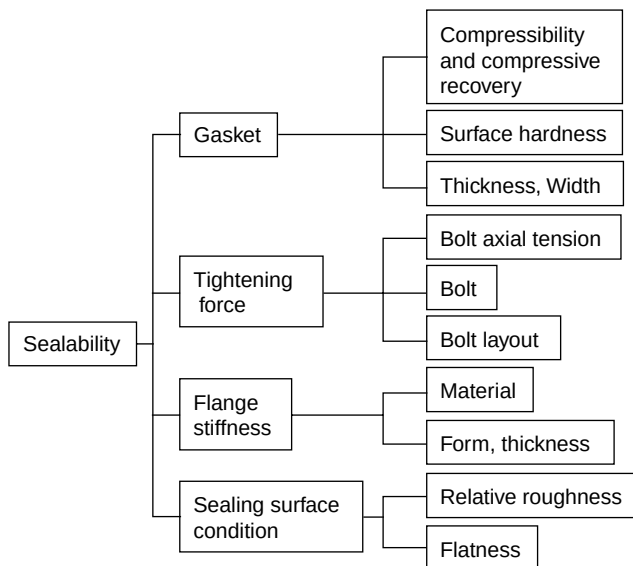


Fig.1 Sealing performance factor analysis chart

such as resins and magnesium alloys primarily in engine air intake system parts in recent years.

2.2. Improving the Compressive Recovery of Gasket Materials

NBR, FKM and other heat-resistant rubbers are currently the most widely used rubbers in automotive engine gasket applications where low hardness and high compressive recovery are required. The authors chose the approach of achieving even greater compressive recovery by intentionally creating air spaces in this type of rubber in order to combine the elasticity of rubber with that of gas. Such structures, generally referred to as foamed rubber, are widely used as fillers and damping materials for electrical appliances.

Fig.2 shows an evaluation model for the compressive recovery of a single cell in foamed rubber. This model, in which a rubber tube is placed between two aluminum alloy plates that simulate the flange surfaces, permits use of the desired gas in the air space and the setting of the desired pressure. Table 1 gives the conditions under which the compressive characteristics of rubber and air were evaluated with this model.

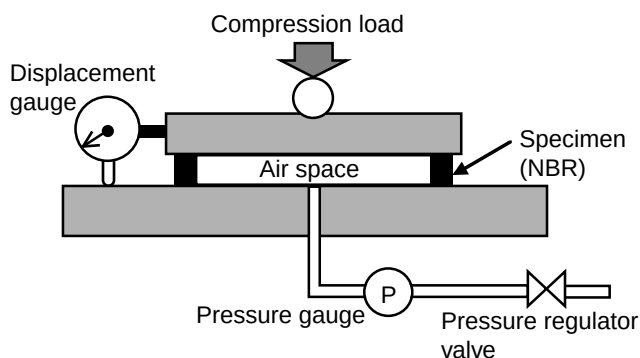


Fig.2 Compressibility evaluation model

Table 1 Compressibility test condition

Specimen specification	
Diameter	40.0 mm
Thickness	6.0 mm
Material	NBR
Rubber hardness	60 Hs
Compression speed	0.1 mm/min

Even at low compression surface pressures, the gas exhibits greater compressibility than the rubber (Fig.3). In particular, at a surface pressure of 5 MPa or less, the gas has more than twice the compressibility of rubber.

Fig.4 shows the change in compressibility at different gas volumes, determined by changing the gas volume in a simple model of rubber and gas, and demonstrates that compressibility increased with gas volume, in almost perfect agreement with the theoretical line.

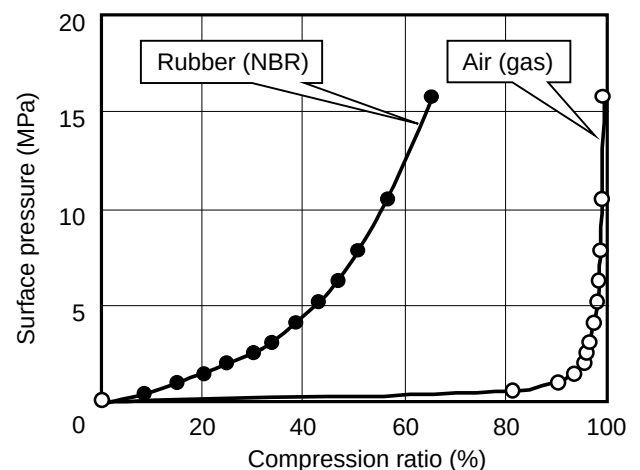


Fig.3 Comparison of compressibility between rubber and air

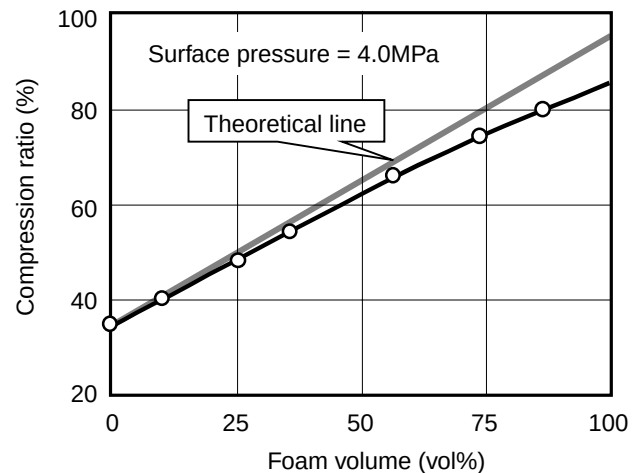


Fig.4 Relationship between volume function of air and compression ratio

The preceding demonstrates that the combining of rubber and gas is a logical method for improving the sealability determinants of low hardness and high compressibility, as the result is a compressibility that increases with gas content per unit of volume.

Next, the authors examined how compression characteristics are affected by the method of fixing gas in the rubber in two types of foam: an open-cell foam, in which cells are exposed to outside air, and a closed-cell foam. These open- and closed-cell models were created by opening or closing an air valve that connects the air spaces in the rubber sample shown in Fig.2. The results (Fig.5) indicate a marked difference in compressive recovery between these two types of foam, i.e., open and closed-cell: closed-cell foam had a compressive recovery rate 1.5 times higher than open-cell foam. This is because in closed-cell foam, the air is unable to escape to the outside and so its elasticity supplements the system's compressive recovery.

Thus, closed-cell foam, having better compressive recovery than open-cell foam, can provide greater pressure-resistant sealability and so is superior as a gasket material.

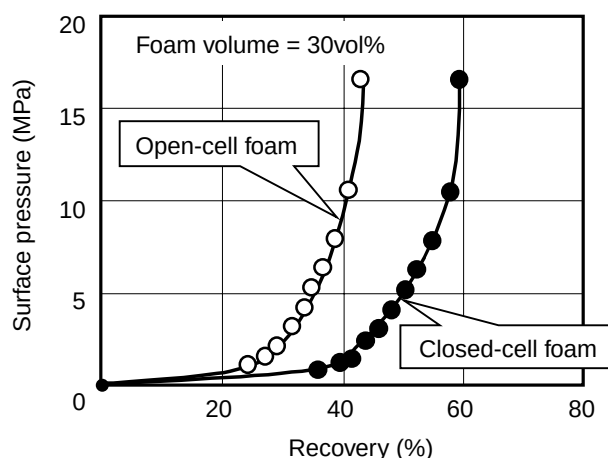


Fig.5 Relationship between foam cell type and compressive recovery

3. Rubber Foaming

The most commonly used method of rubber foaming is chemical foaming, in which a pyrolytic foaming agent such as azodicarbonamide is added to rubber, which is then heated to generate N_2 , CO_2 and other gases that are dispersed throughout the rubber. This technique makes it relatively easy to create a foam structure, and results in foamed rubber with relatively large (100 to 500 μm), cross-linked air bubbles. Fig.6-a is a cross-section photograph of chemical foaming.

As for the suitability of such foamed rubber for sealing parts, the high compressibility promised by Fig.4 is achievable, but the cross-linked air bubbles make it difficult to achieve high compressive recovery (Fig.5) and, moreover, permit the seepage of the sealing medium, eventually culminating in leakage, a fatal defect for a gasket material.

To compensate for this shortcoming, the authors turned to micro-capsule foaming, which disperses relatively small,

closed (i.e., non-cross-linked) cells throughout the rubber. This is done by filling a vinylidene chloride capsule wall with a low-boiling-point hydrocarbon, which is a liquid at room temperature but which expands upon heating, to thus yield foamed rubber. Foamed rubber produced by this method contains closed-cell, microscopic air bubbles as small as 20 μm (Fig.6-b) and has greater compressive recovery than open-cell foamed rubber (Fig.7). The disadvantage of micro-capsule foaming is the difficulty of manufacturing: For example, it is difficult to mix the rubber and the capsule because of the capsule wall's low solvent resistance, while foam properties are highly dependent on foaming conditions. To achieve the consistent production of a gasket material with both high sealability and durability, the authors chose the material specifications described in the following section for a foamed rubber gasket produced by micro-capsule foaming.

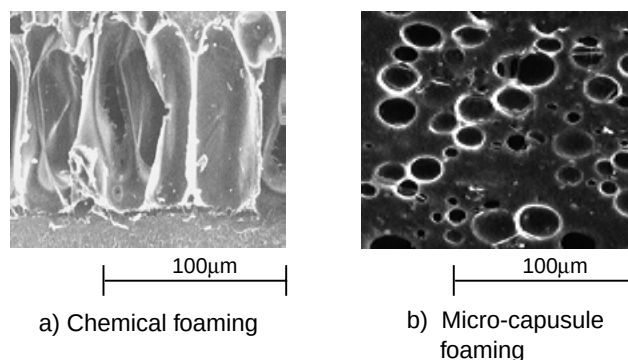


Fig.6 Cross section pictures of foamed rubber

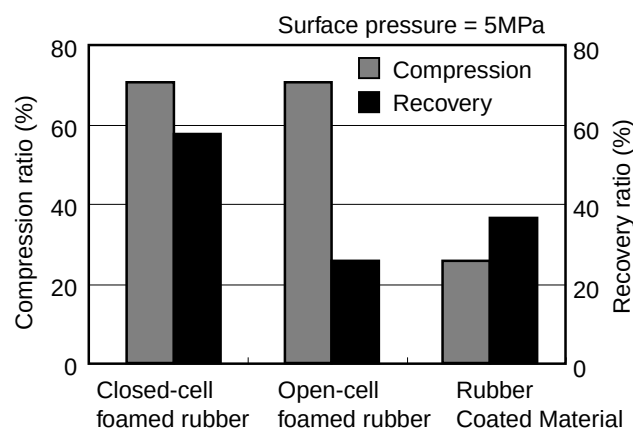


Fig.7 Compression characteristics of foamed rubbers

4. Material Specifications for a Foamed Rubber Gasket

4.1. Elements of a Foamed Rubber Gasket

A foamed rubber gasket produced by micro-capsule foaming has a structure such as that shown in Fig.8, and its primary sealability determinants are cell structure, base rubber, and primer (Fig.9)⁽⁴⁾⁽⁵⁾. The authors determined the optimum specifications for a high-performance gasket for these parameters.

4.2. Base Rubber Properties

With the microcapsule technique, the foam properties of foamed rubber are greatly dependent on the Mooney viscosity of the base rubber. Foaming properties are excellent when the rubber's Mooney viscosity is below 50 poise, but drop sharply as Mooney viscosity exceeds this value (Fig.10).

4.3. Foaming Conditions

To obtain good foaming conditions, micro-capsule expansion must occur below the temperature at which the base rubber begins to vulcanize (Fig.11).

This entails determining the proper foaming temperature by considering the method of rubber vulcanization and the solvent injected (i.e., a low-boiling-point hydrocarbon).

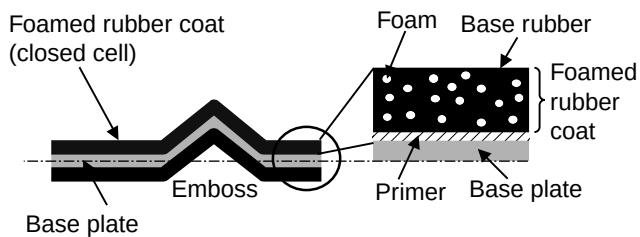


Fig.8 Schematic drawing of foamed rubber gasket

Table 2 Foaming test conditions

Foaming temperature	150°C
Foaming time	30 min
Micro capsule content	10 PHR

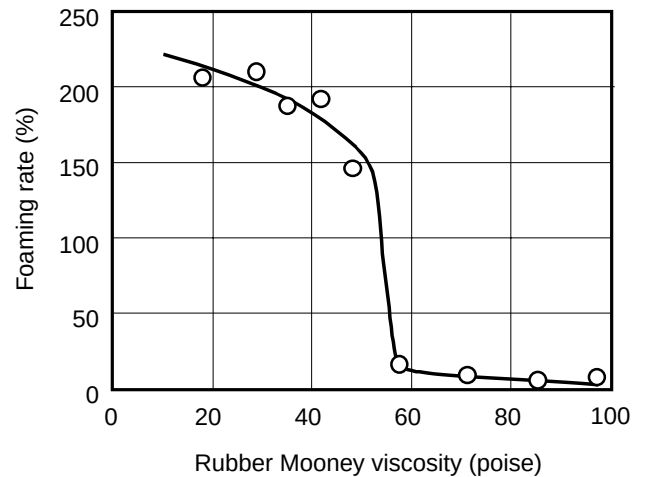


Fig.10 Relationship between rubber Mooney viscosity and volume function of foaming

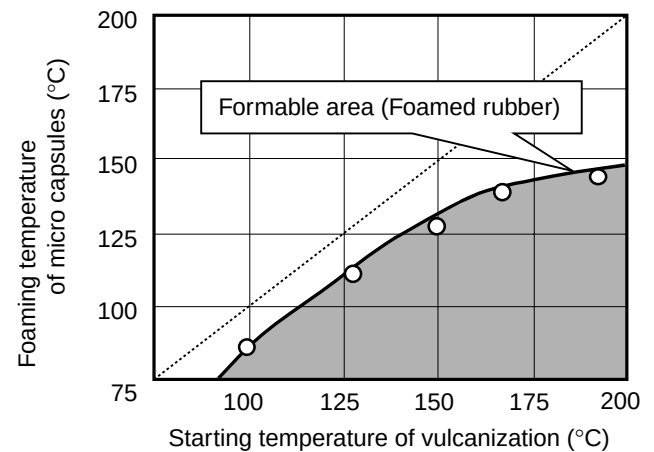


Fig.11 Formable area in foamed rubber production

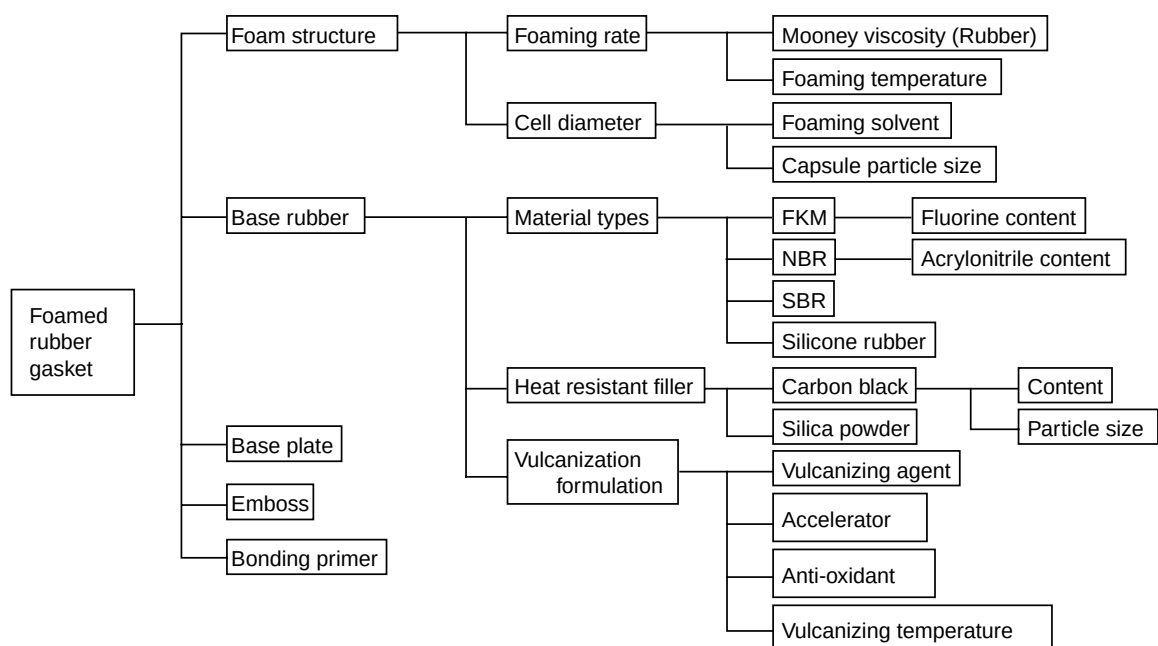


Fig.9 Component parts drawing of foamed rubber

4.4. Defining the Base Rubber Specifications

Base rubber selection must consider the environment in which gasket parts are used. As a representative example, this section considers fluorine rubber, which, among rubbers, offers exceptional thermal and fuel oil resistance. The base rubber to be used as the substrate for a foamed rubber gasket requires excellent durability and reliability in order to assure retention of cell structure, one of the necessary properties.

Fig.12 shows how compression set, which is an indicator of base rubber (fluorine rubber) durability, tends to decrease as fluorine content is increased, and that compression set can be minimized with a fluorine content of 65% or more.

Fig.13 shows the effect of carbon black content, another determinant of the base rubber's thermal resistance. Essentially, a base rubber with higher carbon black content undergoes less compression set and exhibits consistent thermal resistance at a content of 60 PHR or more.

4.5. Defining Cell Structure

In designing gasket parts using the foamed rubber material thus created, the authors focused on the foamed

rubber's cell structure in their investigation of the relationship between sealability and material specifications. Figs.14 through 16 give the results of evaluation of possible sealing pressure and compression set (indicators of sealability and durability, respectively) for different combinations of three determinants of foamed rubber properties: foaming rate, particle size, and coating thickness. The test conditions are given in Table 3.

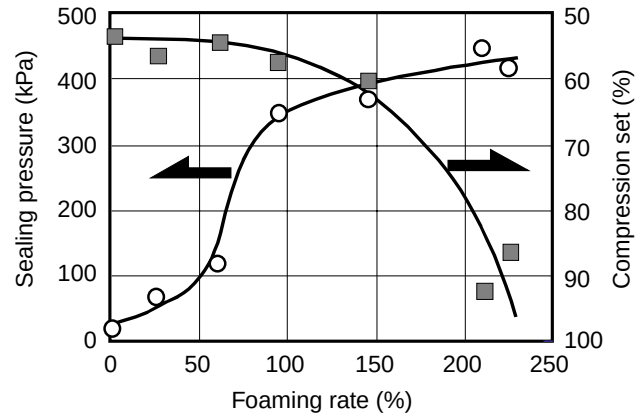


Fig.14 Optimization of foaming volume in relation to sealing performance and compression set

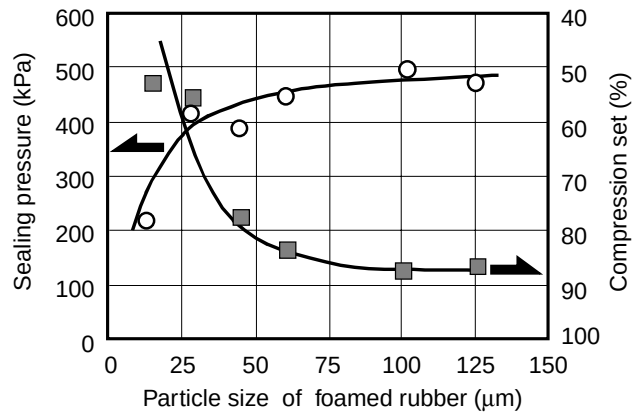


Fig.15 Optimization of particle size in relation to sealing performance and compression set

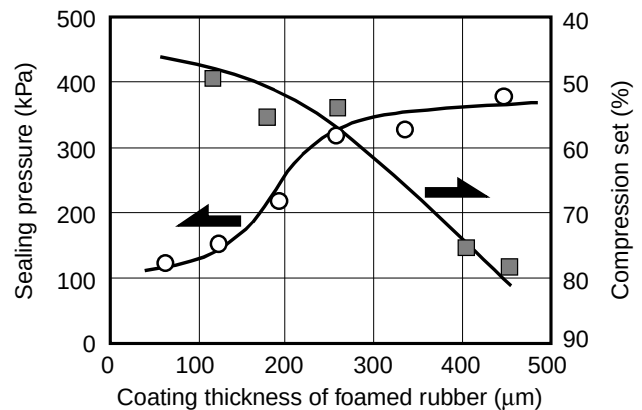


Fig.16 Optimization of coating layer thickness in relation to sealing performance and compression set

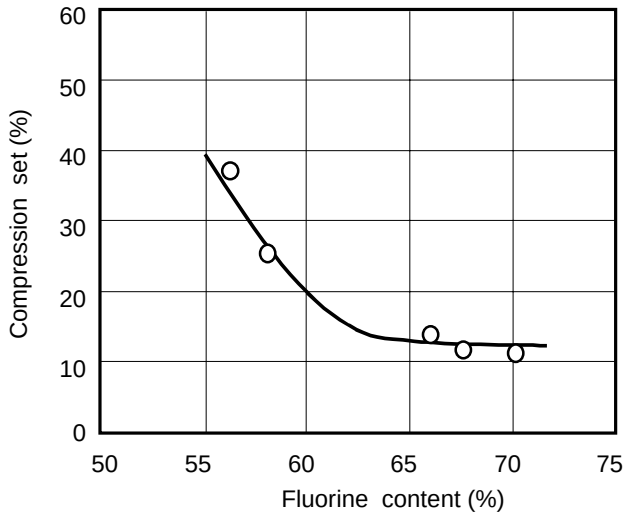


Fig.12 Relationship between fluorine contents and compression set

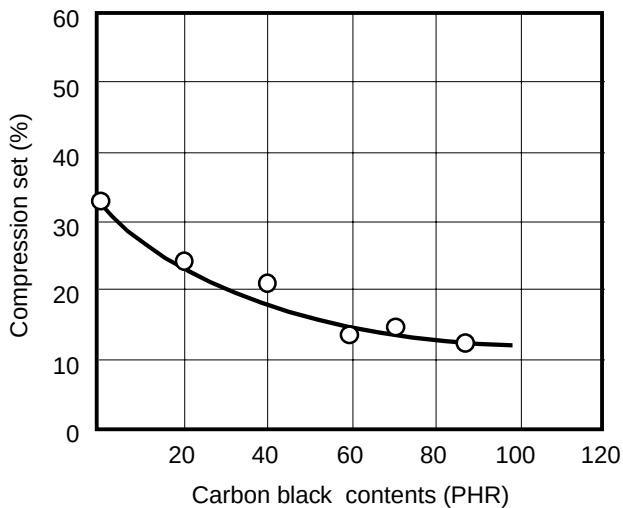


Fig.13 Relationship between carbon black content and compression set

Table 3 Components of foamed rubber and sealing test conditions

Base rubber	FKM (F content=65%)
Carbon black content	60PHR
Foam type	Closed-cell
Foaming temperature	150°C
Vulcanizing temperature	250°C
Foaming rate	120% (basic)
Particle size of foam	25 μ m (basic)
Thickness	80 μ m (basic)
Seal test conditions	Surface pressure=0.5MPa Sealing temp=-30°C
Compression set test conditions	Heating temp=120°C Holding time=24hr

The relationship of foaming rate (Fig.14) to sealability and compression set is such that a higher foaming rate results in better sealability but worse compression set. Larger particle sizes (Fig.15) improve sealability but worsen compression set. A thicker coating layer (Fig.16) also improves sealability while worsening compressive set.

Thus it was verified that a trade-off relationship exists between possible sealing pressure (i.e., sealability) and compressive set (durability and reliability) for each of the determinants of foamed rubber layer properties, and that an optimum value also exists for each.

5. Sealability of Foamed Rubber Gaskets

5.1. Low-Temperature Sealability

The low-temperature sealability of foamed rubber gaskets was evaluated first in view of automotive engine applications. Ordinarily, a rubber gasket exhibits reduced sealability in low-temperature environments due to hardening of the rubber⁽⁶⁾. However, in contrast to the precipitous decline in sealability that gaskets made of molded rubber (typically

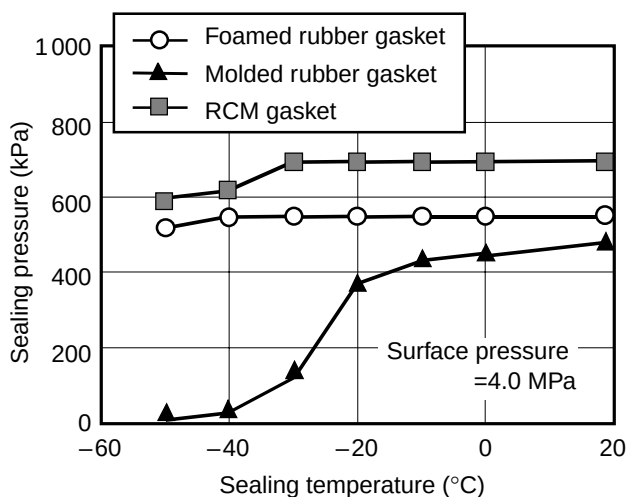


Fig.17 Sealing performance at low temperature

used for resin part sealing) undergo at -20 °C and below (Fig.17), a gasket made of foamed rubber developed by the authors displays consistent sealability, proving that this combination of low hardness and high compressive recovery assures superior sealability even when the same rubber material is used.

5.2. Sealability at Low Surface Pressures

In a light weight, low cost system, the gasket must seal at the lower fastening surface pressures that accompany reduced tightening forced flange stiffness. Commonly used RCM loses its sealability at low surface pressures. However, the authors' foamed rubber gasket, provides four times the sealability of RCM even at surface pressures below 3.0 Mpa (Fig.18), and it maintains its sealability even at a surface pressure as low as 0.5 Mpa. This means that a gasket between an intake manifold and the cylinder heads (Fig.19) would require only 1.5 bolts (Fig.19-b) to assure sealability if made of foamed rubber, instead of the approximately 2.5 fastening bolts per intake port (Fig.19-a) required with current gaskets.

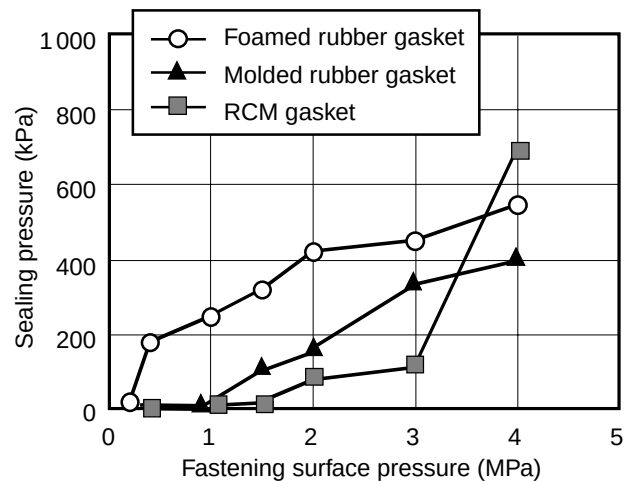


Fig.18 Relationship between fastening pressure and sealing performance

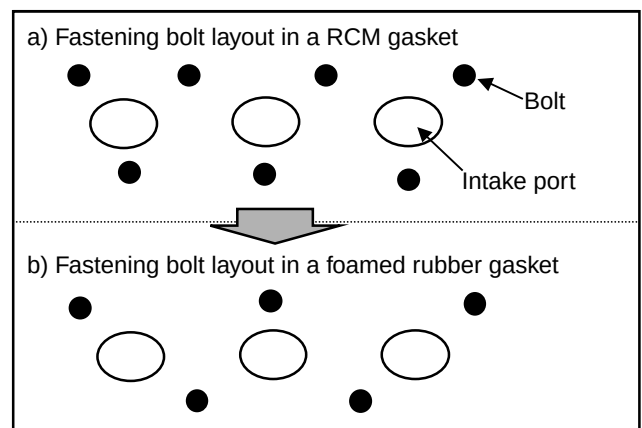


Fig.19 Fastening bolt layout depends on gasket type

5.3. Surface Roughness and Flatness Adaptability

The surface roughness adaptability and flatness adaptability of the seal flange of a foamed rubber gasket are shown in Figs.20 and 21, respectively. As for surface roughness adaptability, sealability is maintained to a surface roughness of R_t -80 μm , suggesting applicability in the unprocessed state for the casting surface of light-metal alloys.

As for flatness adaptability, sealability is maintained to a fastening surface clearance of 0.6 mm, again suggesting applicability, in the unprocessed state, to the deformation and waviness that are generated during, for example, the molding of resin parts. Thus, foamed rubber gaskets possess excellent properties as automotive sealing parts, and can be applied to streamline help fastening seal structures and lower their cost.

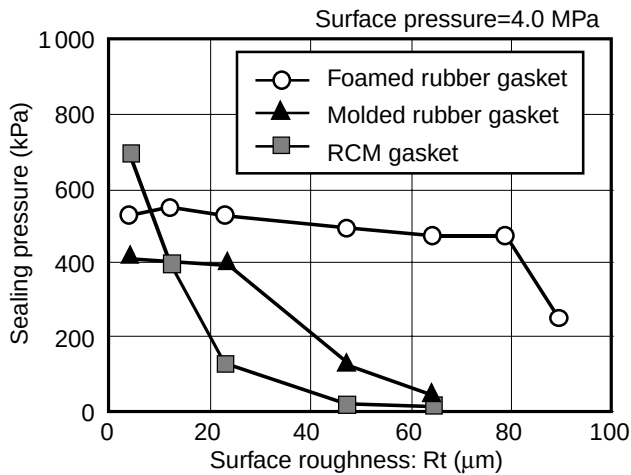


Fig.20 Relationship between surface roughness and sealing performance

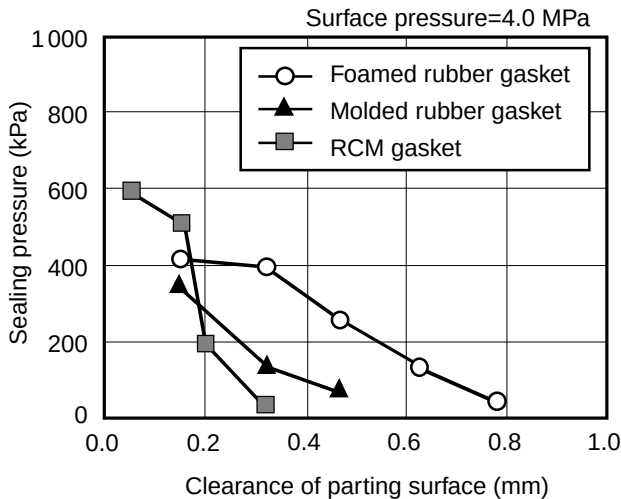


Fig.21 Clearance adaptability of foamed rubber

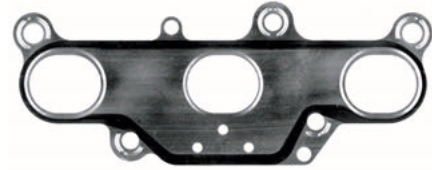
6. Example Applications to Engine Gasket Parts

This section describes applications in the air intake system gaskets of an automobile engine. The gaskets were those in the engine of the Insight (a mass production hybrid car): a gasket between the cylinder head and EGR plate, and

a gasket between the resin intake manifold's throttle body spacer and the throttle body. The gaskets comprise a foamed rubber coating over a thin metal substrate; their shapes are shown in Fig.22.

FKM was used as the base rubber for the EGR plate gasket, a part which is subjected to a thermal load of up to 130°C by the EGR gas. NBR is used as the base rubber for the throttle body gasket as it is subjected to a relatively low thermal load of 100°C or less. Detailed specifications for each gasket are given in Table 4.

A) EGR plate gasket



B) Throttle body gasket



Fig.22 Foamed rubber gasket for a engine air intake systems

Table 4 Foamed rubber gasket specification

Gasket	EGR plate gasket	Throttle body gasket
Base rubber	FKM (F content=65%)	NBR (AN content=40%)
C.B content	60PHR	40PHR
Foaming type	Closed-cell	Closed-cell
Foaming rate	120%	120%
Foaming size	25 μm	25 μm
Thickness	50 μm	100 μm

This engine's fastening seal specifications called for a minimum of two gaskets when designing with RCM, the currently used gasket material. Replacing RCM with foamed rubber made possible a single-layer EGR plate gasket, a thinner EGR plate, and fewer fastening bolts. Similarly, the foamed rubber also permitted a single-gasket design for the throttle body gasket, as well as a thinner plate.

In durability testing with an actual engine, no loss of sealability was exhibited by the foamed rubber gasket. Further, the gasket also possessed sufficient durability and reliability with a sealing pressure of 150 kPa or higher after testing.

7. Conclusion

The authors developed a closed-cell foamed rubber gasket whose high compressive recovery and low surface hardness make it applicable to seal parts for automotive internal-combustion engines. In comparison with conventional RCM or molded rubber gaskets, this foamed rubber gasket offers superior sealability at low surface pressures, and also at low-temperature and on rough-surfaces. The authors demonstrated that the high sealability of this foamed rubber gasket can help streamline fastening seal structure by permitting fewer fastening bolts, thinner seal flanges, and unprocessed seal surfaces.

Foamed rubber gaskets are a potentially effective approach to seals for new components made of the resin materials, magnesium alloys and other new materials that automakers are expected to use to achieve lighter, less-expensive internal combustion engines for their automobiles.

8. Acknowledgments

For the inestimable cooperation with this research, the authors wish to express their most sincere gratitude to Mr. Akira Tanaka, Mr. Kenji Misumi, and everyone else at Japan Leakless Industries.

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Development of Lightweight Oil Pans Made of a Heat-Resistant Magnesium Alloy

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Shigekazu TANAKA*

ABSTRACT

A new heat-resistant magnesium alloy (hereafter referred to as “ACM522”) for die-casting based on the Mg-Al-Ca-RE system has been developed. In the 150°C temperature range, ACM522 yields high creep characteristics which are superior to the conventional AE42 heat-resistant magnesium alloy for die casting. Further it exhibits an excellent resistance to both heat and corrosion which is equivalent to ADC12 general-purpose aluminum alloy. Use of magnesium for oil pans has raised a number of problems such as axial force drop in the bolted areas and, until now, oil pans made of magnesium had not been applied to mass-produced automobiles. The authors applied the ACM522 for oil pans and achieved a 35% weight reduction compared to conventional oil pans made of aluminum.

1. Introduction

Reducing weight to improve the fuel economy of transportation vehicles is important for the effective utilization of limited energy resources and global environmental issues. In the fall of 1999, Honda released its “Insight” hybrid car that achieves the lowest fuel consumption in the world, at 35 km/liter, for a mass-produced, gasoline-engined vehicle. The engine was developed exclusively for use in a hybrid system, and the application of weight-reduction technology throughout the vehicle, including the increased use of lightweight materials, has resulted in the lightest one-liter engine in the world. An external view of this engine is shown in Fig.1. Reducing the weight of the oil pan, one of the engine’s heaviest components, was essential to achieving this light weight. The use of magnesium for the oil pan would usually have resulted in a significant decrease with time in the clamping

force at the bolted joint when using the general-purpose AZ91D and AM60B alloys, so a new, heat-resistant magnesium alloy was needed. This newly developed alloy has as its basic components Mg-Al-Ca-RE and demonstrates outstanding heat resistance and corrosion resistance. In addition, this alloy also possesses suitable molten fluidity and low hot cracking for die-cast production. Oil pans made of this alloy were 35% lighter than conventional aluminum oil pans. This paper describes the material properties of this newly developed ACM522 heat-resistant magnesium alloy and presents an overview of oil pans made from this material.

2. Background

Magnesium alloy is the lightest metallic material for practical use, having a density of 1.78 g/cm³. Its use for automobile parts, portable telephone equipment and home appliances is increasing at a rapid pace due not only to its light weight, but also to its strength-to-weight ratio, specific rigidity, electromagnetic shielding and ease of recycling. However, when applying to automobile engine and transmission parts, it is necessary to use a heat-resistant magnesium alloy that can be easily cast at a low cost. The function of the oil pan is to hold the engine lubricating oil by sealing the bottom of the engine. The design that is typically employed is to bolt the oil pan to the underside of the cylinder block. Since the temperature in that area during engine operation can reach up to 150°C, oil pans have conventionally been made of sheet steel or a die-cast aluminum alloy. When attempting to make an oil pan out of a typical magnesium alloy such as AZ91D, which has balanced mechanical properties, castability and corrosion resistance, there is a substantial decrease with time in the clamping force of the bolted joint, thus creating the need to



Fig.1 Honda integrated motor assist engine system

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develop a new heat-resistant magnesium alloy for die-casting that does not suffer from this defect.

Several types of magnesium alloy have been developed in the past for heat-resisting applications. The AS41 and AS21 alloys offer good creep resistance by crystallizing the intermetallic Mg_2Si compound at the crystal grain boundary. In particular, although the castability of AS21, which has a low aluminum content, is inferior to that of AS41, it has outstanding creep resistance. In addition, although AE42 has creep resistance equal to or better than AS21, it has problems in terms of casting due to its high material cost and susceptibility to hot cracking.⁽¹⁾ In addition, an Mg-Zn-Al-Ca-based heat-resistant alloy has recently been developed in which Zn is the main added element to supplement Al and Ca.⁽²⁾ This alloy has been reported to have better creep resistance, corrosion resistance and die-castability than AE42.

We established the following alloy development objectives for a new heat-resistant magnesium alloy capable of being applied to oil pans.

- (1) Creep resistance and corrosion resistance comparable to die-cast ADC12 aluminum alloy.
- (2) Hot cracking properties equivalent to AZ91 and AE42 to enable mass production of oil pans.

This newly developed alloy is based on an Mg-5Al composition in consideration of die-castability, and contains Ca and Ce-based misch metal for improving the heat resistance. Moreover, a small amount of manganese has been added to improve the corrosion resistance, resulting in the final Mg-5Al-2Ca-2M.M-0.3Mn alloy (ACM522).

3. Experimental Method

3.1. Test Pieces

A master alloy prepared in advance was melted in a melting furnace in a protective atmosphere of SF_6 , CO_2 and air. After melting, the temperature was maintained at about 700°C, and the tensile test pieces and creep test pieces were cast with a 350-ton cold chamber die-casting machine. The resulting test pieces were used while still in the cast state after removing the burrs generated at the parting line of the die. On the other hand, the fatigue test pieces were taken from an oil pan cast with a 1000-ton cold chamber die-casting machine and, the chilled layer on the surface was removed by machining before being used for fatigue testing.

The shapes of the tensile test, creep test and fatigue test pieces are shown in Fig.2. In addition, a sample cut from the tensile test piece was analyzed by ICP for its chemical composition. The results of that analysis are shown in Table 1.

3.2. Mechanical Properties

The tensile test was performed with an Instron type universal tester at a strain rate of 0.5 mm/min. Testing was performed at room temperature, and at 100, 150, 180 and 250°C. The test was started at each temperature after holding at that temperature for 10 minutes. Elongation was measured with an extensometer, while the 0.2% proof stress and maximum tensile strength were determined by preparing stress-strain curves.

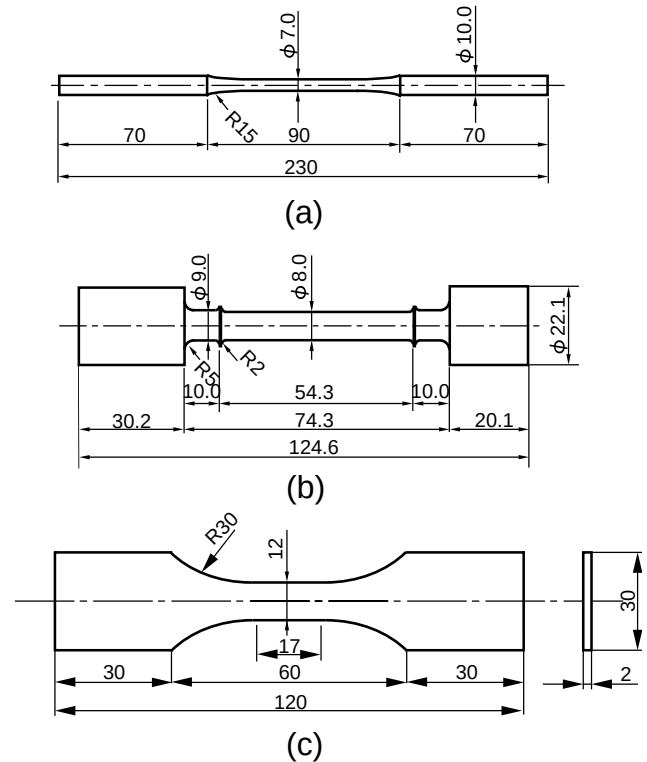


Fig.2 Die casting test specimens:
(a) tensile specimen
(b) creep specimen
(c) fatigue specimen

Table 1 Chemical composition of ACM522 alloy

Al wt%	Rare earth wt%	Ca wt%	Mn wt%	Fe ppm	Cu ppm	Ni ppm
5.3	2.6	2.0	0.17	19	18	< 1

The creep test was performed in compliance with JIS Z 2271. A single lever, 3-ton creep tester having a lever ratio of 10:1 was used for the test. Testing was performed at temperatures of 150, 180 and 200°C. The temperature was held to within $\pm 1.5^\circ C$ by PID control. Creep elongation was determined by pinching a protruding portion of the test piece with an extensometer jig, reading the amount of displacement outside the furnace and recording the mean value of two linear gauges on the left and right sides. A load equivalent to the prescribed tensile stress (50-100 MPa) was applied after allowing the test piece to reach the set temperature and then holding at that temperature for 20 hours.

The fatigue characteristics were evaluated by a pulsating tensile fatigue test at a stress ratio of 0.2 and frequency of 25 Hz. Testing was performed at room temperature, and at 120° and 150°C.

3.3. Corrosion Resistance Test

Corrosion resistance was evaluated by a salt water spray

test. The test piece was obtained by cutting out a plate measuring $25 \times 20 \times 3$ mm from the flat surface of a die-cast magnesium alloy oil pan. The cut surfaces of the test piece were ground with #1500 waterproof abrasive paper. The test method was in accordance with JIS Z 2371, whereby the test conditions include a salt water concentration of 5% NaCl a pH of 6.7-6.8, under the atmospheric conditions of 35°C and 99% humidity. After 400 hours of spraying, the test piece was boiled and washed in a 10% aqueous chromic acid solution before being washed with distilled water. The test piece allowed it to completely dry and then it was weighed.

3.4. Physical Properties

Density was determined according to the Archimedes method. Thermal constants such as the specific heat and thermal conductivity were determined by the laser flash method. Electrical conductivity was measured by the four terminal measurement method, and thermal expansion was measured with a push rod type vertical thermal expansion meter. The vertical elastic modulus and damping capacity were determined according to the resonance method.

3.5. Hot Cracking Test

The ring test method developed by Singer and Jennings was used for measuring hot cracking. This test is suited for evaluating hot cracking in the semi-solid phase. The shape of the ring-die plate is shown in Fig.3, and cast iron was used for the material. The bottom of the casting die was coated with BN as an adiabatic mold release agent, and the casting die was held at the prescribed temperature to cast a 700°C melt. Cracking was evaluated as the total length of cracks that formed in the cast ring test piece.

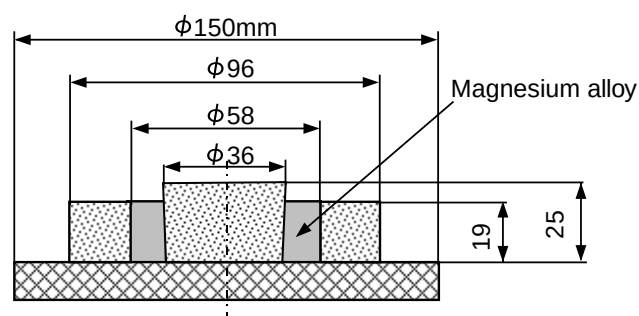


Fig.3 Ring-die plate for hot-cracking test

3.6. Bolt Load Retention Test

The test piece used in the bolt load retention test was sampled from a die-cast biscuit portion. An axial tension bolt incorporating a strain gauge in an M6 bolt (internal thread length of 2.5 D) was used. The test piece was placed in a constant temperature oven at 150°C , and the change in axial tension was measured by the method indicated in the schematic drawing in Fig.4. A similar bolt load retention test was performed on AZ91D alloy for comparative purposes.

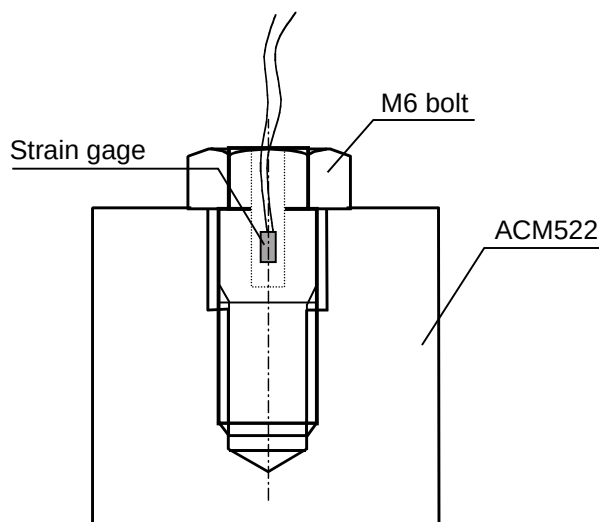


Fig.4 Bolt load retention test assembly

4. Results and Discussion

4.1. Mechanical Properties

The temperature dependence of the strength properties of ACM522 are shown in Figs.5, 6 and 7. AZ91D, AE42 and ADC12 are also given for comparison. As shown in Fig.5, the tensile strength of ACM522 is inferior to that of AZ91D. Although the tensile strength at room temperature is inferior to that of AE42, ACM522 demonstrates superior tensile strength at temperatures of 100°C . and above. In addition, as shown in Fig.6, although the 0.2% proof stress of ACM522 is equal to that of AZ91D at room temperature, it is better than AZ91D at temperatures of 100°C . and above. ACM522 demonstrates better properties than AE42 over the entire temperature range from room temperature to 250°C . As shown in Fig.7, AE42 demonstrates the highest value for elongation. The elongation of ACM522 is about 3% at

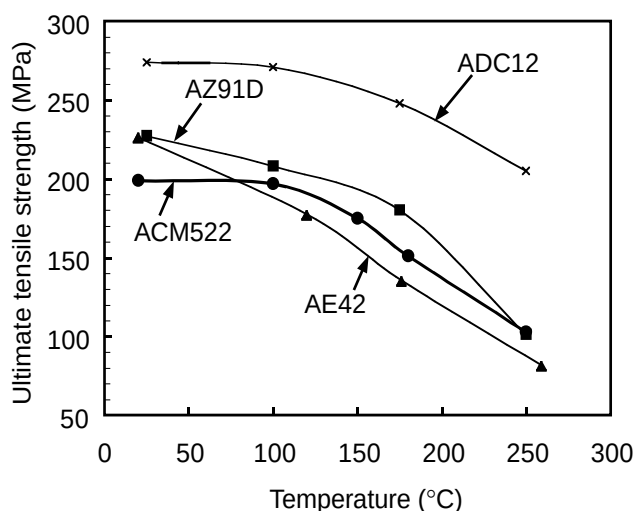


Fig.5 Ultimate tensile strength of ACM522 alloy compared with conventional die casting alloys

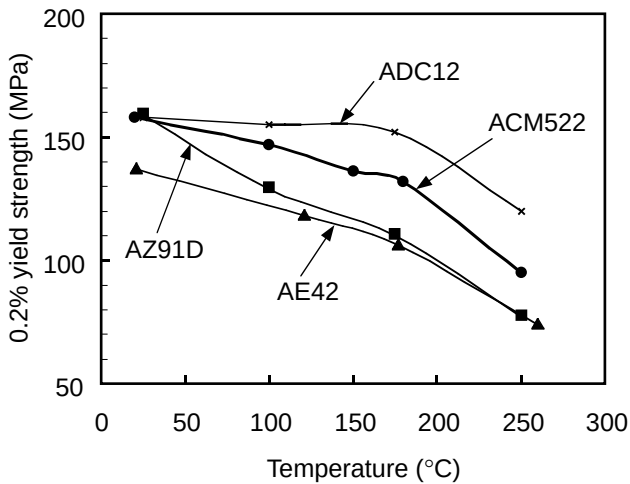


Fig.6 0.2% yield strength of ACM522 alloy compared with conventional die casting alloys

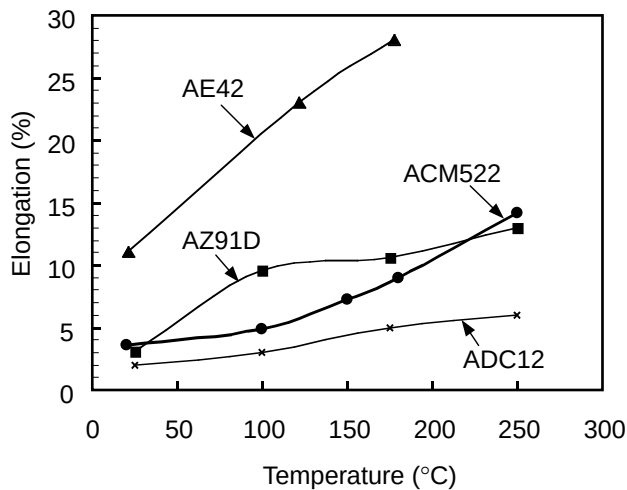


Fig.7 Elongation of ACM522 alloy compared with conventional die casting alloys

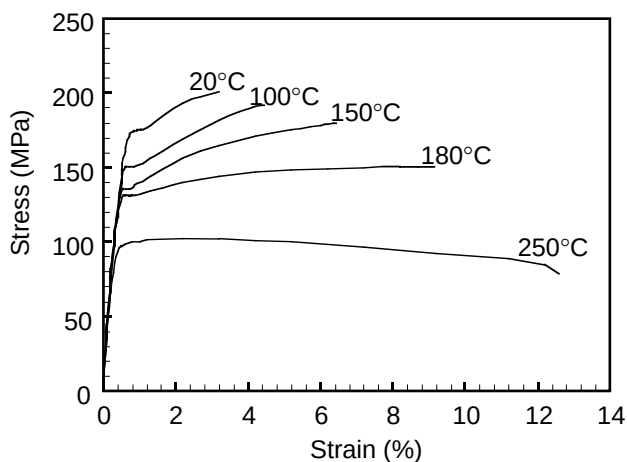


Fig.8 Stress-strain curves of ACM522 alloy

room temperature, which is equal to that of AZ91D, and this elongation increases as the temperature rises. In a comparison with general-purpose die-cast ADC12 aluminum alloy, the tensile strength of ACM522 is inferior over the entire temperature range, while its 0.2% proof stress is equal to that of ADC12 at room temperature, and the decrease in strength at high temperatures is greater for ACM522. The elongation of ACM522 is higher from room temperature to 250°C. Typical stress-strain curves at each temperature are shown in Fig.8. Since the amount of work hardening is low and ACM522 demonstrates a stress-strain curve shape similar to that of AE42, the strengthening mechanism of ACM522 is not in the form of solid solution strengthening or precipitation strengthening, but is believed instead to be the result of a grain boundary compound phase that inhibits slippage of the crystal grains.

The stress at which the creep elongation at 100 hours of ACM522 reaches 0.1% at each test temperature is shown in Fig.9 as compared with various other die-cast alloys. The stress that yields 0.1% creep elongation was calculated by a logarithmic approximation of the creep elongation (X) and stress (Y), rearranging to the formula $Y = a \times \log(X) + b$ and determining the stress at 0.1%. The creep strength of ACM522 is roughly comparable to that of ADC12 over a temperature range of 150-200°C, and the ACM522 alloy demonstrates outstanding creep strength properties that surpass those of AE42.

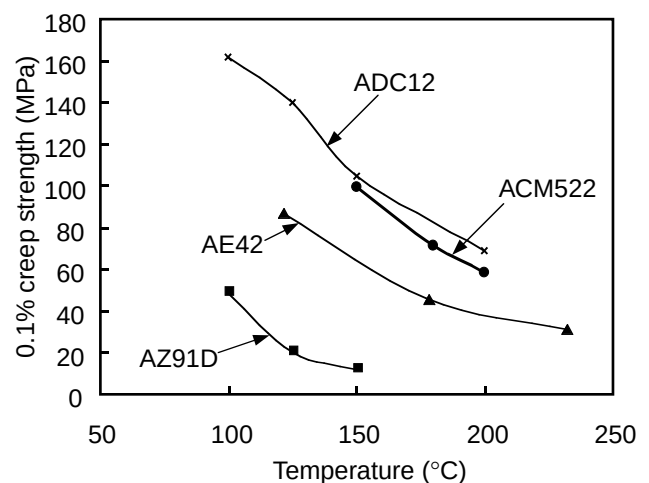


Fig.9 0.1% creep strength of ACM522 alloy compared with conventional die casting alloys

Fig.10 shows the S-N curves at room temperature, 120°C and 150°C for ACM522. The fatigue strength decreases as the temperature rises, the fatigue strength after 10^7 repetitions being 130 MPa at room temperature, 100 MPa at 120°C and 80 MPa at 150°C.

The microstructure of ACM522 was observed in order to examine the mechanism by which it is strengthened. Fig.11 is a photograph of the SEM image microstructure of die-cast ACM522. The phase consisting of black granules dispersed within the pro-eutectic α phase is the Al-Ce compound phase. The lightly colored compounds are

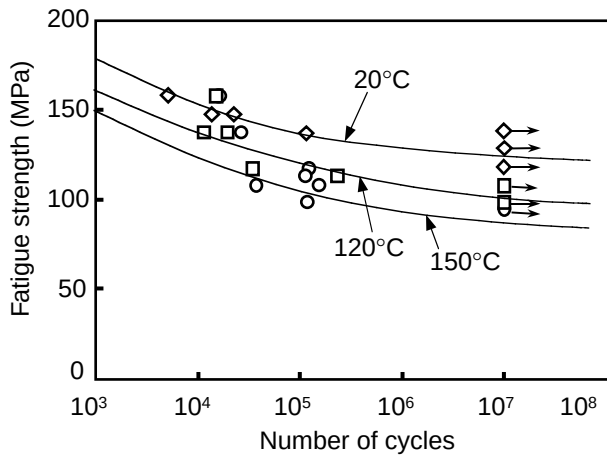


Fig.10 Fatigue strength of ACM522 alloy

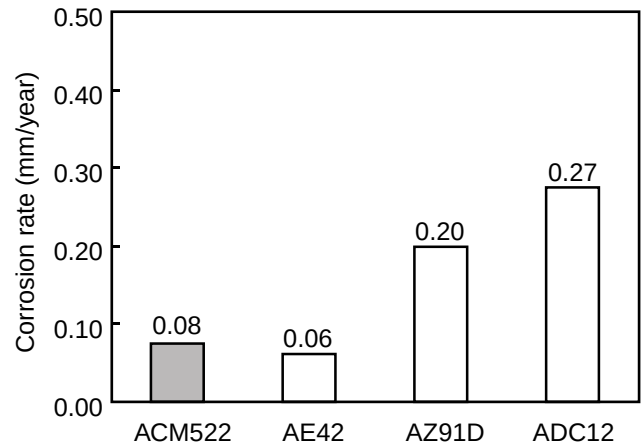


Fig.12 Corrosion rate of each alloy of the salt water spray test in 400 hours for some die casting alloys

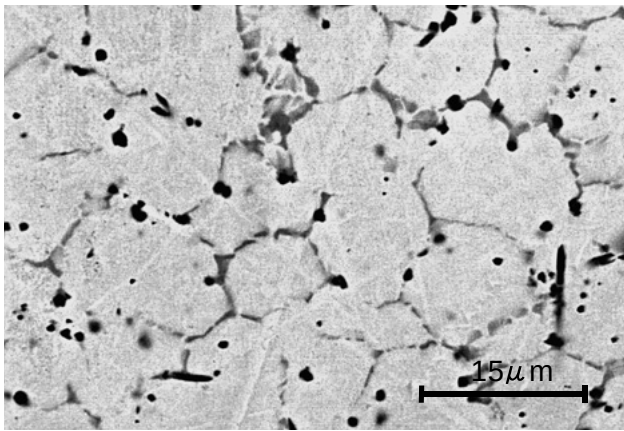


Fig.11 SEM image microstructure of ACM522 alloy

thought to be Al-Ca and Mg-Ca. As a result of this relatively fine intermetallic compound phase being finely dispersed along the grain boundaries of the α phase, the grain boundaries are strengthened. This is considered to have produced the superior creep properties.

4.2. Corrosion Resistance

Fig.12 shows the corrosion rate of ACM522 after 400 hours of the salt water spray test as compared to the AZ91D, AE42 and ADC12 alloys. ACM522 exhibits corrosion resistance that is similar to that of AE42, which is considered to be favorable, and is superior to that of AZ91D and ADC12. The corrosion resistance of magnesium alloys has been reported to be greatly affected by the presence of impurities.⁽³⁾ Typical examples of impurities include Fe, Cu and Ni. It is known that magnesium alloys are able to demonstrate outstanding corrosion resistance if a critical impurity level exists at which the corrosion resistance decreases, and if these impurities are controlled to be below that critical level. This is the reason why the corrosion resistance of ACM522 is superior to that of ADC12. In addition, the form of corrosion of ACM522 involved preferential dissolution of the α phase, indicating corrosion behavior similar to that of AZ91D and AE42.

4.3. Physical Properties

A list of physical properties is shown in Table 2. The physical properties of AZ91D, AE42 and ADC12 are also shown for comparison. Although the thermal conductivity and electrical conductivity of ACM522 are lower than the values for ADC12, they are somewhat higher than those for AE42. ACM522 also exhibits the lowest value for thermal expansion among all the other magnesium alloys. This is advantageous with respect to temperature change in a bolted joint. The damping capacity is equal to that of the other magnesium alloys and die-cast aluminum materials.

Table 2 Physical properties of some die cast alloys

Properties	ACM522	AE42	AZ91D	ADC12
Density (g/cm ³)	1.79	1.81	1.81	2.70
Specific-heat (J/kg·°C)	0.91	0.96	0.96	0.96
Thermal conductivity (W/mK)	86.9	78.6	53.8	100
Electrical conductivity (%ICAS)	22.5	20.3	12.7	27.0
Thermal expansion ($\times 10^{-6}$ /°C)	25.6	27.7	28.5	24.0
Modulus (GPa)	45	45	45	69
Damping capacity (%)	0.325	NA	0.289	0.305

4.4. Hot Cracking

The ring cracking test was performed on AZ91, AM60, AE42 and ADC12. Cracking in AZ91 and AE42 occurred over a maximum of 20 mm, while ring cracking did not occur in ADC12.

The relationship between ring cracking and the composition of ACM522 was investigated by varying the amount of Al, Ca and RE, its main components. Fig.13 indicates the relationship between hot cracking and the amounts of added Al and Ca when the amount of added RE was held constant at 2%.

When the amount of Al was 2-3%, large cracks formed even if a trace amount of Ca was present. As the amount of

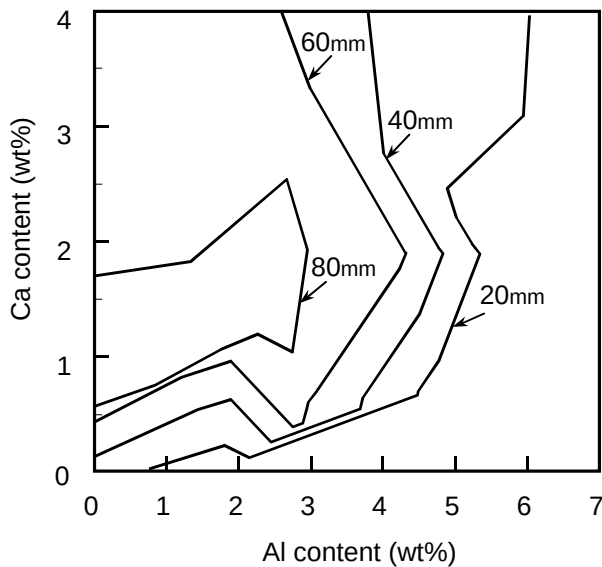


Fig.13 The effect of Al and Ca content on the total hot-crack length for the ACM522 alloy with rare earth = 2wt%

Al was increased, the length of the cracks decreased; when the amount of Al reached about 5% and the amount of Ca was set to approximately 2% or less, hot cracking was at about the same level as that in AZ91 and AE42.

4.5. Bolt Load Retention Test

Fig.14 shows the change in axial tension of a bolted joint at 150°C. The axial tension increases slightly immediately after placing in the high-temperature furnace due to the difference in thermal expansion between ACM522 and the steel bolt. In the case of AZ91D, the axial tension subsequently decreased rapidly with time, and essentially dropped to zero after 1000 hours. In contrast, there was hardly any decrease in the axial tension in the case of ACM522, even after 1000 hours. This is the result of the outstanding creep strength of ACM522 inhibiting set-in fatigue of the bolt seat and internal threads.

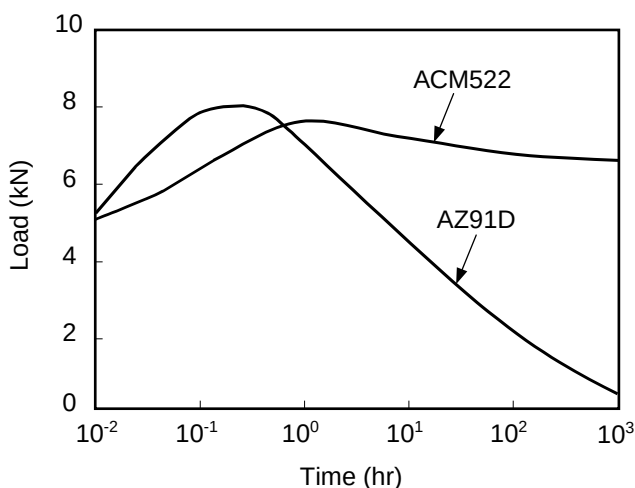


Fig.14 Bolt load retention test at 150°C

5. Overview of Magnesium Alloy Oil Pans

A photograph of the exterior of an oil pan made of ACM522 is shown in Fig.15. This oil pan weighs only 2.5 kg, despite allowing for the direct attachment of an air compressor and oil filter to the side of the oil pan, and weighs 35% less than an aluminum oil pan. An oil pan with a complex shape was created by integrated die-casting to take advantage of the high degree of freedom in shape offered by die-casting. A design was employed to minimize variation in the wall thickness in order to prevent hot cracking during die-cast forming, and the wall thickness of typical sections was set at 2.5-3.0 mm. In addition, ribs were positioned as necessary to ensure adequate rigidity.



Fig.15 Appearance of magnesium oil pan using ACM522 alloy

Since the oil pan is located at the lowest point of the engine, it is subjected to a more severe corrosive environment than other engine parts due to being splashed with water and mud on the road during vehicle travel. In addition, when magnesium alloy is placed in a corrosive environment while in electrical contact with a dissimilar metal such as carbon steel, the magnesium serves as an anode and is selectively corroded. Consequently, it is necessary to take measures to prevent galvanic corrosion by reducing the potential difference through placing a metal such as aluminum having an electrical potential close to that of magnesium between the magnesium and dissimilar metal or by inserting an electrical insulating material between the two. In the development of this new magnesium alloy oil pan, measures to prevent corrosion were employed basically by inserting anodic oxidation-treated aluminum washers under each steel bolt. Anticorrosion measures were also provided for the drain plug and other bolts that are frequently removed during maintenance by providing separate alumite-treated plates.

Although there was initial concern over greater noise and vibration levels that would result from decreased rigidity around the bottom of the engine, this could be compensated for by making small changes to the shape of the oil pan. The results of these changes are shown in Fig.16. As a result of a one-third octave analysis of the vibration noise above and below the engine side mounting bracket,

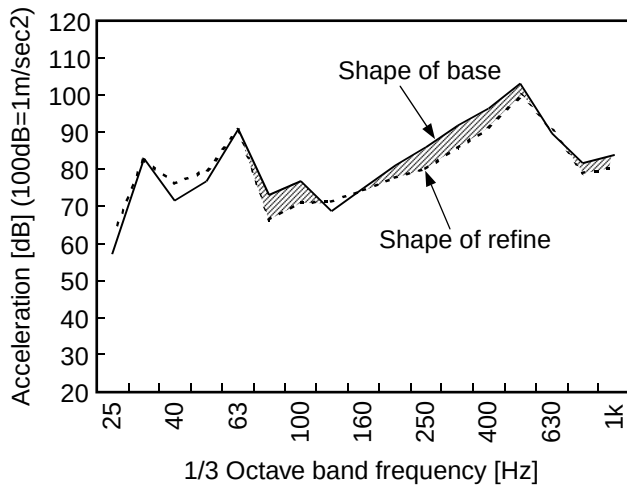


Fig.16 The effect of frequency of side mounting bracket (2000rpm, 250mmHg)

- (2) Luo, A. and Shinoda, T.: A New Magnesium Alloy For Automotive Powertrain Applications, SAE Paper 980086
- (3) Hills, J., E and Reichel, K.N : Controlling the Salt Water Corrosion Performance of Magnesium AZ91 Alloy, SAE Paper 860228

attenuation was obtained in the range of 70-500 Hz by increasing the flange thickness and by reviewing the shape of the curved surfaces and so forth. This was particularly effective in reducing the noise level in the 200-400 Hz band that presents a problem in the form of crankshaft pounding noise and engine acceleration noise.

6. Conclusion

This report has presented an evaluation of the material properties of the newly developed heat-resistant magnesium alloy, and an overview of a lightweight magnesium oil pan made of this alloy.

- (1) The newly developed alloy is based on an Mg-5Al composition for die-castability, and contains Ca and Ce-based misch metal for improved heat resistance. Moreover, a small amount of Mn is added to provide good corrosion resistance, resulting in the final Mg-5Al-2Ca-2Mn-0.3Mn material (ACM522).
- (2) ACM522 demonstrates creep properties and corrosion resistance comparable to those of ADC12, a general-purpose die-cast aluminum alloy.
- (3) A magnesium oil pan made of ACM522 alloy was 35% lighter than a conventional aluminum oil pan.

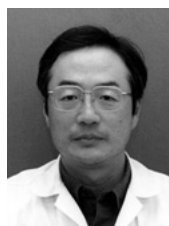
This lightweight magnesium oil pan is employed in the engine installed in the "Insight" hybrid car that has achieved the lowest level of fuel consumption in the world, at 35 km/liter, for a mass-produced, gasoline-engined vehicle. It went into mass production in November 1999.

It is expected that the application of this magnesium alloy will be expanded in the future to include engine and transmission parts requiring a high level of heat resistance.

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The Development of Converter Canning Technology for Thin Wall Substrate

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Jun'ichi NAKADE*

Tatsuya OKAYAMA*

ABSTRACT

By applying a tourniquet type (rolling type) catalytic converter can, which is adaptable to the diameter of the substrate, a stable holding pressure can be successfully obtained. Additionally, by using of a double layer mat material, consisting of an inner layer of a high heat resistant alumina fiber and an outer layer of the current vermiculite containing material, the rate of holding pressure loss due to mat degradation is reduced. A combination of these two techniques allows the canning of a 1200 cell, thin wall substrate without breakage and, further, enables in-use durability requirements to be met.

1. Introduction

With increasing demands for reductions in regulated toxic substances in automobile exhaust for environmental reasons, improving the performance of exhaust cleaning catalysts is a critical area of development. One effective means of improving the performance of exhaust cleaning catalysts is the use of thin wall substrates with high cell densities, enhanced warm-up properties, and faster light-off⁽¹⁾⁽²⁾. However, thin walled substrates are difficult to can with conventional converter technology due to their reduced strength. A typical conventional canning method is to wrap the substrate in a mat material that combines alumina silica fiber and vermiculite and then compress the mat material with a clamshell type case so that the substrate is held by the surface pressure created by the compression of the mat.

One reason for the inapplicability of conventional canning technology to low-strength substrates is a structural problem with the clamshell case, which cannot assure stable surface pressure because of unequal pressure distribution and other factors. Additionally, the considerable decline in pressure due to fiber breakage and other forms of mat degradation necessitated high initial pressures to allow for subsequent degradation⁽³⁾.

In response, the authors both reevaluated converter cases from a structural standpoint and combined this with an improved mat material to develop converter canning technology for application to low-strength, thin-wall substrates. This paper describes said technology.

2. Breaking Strength of Thin Wall Substrates

Thinner walls generally mean reduced strength. Further, cells tend to deform during the extrusion molding of the substrate as a result of higher cell density. This deformation facilitates buckling and subsequently leads to reduced

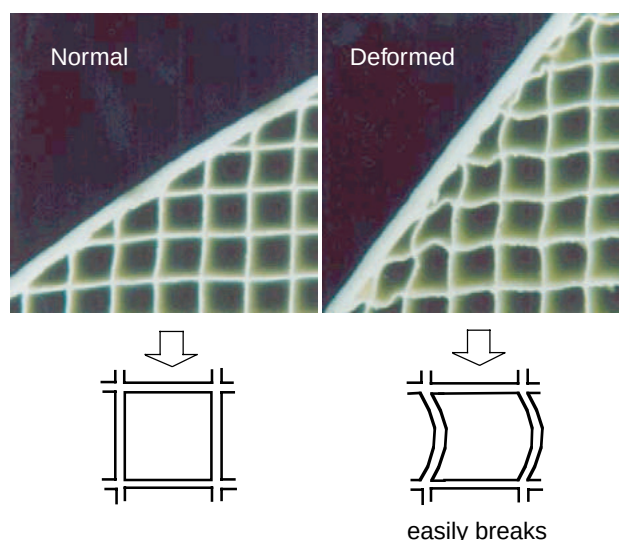


Fig.1 Cell shape of 1200 cell substrate

strength (Fig.1). Cell deformation is particularly likely to occur near the outer wall due to the unequal flow rate of the substrate material when it passes through the slit face plate. Isostatic strength, which is used to assess substrate strength is the pressure at which breakage occurs when the substrate's outer surface is subjected to water pressure or some other type of uniform pressure. Therefore, a cell's weak areas determines the cell's isostatic strength.

Isostatic strength according to wall thickness is shown in Fig.2. A 600-cell substrate has an isostatic strength of at least 20 kgf/cm² and can be canned at a surface pressure of up to 20 kgf/cm² without breaking. The lower isostatic strength of 900- and 1200-cell substrates limits them to a canning pressure of 5 kgf/cm² or less.

* Tochigi R&D Center

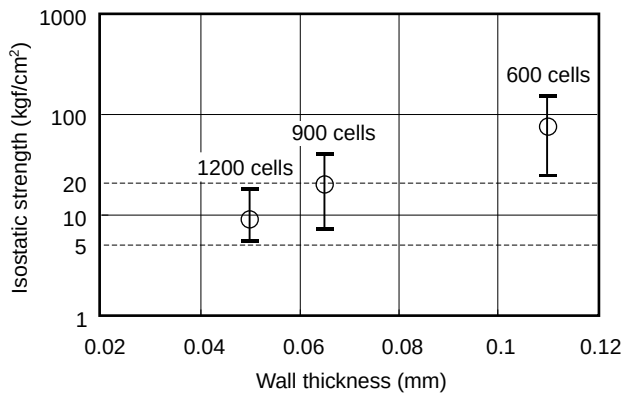


Fig.2 Actual isostatic strength of thin wall substrates

3. Technical Issues in Low Surface Pressure Canning

One method commonly used to hold a ceramic substrate in a stainless steel case consists of arranging a metallic mesh, mat or other holding material between the case and substrate, and utilizing the resultant pressure to hold the substrate in place.

A mat compressed and packed into a converter generates a surface pressure according to its packing density. When the surface pressure generated by the holding seal mat (the "holding pressure") exceeds the strength of the substrate, the substrate breaks. Further, as the substrate is subjected to forces such as vibration and exhaust gas pressure that try to move it ("moving force"), if the holding pressure is insufficient, the substrate will not be held stable. Therefore, surface pressure is the most important specification in converter design and it must always be kept less than the substrate strength yet higher than the minimum pressure needed to withstand moving force.

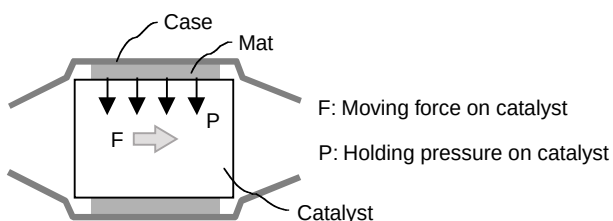
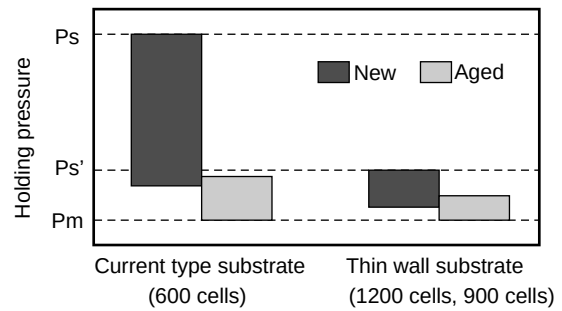


Fig.3 Converter schematic illustrating major design factors

Fig.4 expresses the relationship between substrate strength and holding pressure. The height of the bars in the graph indicate the surface pressure variation that results from uneven surface pressure distribution, dimensional variations in the substrate and case, and variations in mat weight. Maximum surface pressure, including this uneven surface pressure distribution and variations in components, must not exceed the upper limit for surface pressure, as determined by substrate strength. The right-hand bar in each bar pair represents the reduced surface pressure of the degraded holding material, and demonstrates the need for a minimum



Ps : Maximum pressure for minimum strength of 600 cell substrate
 Ps' : Maximum pressure for minimum strength of thin wall substrate
 Pm : Minimum pressure required to keep substrate in case

Fig.4 Required radial pressure range for a thin wall substrate

holding pressure, i.e., the holding pressure that must be maintained to resist moving force after degradation, and which must also allow for variation in components. As stated above, the holding pressure specification must be below substrate strength but high enough to resist moving force. This is a very narrow range for low-strength, high-cell-density, thin wall substrates, making canning difficult. These design guidelines entail three technical issues that must be resolved to achieve low surface pressure canning:

1. Uniform distribution of surface pressure
2. Adaptability to component variation and consistency of mat packing density
3. Minimization of surface pressure loss due to mat degradation

4. Improved Technology for Low Surface Pressure Canning

4.1. Assuring Uniform Distribution of Surface Pressure

Fig.5 is a schematic diagram of the clamshell type

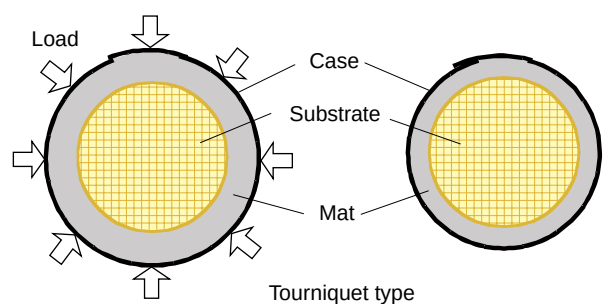
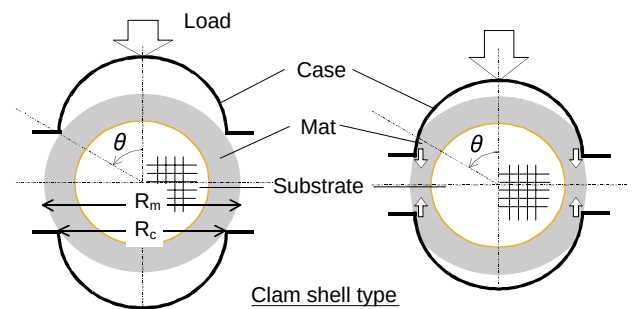


Fig.5 Schematic of canning technology

currently used in mass production and tourniquet type canning.

In clamshell type canning, a substrate is wrapped with a holding mat and canned through a pressurization process with a case having an upper and lower portion. The case outside diameter R_c is designed to be smaller than R_m , the outside diameter of the mat when it is wrapped around the substrate. The mat, once compressed and packed between the case and the substrate, generates a holding pressure. However, the portion of the case that first contacts the mat is the portion from the perpendicular to $\theta = \arcsin(R_c/R_m)$. This portion of the mat is dragged and compressed, by the case, as it moves perpendicularly. This results in variations in the mat's packing density and, consequentially, in an uneven distribution of surface pressure. This tendency toward uneven surface pressure distribution makes the clamshell technique unsuitable for low surface pressure canning.

The authors therefore examined the tourniquet type case as a construction that would assure uniform surface pressure distribution. This technique entails first creating the case by rolling a stainless steel sheet into a cylindrical shape and then inserting a substrate wrapped with a mat into this case, which is then compressed to complete the canning process. Unlike clamshell canning, this technique permits uniformity of surface pressure as the direction of holding seal mat compression during canning is aligned with the center of the honeycombs.

Fig.6 indicates the surface pressure distribution of clamshell and tourniquet type cases, and shows that the low angle area (i.e., $\theta = 30^\circ$) of a clamshell type case has a high pressure area that is generated due to variations in the mat. An area of high surface pressure also exists at the apex where $\theta = 90^\circ$. The higher surface pressure at the apex is most likely due to the substrate being compressed by the low-angle portion of the mat that is dragged by the case. In contrast, the tourniquet type case exhibits uniform surface pressure distribution.

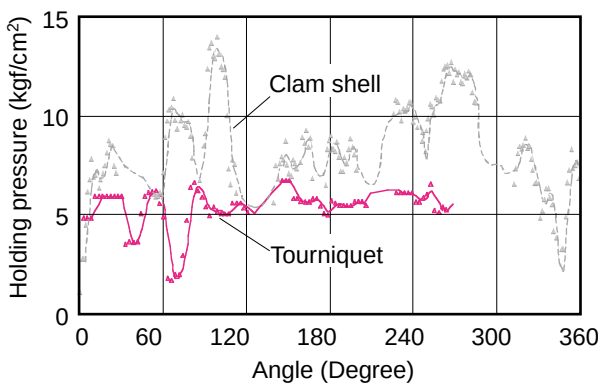


Fig.6 Comparison of clam shell and tourniquet canning techniques

In addition to the tourniquet type, the insertion type case construction also assures uniform surface pressure distribution. Insert type cases are constructed by first forming a cylindrical case of a predetermined diameter and then inserting the mat-wrapped substrate into the case while compressing the mat. Although a uniform distribution of

surface pressure is assured by the cylindrical case and the fact that the mat is compressed in the same direction as the substrate center, a rise in surface pressure occurs in the canning process as the mat must be sufficiently compressed to fit into the case during insertion.

Table 1 summarizes this comparison of case constructions. Based on these results, the tourniquet type case construction to achieve low surface pressure canning. Further, the tourniquet type construction facilitates variable control of case diameter as discussed in section 4.2.

Table 1 Comparison of each case structure to adapt lower level pressure

	Pressure distribution	Increased pressure while canning	Variable case diameter	Lower level pressure absorption
Clamshell type	Unevenly	Higher	Difficult	Impossible
Tourniquet type	Uniform	Lower	Easy	Possible
Insertion type	Uniform	Higher	Difficult	Impossible

4.2. Consistency of Mat Packing Density

Mat surface pressure is determined by packing density during compression. Variation in packing density can result from variations in substrate diameter, case diameter, and mat surface density and subsequently result in changes in surface pressure. Consistent surface pressure thus depends on the ability of the construction to adapt to variations in these components. To arrive at a method capable of efficiently achieving consistent surface pressure, the authors calculated the effect of each component on mat packing density, as determined with the following equation.

$$G.B.D. = \frac{W.A.}{CL} = \frac{2 \times W.A.}{D_{case} - D_{cat}}$$

G.B.D.: Mat packing density

W.A.: Mat surface pressure (weight per unit area)

CL: Clearance between the case and the substrate

D_{case} : Case inner diameter

D_{cat} : Substrate outer diameter

Three parameters were compared in terms of variability by calculating the change in packing density that occurred when one of the three parameters was changed within its range of variation (according to mass-production data for the component in question) while leaving the other two parameters constant. These results, in Fig.7, show how substrate diameter, which changes according to dimensions during extrusion molding and compression during baking, has five times the dimensional variability of the case. Therefore, as the substrate has the greatest impact on mat packing density, adaptability to this dimensional variation is effective.

The preceding shows tourniquet type canning to be a

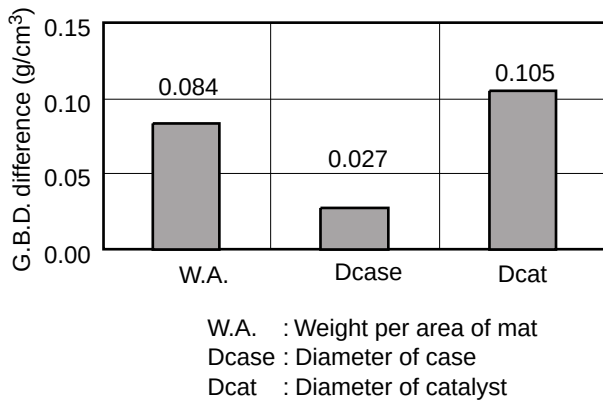


Fig.7 Effect of G.B.D. difference on each parts' variability

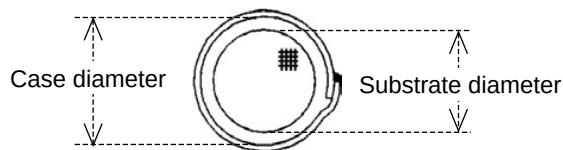
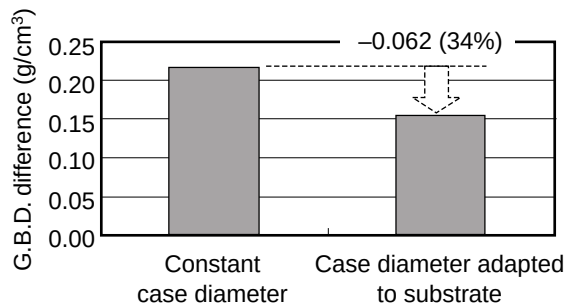


Fig.8 The effect of case diameter adapted to substrate

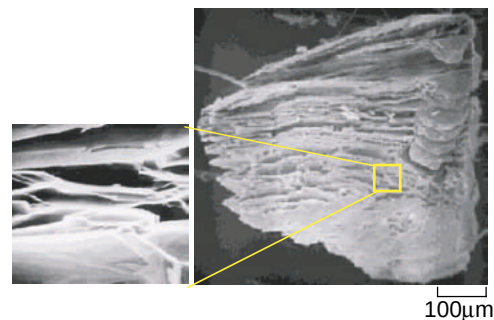
superior canning technique that makes it possible to measure individual substrate diameters and adjust case diameter accordingly. Moreover, this technique reduced variability in mat packing density by 0.062 g/cm³, and reduced mat packing density variation (i.e., surface pressure variation) by 34% beyond what would be possible with a constant case diameter (Fig.8).

Another possible method of tourniquet type canning, aside from controlling dimensions as described above is to wrap a belt around the case and pull with a constant load. Although the constant load is meant to assure uniform holding pressure by absorbing component variation, other variation factors such as case elasticity, friction between the belt and case, and load variations are generated rendering this technique unsuitable for the low surface pressure canning intended for application with a surface pressure of 5 kgf/cm² or less.

4.3. Minimizing Holding Seal Mat Degradation

Lastly, holding seal mat improvements designed to assure the requisite holding pressure after degradation are discussed. Current mass-production holding seal mats are comprised of an alumina silica ceramic fiber to which vermiculite has been added. Vermiculite is a natural mineral having a

multilayered structure of flat mica crystals containing for example metallic ions. Vermiculites possessing suitable high temperatures expansion properties are used in the mat (Fig.9). Vermiculite is added to the mat to prevent the drop in holding pressure that would otherwise occur at high temperatures due to increased clearance between the stainless steel case and the ceramic substrate, each of which undergo different rates of thermal expansion. Alumina silica ceramic fiber is a relatively inexpensive, amorphous ceramic fiber that has a 50% alumina content and which is commonly used as a heat insulator. Ceramic fibers such as this are created by hardening molten alumina silica into a fibrous form by means of compressed gas or centrifugal force, the result being a highly irregular structure, i.e., a vitreous state. A vitreous state is one in which a liquid has been supercooled to achieve a viscosity close to that of a solid without undergoing crystallization. As it is heated, a vitreous substance undergoes changes in state, first softening and then under going fluidization or crystallization. When the fibers such as those used in the mat material are subjected to a strong load at temperatures below 900°C, they start to soften. This leads to contraction that causes the fibers to break or deform. Above 900°C, precipitation of mullite crystals leads to extreme contraction, causing further breakage and deformation that is manifested as a marked functional degradation. Thus, mat improvement must begin with the



Chemical formula of vermiculite (example)

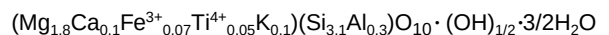


Fig.9 Microscope photograph of expanded vermiculite

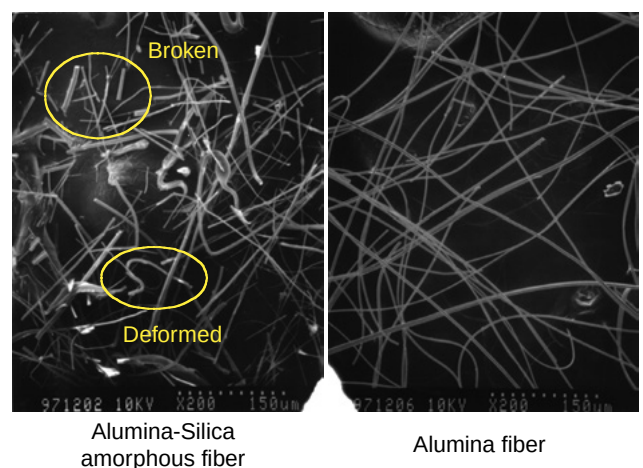


Fig.10 Fiber condition after bench durability test

selection of a fiber having superior thermal resistance. The authors therefore chose to investigate alumina fiber: a fiber that undergoes almost no contraction below 900°C, offers suitable elasticity, and has a high alumina content and a correspondingly low crystallization rate. SEM photographs of fiber conditions after bench durability testing (Fig.10) show that breakage and deformation of the alumina silica fiber occurred in the portions of the mat that had a temperature of 800°C. If extensive, this condition would cause the mat to lose its elasticity, reducing surface pressure to the point that the required holding force could not be obtained. The SEM photo of alumina fiber, however, reveals no such breakage or deformation, demonstrating that the material retains sufficient elasticity even at high temperatures.

Fig.11 shows the results of a comparative pressure retention performance test. In this test, mats were subjected to repeated heating and cooling and to changes in clearance comparable to that resulting from thermal expansion of a case. The residual pressure ratio was calculated from the load determined with a load cell. These results demonstrate the superior pressure retention performance of a mat made with alumina fiber. In addition, observation of the fiber after this bench durability test showed no difference between the pre- and post-test conditions, reflecting an improved thermal resistance.

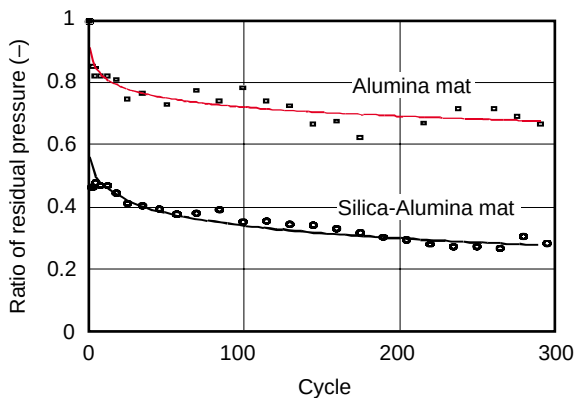


Fig.11 Durability test on thermal cyclic condition

The authors therefore decided to use this alumina fiber to create mats with improved pressure retention performance.

The results of a pressure test at different temperatures and different clearances (Fig.12) show that a mat made of alumina silica fiber containing vermiculite (the conventional mat) generated a maximum surface pressure at the high temperature after one cycle, whereas the alumina fiber mat exhibited maximum surface pressure at an early stage, that is, at a normal temperature. The residual pressure ratio after no more than 5 cycles was calculated relative to this maximum pressure ratio, which was assigned a value of 1 in both cases. The conventional mat underwent a large drop in pressure at the high temperature after two cycles, falling to a residual pressure ratio of 30% after five cycles. In contrast, the vermiculite-free alumina fiber mat underwent an expansion in high temperature clearance and exhibited a 30% residual pressure ratio at a high temperature. Thus, although

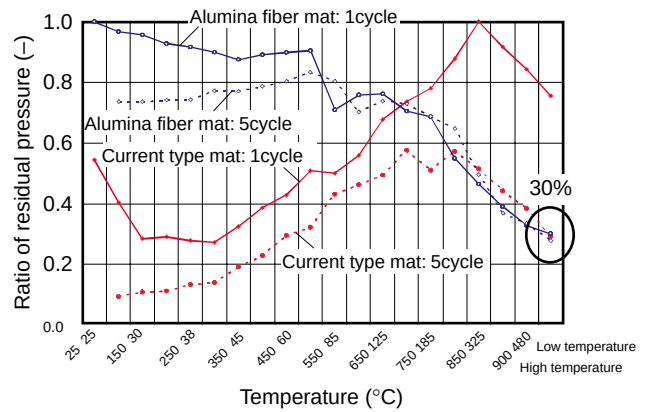


Fig.12 Pressure test on thermal cyclic condition

almost no pressure loss occurred that related to the cycles and ultimately, pressure retention performance was not improved.

To address thermal expansion of the case, the authors considered two-layer constructions such as that in Fig.13, in which a highly thermal resistant alumina fiber mat was placed on the substrate side, which is exposed to high temperatures, and a conventional mat was placed on the case side. The vermiculite in the conventional mat prevents pressure loss during thermal expansion of the case, while the alumina fiber minimizes degradation by acting as a heat insulator. Additionally, the elasticity of the alumina fiber mat provides holding pressure. Thus, this two-layer structure prevents pressure loss at high temperatures due to thermal expansion of the case while also minimizing pressure loss due to degradation.

Next, the authors sought to optimize vermiculite content and the weight per area ratios between the two layers.

Vermiculite, as already stated, is added to prevent pressure loss at high temperatures due to thermal expansion of the case, an appropriate amount must be added: adding too little fails to have the desired effect, while too much results in excessive expansion force. The authors placed a two-layer mat between upper and lower hot plates of different temperatures to determine the relationship of hot plate

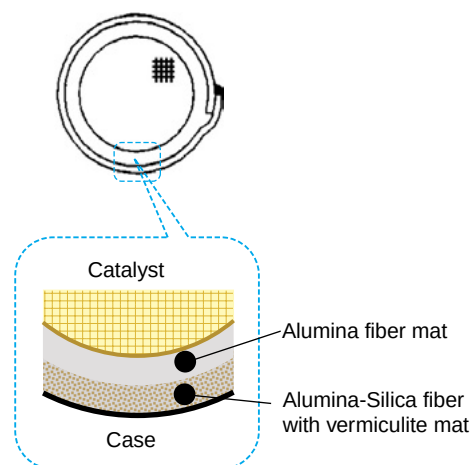


Fig.13 Illustration of double layer mat

temperature to changes in the two-layer mat's pressure under conditions in which clearance increased according to hot plate temperature (Fig.14). Pressure declined gradually prior to vermiculite expansion, but began to increase when the average temperature between the upper and lower plates reached 350°C; maximum pressure was reached at roughly 600°C. Mats differing in vermiculite content were prepared to determine the difference between pressure at normal temperature and maximum pressure at roughly 600°C. This showed that a 10% to 15% vermiculite content in terms of the total weight per area of the two layers resulted in a maximum surface pressure (at high temperatures) comparable to surface pressure at initial room temperature (Fig.15).

Using the vermiculite content thus determined, the authors next investigated the weight per area ratio in the two-layer mat. This was done by preparing a two-layer mat comprising alumina fiber mats of differing weights per area, performing a cyclic test under temperature change and clearance change conditions similar to those in Fig.14, and then determining surface pressure after 600 cycles. The results (Fig.16) show that a ratio between the two layers of 4:6 (alumina fiber mat:conventional mat) yielded the best pressure retention performance. A higher weight per area ratio for the alumina fiber mat resulted in less per-cycle surface pressure loss at high temperatures, but beyond 40%, the effect was diminished. This is believed to be because of the lower contribution ratio of the heat insulator layer to the conventional mat layer. In contrast, a high weight per area

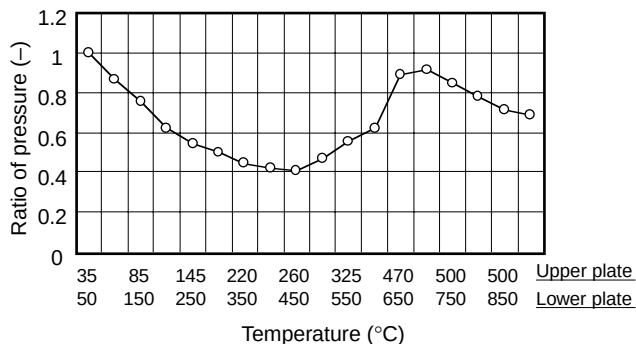


Fig.14 Pressure balance of room temperature and high temperature

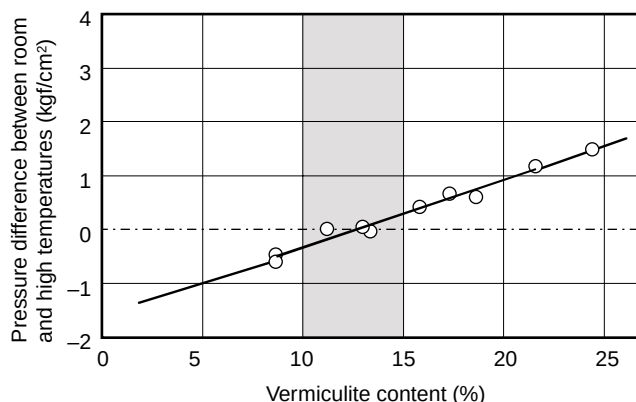


Fig.15 Best vermiculite content

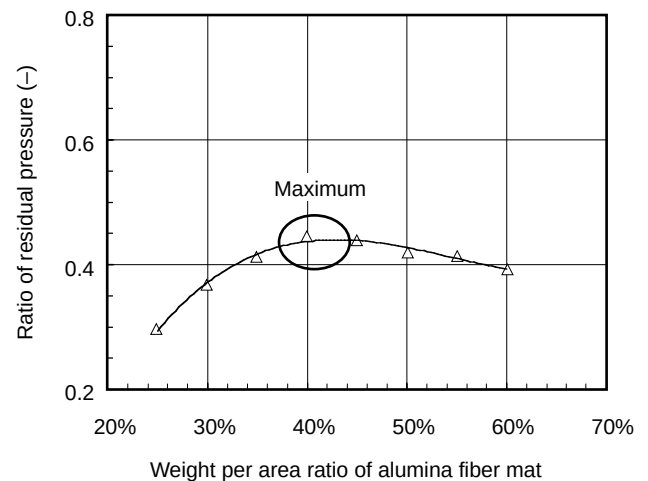


Fig.16 Pressure keeping performance according to the weight per area ratio of the alumina fiber mat

ratio for the alumina fiber mat tends to result in a slightly larger drop in surface pressure after maximum surface pressure at high temperature is reached in the first cycle. A mat's surface pressure normally undergoes a slight drop after reaching its maximum point; this is the mat's creep property. Particularly at high temperature in the first cycle, creep is large because the fiber binder inside the mat burns, freeing the fiber from its constraint, while the fiber itself moves due to thermal expansion, relocating itself so as to weaken the load. For this reason, alumina fiber will most likely undergo less breakage and deformation and have greater creep at a high temperature in the first cycle than the conventional mat. As a result, the per-cycle surface pressure retention performance at high temperature should offset the surface pressure drop due to creep at a high temperature in the first cycle, so that 40% is the highest surface pressure retention performance from initial room temperature through all 600 cycles.

To summarize the preceding, the optimum component ratio for the two-layer mat can be expressed with the following equation.

$$W.A.(total) = W.A.(a) + W.A.(c) \\ = W.A.(a) + W.A.(cf) + W.A.(cv)$$

$$W.A.(a)/W.A.(total) = 40\%(\pm 10\%)$$

$$W.A.(cv)/W.A.(total) = 10-15\%$$

W.A.(a): Alumina fiber mat weight per area

W.A.(c): Conventional mat weight per area

W.A.(cf): Weight per area of the conventional mat's fiber and binder

W.A.(cv): Weight per area of the conventional mat's vermiculite

W.A.(total): Weight per area of the overall two-layer mat

Finally, Fig.17 is a comparison of the surface pressure retention performance of a conventional mat and a two-layer mat with optimized components. This shows that in contrast to the conventional mat, whose surface pressure dropped to

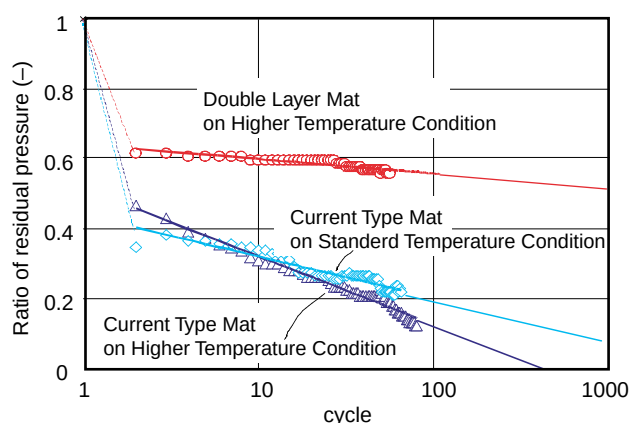


Fig.17 Pressure test under thermal cycle conditions

zero after 400 cycles, the two-layer mat retained roughly 50% of its surface pressure even after 1000 cycles.

5. Conclusion

In developing canning technology for thin wall substrates, the authors arrived at the following conclusions:

- The tourniquet type case structure is capable of achieving a uniform distribution of surface pressure and so is suited to low surface pressure canning.
- In a tourniquet type case, change in mat packing density was reduced 34% by adjusting case diameter to match substrate diameter.
- The drop in surface pressure due to degradation can be minimized with a two-layer mat having an alumina fiber mat on the catalyst side. Furthermore, optimizing the ratio between the two layers and the vermiculite content makes possible a 50% surface pressure retention rate after 1000 thermal cycles.
- Combining the above technologies permits canning at a surface pressure of 5 kgf/cm² or less, thereby allowing the use of low-strength, thin wall substrates, such as 1200-cell substrates.

Acknowledgments

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Development of an ABS/PA6 Polymer Alloy for Body Panel Applications

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ABSTRACT

An ABS/PA6 polymer alloy has been developed as a resin for body panels of vehicles with an aim toward overall vehicle weight reduction. Compared to conventional resins, this resin has improved surface appearance and the balance between dimensional stability and impact resistance. A “Sink Mark Evaluation Standard” was established for surface appearance, and the ratio of acrylonitrile-butadiene-styrene (ABS) resin and Polyamide6 (PA6) was adjusted to satisfy this standard. An ABS resin ratio of 50% or more was found to be effective for sink mark performance. The filler type and amount were important for the balance between dimensional stability and impact resistance with talc possessing particle diameter of $2\mu\text{m}$ being found to be effective target achievement. This resin shows a 45% weight reduction over steel panel products in the front fender.

1. Introduction

The use of resin body panels has been the subject of investigation for more than 50 years as a means of reducing vehicle weight. Fiber Reinforced Plastic (FRP) and Sheet Molding Compound (SMC) made of thermoset resin reinforced with glass fiber have been used for some time. Recently, advances in materials technology and the desire for recyclability have prompted the use of non-fiber-reinforced thermoplastic resins.

Honda has applied various thermoplastic resins to its mass production models, such as polycarbonate (PC)/acrylonitrile butadiene styrene (ABS) resin polymer alloys⁽¹⁾ for front fenders and other parts, and polyphenylether (PPE)/polyamide 6 (PA6) polymer alloys for such parts as the side sill garnish. It is reported that resin materials that have been developed for body panels by other companies include PPE/polyamide 66 (PA66) polymer alloys, PC/polybutylene terephthalate (PBT) polymer alloys, and other thermoplastic resin materials⁽²⁾.

This paper describes a new thermoplastic resin material which offers superior surface appearance and a balance between the coefficient of linear thermal expansion and impact resistance, and which is capable of reducing the weight of body panels.

2. Development Aims and Concept

2.1. Weight Reduction and Superior Aerodynamic Performance

When fuel efficiency goals are given high priority in car development, it is essential to meet the initial goals for vehicle weight and aerodynamic performance. Aluminum is generally considered the most effective light-weight material because of its relative stiffness.

The graph in Fig.1 shows the weight reducing effects with an equivalent surface stiffness⁽³⁾. This graph indicates that when a steel panel is assigned an arbitrary value of 100,

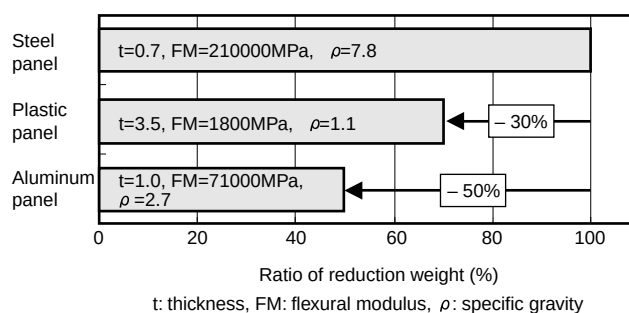


Fig.1 Weight reduction effect at equivalent surface stiffness

resin and aluminum have a weight reducing effects of approximately 30% and 50%, respectively.

However, aluminum has limited formability. Thus, in order to create designs that provide the requisite aerodynamic performance, it is necessary to use an appropriate resins with superior shape flexibility in the appropriate place. Fig.2

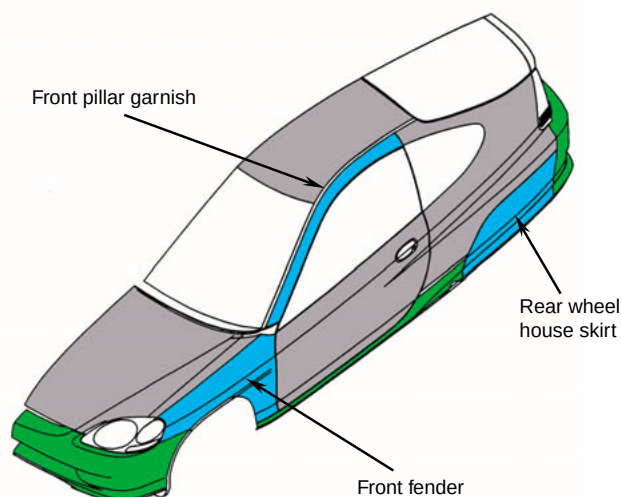


Fig.2 Application of plastic parts for body panel

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provides examples of the shape effects possible with resin components.

2.2. Weight Reduction with Thinner Parts

From the viewpoint of design technology, thinner parts are another possible means of decreasing weight. However, a simply reducing a panel's thickness degrades a part's stiffness, some sort of innovation is needed.

For instance, with resin parts, ribbing is sometimes used on the backside of a part to compensate for the reduction in stiffness due to the reduction in thickness. Such ribbing permits weight reduction by allowing thinner parts that, at the same time, retain the required stiffness.

However, caution is required in creating a design as such ribbing can cause sink marks that degrade the surface appearance of a part.

The authors, by quantifying these sink marks to develop clear standards for product appearance, aimed to develop a resin material that would satisfy surface appearance requirements even when back ribbing is used.

2.3. The Proper Balance Between Dimensional Stability and Impact Resistance

Resin materials exhibit greater dimensional changes under temperature changes than metallic materials and so must have a sufficiently low coefficient of linear thermal expansion in order to minimize the loss of fit and finish quality due to dimensional change. Fillers have generally been considered as a way of reducing the coefficient of linear thermal expansion⁽⁵⁾⁽⁶⁾.

However, because a filler also reduces the impact resistance of resins, the author strove to develop a resin material with both a low coefficient of linear thermal expansion and exceptional impact resistance.

3. Defining Target Value

3.1. Quantification of and Standards for Sink Marks

As there have been no clear criteria for judging sink mark appearance quality, oftentimes judgments are based on visual inspections. This resulted in variations as related to the individual performing the inspection. Therefore, the author decided to define a standard for sink marks that are allowable.

In measuring sink marks in the surfaces of molded parts using a surface roughness tester (Fig.3), depth [V] and width [H] were determined to be the factors necessary to classify sink marks.

Following the classification method in Fig.3, the author created various sink mark samples. It then measured sink

mark depth [V] and width [H] and also had the sink marks visually assessed by a panel of 50 persons to determine the extent of sink mark prominence that is allowable with respect to surface appearance. Based on these results, the range where at least 80% of the panelists judged the sink mark as "allowable with respect to surface appearance" was defined as the "allowable range for sink marks," and the boundary line of that range was defined as the "Sink Mark Evaluation Standard." This allowable range and boundary line are shown in Fig.4. The boundary line is represented as a straight line from the origin to the point with a 7 mm width [H] and 1 mm depth [V].

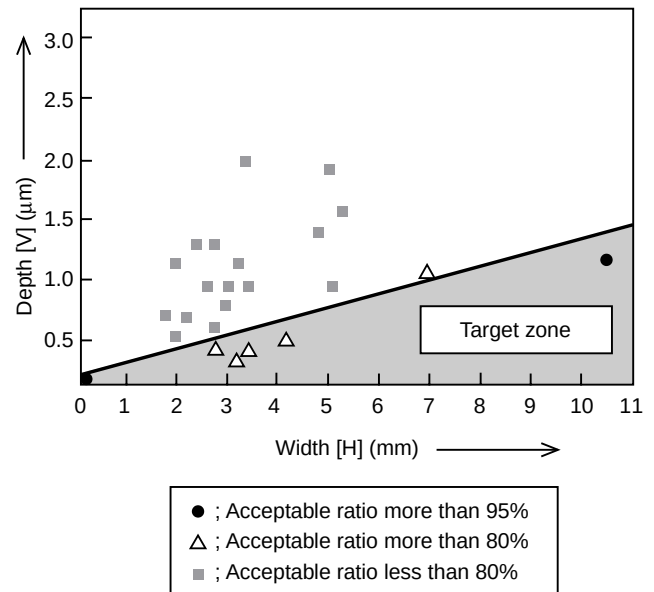


Fig.4 Distribution map of various sink mark profiles

3.2. The Proper Balance Between the Coefficient of Linear Thermal Expansion and Impact Resistance

Table 1 gives, for the parts shown in Fig.2, the maximum length, allowable clearance, the variation in temperature between room temperature and maximum ambient temperature, and the coefficient of linear thermal expansion required in accordance with temperature variation. These data yield a target coefficient of linear thermal expansion for resin materials of 5.5×10^{-5} cm/cm·°C or less.

The target impact resistance was 50J, comparable to that of the ABS/PC polymer alloy, which is being applied to front fenders.

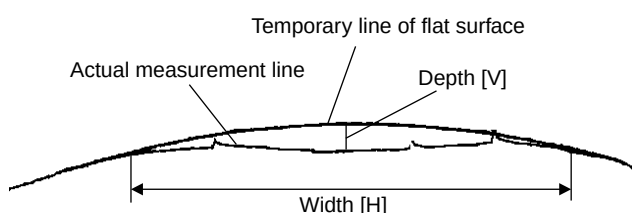


Fig.3 Surface profile of sink mark

Table 1 Limits of CLTE* for applied parts

NO.	Part name	Length (mm)	Allowable dimension change (mm)	Variation in temperature (deg.)	Limits of CLTE ($\times 10^{-5}$ cm/cm·°C)
1	Front fender	850	2.5	50	5.8
2	Front pillar garnish	1 450	4	50	5.5
3	Rear wheel house skirt	700	2	50	5.7

*CLTE: Coefficient Linear Thermal Expansion

4. ABS/PA6 Polymer Alloy

4.1. Reasons for Selecting the ABS/PA6 Polymer Alloy

In the injection molding of a resin, molten resin is injected into a die. It is solidified by cooling as the die with which it is in contact absorbs the resin's heat. It is then removed as a molded part. When the molded part has ribbing, a difference in thermal capacity exists because of the different cross-sectional thickness of the ordinary surface and the rib members. Consequently, the molten resin in each cools and hardens at different rates, resulting in a difference in volumetric shrinkage, which, it is believed, is manifested on the surface as sink marks⁽⁷⁾.

Therefore, resin materials that resist sink mark formation are said to be those that are resistant to the effects of different cooling rates and which have low volumetric shrinkage⁽⁴⁾. As amorphous resin materials have low volumetric shrinkage, the author sought to develop an amorphous ABS resin material offering low cost and broad applicability.

ABS resins have been widely used in interior and exterior automotive parts but normally possess insufficient chemical resistance, impact resistance and dimensional stability for use in body panels. The author therefore considered combining this resin with other resin materials to create improved material performance.

The candidates selected for combination with ABS were PC, PBT, and PA6; the properties of each are shown in Table 2⁽⁸⁾.

The chemical resistance of PC is difficult to improve because of the material's amorphous nature. PBT, a polyester resin with a high specific gravity, is unsuitable for use in

weight reduction applications. PA6 was therefore selected as the component to add to ABS because of its chemical resistance and low specific gravity.

Fig.5 is a transmission electron microscope (TEM) photograph of an ABS/PA6 mixture. The black portions are the ABS resin. As simple mixing results in phase separation (Fig.5-a), a compatibilizer that acts on ABS and PA6 was added to create a polymer alloy with uniform differential scattering (Fig.5-b), thus improving the properties required for use in body panels.

4.2. Effects on Sink Marks

The effects on sink mark formation of this ABS/PA6 polymer alloy were examined using a test die with a general thickness of 2.5 mm and a rib thickness of 1.25 mm. The resin materials examined were ABS/PA6 polymer alloys with an ABS-to-PA6 ratio of 70/30, 50/50, and 40/60. These were compared to polypropylene (PP), which is used in bumpers and instrument panel foundations, and PPE/PA66 polymer alloy, which is used in vehicle body panels. The width [H] and depth [V] of sink marks in samples molded with the test die were measured; the results are in Fig.6.

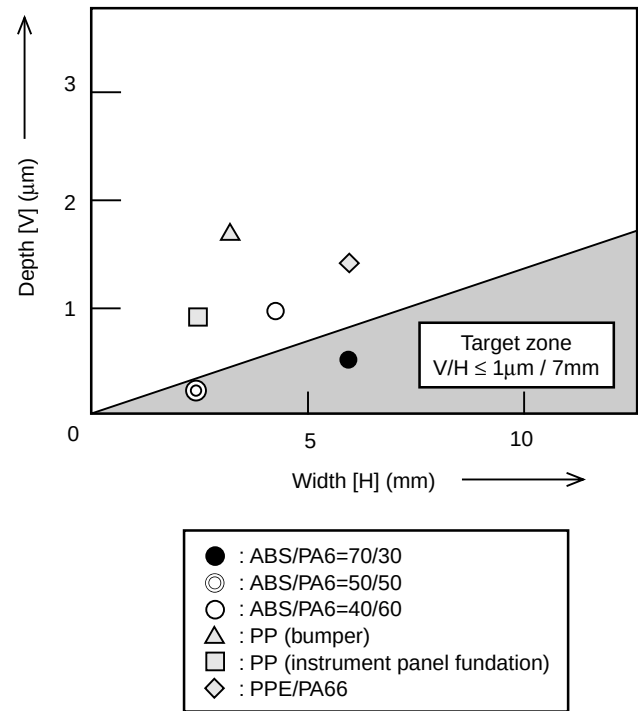


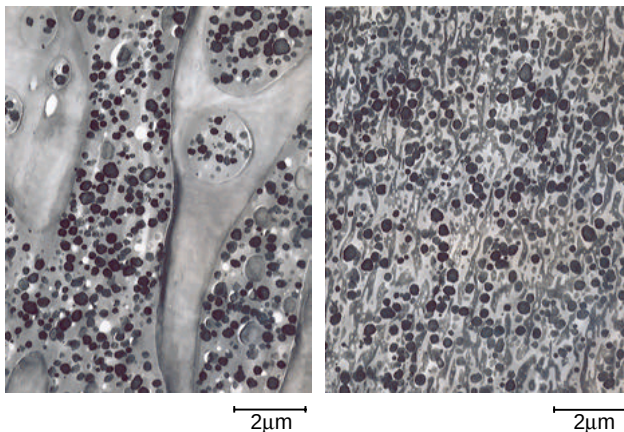
Fig.6 Test results of sink mark measurement

The ABS/PA6 polymer alloys with an ABS resin content of at least 50% were aesthetically effective with respect to sink mark prevention, satisfying the aforementioned sink mark standards. In contrast, PP and the PPE/PA66 polymer alloys fell far short of the standard for sink mark depth.

4.3. The Optimum Balance Between the Coefficient of Linear Thermal Expansion and Impact Resistance

Talc comprising an inorganic mineral filler was added to the ABS/PA6 polymer alloy to determine its effect on the

	PC	PBT	PA6
Chemical resistance	Poor	Good	Good
Specific gravity	1.2	1.4	1.1



(a) Without compatibilizer (b) With compatibilizer

Fig.5 TEM photographs of ABS + PA6 mixture

coefficient of linear thermal expansion. Three types of talc, each with a different particle diameter, were used: 2 mm, 4 mm, and 10 mm. Fig.7 gives the coefficient of linear thermal expansion for different combinations of talc particle diameter and talc quantity.

These results demonstrate how smaller talc particle diameter and greater talc quantity result in a lower coefficient of linear thermal expansion. Talc with a particle diameter of 2 mm was more effective than the other two types in lowering the coefficient of linear thermal expansion, and so was the subject of further investigation.

Fig.8 shows the effect on impact resistance and the coefficient of linear thermal expansion of different quantities of talc having a 2 mm particle diameter. Also shown in Fig.8 for comparison's sake are the properties of an ABS/PC polymer alloy used in actual mass production automobiles.

Impact resistance and the coefficient of linear thermal expansion exhibited a positive correlation. As an impact resistance of 50J or more with a coefficient of linear thermal expansion of $5.5 \times 10^{-5} \text{ cm/cm} \cdot ^\circ\text{C}$ or less was the target for

resin materials for body panels, the composition indicated by an arrow in Fig.8 was selected as the basis for an ABS/PA6 polymer alloy for body panels.

5. Properties of ABS/PA6 Polymer Alloys

Based on the above, the author proceeded with his investigation of ABS/PA6 polymer alloys by focusing on a composition having an ABS content of 50% or more and talc with a particle diameter of 2 mm. Moldability and chemical resistance were optimized by considering ABS and PA6 content and each component's molecular weight, etc., arriving at an ABS/PA6 polymer alloy with the target properties. Table 3 gives the basic properties.

Table 3 Basic properties of the ABS/PA6 polymer alloy

Properties	Typical value
Specific gravity	1.1
Tensile strength	40MPa
Flexural strength	60MPa
Flexural modulus	2200MPa
DuPont impact strength (at -30°C)	53J
Coefficient linear thermal expansion	$5 \times 10^{-5} \text{ cm/cm} \cdot ^\circ\text{C}$

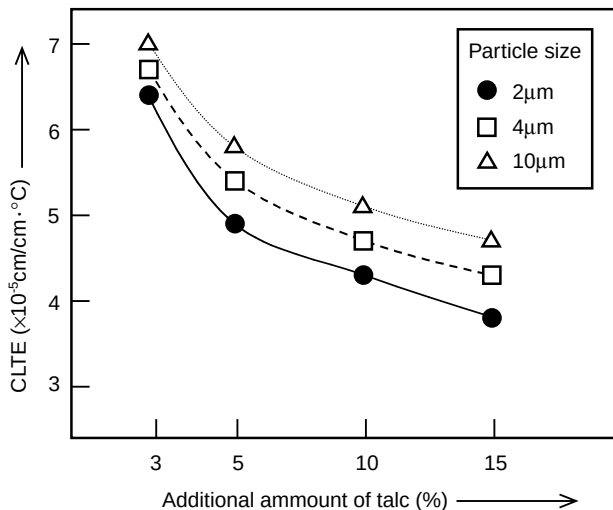


Fig.7 Effect of particle size of talc on CLTE

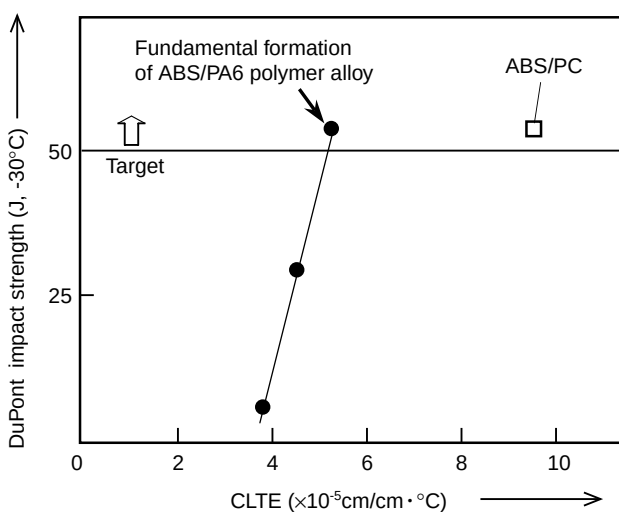


Fig.8 Relationship between DuPont impact strength and CLTE

6. Part Performance

This section discusses the author's investigation of part performance.

6.1. Effects of Rib Sink Marks on Molded Part Surface Appearance

The model in Fig.9-a was used to evaluate the appearance of sink marks caused by ribs in a large molded part (evaluation included deviation). The model used a 1,000 mm x 800 mm model of variable general thickness.

The ribs comprising the rib lattice on the backface of the model had a thickness of 0.8 mm, an intersection of 1.25 mm, and an interval of 100 mm. With general thicknesses of 3.0 mm, 2.5 mm, and 2.2 mm, sink marks were measured at the lattice intersections and in the straight portions between intersections.

Both the depth [V] and width [H] of the sink marks were measured using the method described earlier. The standards for allowable sink marks were as described above: a maximum depth [V] of 1 mm at a width [H] of 7 mm. The measurements for [V] were therefore converted to a 7 mm width equivalent; the mean values and maximum and minimum values of the measurements are shown in Fig.9-b.

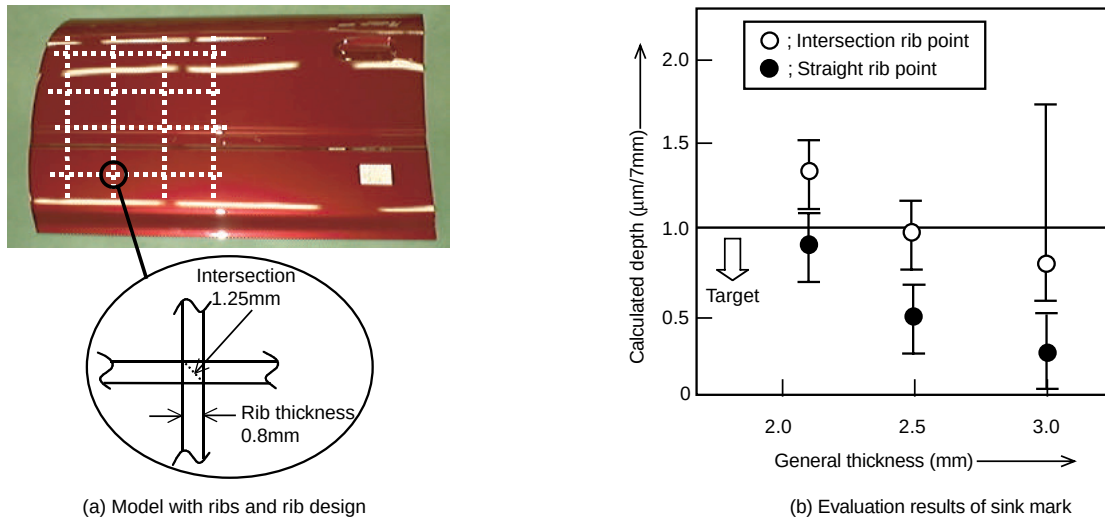


Fig.9 Sink mark appearance evaluation using model with ribs

As the graph in Fig.9-b shows, the aesthetically allowable rib (including variations in the molded parts) when general thickness is 2.5 mm was found to be a straight rib with a thickness of 0.8 mm and no intersections.

6.2. Effect of Ambient Temperature on Product Surface Appearance

A test was carried out to determine the effect of ambient temperature changes on part fit. The results for a front fender, a representative part, are given.

Dimensional change in the front fender was measured after at least four hours of exposure, to an infrared lamp in a 55°C atmosphere thereby raising the fender's surface temperature to 74°C ± 2°C, the high ambient temperature. To develop standards for dimensional changes, measurements were made at seven locations (Fig.10-a) using a hood. Next, dimensional change after at least 4 hours in a -30°C atmosphere, the low ambient temperature, was measured in the same manner as with the high temperature.

The results (Fig.10-b) show that front fender point "a" underwent a maximum dimensional change at the high temperature of 0.5 mm, which was within the allowable

elongation range of 2 mm (the allowable change for point "g" was 0.5 mm).

These results demonstrate the effectiveness of an ABS/PA6 polymer alloy having a low coefficient of linear thermal expansion.

6.3. Impact Resistance at Low Temperatures

The impact resistance of a resin material is normally lower at lower temperatures. Therefore, the author investigated impact resistance in a low-temperature atmosphere (-30°C) likely to be experienced in actual use.

Front fenders mounted to a vehicle were struck with a steel ball attached to a pendulum mechanism in order to evaluate deformation, cracking and other aspects of impact resistance. The impacted part is shown in Fig.11. The steel ball weighed 4 kg and was positioned at a height of 50 cm.

Fig.12 shows the results of this impact resistance test, in which two types of front fender were used: one made of ABS/PA6 polymer alloy, the other made from a steel panel. Test results were assessed on the basis of deformation area.

In contrast to the steel fender, whose extremely large deformation area would have necessitated replacement, the

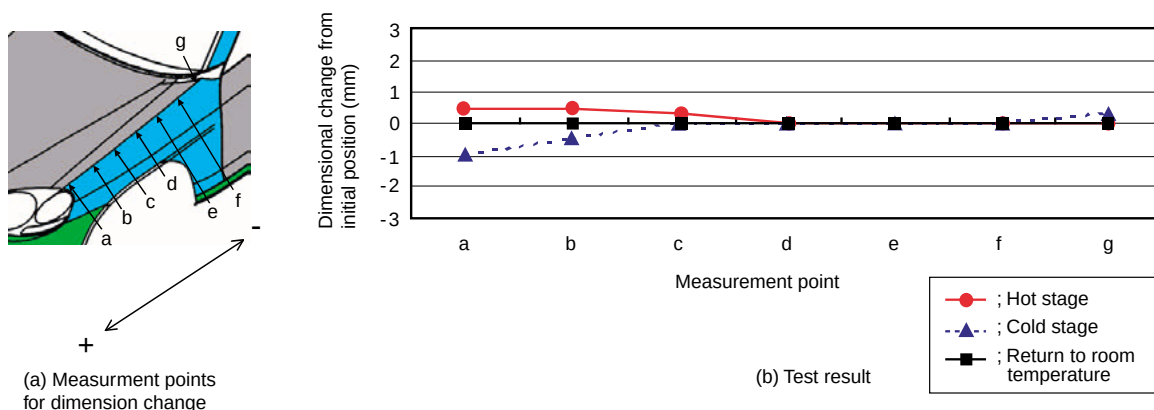


Fig.10 Hot-cold cycle test for front fender

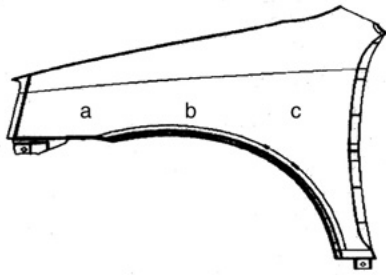


Fig.11 Impact test points on front fender

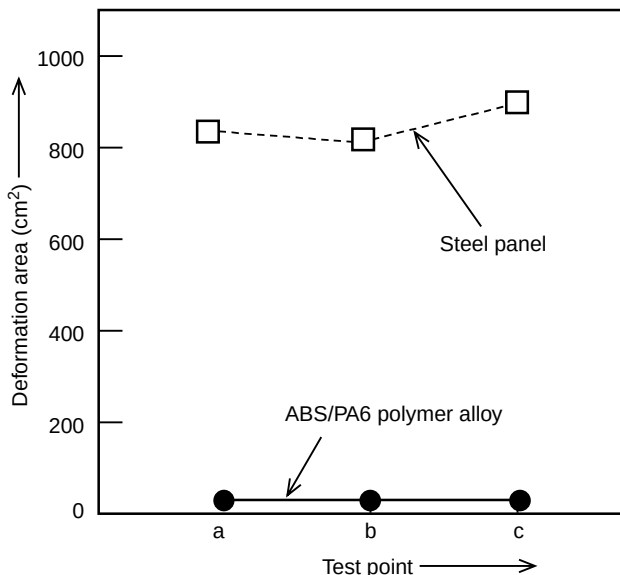


Fig.12 Comparison of deformation area between ABS/PA6 polymer alloy and a steel panel

ABS/PA6 polymer alloy fender showed no cracking and no large deformation, even in a -30°C atmosphere, thus demonstrating its superiority as a flexible resin material.

7. Conclusions

An ABS/PA6 polymer alloy for body panels was developed as a means to decrease automobile weight. The material's properties were enhanced through an investigation of sink marks on a model with ribs and dimensional changes resulting from changes in ambient temperature.

Sink marks were prevented by making use of the properties of amorphous ABS resin; dimensional stability was assured with a filler of optimum type and quantity, and optimum balance between the coefficient of linear thermal expansion and impact resistance.

The author's findings are as follows.

- (1) Maximum allowable sink marks were those with a depth [V] of 1 mm and a width of [H] 7 mm.
- (2) An ABS resin content of at least 50% was effective in preventing sink marks.
- (3) Talc with a particle diameter of 2 mm was effective in lowering the coefficient of linear thermal expansion while

also assuring proper impact resistance.

An ABS/PA6 polymer alloy front fender with a general thickness of 2.5 mm and a weight of 1.17 kg was 45% lighter than a similar steel front fender.

8. Acknowledgments

For their extensive assistance with resin material improvement, part testing and other aspects of this research, the author wishes to express his deepest gratitude to those at Japan A&L, Sumitomo Chemical, and Honda Engineering.

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■ author ■



Atsushi TAKEUCHI

Technical Papers Submitted to Out-of-Company (1999-June to -December)

(English Papers only)

Date	Conference	Title	Author	Organization
Jul.13-16, 1999	JSAE/The 15th Internal Combustion Engine Symposium International (Seoul, Korea)	The Optimization of Catalytic Converter by Using CFD Analysis	Toru Takabayashi Shinichi Tosa	Tochigi R&D Center Tochigi R&D Center
		Reducing Technologies for Exhaust Gas Emissions in ZLEV (Zero Level Engines Emission Vehicle)	Noriyuki Kishi Shinichi Kikuchi Norio Suzuki Tadayoshi Hayashi	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		CNG Operation of a Two-Stroke Two-Cylinder Marine S.I. Engine and Its Characteristics of Self-Ignition Combustion	Herchel T.C.Machacon Seiichi Shiga Takao Karasawa Kazuo Nakamura Takayuki Yamagata Hiroshi Sekita Kozo Uchida	Gunma Univ. Gunma Univ. Gunma Univ. Gunma Univ. Tochigi R&D Center Yamato Motor Co., Ltd. Yamato Motor Co., Ltd.
		A Method of Estimating Gasoline Engine Heat Loss	Takashi Suzuki Yasufumi Oguri Masatake Yoshida Ken Hanayama	Sophia Univ. Sophia Univ. Sophia Univ. Tochigi R&D Center
Aug.30-Sep.3 1999	IAVSD/16th Symposium (Pretoria, South Africa)	Improvement of Vehicle Handling Safety with Vehicle Side-slip Control by Direct Yaw Moment	Yoshimi Furukawa Masato Abe	Tochigi R&D Center Kanagawa Institute of Technology
		Steer-by-Wire-Oriented Steering System Design: Concept and Examination	Naohiro Yuhara Jun Tajima Shoichi Sano Shigenori Takimoto	Nippon Univ. Nippon Univ. Tochigi R&D Center Tochigi R&D Center
Sep.2-3, 1999	AVL LIST GMBH/AVL Congress (Graz, Austria)	Pursuit of the Ultimate Cleanness of Internal Combustion Engines	Tadayoshi Hayashi	Tochigi R&D Center
Sep.16-17 1999	International Congress on Continuously Variable Power Transmission (Eindhoven, Netherlands)	Characterization of a CVT Using a Metal V-Belt at Transitional States	Shigeru Kanehara Toru Fujii Osamu Fujimura	Tochigi R&D Center Doshisha Univ. Doshisha Univ.
Sep.21, 1999	VDA/VDA Technical Congress (Frankfurt, Germany)	A Powertrain Concept for ZLEV Exhaust Emission Legislation	Tadayoshi Hayashi Shinichi Kikuchi Noriyuki Kishi	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
Sep.28-30 1999	International Body Engineering Conference '99 (Detroit, USA)	Application of the Statistical Design Support System Toward Optimization of Vehicle Safety Equipment	Hideoki Yajima Qiang Yu Masaki Shiratori Yasuhiro Dokko Shigeo Ito Keiichi Motoyama	Wako Research Center Yokohama National Univ. Yokohama National Univ. Tochigi R&D Center Tochigi R&D Center Mechanical Dynamics Japan K.K.
Sep.28-30 1999	99 Small Engine Technology Conference (Madison, USA)	Development of a Hybrid System for Single-Seater Commuter	Osamu Suzuki Toru Hayashi	Asaka R&D Center Asaka R&D Center

Date	Conference	Title	Author	Organization
Sep.28-30 1999	99 Small Engine Technology Conference (Madison, USA)	Development of Die Cast Parts for Welded Structure Manufactured by the Transition Flow Filling	Masahiko Nakagawa Eitaro Koya Sadafumi Yoshino Gumpachi Fujimoto Mitsunori Murakami Atsushi Kawauchi	Asaka R&D Center Asaka R&D Center Wako R&D Center Kyushu Yanagawa Seiki Co., Ltd. Kyushu Yanagawa Seiki Co., Ltd. Kyushu Yanagawa Seiki Co., Ltd.
		Development of Directly Actuated Variable Valve Control System	Takaaki Tsukui Takashi Tsutsumizaki Mutsuo Nakajima	Asaka R&D Center Asaka R&D Center Asaka R&D Center
		Development of a CFD System Using PCC Method and Its Application to Exhaust Muffler Design for Motorcycles	Yasushi Takahashi Mutsuo Nakajima	Asaka R&D Center Asaka R&D Center
		Detonation Counter for Production Racing Motorcycle	Hajime Soda Tomonori Sato	Honda Racing Corp. Honda Racing Corp.
	(Continued)			
Oct.1, 1999	JSAE/JSAE Review Vol. 20, No. 4	Method for Measuring Force Transmitted from Road Surface to Tires and Its Applications	Masaaki Minakawa Jiro Ninomiya Yukihiro Orimoto Jun Nakahara	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Improvement of Vehicle Handling Nonlinear Integrated Control of Four Wheel Steering and Four Wheel Torque	Shinichiro Horiuchi Kazuyuki Okada Shinya Notomi	Nippon Univ. Nippon Univ. Tochigi R&D Center
		Two-Degree-of- Freedom Hydraulic Pressure Controller Design for Direct Yaw Moment Control System	Takatoshi Nishimaki Naohiro Yuhara Yasuji Shibahata Nobuharu Kuriki	Nippon Univ. Nippon Univ. Tochigi R&D Center Tochigi R&D Center
Oct.5-7, 1999	1999 GLOBAL POWERTRAIN CONGRESS (Stuttgart, Germany)	Application of High Cell Density and Thin Wall Substrate for High Performance Catalyst	Shinichi Kikuchi Tatsuya Okayama Shinichi Tosa	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Research of Ultimate Low Emission Technologies for Gasoline Engines	Noriyuki Kishi Shinichi Kikuchi Norio Suzuki Tadayoshi Hayashi	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Estimation of Ozone Reduction Catalyst Effect at On-road Condition	Tatsuya Okayama Seiji Hatcho Yoshihiro Yamamoto Noriyuki Kishi Tadayoshi Hayashi	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center

Date	Conference	Title	Author	Organization
Oct.5-7, 1999	1999 GLOBAL POWERTRAIN CONGRESS (Stuttgart, Germany) (Continued)	Development of A Rocker Arm for Variable Valve Timing Mechanism by Applying Metal Injection Molding (MIM) Technology	Toshihiro Kamiyama Toshiyuki Sato Shunsuke Takeguchi	Tochigi R&D Center Tochigi R&D Center Nippon Piston Ring Co., Ltd.
		Application of Thixotropic Technology for Automobile Parts	Kazuya Sakamoto Hisao Hirono Takashi Idegomori Eiji Masuda Nobuhiro Saito	Honda Engineering Co., Ltd. Honda Engineering Co., Ltd. Honda Engineering Co., Ltd. Honda Engineering Co., Ltd. Tochigi R&D Center
Oct.5-8, 1999	ITSC IEEEJ JSAI/1999 IEEE/IEEJ/JSAI International Conference on Intelligent Transportation Systems (Tokyo, Japan)	A Study of System Architecture for the Advanced Cruise-Assist Highway System (AHS)	Kouji Kubota Kazuhiko Yamaguchi Hiroshi Mashimo	Nippon Telegraph and Telephone Corporation DENSO Co., Ltd. Tochigi R&D Center
Oct.7-8, 1999	ESI/EURO —PAM '99 (Darmstadt, Germany)	Reducing a Process to Generate a Model with CATIA Interface for PAM-FLOW	Akio Takamura	Tochigi R&D Center
Oct.12-16 1999	16th International Electric Vehicle Symposium (Beijing, China)	Development of High Performance and Compact Ni-MH Batteries for HONDA City Pal	Takashi Ishikura Kazuhiko Yagi Noboru Sato	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Development of the 2-Seater Small Electric Vehicle HONDA City Pal	Masami Ogura Yoshinori Mita Tatsuo Takahashi	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Electrically Powered Small Commuter Vehicles	Chiaki Kumagai Toru Hayashi Kaoru Hatanaka	Asaka R&D Center Asaka R&D Center Asaka R&D Center
		Development of VRLA Batteries for HONDA ICVS	Masaaki Hosokawa Sinji Yamada Keiichi Hasegawa Yasuhide Nakayama Masahide Yokoo Kyoichi Ariga Toru Takeda	YUASA Corporation YUASA Corporation YUASA Corporation YUASA Corporation Asaka R&D Center Asaka R&D Center Asaka R&D Center
Oct.23-28 1999	Society for Neuroscience/29th Annual Meeting (Miami, USA)	On the Perception of Series of Vowels Based on the Dynamic Information	Masashi Ito	Wako Research Center
		Intuitive Global Decision Can Be Made by a Simple Parallel Processing	Hiroshi Tsujino	Wako Research Center

Date	Conference	Title	Author	Organization
Oct.25-27 1999	ECM/TPOs IN AUTOMOTIVE '99 (Novi, USA)	High Performance TPO for Bumper Facia: 1999 Honda Odyssey	Motoki Yamada Yoshihisa Saeki Hiroyuki Yamamoto	Idemitsu Petrochemical Co.,Ltd. Tochigi R&D Center Honda Engineering Co., Ltd.
Oct.25-28 1999	SAE International Fall Fuels and Lubricants Meeting and Exposition (Tronto, Canada)	The Effect of 0W-20 Low Viscosity Engine Oil on Fuel Economy	Hideki Tanaka Takashi Nagashima Takayuki Sato Shoji Kawauchi	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
Oct.29, 1999	SAMPLE/6th Japan International SAMPLE Symposium & Exhibition	Application of Metal Matrix Composites to Automotive Manufacturing	Seiichi Koike	Tochigi R&D Center
Nov.1, 1999	“Neural Networks” Journal, Pergamon Press/Nov.1999	A Model of Computation in Neocortical Architecture	Edgar Koerner	Honda R&D Europe
Nov.8-12, 1999	6th World Congress on Intelligent Transport Systems (Tronto, Canada)	Advanced Safety Vehicle (Panel Session)	Yukinobu Nakamura	Tochigi R&D Center
		An Adaptive Automatic Steering Control System Using Generalized Predictive Control Theory	Shinya Notomi Shinichiro Horiuchi Toshiyuki Tamatsukuri	Tochigi R&D Center Nippon Univ. Nippon Univ.
		Evaluation of Driver Assistance System	Satoshi Kondo Nobuyoshi Asanuma Shinnosuke Ishida Manabu Ikegaya Jun Tanaka	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Autonomous Vehicle Control System Equipped with a Navigation System	Kazuya Tamura Akihiko Takei Tooru Saitoh	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Cognitive Performance of Visual Messages on In-Vehicle Display and Driving Behavior	Tsutomu Asoh Hiroyuki Kamiya Hajime Ito	Japan Automobile Research Institute Tochigi R&D Center YAZAKI Corp.
Nov.14-19 1999	ASME/1999 International Mechanical Engineering Congress & Exposition (Nashville, USA)	How to Measure the Internal Forces of a Combustion Engines	Yukitaka Takahashi Toshibumi Suzuki Naoki Kamimura	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
Dec.1, 1999	Engine Technology International/Electric & Hybrid Vehicle Technology '99	Decisions, decisions	Noboru Sato	Tochigi R&D Center

Editor's Notes

Vol.12 No.1

The memorable celebration of Honda's 50th anniversary just one year ago has made us all appreciate the immensity of Honda's cumulative contributions to technology. The Honda R&D Technical Review has just celebrated its 10th anniversary and is now being published twice a year. The last issue, Vol. 12, No. 1 was published in the second year of the new semiannual system and the editorial board members continues to become more adept at its editorial tasks. The understanding and cooperation we have received from the employees in regard to our technical paper related activities have been tremendous and are greatly appreciated.

The key words "environment" and "safety" characterize many of the papers we have published in recent years. The papers included in this edition have such themes as "reduced resource consumption," "reduced energy consumption," "weight reduction," "downsizing," "recycling," "global warming," "low fuel consumption," "low exhaust emissions," "low noise," "secondary cells and electric energy," "electric propulsion," and "crash safety." Each of these papers came into being as a result of applying remarkable advances in electronics and computer technology as well as the efforts required to "create products appropriate for the times."

This year represents the start of a new millennium. To each of our readers in his or her place of research, it is our wish that the *Honda R&D Technical Review* be used as a tool that generates synergism between the motorcycle, automobile, power product, and fundamental research divisions, leading to the birth of many new areas of technology. And we, the members of the Committee for Technical Papers, look forward to the day when so many papers are submitted for publication that we find it difficult to select those that will be published in future Technical Reviews.

In closing, we ask for your continued support of and cooperation with the *Honda R&D Technical Review.
(Shinichi Koizumi: Asaka R&D Center)

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Editor's Notes for English Version

The 1st edition of the English version of the Honda R&D Technical Review has been completed.

We had considered issuing an English version of the journal over the years, and now it has finally come about.

In preparation for the English version, we set English quality as a high priority. In order to achieve the level we were aiming for, we requested that various North American associates review these technical papers. I'd like to express my thanks to all the people who helped us with their cooperation.

The following North American associates provided us with their cooperation and expertise:

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