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Technical Review

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Introduction of HONDA ASV-2 (Passenger cars)

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ABSTRACT

In response to the Advanced Safety Vehicle Project Phase 2 (ASV-2) proposed by the Ministry of Transport, we have addressed the two topics below with reference to the results from analysis of actual traffic accident conditions. After selecting effective technologies, we fabricated test vehicles for each of the objectives and moved forward with the research.

- Safety-enhancement through the reduction of both driver operating burden and rear-end collision damage through driving and operational support using CCD camera, radars, and related technology (Honda ASV-2, Vehicle 1).
- Enhancement of visual recognition to thereby provide information on the presence of motorcycles and pedestrians through information and visual support using communications and control devices, infrared cameras, and other technology (Honda ASV-2, Vehicle 2).

Research is proceeding on installation and implementation of each system, taking the basic approach of developing systems that will not interfere with driving operations. The test results on these systems have shown that they can be expected to produce accident-reducing effects. Minister of Transport approval was granted and road tests are being conducted on some of these technologies.

1. Introduction

The Advanced Safety Vehicle Phase 2 (ASV-2) Development Promotion Project proposed by the Road Transport Bureau of the Ministry of Transport has been implemented as a 5-Year Project starting from fiscal year 1996 with the participation of 13 domestic manufacturers. The ASV project aims to increase the intelligence of vehicles by utilizing electronics and other new technologies that are advancing rapidly, and to research and develop vehicles to enhance safety. Technical items have been set for a total of 32 systems in the following six fields: (1) Prediction and collision avoidance safety technology, (2) Accident evasion technology, (3) Fully automatic driving technology, (4) Collision avoidance technology, (5) Reduction of secondary effects from collisions, and (6) Automobile infrastructure technology. Each participating company was left to its own devices in dealing with each of the items. In Phase 2 the project was expanded to include trucks, buses and motorcycles in addition to the automobile research initiated in Phase 1 (implemented from 1991 to 1995).⁽¹⁾ In addition, joint proving tests (Smart Cruise 21) with the Advanced Cruise-Assist Highway Systems (AHS) promoted by the Ministry of Construction are scheduled for the end of this year.

2. Background

The types of accidents that would be most effective to research and the technologies needed to avoid accidents served as the starting points for safety research for the Honda ASV. Fig. 1 shows the transition in the number of fatalities and casualties over the last ten years.⁽²⁾ This graph shows that the casualties are still increasing, and it is thought that research and development on collision avoidance technology will become even more important for reducing traffic accidents in the future.

Fig. 2 shows the percentage of traffic accidents according to the parties involved. These graphs show that collisions between automobiles account for 50% of accidents involving casualties. Likewise, 50% of fatal accidents involve vulnerable parties such as pedestrians, motorcycles and bicycles.

Analyzing this data in greater detail shows that the number one cause of accidents between vehicles is rear-end collisions, accounting for 30% of the total, while most single vehicle accidents involve colliding with objects or the vehicle leaving the designated lane due to driver recognition or judgment errors. On the other hand, 70% of accidents resulting in the death of vulnerable parties occurred at night, with most occurring while pedestrians were crossing streets.

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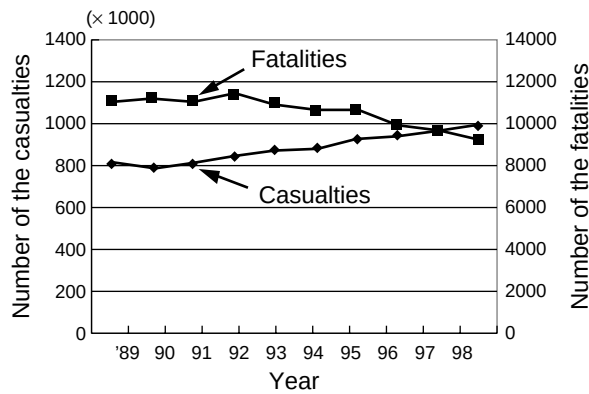


Fig.1 History of fatalities and casualties in traffic accidents

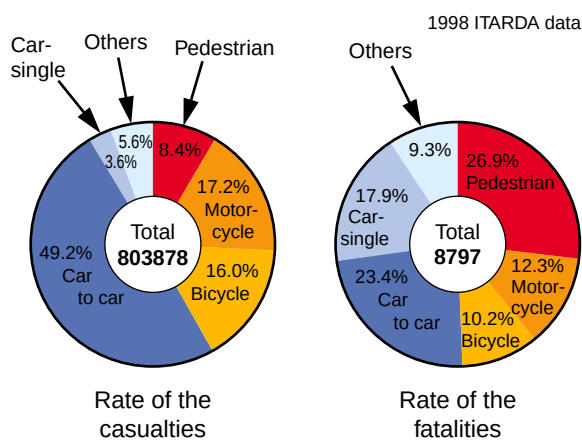


Fig.2 Percentage of the traffic accident details

In addition, 70% of motorcycle accidents are accounted for by intersection collisions and collisions involving vehicles making a cross-traffic turn (right-turn in Japan, left-turn in America) in front of an oncoming motorcycle.

Based on the above, we selected the following three topics:

- (1) The tracking of vehicles in front, as a way of reducing rear-end accidents between vehicles.
- (2) The avoidance of departure from a lane, as a way of reducing single vehicle accidents.
- (3) The detection of pedestrians at night and motorcycles at intersections, etc., as ways of reducing traffic accidents involving vulnerable parties.

Experimental vehicles were divided by topic as follows to allow research to proceed in an efficient manner.

- Honda ASV-2 experimental vehicle 1: Research on driving support
- Honda ASV-2 experimental vehicle 2: Research on improving recognition of vulnerable parties

Fig. 3 shows the experimental vehicles that were fabricated for the purpose of the research.

Research proceeded while bearing in mind the fact that "increasing human comfort reduces driver load," and the following themes were set as goals for the realization of our objectives:

- (1) Establishing guidelines for the development of new safety technologies
- (2) Real world tests
- (3) Verifying user compatibility
- (4) Improving system reliability



Fig.3 HONDA ASV-2

3. Technologies Applied to the Honda ASV-2 Experimental Vehicle 1

3.1. Outline

An analysis of traffic accident data revealed that the major factors involved in traffic accidents were human factors such as driver carelessness and/or errors in judgment. Therefore, the following three functions were incorporated into experimental vehicle 1:

- (1) Honda Lane Keep Assistance System (HLKAS), a system to help drivers keep their lane position and help avoid departure from a lane.

This system uses a CCD camera to recognize the lane markers and makes lane keep assistance.

- (2) Adaptive Cruise Control (ACC) with Stop & Go System, an ACC system with tracking control for congested traffic situations.

This system uses a millimeter-wave radar sensor to track the vehicle in front and keep the appropriate distance between the two cars during operation from start to stop.

- (3) Honda Collision Velocity Reduction System (HCVRs), a system for reducing rear-end collision velocity.

This system applies the brakes sharply to reduce the vehicle's collision velocity when the system has judged that a collision with the vehicle in front is likely.

Note that the system combining (1) HLKAS and (2) ACC⁽³⁾ is named the Honda Intelligent Driver Support System (HIDS)⁽⁴⁾. This is the first such domestic system to receive the approval of the Minister of Transport. Expressway running tests are currently underway with the aim of achieving actual application and use.

3.2. System Configuration

Fig. 4 shows the installation location of components used to achieve the above three functions in experimental vehicle 1, and Fig. 5 is a block diagram of the system's control. The CCD camera is positioned at the top of the windshield and the millimeter-wave radar is placed inside the front bumper

to detect the lane markers and the vehicle in front. The electric power steering (EPS) system, throttle actuator, electronically controlled master power (E-M/P) cylinder system, etc., are activated to control lane position and distance between vehicles.

3.3. Honda Lane Keep Assistance System (HLKAS)

This system provides the following lane keeping assistance: it recognizes the lane markers on the roadway by processing the images obtained from the CCD camera⁽⁵⁾, establishes the target point that the vehicle should reach along the target route to keep itself in the center of its own lane, and calculates the angular difference between the target point and the vehicle's predicted position. The system then calculates the optimum steering assist torque to steer the vehicle toward the target point, and controls the current supplied to the vehicle's EPS system to steer the vehicle

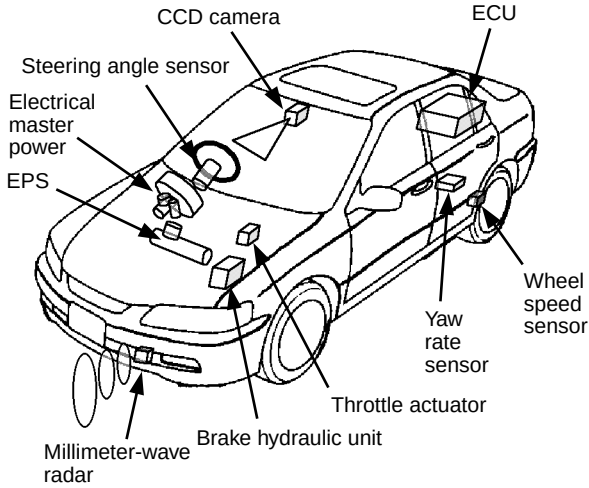


Fig.4 Configuration of ASV-2 experimental vehicle 1

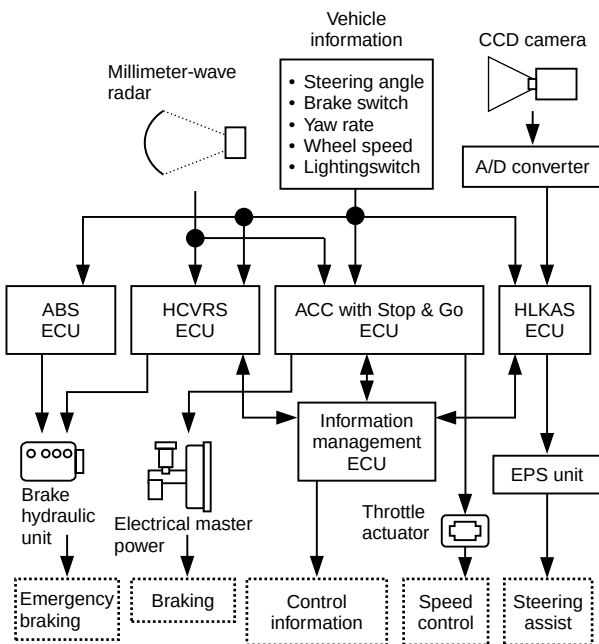


Fig.5 Block diagram of ASV-2 experimental vehicle 1

and keep it in its own lane. The system also provides a warning when the vehicle departs from the lane recognized by the system.

3.3.1. Lane Keeping Control Algorithm

As indicated by the geometric model shown in Fig. 6, the basic approach to controlling vehicle lane keeping consists of two functions.

The first function, feed-forward torque T_f , has the objective of stabilizing the vehicle for steady-state cornering. Steady-state cornering is made possible by calculating, the steering torque which will generate cornering force C_f that is optimal for centrifugal force $m \cdot V^2/R$ based on the road radius R , which is determined by image processing, and by steering the vehicle at this steering angle.

The second function, feedback torque T_b , has the objective of allowing stable cruising in the center of the lane. Cruising in the center of the lane is made possible by reducing the vehicle deviations to zero (in lateral position Y_d and vehicle heading angle θ_h) from the target route.

By generating a steering torque that optimally corrects these deviations, it is possible for the vehicle to follow the target route. Simultaneously, the appropriate steering torque is also generated to stabilize the vehicle-heading angle whenever this angle is suddenly altered by external disturbances such as road unevenness or crosswinds.

The combined torque of these two elements is calculated as lane keep assistance torque T_a , and lane keep control is performed for the vehicle.⁽⁶⁾

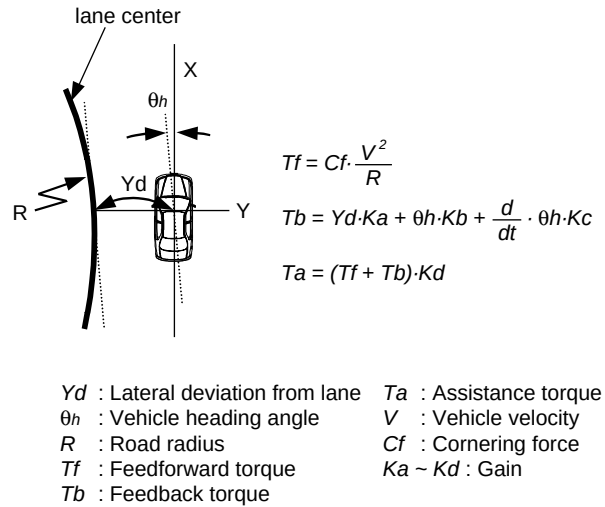


Fig.6 Geometric model

3.3.2. Driver Torque Interface Structure⁽⁷⁾

Within the EPS system, changeover between the driver's torque and the HLKAS torque is done as shown in Fig. 7-a to prevent steering wheel torque interaction with the driver. This is done by lowering the HLKAS torque in response to the amount of the torque input when the driver intentionally provides steering input during HLKAS control as shown in MAP1 in Fig. 7-b. At the same time, steering with little torque fluctuations is realized by restoring the assist characteristics of conventional power steering where there is no control system as shown in MAP2 in Fig. 7-b.

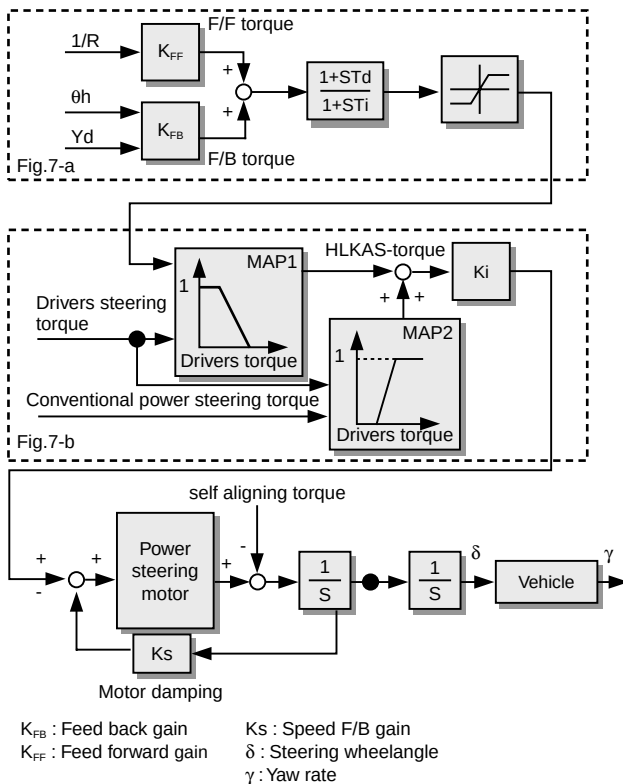


Fig. 7 Block diagram for HLKAS control

3.3.3. Steering Torque and Steering Angle Frequency Test Results

We compared the steering characteristics for a vehicle in which the system was installed with a conventional vehicle with no control system (using the travel data obtained from the lanes on the Tohoku Expressway, with a cruising speed of 100 km/h and a travel distance of approximately 50 km). Fig. 8 shows a comparison of the steering torque distribution. These results revealed that the high steering torque change frequency was reduced by HLKAS compared with a conventional vehicle without the control system. In addition, Fig. 9 shows a comparison of the steering angle distribution. This figure shows that HLKAS has a higher frequency near the steering angle center and smaller fine adjustment steering angles compared with a conventional vehicle without the system, thus confirming that HLKAS reduces the driver load.⁽⁸⁾

3.4. ACC with Stop & Go System

This is a driver load reduction system that assists drivers in keeping a set distance from the vehicle in front. In congested traffic, for example, the system decelerates the vehicle and, if necessary, stops the vehicle automatically. In this way, it keeps track of the vehicle ahead under a wide range of driving speeds, thereby reducing the driver load. Using a millimeter-wave radar sensor, the system detects the driving conditions of the vehicle traveling ahead, it senses the condition of the vehicle in which it is installed using the vehicle speed sensor and yaw rate sensor, and in this way focuses on its relative position to the vehicle ahead in the same lane. If the distance between the two vehicles becomes too short, the system reduces the vehicle's speed primarily

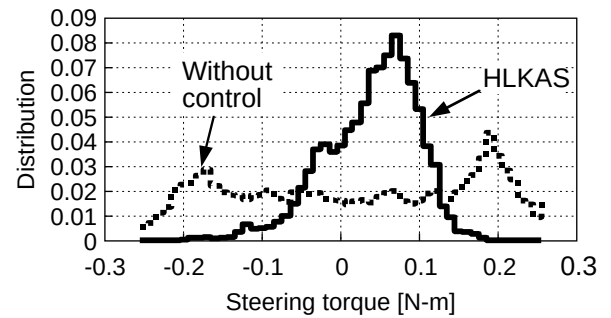


Fig. 8 Normalized steering torque distribution

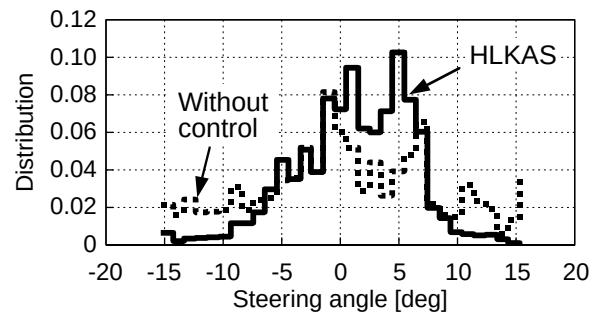


Fig. 9 Normalized steering angle distribution

by releasing the throttle. If this speed reduction is not sufficient, it further reduces the speed using the main brakes. In this way, the appropriate distance between the vehicles is maintained. At the same time, warning devices are used to alert the driver. Conversely, when the vehicle in front is traveling faster and the distance between the vehicles is becoming progressively greater, the system functions as a regular cruise control.

3.5 Honda Collision Velocity Reduction System (HCVR)

When the distance from the vehicle in front is progressively decreasing, this system urges the driver to exercise caution by way of a warning sound. If the driver fails to respond and proceeds to close in on the vehicle in front to the point where the distance between the vehicles is dangerously short, the system sounds a warning and begins braking. This encourages the driver to avoid the vehicle ahead by steering and applying the brakes. Furthermore, if the system judges that a collision is unavoidable, it reduces the collision velocity using the emergency brake which assists the driver's braking operation in order to reduce the severity of impact damage.

4. Technologies Applied to the Honda ASV-2 Experimental Vehicle 2

4.1. Outline

An analysis of traffic accident data revealed that poor visual identification of the vulnerable parties in traffic accidents is a significant factor in pedestrian accidents occurring at night and collisions with motorcycles at intersections.

In view of the above, the following three functions were incorporated into experimental vehicle 2:

- (1) Honda Night Vision System, a system which uses infrared cameras to detect pedestrians walking at night.
- (2) Active Headlight System, a system which synchronizes with steering angle, etc. to illuminate the path of the vehicle and thereby secure the driver's field of view during the night.
- (3) Inter-Vehicle Motorcycle-Automobile Communication System, a system to help enhance driver awareness through communication via systems installed in both motorcycles and automobiles.

4.2. System Configuration

Fig. 10 shows the installation location of the components of the system that performs the above three functions in experimental vehicle 2, and Fig. 11 is a block diagram of the system's control. The two infrared cameras installed inside the front grille detect pedestrians while the

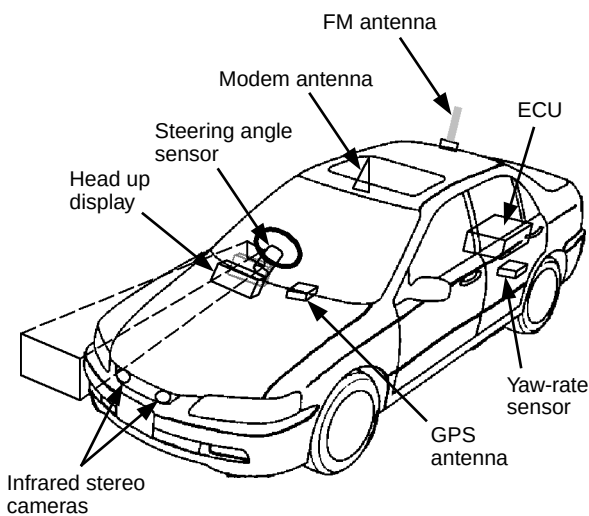


Fig.10 Configuration of ASV-2 experimental vehicle 2

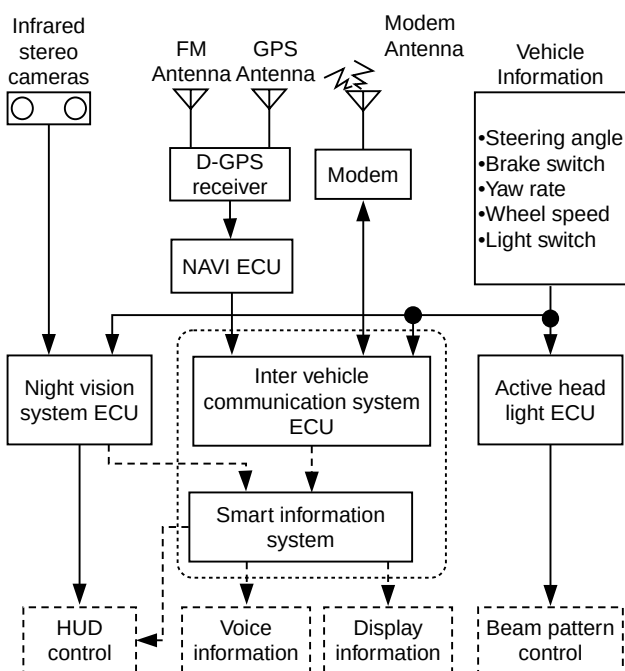


Fig.11 Block diagram of ASV-2 experimental vehicle 2

communication modem antenna on the roof enables communication with motorcycles in the vicinity of the vehicle. Depending on the conditions, warning sounds are emitted and images are shown on the head-up display (HUD) and navigation screen. The HUD has color image functions and is capable of displaying images during the day or night. The light distribution control of the Active Headlight System is coupled with the operation of the steering wheel.

4.3. Honda Night Vision System

This is a visual identification assistance system with the main objective of reducing accidents involving pedestrians walking at night. It helps the driver be aware of the presence of pedestrians using voice guidance and highlighted infrared images of pedestrians on the HUD. Fig. 12 shows the system's operating condition when the system supplies such information.

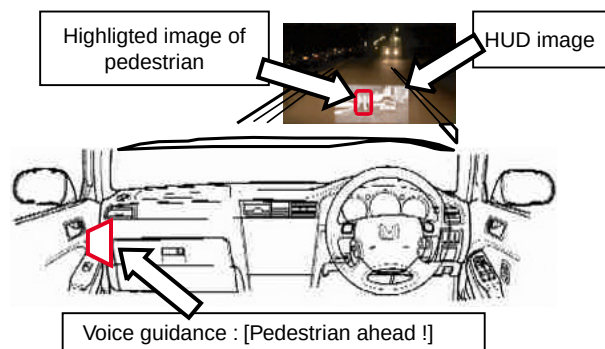


Fig.12 Example of HUD image and voice guidance

4.3.1. Pedestrians about which Information Should be Provided

As shown in Fig. 13, this system aims to inform drivers of pedestrians walking on roadways or attempting to cross streets at night at a distance whereby the vehicle will be able to stop in time.

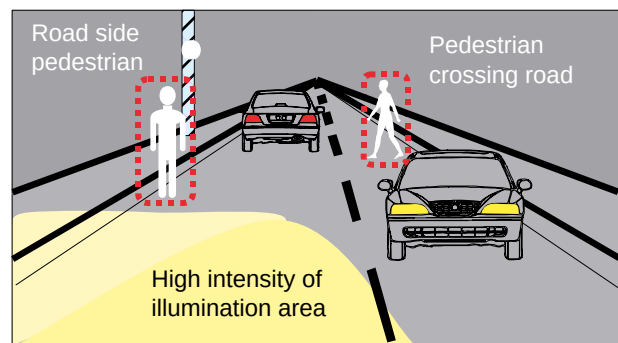


Fig.13 Example of detected pedestrians

4.3.2. Pedestrian Detection Method⁽⁹⁾

Pedestrians and other high-temperature objects are highlighted in white on the infrared camera image. However, shape differentiation with other high-temperature objects is difficult. Therefore, this system uses stereoscopic images and processing to calculate the distance to the objects. Then it identifies the objects and judges the possibility of collision

based on the speed by obtaining the relative moving vectors from the changes occurring over time. Fig. 14 shows the processing flow.

4.3.3. Pedestrian Presence Information Range

The area for which information is provided was set broader than the high intensity illumination area of the headlights as shown in Fig. 15, and the system operates to provide information at a vehicle speed of 40 km/h or higher. In addition, the lateral range was set at 12 degrees or less for entering objects, and the vehicle lane width for approaching objects. Information is provided 3.5 seconds before the estimated collision time as calculated from the steering evasion success rate and stopping distance with respect to the advance information provision time.⁽¹⁰⁾

4.3.4. Verification of Effects

This system incorporates voice guidance and highlighted frame displays of pedestrian areas as a means of compensating for poor visual identification of pedestrians.

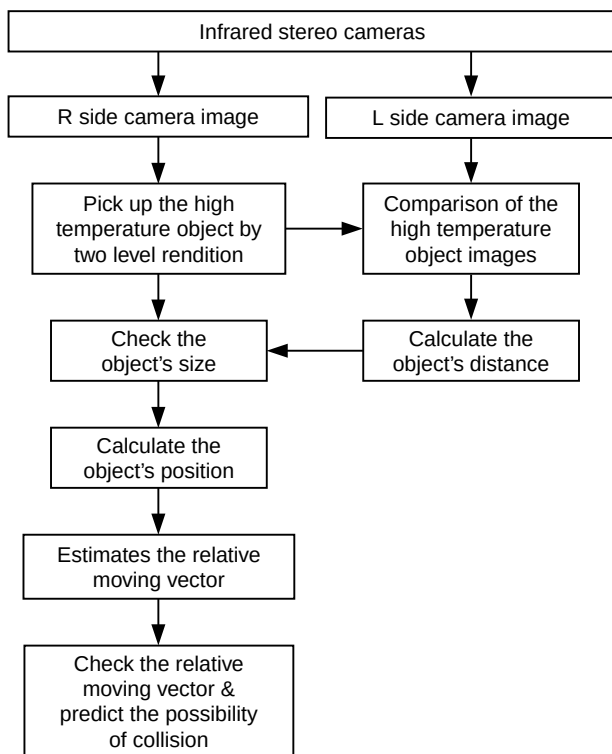


Fig.14 Pedestrian detection processing flow

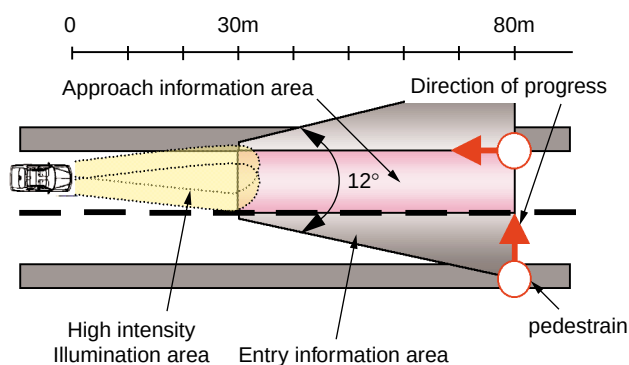


Fig.15 Pedestrian detection area

We compared the effectiveness of these functions with constant displays of images for verification purposes.

(1) Test method

As shown in Fig. 16, a number from 0 to 9 that changed at 1.5-second intervals was displayed on the screen in front. By having the test drivers read off these numbers, their ability to keep their eyes "on the road" was restricted to reproduce a state of visual attention resembling actual driving conditions. Images simulating driving were then shown on the HUD. When the test drivers identified a pedestrian crossing the road in the image, they were instructed to press a switch either on the left or right of the steering wheel, depending on the direction in which the pedestrian appeared, and their response time was measured. A total of 10 test drivers consisting of 8 males and 2 females ranging in age from their twenties to fifties was selected for the test.

The test patterns compared this system with a system which only displayed a constant image as shown in Table 1. Pedestrians crossed the road 10 times in the travel images shown over a 5-minute period.

(2) Test results

Fig. 17 shows the results obtained from measuring the average response time of the test drivers. When test drivers were placed in the near-driving state of visual attention in the forward direction, the system yielded a response time approximately 50% or less compared with the image displays alone. This proved that the system's voice guidance and highlighted displays are effective in improving the visual identification of pedestrians.



Fig.16 Measuring conditions

Table 1 Test pattern

	Image only	HONDA night vision
Voice guidance	×	○
Highlighted image of pedestrian	×	○
HUD Image	A	P

○ : Available A : All the time
 × : Not Available P : Pop-Up

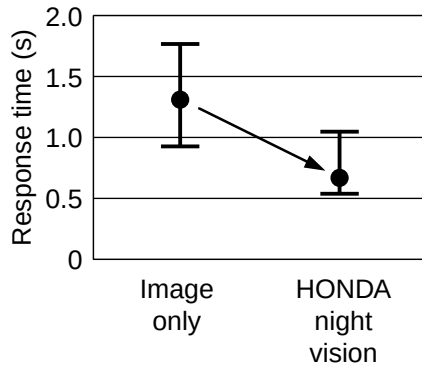


Fig.17 Comparison of response time

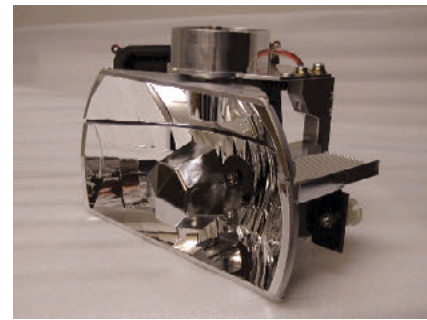
4.4. Active Headlight System

At intersections or curves, this system moves the upper reflectors of the headlights according to the steering angle and vehicle speed, and it produces light distribution appropriate to the roadway conditions. Fig. 18 shows a comparison of the light distribution with normal headlights. By enabling the driver to gain an accurate grasp of the roadway conditions and by having the effect of drawing the driver's visual attention into the curves, the system helps drivers to drive more safely and it enables pedestrians to be identified sooner.

Fig. 19 shows the internal configuration of the active headlights and their operating modes. The upper reflectors are rotated up to 40 degrees outward and up to 20 degrees inward by a stepping motor which is mounted at the rear. The ratio of the light distribution between the upper moving reflectors and lower fixed reflectors is approximately 4:6.

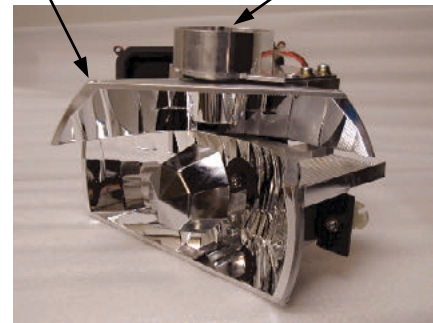
4.5. Inter-Vehicle Motorcycle-Automobile Communication System

This communication system, which is installed on both motorcycles and automobiles, exchanges information on, for example, the types, positions, speeds and directions of the respective vehicles, thus assisting the awareness of each of the drivers. In this way, it helps to reduce failures to spot the other vehicle and it decreases the number of errors in judging approach speed and relative distance. As shown in Fig. 20,



Straight driving condition

Upper reflector Stepping motor



Turning driving condition

Fig.19 Active head-light mechanism

it prevents intersection collisions and cross-traffic turn (right-turn in Japan and a left-turn in America) collisions which are the most typical accidents involving motorcycles and automobiles.

Fig. 21 shows an outline of the system. The vehicle's own position is determined by a navigation system using a high-accuracy GPS, and information such as the vehicle speed and turn signal operation is obtained from the vehicle. The modems installed on both vehicles then perform two-way communication based on this information. When the system judges that a collision is likely at an intersection, it displays the symbol, etc., of the other vehicle on the navigation or head-up display and alerts the driver by voice announcements and warning sounds. The details of this system are presented separately.⁽¹¹⁾

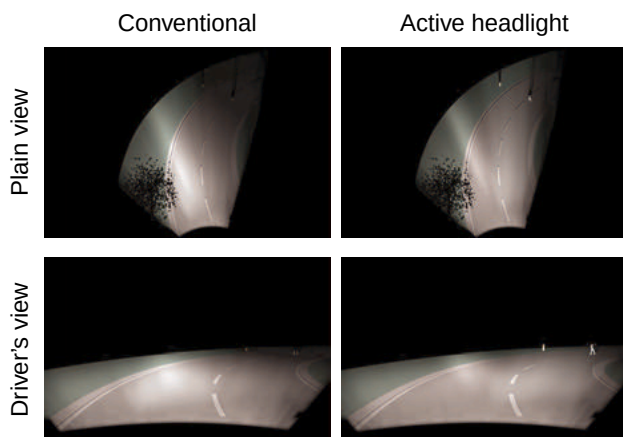


Fig.18 Comparison of vehicles with and without control

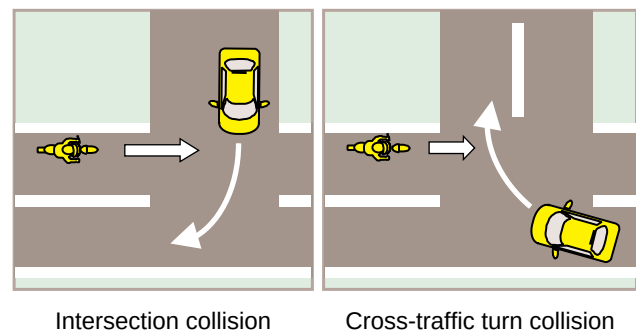


Fig.20 Typical traffic accidents between automobile and motorcycle

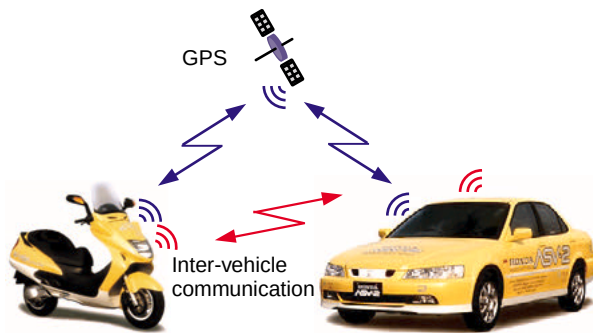


Fig.21 Inter-vehicle communication system

5. Conclusion

In response to the ASV-2 project, we have conducted research on advancing driving operation assistance and enhancing the identification of vulnerable parties in traffic accidents, and we fabricated experimental vehicles to validate the objectives of each of our research projects. As a result, we were able to validate the functions and performance of the various systems installed in the experimental vehicles. Moreover, we conducted validation tests and confirmed the effectiveness of these systems on reducing accidents.

Research is currently proceeding for some of these technologies with the aim of achieving actual application and use.

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Introduction of Honda ASV-2 (Motorcycles)

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ABSTRACT

For the Advanced Safety Vehicle Project Phase 2 (ASV-2) as conducted by the Ministry of Transport of Japan, two types of prototype “ASV motorcycles” were produced to promote studies on safety. The research topics for the two types of prototype “ASV motorcycles” were selected from two viewpoints:

- Active safety technologies:
 - (1) “Inter-vehicle communication system” which provides information on the mutual presence of motorcycles and automobiles.
 - (2) “Discharge headlight for motorcycles” taking into consideration the high voltage unit and optimizing light distribution.
 - (3) “TUFFUP tires” applied to minimize pressure leaks due to punctures of tubeless tires with a self-seal gel sealed in their double structure chambers.
 - (4) “Air pressure monitor system” which will display air pressure and give an alarm when certain conditions exist.
- Passive safety technologies:
 - (5) “Airbag system for motorcycles” including an airbag, inflator and sensor system.
 - (6) “Advanced safety helmet” to reduce brain damage resulting from rotational acceleration of the head.

It is believed that potential accident reduction effects may be anticipated through the application of these technologies.

1. Introduction

Motorcycles were added as a research subject for Phase 2 (1996 to 2000) of the Advanced Safety Vehicle (ASV) Development Promotion Project. In order to study motorcycle safety, a detailed analysis was made of traffic accidents in Japan, and the research topics were selected from the dual viewpoints of “active safety technologies” and “passive safety technologies”.

The experimental vehicle for studying active safety technologies was a 250cm³ displacement scooter (Foresight) equipped with the inter-vehicle communication system, discharge headlight for motorcycles, self-seal (TUFFUP) tires and the air pressure monitor system. In addition, ABS and turn signal position lights were also applied (Honda ASV-2 experimental vehicle 3).

The experimental vehicle for studying passive safety technologies was a large touring model GL1500 equipped with the airbag system for motorcycles, and an enhanced impact absorption helmet was also employed (Honda ASV-2 experimental vehicle 4). Fig. 1 shows the exterior view of the ASV-2 experimental vehicles 3 and 4. Note that there are also ASV-2 experimental vehicles (automobiles) 1 and 2, which are being researched separately. Details on the Advanced Safety Vehicle (ASV) Development Promotion Project are presented in the separate paper “Introduction of HONDA ASV-2 (Automobiles).⁽¹⁾



Fig.1 Passive (left) and active safety research motorcycles

2. Honda ASV-2 Experimental Vehicle 3 (Active Safety) Technologies

2.1. Outline

An analysis of traffic accident data revealed that intersection collisions and right-turn (cross-traffic) collisions account for the majority of accidents between motorcycles and automobiles involving fatalities. These results also showed that one of the main factors causing accidents was recognition mistakes on the part of automobile drivers. Therefore, experimental vehicle 3 was equipped with the following functions to assist in accident prevention by

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improving visibility at night and pressure-loss resistance due to tire punctures:

- (1) Inter-vehicle communication system, which provides information using a communication system installed on both motorcycles and automobiles to inform riders and drivers of each other's presence.
- (2) Discharge headlight for motorcycles, which improves visibility during nighttime driving.
- (3) TUFFUP tires and air pressure monitor system, which improve resistance pressure-loss due to tire punctures and inform riders of abnormal air pressure.

Fig. 2 shows where these systems are installed in experimental vehicle 3.

2.2. Inter-Vehicle Communication System

2.2.1. Characteristics and Factors of Accidents between Motorcycles and Automobiles

According to Japanese accident data⁽²⁾, accidents resulting in fatalities while riding motorcycles account for 18% of all accident fatalities as shown in the upper left graph of Fig. 3. In addition, the upper right graph of Fig. 3 shows that 63% of fatal motorcycle accidents involve collisions with automobiles. Furthermore, as shown in the bottom graph of Fig. 3, detailed analyses of right-turn collisions and intersection collisions revealed that the main factors were distance and speed recognition mistakes, oversight or the inability to see motorcycles hidden by other automobiles on the part of the first party (automobile). Of these factors,

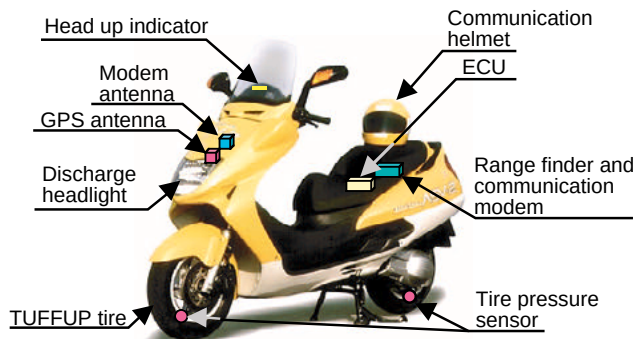


Fig.2 HONDA ASV-2 Active safety vehicle 3

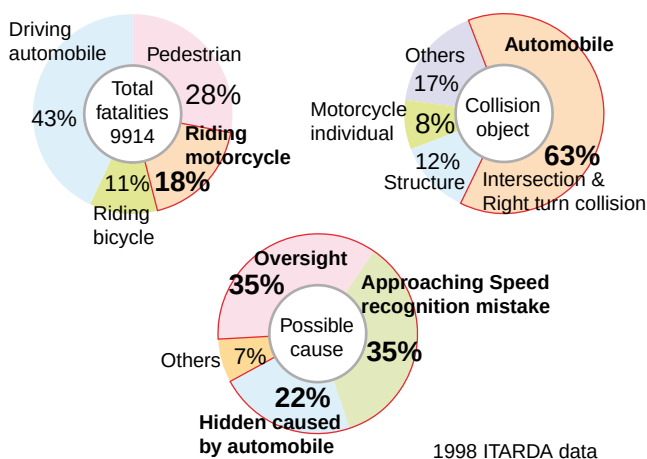


Fig.3 Rate of motorcycle accident in fatalities

approaching speed and distance recognition mistakes accounted for 35% of the total.

Fig. 4 shows the accident modes caused by these factors. Here, right-turn collisions account for 23% of the accidents, and intersection collisions 40%. The frontal projected area of motorcycles is generally smaller than that of automobiles, and it is known that drivers tend to recognize motorcycles as being further away and traveling slower than the actual distance and speed.⁽³⁾

Oversight, such as when a truck, bus or other large-sized vehicle is cruising behind a motorcycle, causing the motorcycle to easily blend into the background, is also thought to be a factor. Being hidden by automobiles is also a typical cause of accidents such as when the forward field of vision of the driver of a vehicle making a right-hand turn is obstructed by a large-sized vehicle stopped or cruising in the opposing right-turn lane, making it impossible to know or predict the presence of a motorcycle cruising in that blind area.

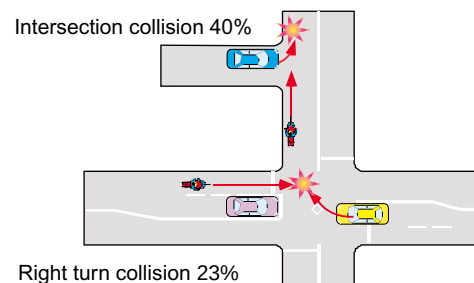


Fig.4 Typical traffic accident between automobiles and motorcycles

2.2.2. Goal of the Recognition Assistance System

We researched a recognition support system by providing various means of assisting motorcycle and automobile drivers to recognize each other's presence. This system aimed to have drivers of automobiles making right-hand turns accurately, recognize motorcycles and to have riders of motorcycles proceeding straight ahead recognize the presence of vehicles which might make a right turn in front of them.

2.2.3. Systems

The recognition support system consists of an electrical control unit (ECU), which performs vehicle information control and communication control, an inter-vehicle communication and range finder modem, and a human-machine interface (HMI). The system configuration is roughly the same for both motorcycles and automobiles.

(1) AVS-2 experimental vehicle 2 (automobile: protection of vulnerable parties in traffic accidents) systems:

A high accuracy GPS, communication and range finder modem, and navigation system are connected to an electronic control unit (ECU), which performs vehicle information control and communication control processing, and the positions of motorcycles, etc., are shown on the navigation display installed on the vehicle in real time. In addition, a high intensity head-up display (HUD) mounted in front of the driver's seat projects symbols onto the windshield and provides voice attention attracting information as necessary.

Fig. 5 shows the block diagram of this system.

An algorithm for calculating the most recent position of the driver's own vehicle based on the GPS position information, vehicle speed, acceleration/deceleration and yaw rate was developed to minimize the estimated positioning error. As a result, a real-time positioning accuracy of ± 5 m was obtained. Fig. 6 shows this positioning algorithm.

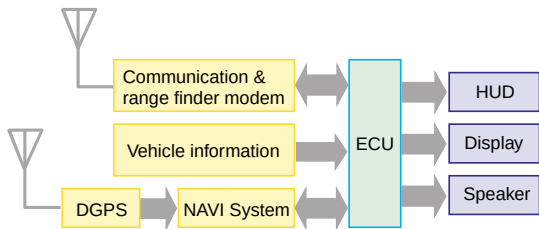


Fig.5 Block diagram of automobile system

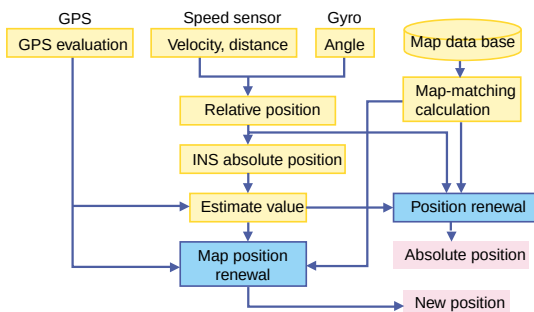


Fig.6 Estimated positioning algorithm

The system detects motorcycles and automobiles around the intersection at a distance of approximately 200 m before the intersection, and extracts vehicles likely to intersect with the future course of one's own vehicle from the cruising vectors of one's own vehicle and the other vehicles.

In particular, when a motorcycle is extracted, an enhanced symbol appears on the navigation display as shown in Fig. 7.

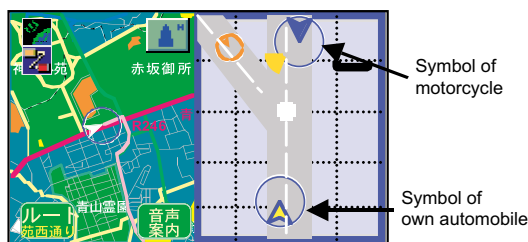


Fig.7 NAVI display information

When the course intersecting conditions obtained from the time required for the other vehicle to reach the intersection and the approach time for one's own and the other vehicles match, the system displays a caution symbol on the HUD as shown in Fig. 8 and provides a voice message.

The main contents of the inter-vehicle communication are identification (motorcycle/automobile vehicle type identification), current position (latitude and longitude), vehicle speed, estimated position, position of next intersection, arrival time at next intersection, number of

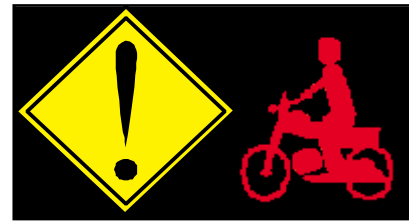


Fig.8 HUD Caution symbol

vehicles from which signals are received, one's own vehicle's state (brake, turn signal, etc.) and so on for a total data communication volume of approximately 40 bytes. The data update cycle is automatically varied depending on the speed, intersection approach and other states of one's own vehicle to comprise an autonomous decentralized communication network.

(2) AVS-2 experimental vehicle 3 (active safety) systems:

Motorcycles require lightweight and compact components; so experimental vehicle 3 was not equipped with a navigation system. Therefore, in place of referencing digital map data, the latitude and longitude of intersections in the direction of the automobile's path are obtained via inter-vehicle communication. The motorcycle system judges whether the future cruising courses intersect based on the position of one's own vehicle as determined by the GPS, the cruising vector calculated with the azimuth data and the automobile cruising vectors. Fig. 9 shows the system block diagram. In consideration of the characteristics of motorcycles, the HMI was provided with two notification systems, a head-up indicator (HUI), which allows for confirmation within the peripheral vision of the driver, and a voice message from a speaker built into the helmet. Fig. 10 shows where this equipment is installed on experimental vehicle 3.

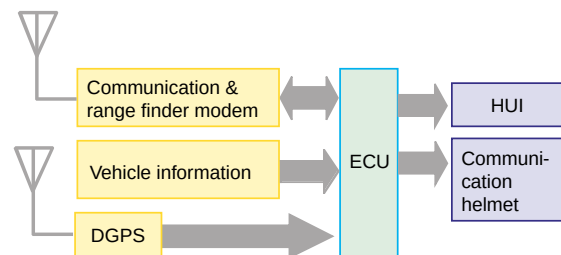


Fig.9 Block diagram of motorcycle system

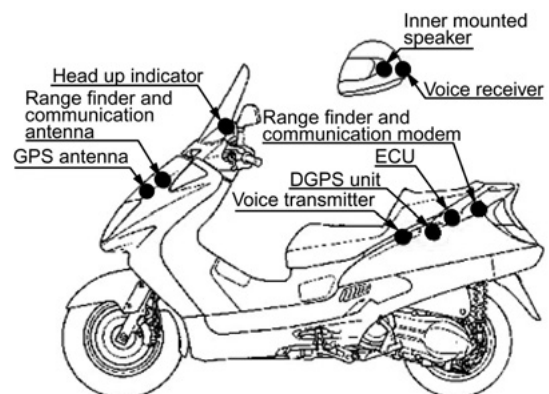


Fig.10 Outline of motorcycle system

The motorcycle path is determined while maintaining balance through two inputs, steering angle and body weight shifting, so the steering mode differs from automobiles in that the roll and yaw angle variation are detected, mainly using visual information, and which are then fed back to the path. Particularly around intersections and in other cases where caution must be given to the surrounding circumstances, providing alphanumeric information or diagrams may draw attention toward these displays and cause riders to shift their line of sight from the outside world.

Therefore, visual information is provided using a method which reflects a high intensity LED (yellow) with a wide horizontal emitting surface to the center of the windshield in a manner which can easily be recognized even with peripheral vision, and the amount of information was limited to the minimum necessary, 1 bit. Fig. 11 shows an example of the reflected image. In addition, a communication helmet was developed which provides high quality voice sound by a transmitter installed on the motorcycle and a receiver installed on the exterior of the helmet. Fig. 12 shows a photograph of this helmet. Phrases, which could be readily understood and were completed within 2 seconds, were created for the voice message.

(3) Inter-vehicle communication and range finder modem:

The radio waves from GPS satellites are cut off among high-rise buildings or under bridges, etc., and the accuracy of location often drops due to the effects of multipass, etc. Therefore, we developed a spread spectrum (SS) communication range finder modem that allows measurement of relative distance during inter-vehicle communication under these types of conditions. The spread spectrum communication system provided anti-multipass characteristics and can measure the distance to

communicating parties while communicating information, thus ensuring redundancy without adding new equipment when used together with a high accuracy GPS. This inter-vehicle communication and range-finder modem is mounted on both motorcycles and automobiles, and communication control uses carrier sense multiple access (CSMA). Future improvements in communication control, communication speed and processing speed rate are expected to allow inter-vehicle communication between all vehicles (approximately 1,000 vehicles) within a 300 m radius centering on intersections.

2.3. Discharge Headlight for Motorcycles

A discharge headlight like those used on automobiles was also newly developed for motorcycles to improve visual recognition when driving at night. Discharge headlights have the characteristics of a luminous intensity approximately two times, a life five times or more, and power consumption of approximately 70% of conventional halogen type headlights.

However, employing discharge headlights on motorcycles presented the following technical issues:

- (1) Discharge headlights are ignited by high voltages of approximately 20 kV, and the high voltage parts are close to the rider, so particular care must be given to safety when using discharge headlights on motorcycles.
- (2) The rider's eye point is higher than for automobiles, so riders look down on the boundary between light and dark from above. In addition, the difference in brightness appears greater as the center area becomes brighter.

These technical issues were resolved as follows, making it possible to mount the optimum discharge headlights on motorcycles.

2.3.1. Assuring Safety of High Voltage Parts

The headlight and igniter were integrated to enclose the high voltage parts completely inside the headlight housing, and the construction incorporates thorough electrical leak prevention measures to prevent riders from being subjected to electric shocks even when the unit is submerged. Fig. 13 shows the discharge headlight construction.

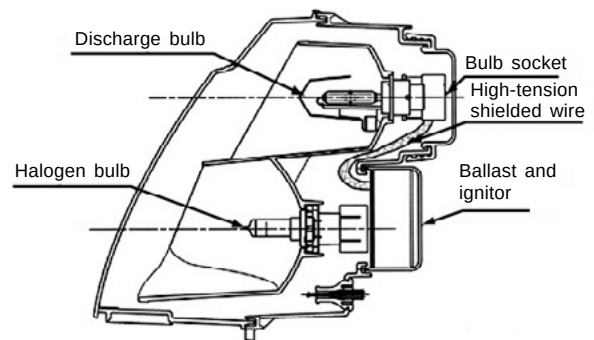


Fig.13 Cross section of discharge headlight

2.3.2. Setting a Light Distribution that Provides a Sense of Brightness

The bulb light flux of discharge bulbs increases compared to halogen bulb headlights, so the center area becomes brighter. However, this makes more distant and peripheral areas seem darker.

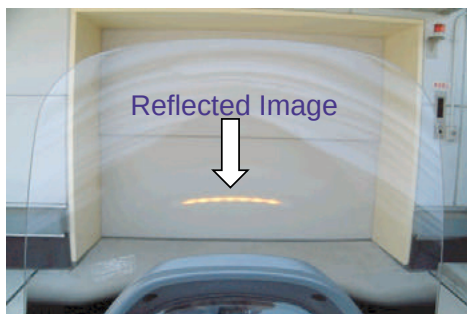


Fig.11 HUI image



Fig.12 Communication helmet

The above-mentioned contents were taken into account when setting brightness targets, and confirmation by running tests showed that the sense of brightness is greatly affected by the brightness of the range which riders normally view and the peripheral brightness. When the light was distributed focusing on this point, a sense of brightness double or more than that of conventional halogen headlights was obtained.

2.3.3. Setting an Illumination Gradient which does not Produce a Sense of Brightness Contrast

As the brightness of the center area increases, more distant areas seem darker when the vehicle body undergoes pitching, etc. When the vehicle body tilts forward, the light distribution shifts toward the vehicle and the driver's eyes are unable to follow the rapid change in illumination from the center area to the cut line. Therefore, by setting the same gradual change in illumination as conventional halogen headlights which do not produce a sense of brightness contrast, we achieved a light distribution with the same lack of brightness contrast as conventional headlights.

2.3.4. Light Distribution when Mounted on the ASV

The discharge headlight provided an enhanced light distribution with 2 times the sense of brightness, 1.3 times the range and 1.5 times the light distribution range as conventional halogen headlights, thus increasing the visual range when cruising at night. In addition, the potential for glare affecting occupants in opposing vehicles or the vehicle in front was reduced to near that of halogen headlights without adding a levelizer by setting a gradual change in illumination near the cut line. Fig. 14 shows the light distribution performance for the developed discharge headlight.

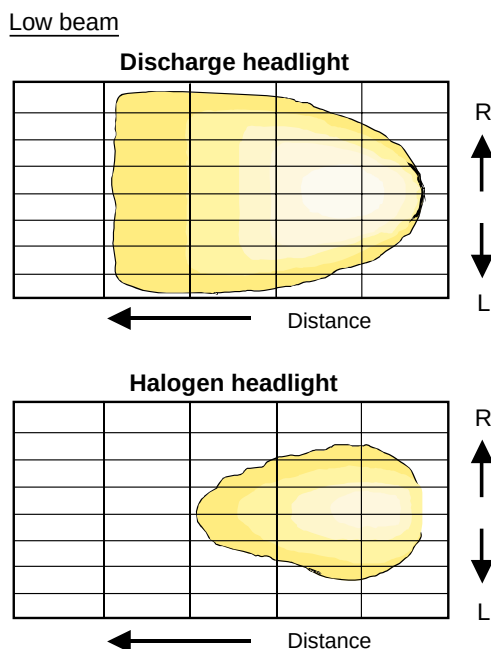


Fig.14 Headlight distribution performance

2.4. Tires with Pressure-Loss Resistance Functions (TUFFUP Tires)

As motorcycles are not generally equipped with spare tires and to help reduce pressure loss from punctures we developed a tubeless tire with a dual structure on the tire air

chamber side, and a newly developed special self-sealing gel sealed in a chamber on the tread side. This gel consists of various types of fibers and ceramic powder mixed in a highly viscous gel containing propylene glycol and adhesives, etc., and is sealed in a 1.2 mm thick chamber in the upper portion of the tire. This gel is constantly pressed by the tire air pressure, so if a hole opens in the chamber, the gel is extruded to the outside, and the fibers, etc. in the gel are pinched by the elasticity of the tire rubber to prevent air leaks.

This is the same principle used by the previously developed TUFFUP tube, which boasts mass production results of over one million units. The performance target is the ability to run up to the tire life even with a hole made by a 3 mm diameter nail. Fig. 15 shows this tire construction, and Fig. 16 shows the mechanism for preventing air leaks.

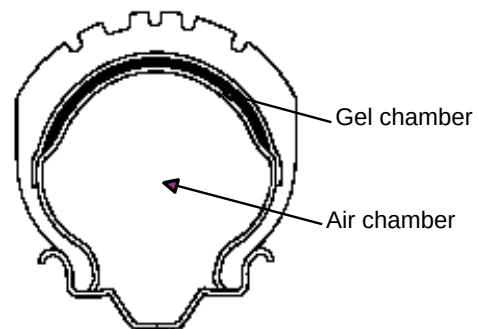


Fig.15 Construction of TUFFUP tire

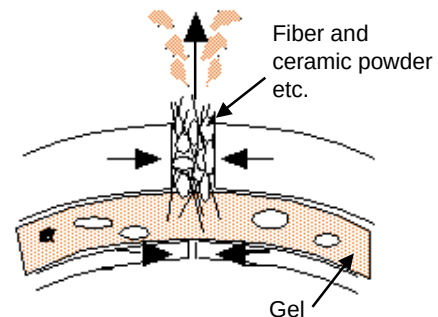


Fig.16 Puncture-plugging mechanism

2.4.1. Air Pressure Monitor System

Air pressure monitor systems are a pending technology that have been desired as a technology for notifying motorcycle riders in advance of drops in air pressure while cruising. Fig. 17 shows where each device is installed on experimental vehicle 3. As shown in Fig. 18, air pressure and temperature sensors are integrated with the tire valve inside the tire, and periodic radio wave signals are sent using power from the built-in battery. The battery life is set at six years or more. The radio waves are sent to a receiver installed in the vehicle body, and are then processed and shown on the display.

Air leaks must be recognized before significant pressure loss occurs so that the rider can stop the motorcycle. Therefore, the system shortens the interval for calculating and transmitting the air leak speed in accordance with the leak rate in order to notify the rider of the air leak as quickly

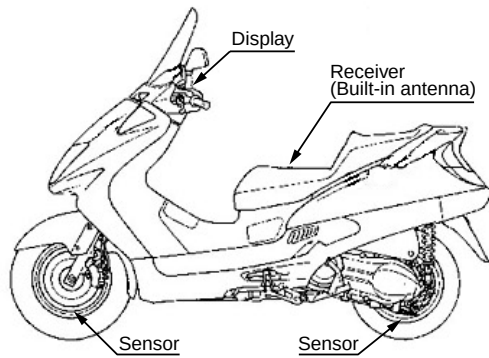


Fig.17 Layout of air-pressure monitor system

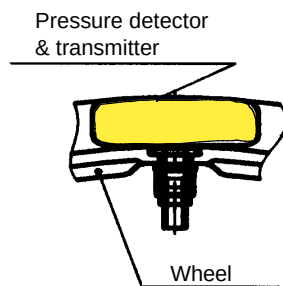


Fig.18 Construction of air-pressure sensor

as possible. The system also monitors the temperature inside the tire to predict tire bursting due to abnormally high temperatures of 100°C to 150°C, and warns the riders in advance.

Drops in the air pressure due to normal air permeation leakage from the tire rubber can also be monitored, so riders can be advised to maintain the appropriate air pressure. This is also effective for improving fuel economy and tire life, and for ensuring cruising performance.

3. Honda ASV-2 Experimental Vehicle 4 (Passive Safety) Technologies

The experimental vehicle shown in Fig. 19 was used to study an airbag for motorcycles and a helmet with improved performance in impact absorption as passive safety technologies.

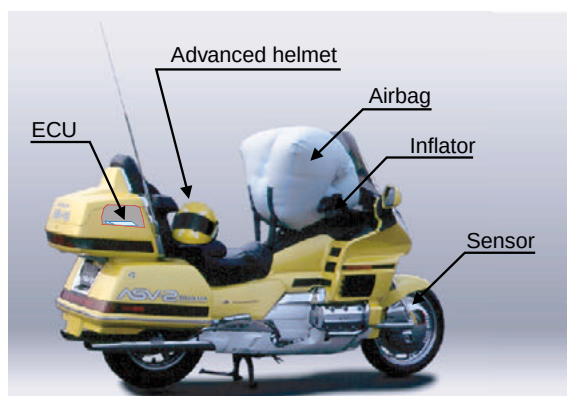


Fig.19 "ASV-2": Motorcycle for passive safety research

3.1. Airbag System for Motorcycles

The airbag system for motorcycles aims to reduce the rider's injuries caused by hitting the opposing vehicle or another object by absorbing the rider's energy in a forward collision using an airbag.

3.1.1. Overview of Prototype Airbag System

Based on the results of initial fundamental research, we decided to use the specific large-sized model (GL1500) shown in Fig. 19 for our study. The bag and inflator were determined through static inflation tests and sled tests, etc. Then the sensor system was investigated with various running tests and crash tests using a number of cases, and a prototype airbag system was fabricated.

The bag specifications were determined as shown in Table 1 to achieve high performance in the areas of restraining the rider and absorbing energy and impact. The features of the airbag are a V-shaped airbag rear surface, which the rider contacts during a collision, and belt supports connected to the frame below the seat, etc.

Table 1 Specifications of prototype airbag

Volume	(liter)	120
Height × width × length	(mm)	630 × 500 × 615
Vent holes	(mm)	φ50 × 2 holes
Bag mounting point	1) module box 2) motorcycle frame, via 2 connectiong belts	
Inflator:	Pyrotechnic, passenger car type, 450kPa maximum pressure	

The sensor system consists of acceleration sensors attached on the right and left of the bottom case of the front fork and an ECU which processes and evaluates these acceleration values. The faster of the right and left acceleration value processing results activates the inflator and inflates the airbag. Fig. 20 shows the appearance of the airbag.



Fig.20 Prototype airbag

3.1.2. Full-scale Crash Tests for Prototype Airbag System

Full-scale crash tests were carried out using real vehicles under various crash modes to ascertain system operation and the injury reducing effects of the airbag for riders.

These full-scale crash tests were carried out as prescribed by ISO 13232 (Motorcycles - Test and Analysis Procedures for Research Evaluation of Rider Crash Protective Devices Fitted to Motorcycles)⁽⁴⁾. One of the seven full-scale crash

test modes prescribed by ISO 13232 which was unrelated to this airbag was eliminated. On the other hand, five other modes such as impact at a higher speed (72 km/h), with a passenger, and in the forward leaning position were added, and the full-scale crash tests shown in Fig. 21 were carried out.

6 Configurations from ISO full scale test standard					
Test 1	Test 2	Test 3	Test 4	Test 5	Test 6
48 km/h	48 km/h	48 km/h	48 km/h	48 km/h	48 km/h
24 km/h	24 km/h	24 km/h	24 km/h	Parked	Parked
5 Configurations added for airbag feasibility research					
Test 7	Test 8	Test 9	Test 10	Test 11	
48 km/h	72 km/h	72 km/h	With passenger 48 km/h	Forward leaning 48 km/h	
Parked	Parked	Parked	Parked	Parked	

Fig.21 Configurations of full scale impact tests

The dummies were ISO 13232-MATD (Motorcyclist Anthropometric Test Device), and measurements, etc. were taken to evaluate the injury to the head, neck, chest, abdomen and leg portions.

The opposing vehicles used in the tests were 1988 to 1990 year model Toyota Corollas based on ISO 13232.

Fig. 22 shows the airbag functioning time span in the full-scale crash tests.

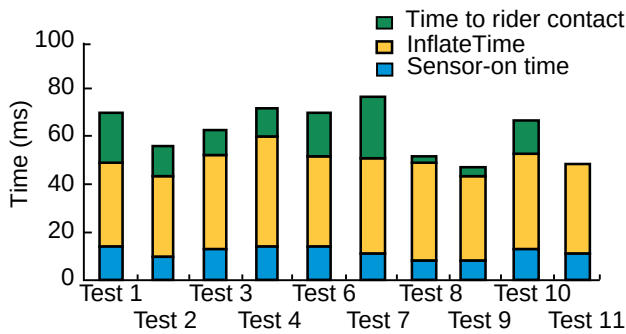


Fig.22 Airbag timing data from full scale tests

The sensor-on time is the time from the start of the crash until the ECU outputs the ON signal, the inflation time is the time from when the ON signal is output until the airbag completes inflation, and the time to rider contact is the time from when the airbag is completely inflated until the rider contacts the bag. In the Test 5 impact mode, it was found that the prescribed impact acceleration was not produced so the airbag system did not activate. Except for Test 5 and the forward leaning position Test 11, the rider arrived at the airbag after it was completely inflated, indicating that the airbag activated on time and achieved inflation.

The effects of the airbag on the injury were obtained by comparing each of the nine test modes with and without the airbag based on the ISO 13232 injury evaluation method from the results of dummy measurements in the full scale crash tests. Fig. 23 shows these results, and indicates the

change in the Abbreviated Injury Scale (AIS) for each body region. The total effect on the whole body is indicated by the change in the Normalized Injury Cost (NIC). Here, NIC = 1 is equivalent to death. When injury is reduced by the vehicle equipped with the airbag, this is shown as a benefit. Conversely, when injury increases this is shown as a risk.

These effects of the airbag on injury mainly appear in the head and neck portions. Examining the total NIC values for the whole body, the airbag reduced injury in Test 1, Test 2, Test 8 and Test 9. On the other hand, the airbag had adverse effects in Test 3 and Test 4. In addition, the airbag had no effect in Test 5, Test 6 and Test 7.

In other test results, an injury measurement value expressed by AIS did not occur for the rider on the front of the seat in Test 10 with a passenger. Also, a value suggesting neck injury was measured in Test 11 for forward leaning position impact.

The rider protective effects of the airbag were understood with this study. In the future, countermeasures must be taken to counteract the adverse effect phenomena, and tests must be carried out to confirm the effects under even more diverse impact conditions.

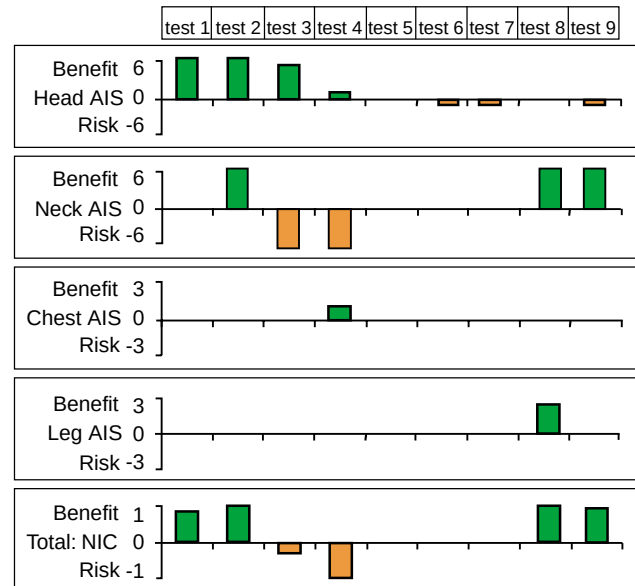


Fig.23 Airbag injury risks and benefits

3.2. Advanced Safety Helmet

Helmets exhibit significant head protection effects for motorcycle riders during accidents, but even when wearing a helmet the greatest cause of death is still head injuries as shown by the accident data⁽²⁾ in Fig. 24. Moreover, in recent investigations of accidents involving motorcycle rider fatalities it is said that 20 to 30% of head injuries are caused by impacts involving mainly rotational acceleration. This study pursued the new viewpoint of reducing brain injury caused mainly by rotational acceleration of the head.

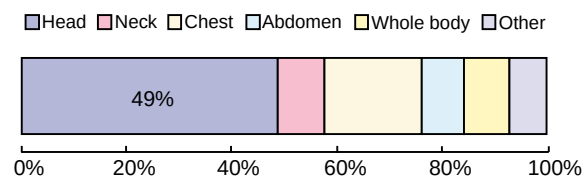


Fig.24 Body region of fatal injury

3.2.1. Outline of Prototype Helmet

A new impact absorption layer was set between a double liner as shown in Fig. 25, and the helmet construction allowed both liners to move around this layer. This construction reduces the rotational movement transmitted to the head when impact type rotational force acts on the helmet.

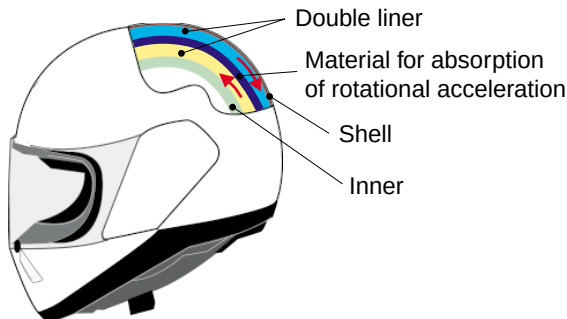


Fig.25 Prototype helmet

3.2.2. Preliminary Test Results for the Prototype Helmet

Fig. 26 shows the results of measuring the rotational acceleration produced on the head portion when the head of a Hybrid III dummy wearing a helmet is dropped from a height of 2.5 m onto a 45° incline concrete surface.

The prototype helmet reduced the maximum rotational acceleration value compared to the base helmet, indicating the feasibility of countermeasures.

In the future, optimization of construction and materials must be studied, and research on methods for evaluating performance from this perspective is also necessary.

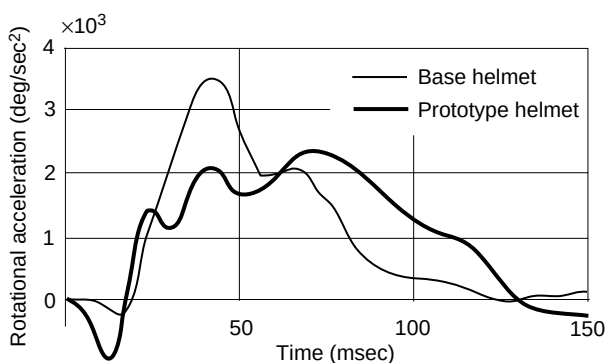


Fig.26 Test result of prototype helmet

4. Conclusion

In response to the ASV-2 project, we have conducted research on active and passive safety technologies for motorcycles, calculated the accident reduction effects of the developed technologies based on the accident effect estimation method of the Japan Automobile Research Institute, and believe that the potential accident reduction effects shown in Table 2 may be expected.

The motorcycle safety technologies introduced in this paper are still in the feasibility study phase, but research is currently proceeding for some of these technologies with the aim of actual application.

Table 2 Effect of motorcycle accident decrease rate

ASV vehicle 3 systems	Fatalities	Casualties
Inter-Vehicle communication system	– 12.2%	– 8.9%
Discharge headlight for motorcycles	– 0.3%	– 0.03%
TUFFUP tires	– 0.3%	– 0.04%
Air pressure monitor system	– 0.5%	– 0.06%
Total	– 13.3%	– 9.0%

ASV vehicle 4 systems	Fatalities	Casualties
Air bags for motorcycles	– 6.1%	– 7.6%
Helmet with improved performance	– 7.5%	– 1.2%
Total	– 10.6%	– 7.6%

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Research for Emissions Reduction Systems for Small Motorcycles

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ABSTRACT

Emissions regulations for small motorcycles are becoming increasingly stringent every year in various countries throughout the world. Therefore, we conducted research on high-precision control of the air-fuel ratio with an emissions reduction system for small, fuel-injection motorcycles, in particular, for motorcycles equipped with a 100cm³ four-stroke engine. Fuel injection is controlled by two injection amount maps applied according to throttle opening, engine speed and intake pressure. The control index used in this control system maintains the engine intake pressure peak when the intake valve is open in order to eliminate the major impact of intake pressure fluctuation exhibited by single-cylinder engines. Moreover, O₂ feedback control and three-way catalyst systems were applied for exhaust emissions purification, and a small, power-saving fuel pump was developed to improve fuel economy.

In addition, a control system to obtain excellent starting ability with a kick starter was also developed. By adopting these systems, hydrocarbons were decreased to about 1/30 of the regulation value provided in the European motorcycle emissions regulation "Directive 97/24/EC chapter 5" (EURO-1), and the fuel economy was improved by approximately 20% compared to the base engine.

1. Introduction

Emissions regulations for small motorcycles are becoming increasingly strict every year. For example, in Thailand, a major market for 100cm³ small motorcycles, the fourth emissions regulation is currently in force, and the even more restrictive fifth emissions regulation is scheduled to be introduced in 2003. Table 1 shows emissions limits for motorcycles in major regions.

To this point, most emissions reduction systems for small motorcycles have been based on the use of a carburetor, air injection, and catalyst in combination.

When fuel is supplied by the carburetor, there is a limit to the precision with which the air-fuel ratio can be controlled, and in recent years it has become difficult to both improve emissions purification and maintain driveability.

Therefore, research has been conducted on the incorporation of fuel injection in 100cm³ four-stroke engines as an effective means of reducing emissions for small motorcycles.

This paper describes how we adapted a fuel injection system and an exhaust-gas conversion device for small motorcycle applications, and the results that we achieved.

2. Target Emissions Levels

One focus of this research was to reduce THC emissions, so we set our THC target level to the same level mandated for passenger vehicles under Japan's 2000 emissions regulations. This level is approximately 1/30 of the THC standard in Europe's "Directive 97/24/EC chapter 5" (EURO-1) motorcycle emissions regulations. We also set the same levels called for in the above regulations for CO and NOx emissions. Table 2 summarizes the target emissions levels that we set for this research.

Table 1 Current emission limits for motorcycles

	CO (g/km)	THC (g/km)	NOx (g/km)
EURO-1 (4 stroke)	13	3	0.3
Japan (4 stroke)	13	2	0.3
Thailand Level 4	4.5	3 (combined)	

Table 2 Targets emissions

	CO (g/km)	THC (g/km)	NOx (g/km)
Targets	0.67	0.08	0.08

* Asaka R&D Center

3. Overview of Test Engine

Our test engine was a Honda 100cm³ four-stroke engine used widely around the world. Table 3 lists the major specifications for this engine.

Table 3 Major specifications of the test engine

Items		Specifications
Bore × Stroke	(mm)	50.0 × 49.5
Displacement	(cm ³)	97.2
Compression ratio		8.8
Valve system		OHC 2 Valve
Max. power	(kW / r / min)	5.32/8000
Max. torque	(Nm / r / min)	7.23/6000

4. Issues Concerning the Utilization of Fuel Injection

Some of the technologies that are commonly used for emissions reduction are listed below.

- (1) Use of fuel injection in the fuel supply devices, with precise control of the air-fuel ratio.
- (2) O₂ feedback control that monitors the density of O₂ in the exhaust gas and adjusts the air-fuel ratio to the stoichiometric air-fuel ratio.
- (3) Installation of a three-way catalyst in the exhaust pipe, purifying the exhaust gas through oxidation and reduction.
- (4) Introduction of secondary air in the exhaust port to promote the purification of unburned gases.

In order to apply such devices in a small-displacement single-cylinder engine used in a small motorcycle, it is necessary to be able to control minute quantities of fuel, keep the weight light, prevent a decrease in engine output, and keep costs low.

The following issues needed to be addressed:

- (1) The fuel supply control would have to be suitable for single-cylinder engines, which exhibit major fluctuations in engine intake pressure.
- (2) The installed catalyst must not lower engine output.
- (3) The system must reduce the electrical load on the engine as the electrical load has a negative effect on fuel economy.
- (4) The engine must still be startable by means of a kick starter.

5. Overview of the Emissions Reduction System

Fig. 1 shows the emissions reduction system that we adopted. A 16-bit CPU is used for the fuel injection ECU. The ECU controls the fuel injection amount, the fuel injection timing, and the ignition timing. The ECU controls the fuel injection amount in response to the O₂ density that is detected in the exhaust gas by an oxygen sensor installed between the engine and the catalyst. Fuel is delivered to the injector under pressure by a power-saving internal gear-type fuel pump located in the fuel tank. In order to reduce the electrical

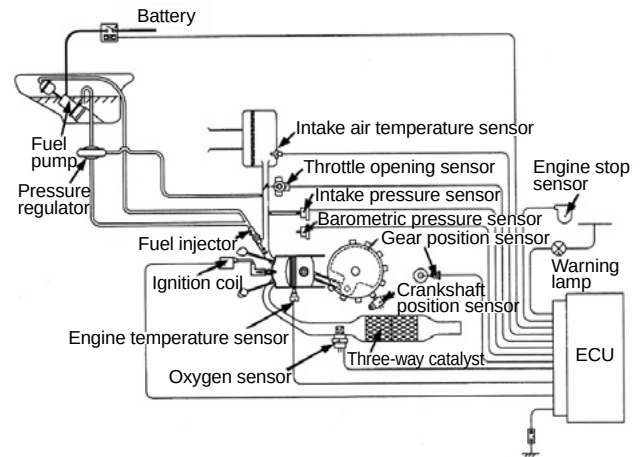


Fig.1 Emission reduction system

load of the fuel pump, the fuel supply pressure is set at 250kPa. By adjusting both the carburetor specifications and the intake pipe length, we were able to maintain the engine torque characteristics and by reducing the number of curves in the exhaust system, we were able to lower the exhaust resistance. The injectors are positioned so that the fuel injection is directed at the back of the inlet valve. Fig. 2 shows the location of the inlet system. The exhaust-gas conversion system has the metal substrate of the three-way catalyst located inside the muffler.

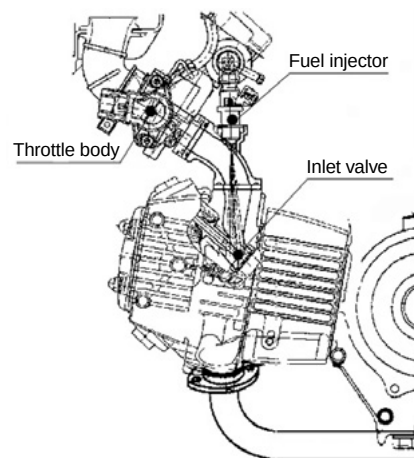


Fig.2 Air intake system

6. Fuel Supply Devices

6.1. Fuel Injection Control

We decided to use a fuel injection system with two injection amount maps. The switching between the two maps is controlled in accordance with the throttle opening and the engine speed.

- (1) At low loads, slight changes in the throttle opening are detected by the engine intake pressure, and an engine intake pressure injection amount map based on the engine intake pressure and the engine speed is used.
- (2) At high loads, a throttle injection amount map that is

based on the throttle opening and the engine speed is used, allowing continual detection of changes in the throttle opening.

Although it is necessary to know the engine intake pressure as an indicator for controlling the amount of fuel injection, the combustion interval in a single-cylinder engine is 720 degrees, so there are large fluctuations in the intake pressure on which the fuel injection amount control is based.

As shown in Fig. 3, the engine intake pressure varies during the combustion interval. Therefore, the value of the engine intake pressure sensor output varies according to when the output is checked, which means that it cannot be used for controlling the amount of fuel injection. In the system that we developed, the ECU stores the peak engine intake pressure while the inlet valve is open, and then uses that value as the indicator for controlling the amount of fuel injection.

In a conventional fuel injection system, reluctors are located every 30 degrees on the crankshaft in order to determine the rotational angle of the crankshaft. These reluctors were used in combination with the signal from the camshaft position sensor in order to determine the rotational angle and stroke of the crankshaft. In our system, we have eliminated the camshaft position sensors. Instead, we left a 120-degree interval between the reluctors in just one location. The system is then able to determine the position of the crankshaft by detecting the location of this longer interval. The stroke is determined on the basis of the difference in intake pressure between the intake stroke and the expansion stroke.

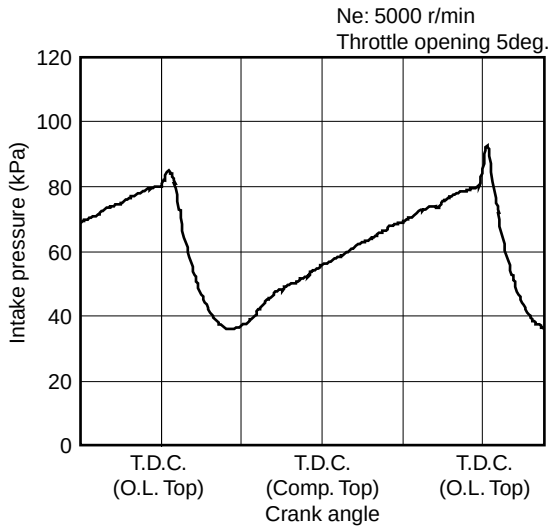


Fig.3 Change of engine intake pressure

6.2. Fuel Injector

In order to use fuel injection in a small displacement engine, it was necessary to adopt a small-flow injector that was suited to measuring out minute quantities of fuel. Table 4 lists the fuel flow specifications for two injectors.

We used three types of injectors to study the effect of fuel particle diameter on emissions: pintle nozzle injectors, four-hole plate injectors, and air assist injectors.

Fig. 4 shows the injection forms and Sauter's Mean

Table 4 Fuel flow specifications of injectors

Items	Developed Injector	HONDA VFR800FI
Dynamic Injection quantity at 2 msec ($\text{mm}^3/\text{stroke}$)	1.68	4.38
Static Injection quantity(cm^3/min)	107.5	248.5

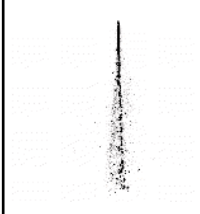
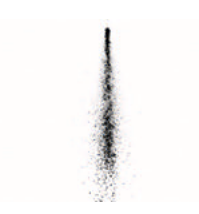
Pintle nozzle	4-hole plate	Air assist
		
S.M.D.:125 μm	S.M.D.:95 μm	S.M.D.:40 μm
Measurement conditions: Fuel pressure: 250kPa Assist air pressure: 26kPa S.M.D.: Measured by LDSA		

Fig.4 Forms of injectors and Sauter's mean diameter

Diameter (S.M.D.) for each of the injectors that we tested.

Because small motorcycle engines usually operate at high revolutions per minute, the inlet valve is usually open for only a short period of time. As a result, some of the injected fuel does not directly enter the combustion chamber; instead, it stays around the back of the inlet valve which has become heated. The fuel is vaporized by the heat and then enters the combustion chamber. Therefore, in testing of the ECE40 mode, which has a warm-up cycle, we were not able to determine whether the atomization produced by different injectors was effective in reducing emissions or not. As a result, we performed an emissions comparison test in LA-4 mode which does not have a warm-up cycle.

Our results indicated that LA-4 mode is more suitable than ECE40 mode as a means of evaluating injector particle diameters. Fig. 5 shows that in LA-4 mode, when using a pintle nozzle, greater atomization produced a decrease in CO and THC, but an increase in NOx. When these results

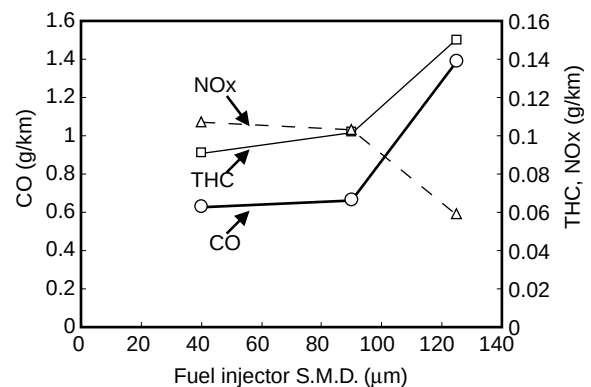


Fig.5 Emission results in LA-4 mode

were compared with results obtained using a four-hole plate injector and an air assist injector, no major differences in emissions were found. As a result of these findings, we elected to utilize the four-hole plate injector due to its simple structure and low cost.

6.3. O₂ Feedback System

To keep the air-fuel mixture at the stoichiometric air-fuel ratio, we adopted a feedback system that uses an oxygen sensor in order to efficiently utilize the performance characteristics of the three-way catalyst installed in the exhaust system. In a single-cylinder engine, fuel is injected into the cylinder once for every 720 degrees of crank rotation. As a result, there was concern that the effects of slow injection control and the exhaust pulse would prevent the oxygen sensor from generating a stable output. Because the relationship between the oxygen sensor output and the air-fuel ratio is not linear, we artificially increased and decreased the fuel supply and used feedback control to control the air-fuel ratio. We were able to implement O₂ feedback control effectively by shortening the cycle as compared with a conventional four-cylinder engine.

7. Exhaust-gas Conversion Devices

7.1. Catalyst Location and Early Activation

In order to activate the three-way catalyst early and promote purification, it is necessary to locate the catalyst in a position where the exhaust gas is hot. However, the exhaust resistance at such a position is high. In order to attain the required output, it is necessary to reduce the exhaust resistance. We studied the placement of the catalyst in a location that would be able to satisfy these two opposing requirements. Fig. 6 shows the exhaust gas temperature at different points in ECE40 emissions measurement mode.

In ECE40 mode, there are two warm-up cycles prior to the measurement cycle; it is essential that the catalyst become fully activated during these two cycles. Therefore, we set the target catalyst activation temperature at the third peak of the first cycle in ECE40 mode to at least 350°C. In our test engine, we were able to achieve the target temperature by locating the catalyst at a position within 480mm of the exhaust valve. After performing computer simulations with the purification rate required by the exhaust gas volume, we settled on a catalyst substrate cell number of 300, and a

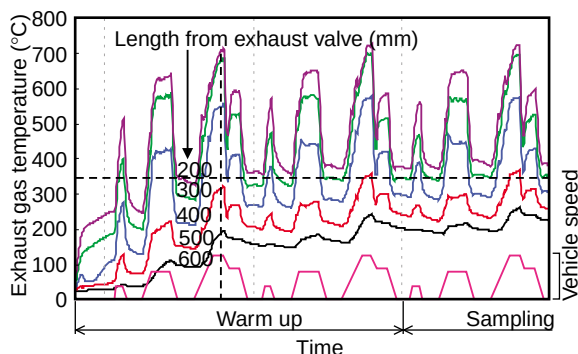


Fig.6 Exhaust gas temperature (ECE40 mode)

volume of 250cm³. Regarding catalyst placement, we studied two cases with catalysts of different diameters, as shown in Table 5.

Figs. 7 and 8 show the results of THC emissions and output measurement in ECE40 mode.

Greater output is obtained when using the catalyst with the larger diameter because that catalyst permits utilization of the exhaust pulse in the pre-catalyst chamber, and because it has lower output resistance. However, because it permits little flexibility in setting the installation position, the catalyst ends up being positioned towards the rear of the exhaust system. This results in a lower catalyst temperature, which in turn results in a lower purification rate than with the smaller-diameter catalyst. In our system, we adopted a large-diameter (60mm) catalyst located 460mm behind the exhaust port, minimizing the decrease in output while still maintaining the catalyst activation temperature.

Table 5 Specification of Test catalyst

Items	Spec. A	Spec. B
Substrate material	20 Cr 5 Al	
Diameter (mm)	60	40
Length (mm)	90	90
Number of catalyst	1	2
Volume of catalyst (cm ³)	254	226
Substrate cell number (cpsi)	300	
Wall thickness (μm)	50	
Precious metal	Pt/Rh	

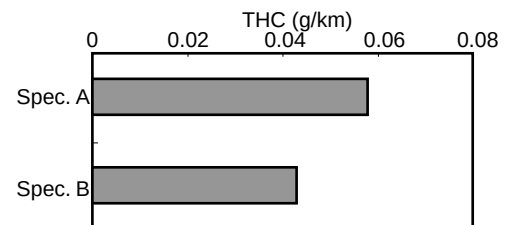


Fig.7 Result of THC emissions with different catalysts

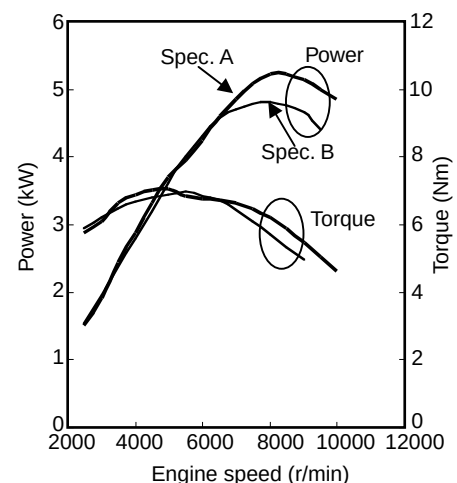


Fig.8 Engine performance with different catalysts

7.2. Secondary Air Induction System (Air Injection System)

We studied the effectiveness of reducing emissions by means of secondary air induction in the exhaust port. The ECU controls air injection by means of a solenoid valve so that air injection occurs when O_2 feedback is not functioning while the engine is cold and while the engine is idling. As Fig. 9 shows, one effect of early activation of the catalyst is an increase in temperature at the first peak in the initial stage of ECE40 mode, reducing the time until catalyst activation by 19 seconds. However, there was no difference in catalyst temperature from when emissions were measured.

Note that because the emissions reduction system that we developed met the emissions targets without air injection, we did not adopt air injection in our system.

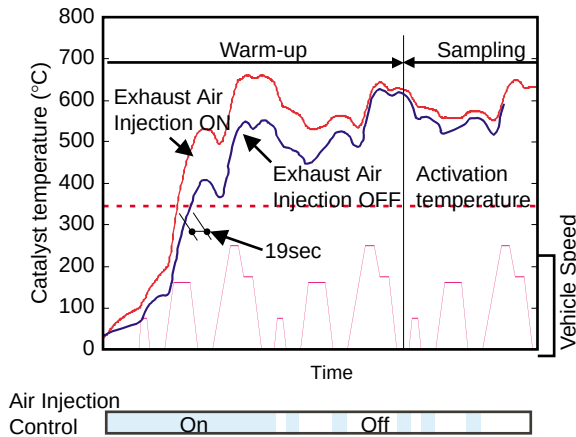


Fig.9 Catalyst temperature in ECE40 mode

8. Technologies for Improving Fuel Economy

8.1. Optimization of Ignition Timing

Because the ignition timing has a large effect on emissions, fuel economy, and driveability, we maintained the optimum ignition timing by using a three-dimensional map based on the throttle opening and the engine speed. Fig. 10 shows the ignition timing 3D map. In the partial area, we advanced the ignition timing by a maximum of 10 degrees versus the base engine in accordance with the throttle opening. The effect of this advance was a 5% improvement in fuel economy in ECE40 mode.

8.2. Low-current Fuel Pump

Because of the constraints on power generating capacity, a power-saving fuel pump is needed in small motorcycles. Therefore, we selected a high-efficiency capacity internal gear pump. Although we initially studied the use of automobile-application internal gear pumps, it was difficult to reduce the current because the large sliding surface area resulted in large friction loss. Therefore, in order to reduce the driving torque, we developed an internal gear with a smaller outer diameter and thickness. We also adopted a high-performance motor with rotors wrapped with a large

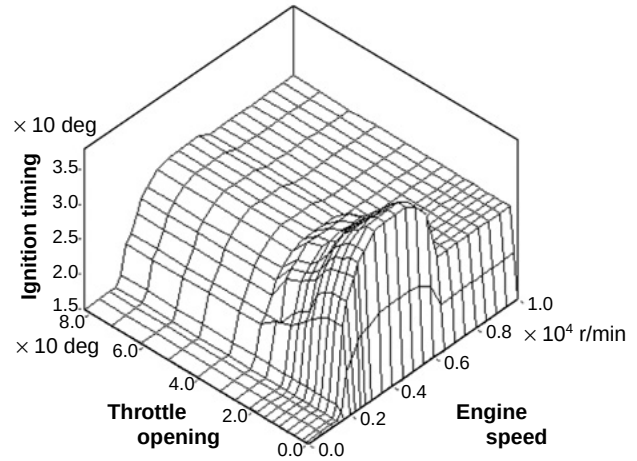


Fig.10 Ignition timing 3-D map

number of thin copper wires and magnets that generated a large magnetic flux density. We were able to reduce the current as a result and produce a power-saving fuel pump. Fig. 11 shows the characteristics of the automobile pump and the developed pump for motorcycles.

As shown in Fig. 12, by utilizing optimum control of the air-fuel ratio through fuel injection and by cutting down the fuel supply during deceleration, we were able to improve fuel economy in ECE40 mode by 20% versus the base vehicle.

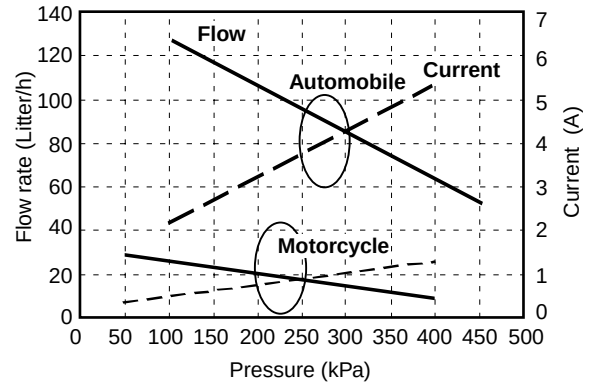


Fig.11 Features of pump developed for small motorcycles

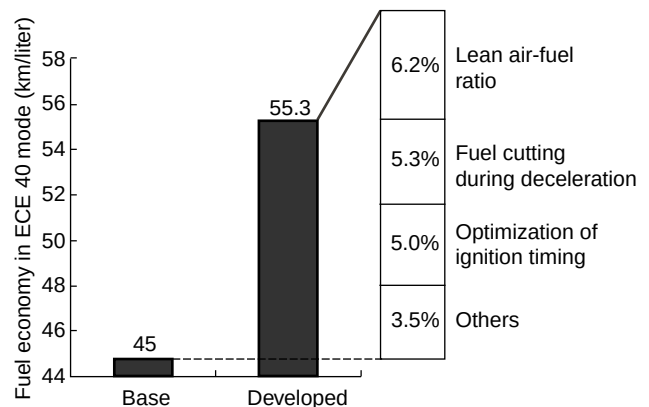


Fig.12 Fuel economy improvement

9. Control of the Kick Start

In small motorcycles, the startability of the engine with a kick starter is extremely important. Kick starting only rotates the crankshaft five or six times with one kick. Furthermore, fuel injection control requires about three rotations of the crankshaft from the start of rotation until fuel is injected in order to allow for detection of the injection interval and for operation time.

After eliminating the rotations in which there is no chance of ignition, there are only one or two rotations after a kick start during which ignition is possible. Therefore, startup can easily fail. In order to improve this situation, we introduced a special startup control system that detects the start of the rotation of the crankshaft by means of the signals from the crank position sensors; fuel injection begins when five signal pulses are detected. This control system allowed our system to start up at the same number of crankshaft rotations as the base vehicle.

10. Test Results

The results of emissions, fuel economy and output tests of our 100cm³, single-cylinder, four-stroke engine using a fuel injection system, O₂ feedback system and ignition timing control system are shown below.

- (1) Table 6 shows the ECE40 mode emissions targets and the test results.
- (2) Table 7 shows the fuel economy in ECE40 mode. Compared to the base motorcycle, fuel economy was improved by about 20%, from 45.0km/l to 55.3km/l.
- (3) Although engine output decreased from 5.32kW for the base engine to 5.22kW for our engine, Table 8 shows that our engine could still achieve a maximum speed and acceleration equivalent to the base engine.

Table 6 Test results of ECE40 mode emissions

	CO (g/km)	THC (g/km)	NOx (g/km)
Result	0.44	0.058	0.023
Targets	0.67	0.08	0.08

Table 7 Test results of fuel economy by ECE40 mode

	Fuel economy (km/l)
Result	55.3
Base	45.0

Table 8 Test results of power performance

	Maximum speed (km/h)	0-200m Acceleration time (sec)
Result	87	13.95
Base	86	13.95

11. Conclusion

In order to decrease small motorcycle emissions, we developed a fuel injection system, O₂ feedback system, three-way metal catalyst system, and ignition timing control system, all suited for a 100cm³ single-cylinder four-stroke engine. Adopting these systems made the following achievements possible:

- (1) By adopting a three-way catalyst and O₂ feedback, we were able to reduce emissions to 1/30 of the current EURO-1 regulations. The emissions in ECE40 mode are CO: 0.44g/km; THC: 0.058g/km; and NOx: 0.023g/km.
- (2) By using fuel injection to obtain a leaner air-fuel mixture, optimizing the ignition timing and adopting a low-current fuel pump, we were able to improve fuel economy by 20% compared to the base vehicle.
- (3) Our selection of the three-way catalyst installation position and size allowed us to achieve early activation of the catalyst while maintaining engine output.
- (4) By eliminating the camshaft position sensors and using engine intake pressure sensors, we were able to develop an inexpensive fuel injection system for small motorcycles.

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- (1) M. Abe, et al, Exhaust emission reduction for the VFR800FI, International Motorcycle Conference (Ifz Conference), Germany, 1997

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Development of General Purpose V-Twin Cylinder Engine GCV530/520

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ABSTRACT

A 90-degree V-twin OHC vertical shaft engine for riding mowers has been developed. The V-twin cylinder layout and the aluminum sintered alloy connecting rod enabled the offset between cylinders to be decreased, the weight of the reciprocating parts to be reduced and the vibration level to be reduced by 50% compared to conventional single cylinder engines. The OHC construction with built-in timing belts, plastic cam pulleys and a turbo shaped cooling fan facilitated a noise level reduction of 4 dB compared to conventional single cylinder engines. Further, ease of assembly was improved with the application of uni-block cylinders, common parts for both cylinders and sub-assembly constructions. The engine size and cost are comparable to those of conventional single cylinder engines.

1. Introduction

General purpose vertical shaft engines have a world-wide market scale of approximately 1.5 million units per year, and of these approximately 90% are mounted on home use riding mowers. Price competition in this market is extremely fierce, and low-cost single cylinder engines with a simple construction dominate the market.

Home use riding mowers are generally used for about two hours once or twice per week from spring to autumn. The most common request from users is to “make work easier,” so the engine specifications include ample power and low vibration and noise. In addition, compliance with environmental regulations is also necessary.

The 16 HP general purpose V-twin engine GCV530 shown in Fig. 1 was developed as a new engine to meet these needs. The 14HP specification GCV520 was also

developed to meet a wider range of needs. At the same time, an engine for professional use riding mowers that require additional durability compared to home use has been developed.

2. Development Concept

Development of this engine was based on the concept of realizing the optimum vertical shaft engine for use on riding mowers. The price to power positioning was set as shown in Fig. 2, and a market-competitive engine was developed focusing on the following points:

- (1) Easy to use and not tiring
 - Ample power
 - Low vibration
 - Low noise
- (2) Optimal for the application



Fig.1 External view of GCV530

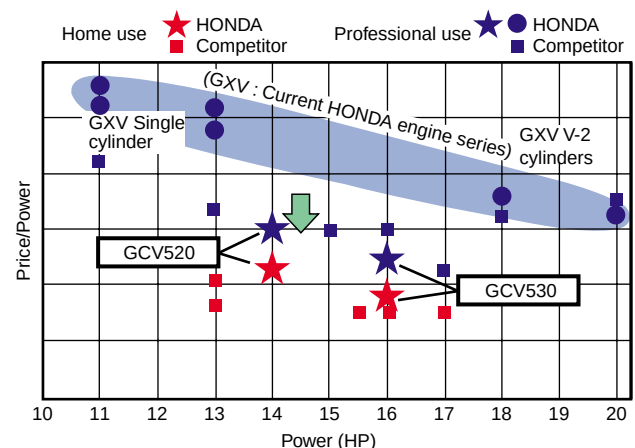


Fig.2 Positioning of GCV530/520

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- Durability (home/professional use)
 - Interchangeability (with single cylinder engines)
- (3) Compliance with environmental regulations
- California Air Resources Board (CARB) and Environmental Protection Agency (EPA) exhaust emissions regulations
 - European Norm (EN) noise regulations

3. Main Specifications and Basic Construction

The basic construction of the GCV530/520 is a vertical shaft type engine with a vertically positioned crankshaft. The displacement was set at 530 cc to provide ample power compared to competitors' 500 cc class single cylinder engines, and a 90° V-twin cylinder construction was adopted to achieve low vibration and noise. The main technologies used to achieve a size comparable to single cylinder engines and ensure market competitiveness while still maintaining low vibration and noise performance are: (1) uni-block cylinders, (2) a valve train with built-in timing belts, (3) aluminum sintered alloy connecting rods, (4) a turbo shaped cooling fan, (5) parts constructions for subassembly, and (6) exhaust emissions reduction technology, etc. In addition, a left-right asymmetrical cylinder layout was used to enable interchangeability with single cylinder engines without changing the mower body construction. GCV520 has a smaller bore carburetor, and the professional use engine has cast-iron sleeves instead of sleeveless aluminum cylinders. Table 1 gives the main specifications of the GCV530/520, and Fig. 3 shows the sectional drawing.

4. Technologies Used to Achieve Target Performance

4.1. Uni-block Cylinders

The uni-block cylinder technology⁽¹⁾ used in the general purpose engine GCV160/135 to achieve compact, light work equipment was adapted to reduce both the size and the number of parts. The cylinder heads and cylinders were integrated, the cylinders were bolted to the crankcase, and the #1 and #2 cylinders shared a common design. As no

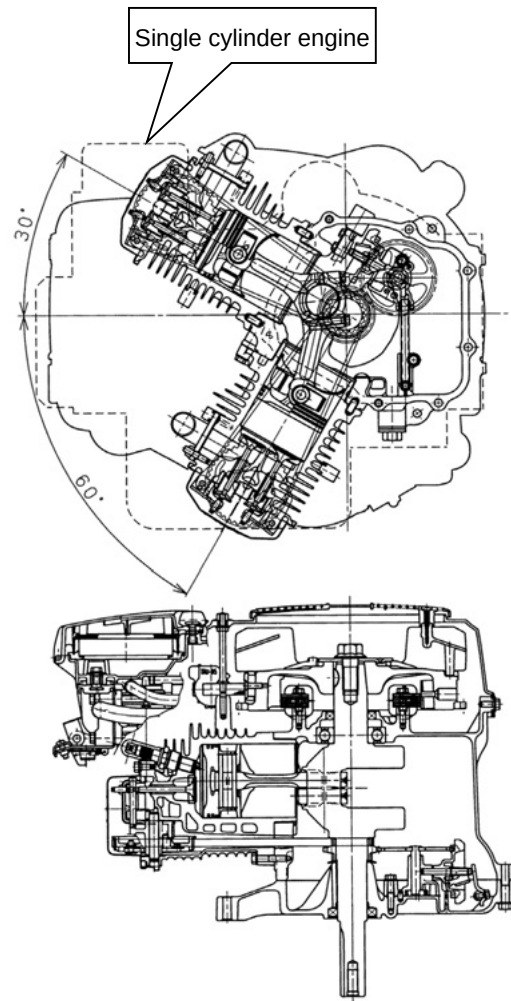


Fig.3 Layout of GCV530/520

Table 1 Specifications

Model name	GCV530	GCV520
Engine type	Air cooled 4-stroke OHC V-90° Twin-cylinder	
Bore × Stroke (mm)	77 × 57	
Displacement (cm ³)	530	
Compression ratio	8.0	
Maximum power (kW/rpm)	11.3/3600	9.7/3600
Maximum torque (Nm/rpm)	34.3/2500	33.0/2500
Lubrication	Splash	
Carburetor type	Twin bore, float, butterfly, side draft	
Carburetor venturi bore (mm)	18	14
Ignition system	Transistorized magneto	
Starting system	Electric/Recoil	
Dimensions (L × W × H) (mm)	456 × 427 × 331	
Dry weight (kg)	30.5	
Cylinder sleeve	Home-use : Aluminum (Sleeveless)	
	Pro-use : Cast-iron	

head bolts are used, it is possible to position the spark plugs roughly in the center, thus improving combustion efficiency. In addition, an OHC valve construction was used to eliminate the cam pulleys inside the crankcase thus reducing the crankcase size and weight. Fig. 4 compares the new construction with the conventional construction.

In terms of specifications, two types of cylinders were prepared: sleeveless aluminum cylinders and cast-iron sleeve molded cylinders, with the former applied to home use and the latter applied to professional use mowers which require higher durability. Furthermore, this construction allowed the specifications to easily be changed according to the application. Fig. 5 shows sectional drawings of the cylinders.

This construction reduced the overall weight of the aluminum die cast parts by approximately 45%. Fig. 6 shows the weight comparison with a conventional twin cylinder engine. Furthermore, in terms of manufacturing efficiency as well, two types of cylinder die cast molds were previously required, but these were integrated into a single type. Also, production was made possible with the same size casting machine as for conventional engines. The crankcase die cast machine size was also reduced from 1,250 tons to 800 tons, thus greatly improving manufacturing efficiency and reducing die investment costs.

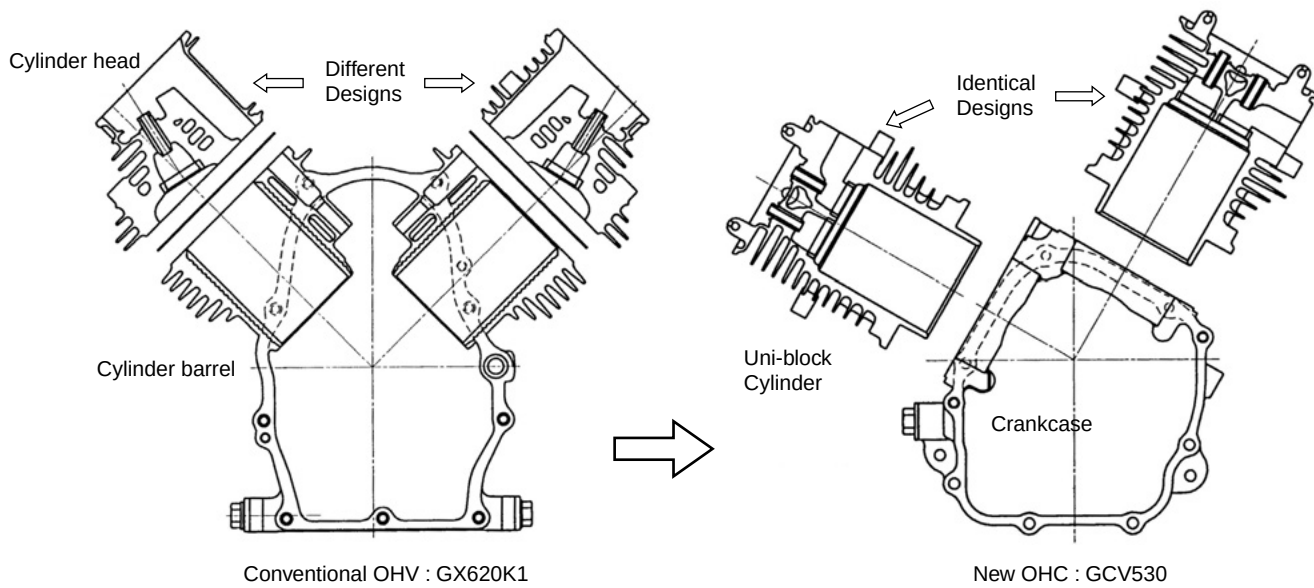


Fig.4 Comparison of cylinder and crankcase constructions

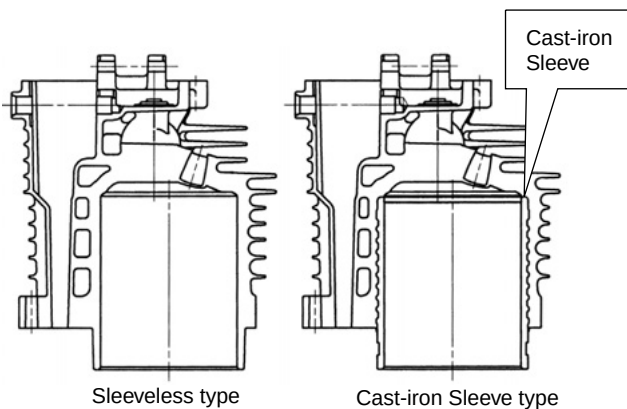


Fig.5 Comparison of cylinder sleeves

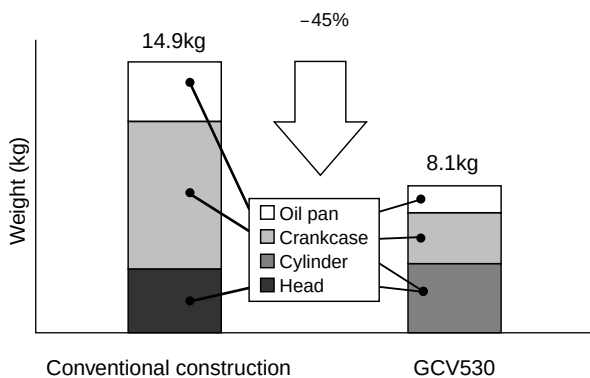


Fig.6 Weight comparison of aluminum die cast parts

4.2. Valve Train with Built-in Timing Belts

Achievements in noise, size, weight and cost reduction were made based on the OHC mechanism parts configuration consisting of timing belts built into the engine, L-shaped pressed rocker arms, and a plastic one lobe cam and pulley used first in the GC/GCV135-160⁽¹⁾ series which has been mass produced since 1997. The new engine attached two timing belts to the one piece sintered timing pulley press-fit onto the crankshaft, and used an L-shaped rocker arm to allow the cam pulleys to be positioned on the intake and exhaust valve sides. Fig. 7 shows this construction, and Fig. 8 shows the sectional drawing.

The timing belts being built into the engine eliminates the dust and water contamination that affect general purpose engines and limit the belt environment to an oil-only atmosphere, thus improving timing belt reliability and

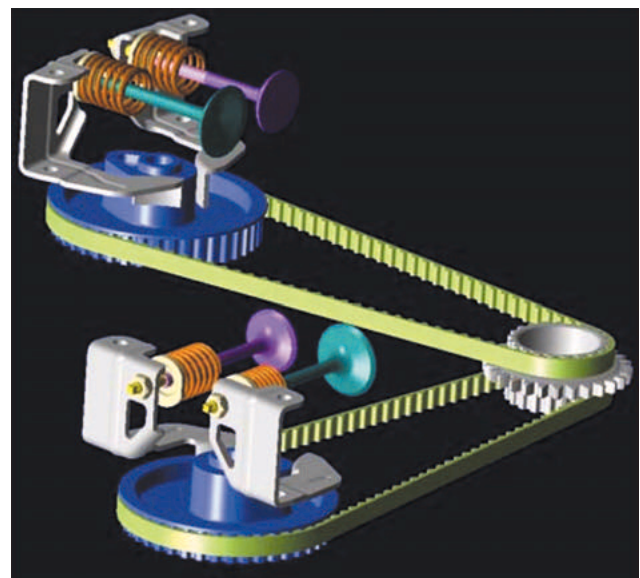


Fig.7 Valve train

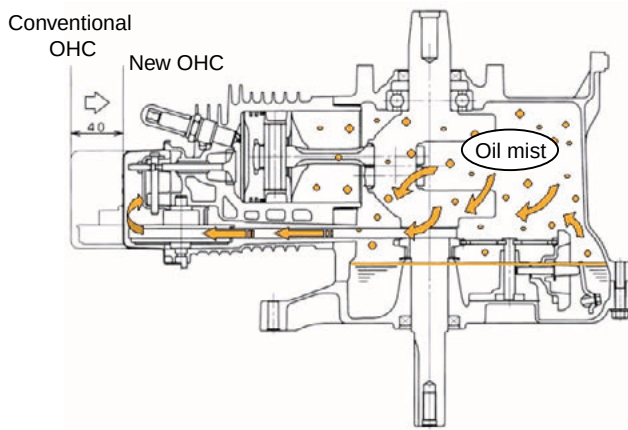


Fig.8 Lubrication system

durability. These timing belts made it possible to: (1) Greatly reduce the mechanical noise produced by the valve train and cam shaft drive train compared to regular gear-driven general purpose engines. (2) Supply oil to areas around the cam pulley and valves using an oil slinger and the rotation of the timing belts without increasing the dedicated lubricating parts such as oil pumps. (3) Replace the valve lifters, push rods and other parts with timing belts, thus reducing the number of parts and the weight.

The L-shaped pressed rocker arms decrease the dimension in the cylinder direction by approximately 40 mm compared to the conventional OHV, making it possible to keep the same dimensions as the conventional OHC. The elimination of machine tooling for the pressed rocker arms and plastic cam pulleys and completely common parts for both cylinders achieved weight and cost reductions of approximately 30% compared to conventional twin cylinder engines.

4.3. Aluminum Sintered Alloy Connecting Rod

The GCV530/520 uses aluminum sintered alloy connecting rods⁽²⁾ to achieve a size comparable to single cylinder engines and reduce vibration. These connecting rods were improved as follows compared to conventional products. (1) The fatigue strength was doubled. (2) The surface pressure limit for wear was approximately doubled by dispersing the pro-eutectic Si and optimizing the grain diameter. (3) The same high material dimensional accuracy as ferrous sintered products was ensured.

The basic twin cylinder, built-in timing belt construction causes the engine height to increase, and this height must be suppressed to achieve a comparable size to single cylinder engines. However, strength requirements made it difficult to reduce the sectional shape of the connecting rod or the width from the bearing surface to the connecting rod bearing for the die casting aluminum connecting rods used in general purpose engines.

Adopting aluminum sintered alloy connecting rods made it possible to reduce the connecting rod width from 28 mm to 19 mm, which allowed the engine height to be reduced by 18 mm. The weight of the connecting rod alone was reduced by 37%. The vibration level was reduced by 50% compared

to conventional single cylinder engines by reducing the weight of reciprocating parts and using an offset between cylinders of just 16 mm, which is the smallest in its class. Fig. 9 shows a photograph of a conventional die casting connecting rod and the sintered alloy connecting rod.

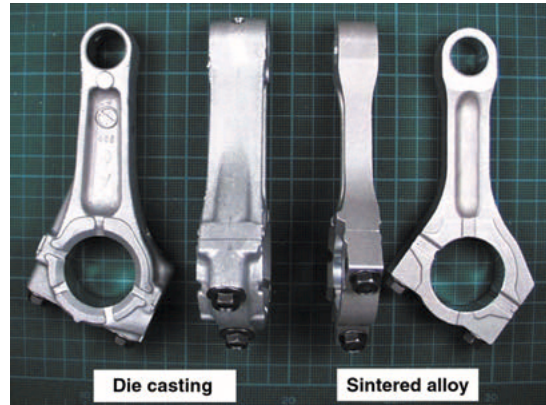


Fig.9 Comparison of connecting-rod

4.4. Turbo Shaped Cooling Fan

The normal rotating speed for air cooled general purpose engines is approximately 3,000 to 3,600 rpm, and sirocco shaped cooling fans are often used to obtain the air flow needed for cooling at this speed.

Turbo shaped cooling fans have a lower noise level than sirocco shaped cooling fans, but it is difficult to obtain a sufficient air flow with the same size fan. This development optimized the fan outer diameter, height, and the inlet and outlet angle of the vanes for turbo shaped cooling fans. Furthermore, a separator was set to increase the outlet air speed as a means of securing the required cooling air flow, and this both improved cooling performance and reduced noise. Fig. 10 shows a comparison of fan shapes, and Fig. 11 shows the degree of noise effects.

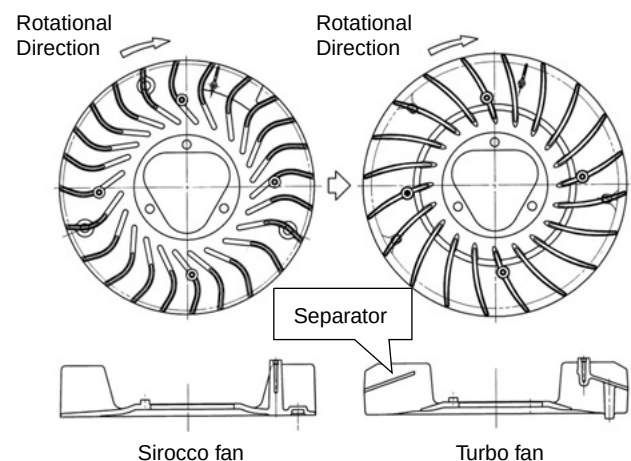


Fig.10 Comparison of cooling fan types

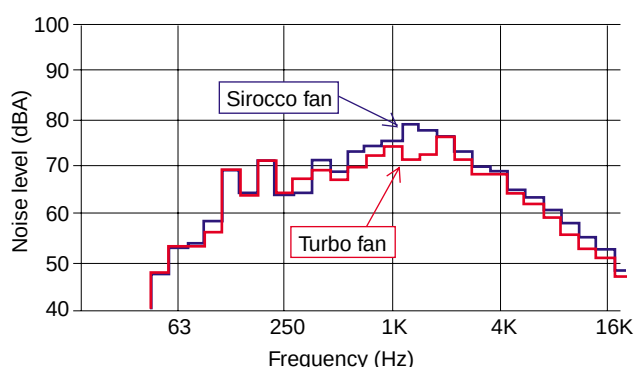


Fig.11 Noise level spectrum

4.5. Parts Constructions for Subassembly

Ease of assembly was improved by adopting the following parts constructions to allow for subassembly.

- Subassembly of the piston, connecting rod, valve train, etc. to the cylinders was made possible by using split cylinders.
- Subassembly of the intake system parts was made possible by using upward-facing inlet ports (both cylinders on the same level).

Fig. 12 shows these constructions.

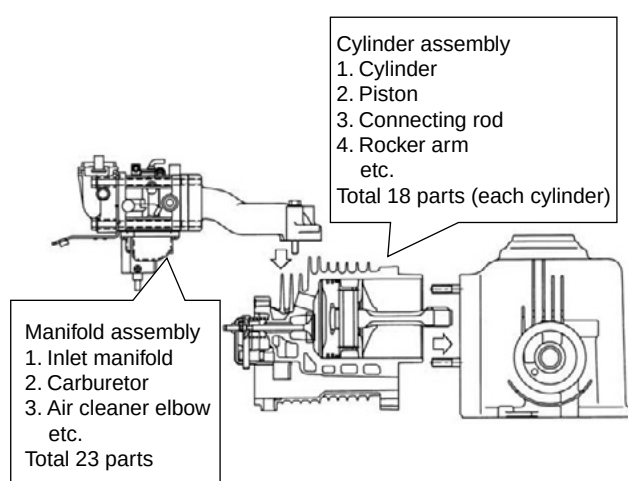


Fig.12 Parts construction of subassembly

4.6. Exhaust Emissions Reduction Technology

The following technologies were used to ensure compliance with the 2001 EPA Phase II and 2002 CARB Tier II standards.

- The spark plugs were positioned as close to the center as possible to improve combustion efficiency.
- The quenching areas in the corners of the uni-block cylinders were minimized.
- A twin bore carburetor optimizes the supply of the fuel-air mixture to each cylinder.

4.7. Effects of the Adopted Technologies

- (1) The 90° V-twin cylinder construction and aluminum sintered alloy connecting rods achieved a 50% reduction in the vibration level compared to conventional single cylinder engines as shown in Fig. 13.
- (2) The built-in timing belt OHC construction, plastic cam pulleys, turbo shaped cooling fan and other improvements achieved a 4 dB reduction in the noise level compared to conventional single cylinder engines as shown in Fig. 14.
- (3) Overall, the adopted technologies achieved a weight reduction of approximately 20% over conventional twin cylinder engines.

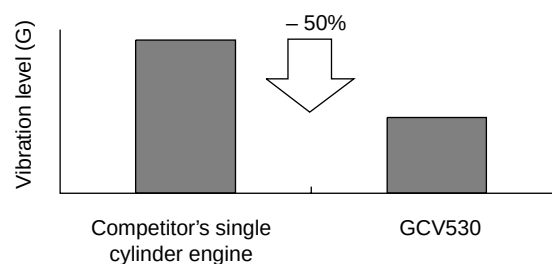


Fig.13 Comparison of vibration

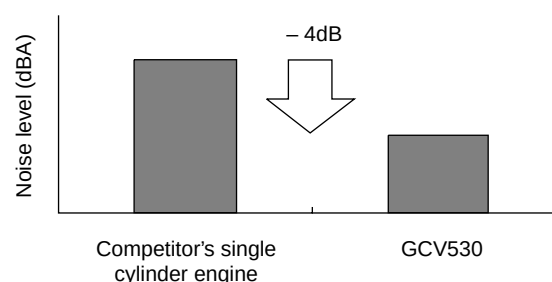


Fig.14 Comparison of noise

5. Conclusions

The 90° V-twin cylinder engines GCV530/520 for riding mowers were developed for a market that is dominated by single cylinder engines.

- (1) Ample power, low vibration and low noise were achieved by combining a displacement setting of 530 cc with various new technologies.
- (2) Interchangeability with single cylinder engines was ensured by a left-right asymmetrical cylinder layout, and the engine was given strong specifications and price competitiveness.
- (3) Compliance with the 2001 EPA Phase II and 2002 CARB Tier II standards was assured.
- (4) Home use and professional use specifications were set for the GCV530 and GCV520, respectively, to support a wide range of needs.

Honda is confident that these engines will be accepted in the riding mower market centered in the US and Europe. The authors wish to express their deep thanks to all parties

involved for their immense cooperation in developing these engines.

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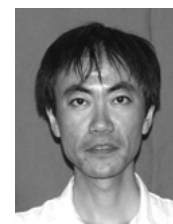
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Development of a 1.5L Lean-burn Engine for the 2001 Civic

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ABSTRACT

A 1.5L SOHC-VTEC lean burn engine designed for the 2001 model Civic achieves higher fuel economy (20km/liter based on the 10-15 mode) and an emissions reduction of 50% over the Japan 2000 emission standards. Engine friction is reduced and a newly designed intake-port and combustion chamber are utilized for improved combustion characteristics. In addition, a NOx adsorption reduction type three way catalyst is used with the new exhaust system to achieve low-emissions.

1. Introduction

Protecting the global environment and conserving natural resources have increased the demand for higher fuel economy automobiles with reduced exhaust emissions. There are a variety of gasoline engines, in addition to the lean-burn engine, such as the direct injection and Miller cycle types, that have been researched and commercialized. Honda is continuing the development of lean-burn engines that apply the VTEC mechanism (VTEC: variable valve timing and lift electronic control) such as the 1.5 L VTEC-E lean-burn engine⁽¹⁾ in 1992 for the Civic, the 1.5 L 3- stage (VTEC) lean-burn engine⁽²⁾ that was combined with continuously variable transmission (CVT) in 1996, and the 1.8 L VTEC lean-burn engine⁽³⁾ for the Accord in 1998. This technical review introduces the development of a new power train incorporating a further modified 1.5L SOHC – VTEC lean-burn engine with a CVT for the new Civic model.

2. Aims of the Development

The basic configuration of the power train was reviewed in order to realize the developmental concepts of the new Civic model, namely “creating new values through innovative packaging,” “designing for safety and environmental compatibility” and “suitability for use almost anywhere in the world”. Based on a compact design to match the packaging of the chassis, this research aimed at improved fuel economy by better engine combustion, reduced engine friction and improved CVT efficiency, and at lower emissions by adopting a lean NOx catalyst and improved exhaust system.

The detailed objectives for the development were as

follows:

- (1) The best fuel economy in the domestic popular car class.
- (2) Low-emission performance that conforms to global standards (potential compatibility with the U.S. Ultra-low Emission Vehicle (ULEV) standard).
- (3) Engine Vibration and Noise Reduction.
- (4) 10% smaller engine compared to the current engine.
- (5) Better response under normal driving conditions.

3. Outline and Main Specification of the New Engine Design

Fig. 1 shows an external view of the engine and Table 1 describes the main engine specification.

The more compact design, involving position changes to several external components, involves a re-designed cylinder block and a substantial change in the layout of the induction system, including the use of a resin intake manifold. As can be seen in the overhead view of the engine (Fig. 2), the air cleaner and intake resonator chambers are both located above the engine and transmission, giving a more compact design for the entire power train and achieving a modular concept.

Almost all of the engine parts had to be re-designed to achieve the optimum combustion chamber shape, including the crown shape of the piston to reduce the friction of each component, and to further reduce emissions.

Fig. 3 shows the engine performance curves. Importance was placed on better low-speed torque to match the CVT characteristics. Compatibility has been achieved between high fuel economy and the feeling of linear drive in normal use.

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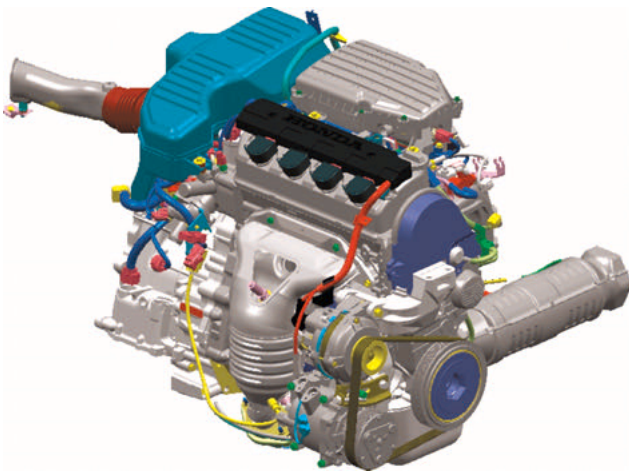
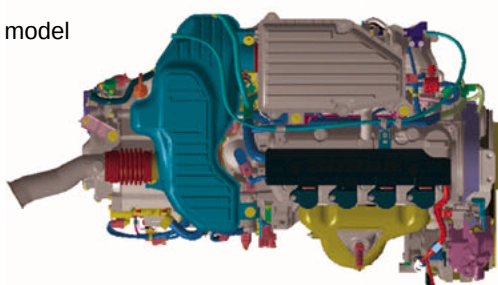


Fig.1 View of 2001 model lean-burn engine

Table 1 Engine specification

Engine code	D15B	
Cylinder configuration	In-line 4-cylinder	
Bore × stroke (mm)	75 × 84.5	
Displacement (cm ³)	1493	
Compression ratio	10.2 : 1	
Valve train	SOHC VTEC	
Number of valves	4 per cylinder	
Valve diameter (mm)	In.	30
	Ex.	26
Fuel supply system	Programed fuel injection	
Ignition system	Programed ignition	
Max. power (kW/rpm)	77 / 5800	
Max. torque (N·m/rpm)	139 / 4200	

2001 model



1999 model

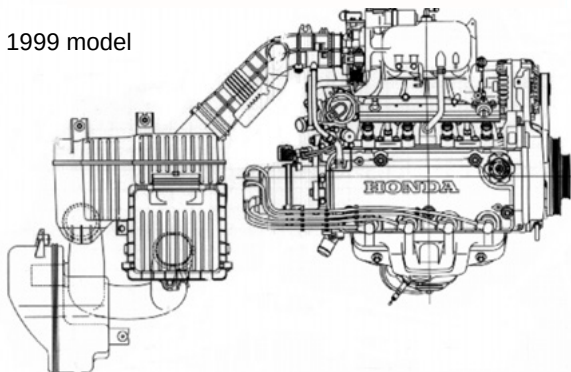


Fig.2 Overhead view of engine

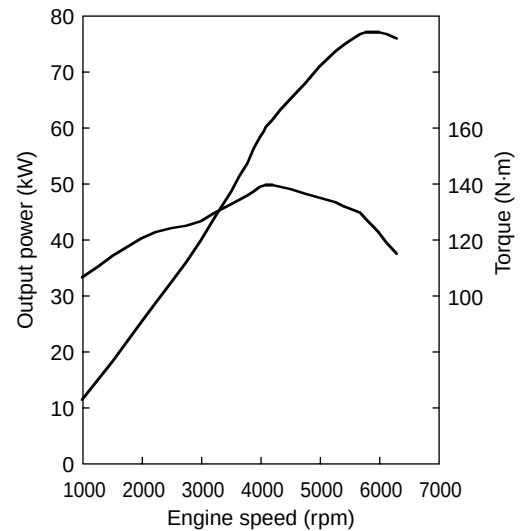


Fig.3 Engine performance

4. Low Fuel Consumption Technology

Higher targets were set for the engine, CVT and chassis in order to develop the technology required for high fuel economy (Fig. 4). As a result, we have achieved a fuel economy performance of 20 km/L in the 10-15 mode.

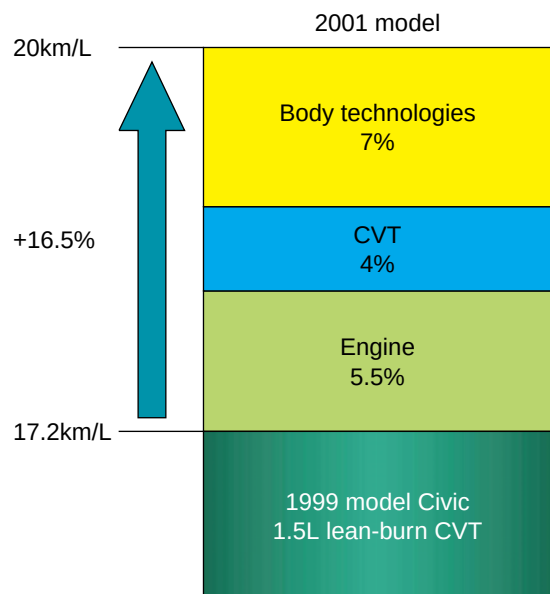


Fig.4 Ratio of contributions for improved 10-15 mode fuel economy (4door)

4.1. Fuel Economy Mode Positioning

Fig. 5 shows the relationship between the vehicle weight and domestic 10-15 mode fuel economy. In comparison to the current vehicles with a stoichiometric air-fuel ratio engine, vehicles with a lean-burn engine have approximately 10% better fuel economy. Direct injection (DI) provides a further 8% improvement in fuel economy for lean-burn engine vehicles, indicating the advantage of direct injection.

The newly developed lean-burn engine for the Civic offers approximately 8 - 12% better fuel economy than that of a DI engine vehicle.

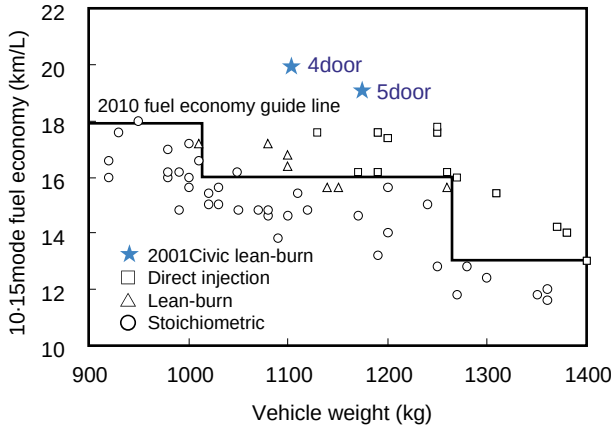


Fig.5 Relationship between vehicle weight and 10- 15 mode fuel economy in production gasoline vehicles

4.2. Methods for Improving the Engine Fuel Economy

Fig. 6 shows the relationship between the total displacement and brake specific fuel consumption (BSFC). A BSFC reduction of 8%, with regard to current engine types, with stoichiometric combustion was attained with this engine. With lean burn combustion, another 10% BSFC reduction was attained for a total BSFC reduction of 18%. This value is comparable to the Insight engine when differences in displacement are taken into consideration⁽⁴⁾⁽⁵⁾.

The improved fuel economy of this engine involved modification of the lean-burn characteristics and significant reductions in engine friction. The lean burn characteristics have been modified by a new swirl port, which further enhances the turbulence of the air-fuel mixture inside the cylinder, and the use of a more compact combustion chamber, which improves the thermal efficiency. Engine friction has been reduced by a comprehensive review of the

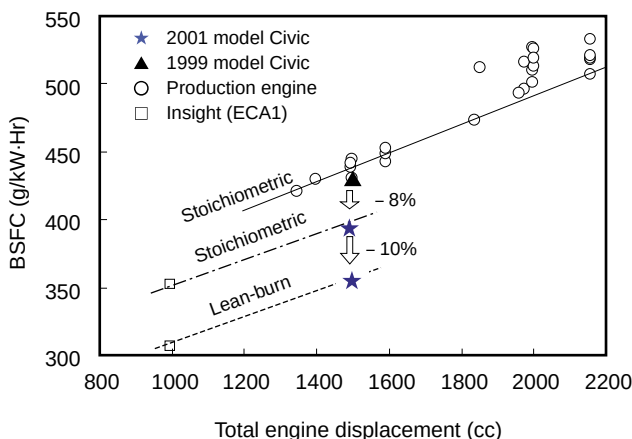


Fig.6 Improvement of engine thermal efficiency

shape of all components in the engine, changes in materials, and modification of the forming processes.

The next section discusses the technological elements applied to achieve the improved fuel economy.

4.2.1. Shapes of the Intake Port and Combustion Chamber

To stabilize the burn of a lean mixture flowing into the cylinder, it is necessary to form an easily ignitable air-fuel mixture around the spark plug with a flame front that quickly propagates throughout the combustion chambers. It is important to have a good balance between the swirl (horizontal vortex) and tumble (vertical vortex) inside the combustion chambers to achieve this combustion. The newly developed intake port enhances the tumble components of turbulence to achieve a swirl ratio of 2.6, tumble ratio of 2.0 and swirl inclination angle⁽⁵⁾ of 37 degrees (Fig.7). This swirl inclination angle shows good correlation with the lean-burn limit in the test results of the 1.0L to 1.8L lean-burn engines and has provided the authors with an effective index for designing the shape of the intake port.

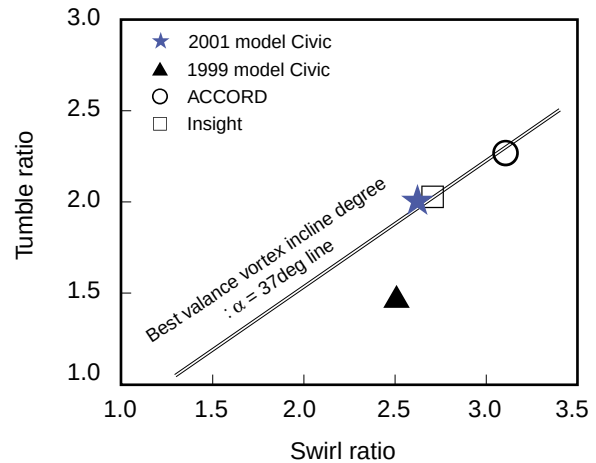


Fig.7 Relation between swirl and tumble intensity

Another important factor in lean-mixture combustion is the distribution of the air-fuel mixture inside the combustion chamber. This distribution was studied by simulating the shape of the intake port under the following conditions:

Average air-fuel ratio $A/F = 22$ (excess air ratio: 1.5)

Fuel injection timing: 40 degrees after TDC of the intake stroke.

The simulation showed the horizontal sectional view and vertical sectional view at the position of the electrode in the center of the spark plug. Fig. 8 shows the distribution of the air-fuel mixture in the combustion chamber 40 degrees before the top dead center of the compression stroke equivalent to the ignition timing of an actual engine. The new port produces a richer air-fuel mixture near the electrode in the center of the plug, than the current port. A good correlation was confirmed with the test results (Fig. 9) for the improved lean-burn limit.

The same simulations and tests were performed to identify the optimum combustion chamber shape. The results were used to select the bowl-in-crown shape of the piston and the squish area.

Ne = 1500rpm
 P intake = 48kPa
 A/F (averaged) = 22
 Crank angle = 40deg BTDC.

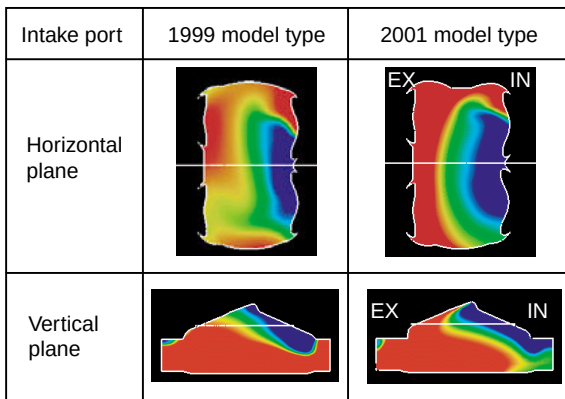


Fig.8 Effect of intake port shape on in-cylinder A/F distribution

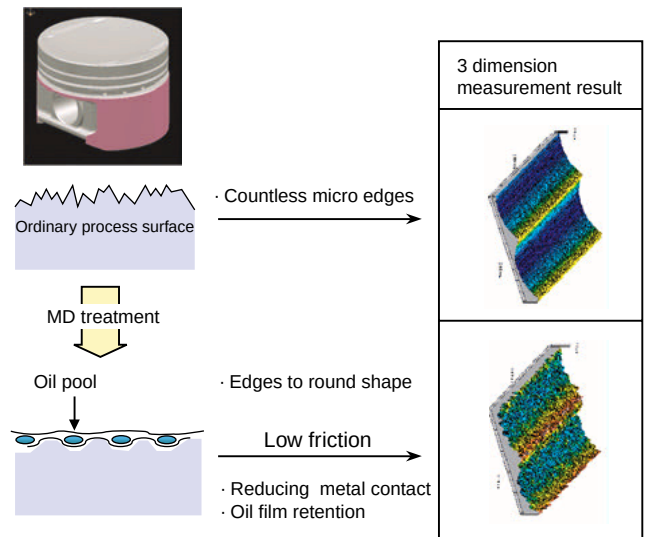


Fig.10 Face treatment of piston skirt portion

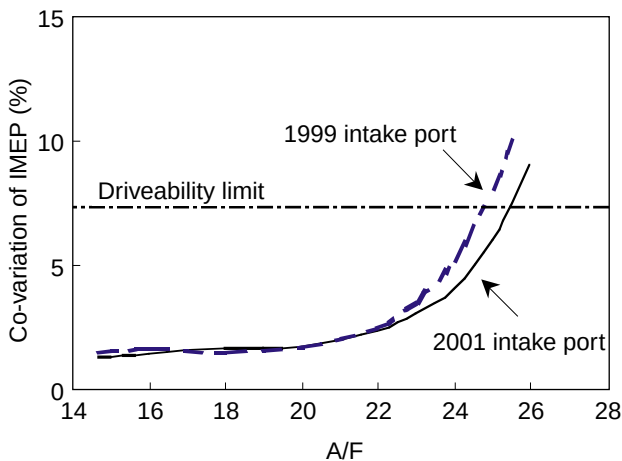


Fig.9 Comparison of lean-limit characteristics based on intake port shape

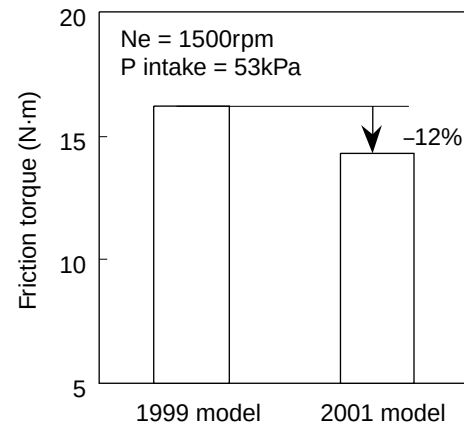


Fig.11 Friction loss comparison

4.2.2. Reduction of Friction

Together with the improved combustion characteristics, it was also important to reduce engine friction to achieve improved fuel economy.

The following modifications were taken to reduce engine friction:

- Micro dimple (MD) processing⁽⁶⁾ of the pistons
- Low-tension piston rings
- Reduced crank pin width
- Low-viscosity engine oil (0W-20)
- Roller cam followers

Fig.10 describes the MD processing of the piston skirt. This processing uses basically round ceramic beads with an average granule diameter of 40 mm to impact the surface of the piston skirt at a speed of 100 m/s or more to create microscopic indentations, or dimples. These enhance the performance of the oil film retention between the piston and

the cylinder to reduce piston wear, decrease piston friction loss and increase the load for seizure.

In addition to this low-friction treatment of the pistons, technologies for valve spring load reductions, piston ring tension reduction, crankshaft pin width reduction, and low-viscosity engine oil⁽⁷⁾ (0W-20/SJ/GF2) were adopted.

The friction loss of the engine is shown in Fig. 11, indicating a reduction of 12% compared with the current engine.

4.3. CVT

CVT enables the engine to be operated within the range of low fuel consumption (BSFC) to improve the fuel economy. Honda has been mass-producing CVT for the Civic since 1996. Enhanced hydraulic control of the CVT and modified belt side pressure resulted in improved transmission efficiency of the power train.

Fig. 12 shows the running characteristics of the improved CVT in comparison with those of the 4AT. The engine speed was controlled to maintain the operating envelope in the low-speed zone and achieve a low BSFC.

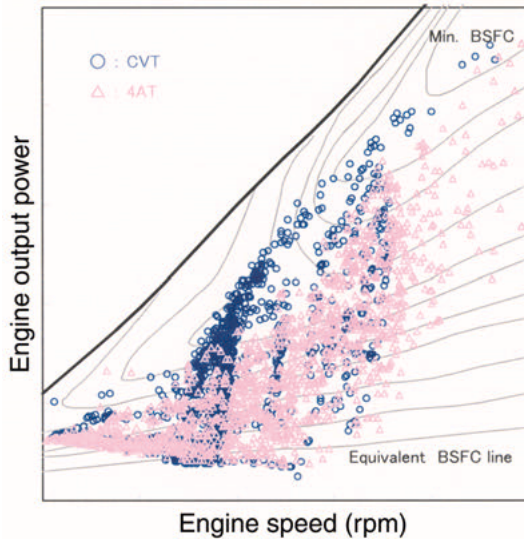


Fig.12 Comparison of the CVT and 4AT in 10- 15 mode driving range

5. Low Emission Technology

The three-way catalyst (TWC) unit could not be expected to reduce NO_x in the lean-burn condition involving excess-air combustion, but we nevertheless had to achieve low emissions.

A target figure of 50% of the 2000 emission standard (certified as a superior low-emission exhaust) was selected for the domestic sales model. At the same time, studies were conducted for each specification based on the premise that the component would have the potential to clear the US ULEV standard, which is considered the global standard.

5.1. Exhaust System

The exhaust system of this engine is shown in Fig.13. A three-way catalyst is used for exhaust conversion in cold and stoichiometric air-fuel ratio ranges. This TWC unit is close coupled to the exhaust manifold, and a NO_x absorption reduction type three-way catalyst⁽⁸⁾ is situated under the floor for NO_x reduction in the lean-burn range.

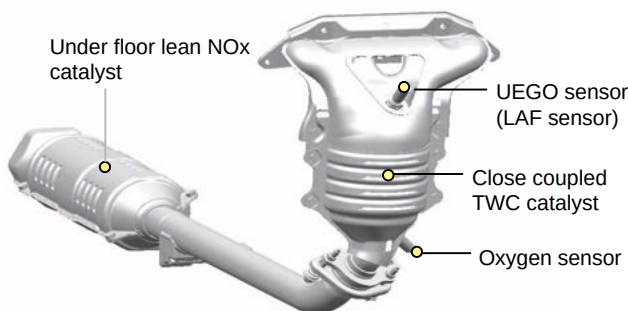


Fig.13 Layout of NO_x adsorption reduction type three way catalyst system

5.2. Multi-hole Injector

The 4-hole injector used in this engine is shown in Fig. 14. The use of this injector has helped to achieve a lighter and more compact injector with 33% of the length and 65% of the weight of the current injector. Also, 4-hole type injection pores were used instead of the pintle types on current engines. Better atomization of the spray and reduced gasoline coating on the air intake port were achieved so that hydrocarbon emission could be reduced by 14% at cold starts (LA-4 cold transient mode) as shown in Fig.15.

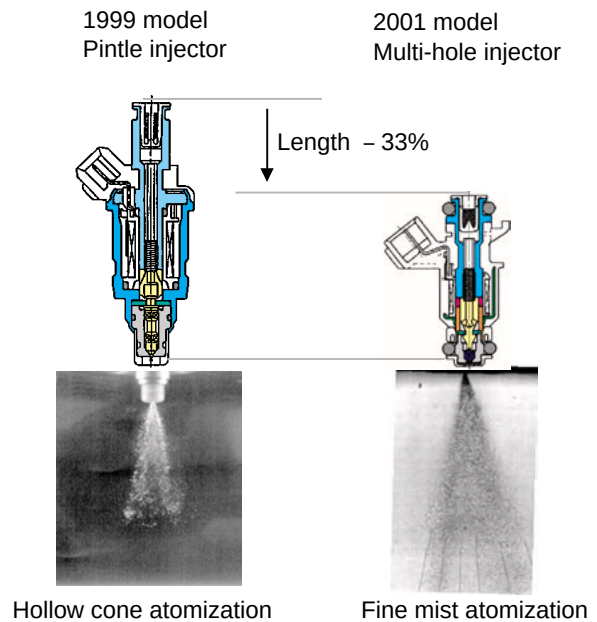


Fig.14 Small fuel injector delivering efficient atomization

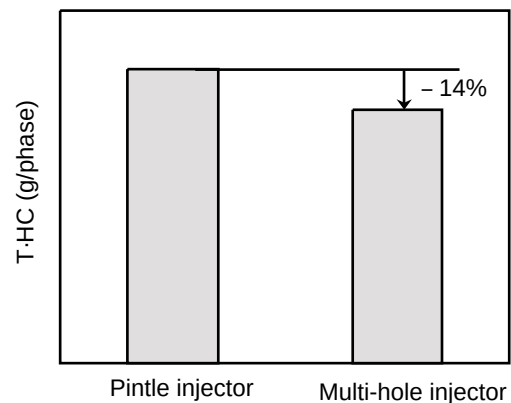


Fig.15 LA-4 cold transient mode emission

5.3. Achieved Emissions

In addition to the reduction of emissions just described, engine control was optimized when shifting between lean-burn combustion and stoichiometric air-fuel ratio ranges, and in the EGR range. A fuel consumption of 20 km/L in the 10-15 mode and an emission level of 50% of the 2000 emission standard have been achieved. (Fig. 16)

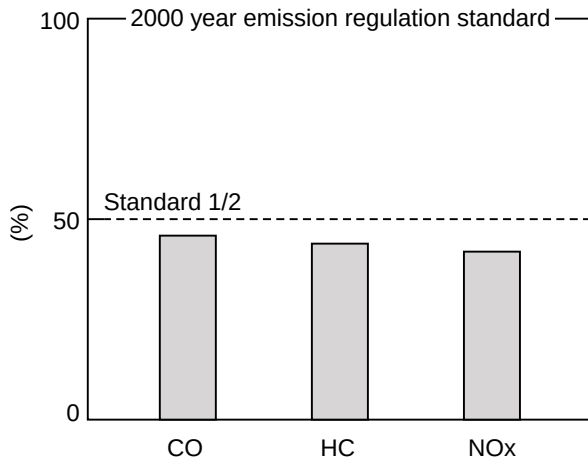


Fig.16 11 mode emission

6. Engine Vibration and Noise Reduction

Improving the quietness of the engine was one of the main development objectives, in addition to improved fuel economy and reduced emissions. The main aims were to reduce the engine noise when accelerating and at high engine speeds. A highly-rigid engine side mount bracket is used, the crankshaft rigidity is improved, and intake rumble noise is reduced by employing equal length intake manifold pipes.

A comparison of the radiated engine noise is shown in Fig. 17. A reduction in noise of 2 to 3 dB in the high speed operation range was attained.

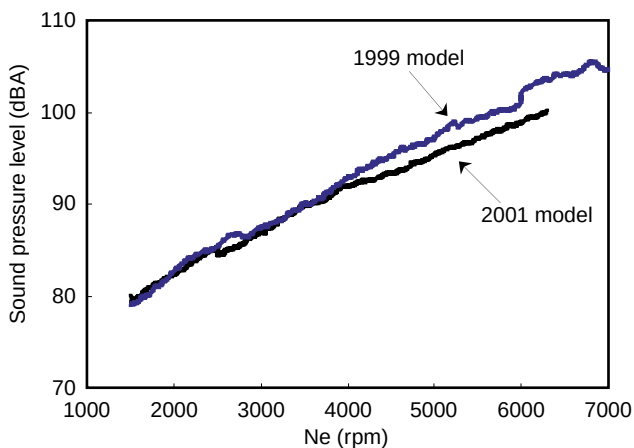


Fig.17 Reduction of engine radiation noise

6.1. Engine Side Mount Bracket

Fig. 18 shows a comparison between the new and old engine side-mounting brackets. Material for this bracket is changed from cast iron to aluminum, and it is integrated with the ACG stay to attain both high rigidity and weight reduction. This reduces vibration transmission from the engine to the chassis.

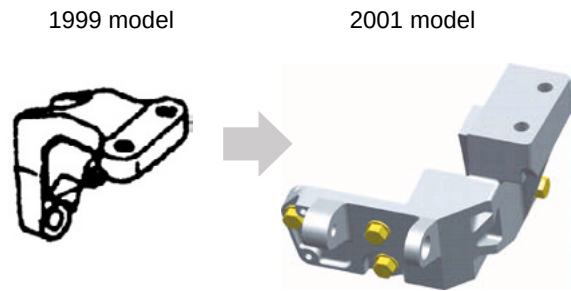


Fig.18 Improving the rigidity of the engine side mount bracket

6.2. Crankshaft

As shown in Fig. 19, the width of the crankshaft pin has been reduced from 23 mm to 20 mm to enable the crankshaft web to be increased by 1.5 mm. This increased the overall rigidity of the crankshaft to reduce the rumble noise. Furthermore, the narrower width of the crankshaft pin has reduced the friction of the big-end bearings.

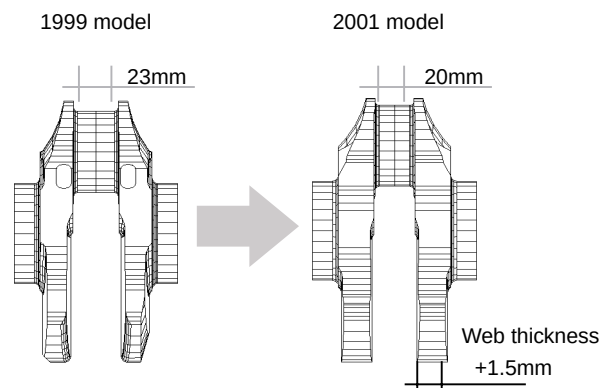


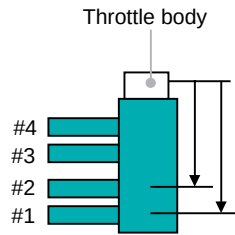
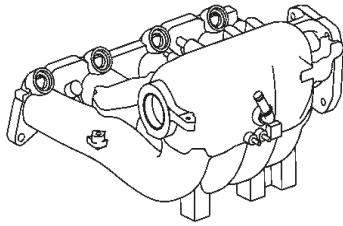
Fig.19 Improving the rigidity of the crank shaft

6.3. Intake Manifold

One of the sources of the rumble noise in the induction system is caused by a half-order harmonic from the intake port. This occurs because the lengths from the throttle body to the intake open ends of individual cylinders are not equal. The material used for the intake manifold on this engine is changed from aluminum to resin, and the greater forming freedom for this material has enabled each intake pipe to have equal length (Fig. 20).

Fig. 21 shows the sound spectrum of the induction noise from the engine's intake ports. It is clear from a comparison between the new and old models that the half-order harmonic has been substantially reduced resulting in reduction of the rumble noise.

1999 model



2001 model

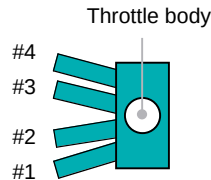
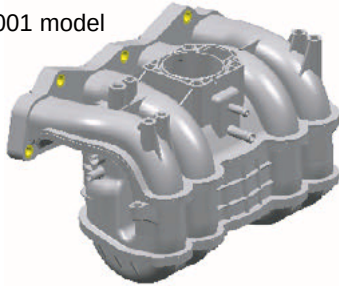
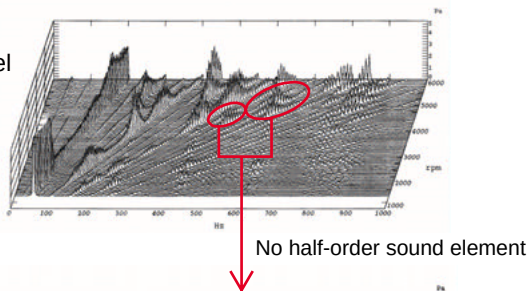


Fig.20 Adopting pipes of equal length

1999 model



2001 model

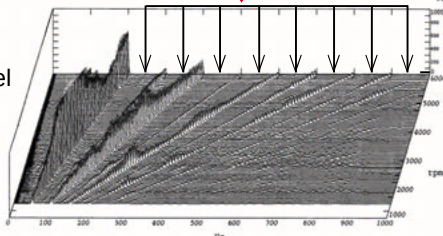


Fig.21 No air-sucking-induced rumbling noise

7. Conclusion

The enhanced thermal efficiency of the new D15B lean-burn engine achieved by using improved combustion and reduced friction, enhanced CVT transmission efficiency, and the improved aerodynamics and rolling resistance of the body have led to the achievement of a fuel consumption of 20 km/L in the 10-15 mode. The use of a combined NO_x adsorption reduction type three-way catalyst system that compliments the lean-burn exhaust has enabled the low emission level of 50% of the Japan 2000 emission standard (certified as a superior low-emission exhaust) to be achieved.

8. Postscript

The lean-burn system developed for this engine is used on the 1.7L model being sold in the United States. The high fuel economy from the lean-burn system conforms to the ULEV standard, so that the original aim of a “low-emission system conforming to global standards” has been achieved.

This new power train system is one answer to the serious issue of compatibility between low emissions and fuel economy which will become ever more important in the 21st century. More modifications will be made in the future to develop a highly efficient power train system that can meet the demands of the time.

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Super Ultra Low Emission System for the Accord SULEV

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ABSTRACT

We developed the Accord SULEV (Super Ultra Low Emissions Vehicle) in order to comply with California's LEV-II SULEV regulations. The emissions system employed is characterized by applying only an under-floor catalyst to attain the super ultra low emissions levels through sophisticated control technology. The improved three-way catalyst achieves both high conversion efficiency and low back pressure while using the same amount of precious metal as was used in the Accord ULEV. ⁽¹⁾

To satisfy SULEV standards using only the under floor catalyst, we developed technologies to increase emissions temperatures by controlling the amount of intake air and the ignition timing before the catalyst is active. This promoted quick catalyst light-off and enabled us to obtain leaner air/fuel ratios when the engine is started than was possible with the previous catalyst system.

Also, in order to further improve the conversion performance of the catalyst once it is active, we developed a New Secondary O₂ Feedback Control System. This is an extremely high precision system that identifies the dynamic changes of the catalyst in real time and utilizes a sliding mode controller mechanism to predict changes in the post-catalyst sensor output.

The implementation of these technologies makes it possible to achieve the most rigorous SULEV standards, with engine output performance equivalent to our ULEV vehicle.

1. Introduction

Worsening air pollution is making it increasingly important to address environmental issues and to find solutions. Automobile emissions are no exception; governments around the world are enacting ever more stringent emission regulations, and the state of California will be enacting the new and more rigorous LEV-II regulations in 2004.

Table 1 indicates emission standards by category as set forth under the LEV-II regulations. Even durability requirements have been revised under the LEV-II regulations so that emissions through 120,000 miles will be regulated.

Fig. 1 indicates the emission standards for NMOG (Non

Methane Organic Gas) under the LEV-I and LEV-II regulations. The new SULEV category has been added in the LEV-II regulations. The SULEV emission standards are extremely strict, being less than 1/5 of the standards under the LEV-I ULEV (Ultra Low Emission Vehicle) regulations, and have been expanded to include emission standards from 50,000 miles up to 120,000 miles. Fig. 2 indicates the NOx emission standards under the LEV-I and LEV-II regulations. Under LEV-II, the NOx standards are more strict, with the

Table 1 LEV-II emission standards (g/mile)

Durability	Category	NMOG	CO	NOx	HCHO
50 kmile	LEV	0.075	3.4	0.05	0.015
	ULEV	0.040	1.7	0.05	0.008
120 kmile	LEV	0.090	4.2	0.07	0.018
	ULEV	0.055	2.1	0.07	0.011
	SULEV	0.010	1.0	0.02	0.004

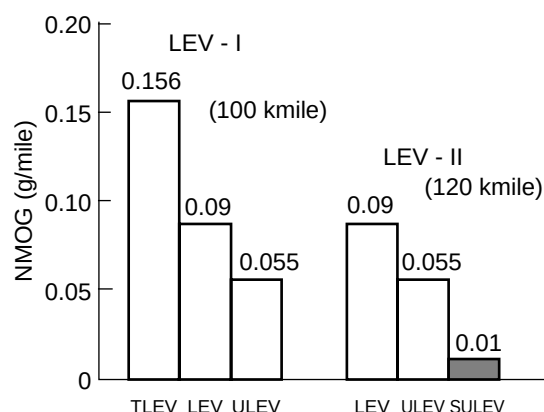


Fig.1 LEV-I and LEV-II NMOG regulations

* Tochigi R&D Center

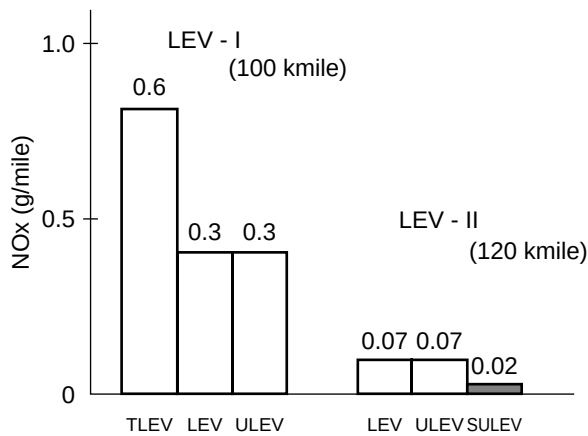


Fig.2 LEV-I and LEV-II NOx regulations

SULEV standard calling for emissions that are less than 1/10 of the standard allowed under LEV-I ULEV.

Honda has developed a Super Ultra Low Emission Vehicle, based on the Accord ULEV, that satisfies the SULEV regulations and has the lowest emissions of gasoline-powered production vehicles in the world. This paper describes the emission system in the Accord SULEV.

2. Technical Concepts Behind the SULEV

As we undertook the development of the SULEV technology, we re-examined the emission performance of the Accord ULEV that is currently being mass produced by Honda, and decided on clear targets for reducing emissions.

Fig. 3 shows the measured emissions from an Accord ULEV, with the FTP (Federal Test Procedure) cycle broken down into three periods. Period A extends from engine startup to the start of driving, period B extends from the end of period A until the engine stops at the end of the LA-4 driving cycle, and period C extends from restart after the hot soak until the end of the test.

In period A, the measured emissions of NMHC (Non Methane Hydrocarbon) and NOx were both twice the level allowed under the SULEV regulations. This period encompasses a cold start and warm-up of the catalyst. Since the catalyst is not fully active during this period, a technology that reduces the engine emissions during this time and that rapidly raises the catalyst temperature for early light-off are both needed. In period B, NMHC emissions were below the SULEV standard, but NOx emissions were at roughly the same level as the SULEV standard. It became clear that in order to satisfy the SULEV regulations, it would be necessary to reduce NMHC and NOx emissions not only in period A but also in period B. Since the catalyst is fully active in this period, the problem at hand was to develop new control technology that would further increase the catalyst conversion efficiency.

The engine is already warmed up in period C, so any technology that can reduce emissions in periods A and B would also be able to reduce emissions in period C.

Although there are a variety of approaches⁽²⁾ available for solving the problem at hand, "simplicity" had been stipulated as one of the basic principles behind the

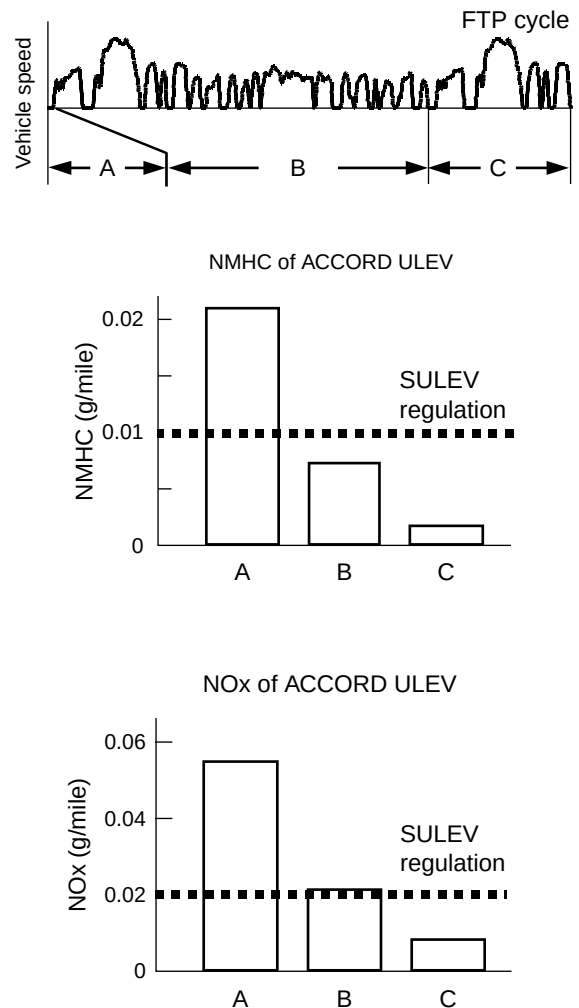


Fig.3 ACCORD ULEV Emissions at A-C period

development of the Accord SULEV. This emphasis on simplicity as a system concept had been carried over from the development of the Accord ULEV, a gasoline-powered car designed by Honda to satisfy California's LEV-I ULEV regulations of September 1997. The Accord ULEV system configuration manifested this simplicity, since the system was built around VTEC-E and a high-precision air-fuel ratio control system, and it utilized only an underfloor three-way catalyst.

In order to meet the stringent SULEV emission regulations, we considered means such as using a close-coupled catalyst or hydrocarbon trap, but there was concern that such means would raise costs and reduce engine output. By utilizing only an underfloor three-way catalyst that could achieve early light-off and provide high conversion efficiency as the result of a high cell density and thin walls, and by developing control technology that could obtain maximum performance from the catalyst, we were able to achieve a low-cost system with low back pressure.

Fig. 4 shows the overall configuration of the system designed to meet the SULEV regulations. The current Accord ULEV system comprises components (a) through (h).

The new technologies that were adopted in the Accord

SULEV system are as follows:

- A catalyst with high cell density and thin walls (1200-cell, 2.0-mil, the precious metal content being equivalent to that of the catalyst used in the Accord ULEV)
- A quick catalyst warm-up control system to achieve early light-off of the catalyst
- Improved AAI (Air Assisted Injector) that boosts the assist air pressure
- Lean burn control after startup that produces an air/fuel ratio that is even leaner than that achieved in the Accord ULEV
- New secondary oxygen feedback control (PRISM: Prediction and Identification type Sliding Mode Control) that maximizes the conversion performance of the catalyst

3. Three-way Catalyst with High Cell Density and Thinner Walls

It is already known that increasing the cell density and reducing the wall thickness effectively improves the conversion performance of a catalyst.

Fig. 5 shows the relationships between the cell density of the newly developed 1200-cell, 2.0-mil catalyst and the surface area and bulk density. Compared with the 600-cell, 4.3-mil catalyst used in the ULEV, the newly developed catalyst has 50% more surface area which provides improved conversion performance.

In addition, the bulk density has been decreased by 35%, which results in improved catalyst light-off performance.

Fig. 6 shows two types of canning construction. The conventional method uses clamshell-type canning whereby the catalyst and holding seal mat are enclosed within an upper and lower case. This construction has a fixed inside case

diameter, so any variation in the diameter of the catalyst and holding seal mat can lead to a significant surface pressure difference being generated during the canning process. Therefore, a catalyst with an isostatic strength of 15kg/cm² or more is required for this type of canning.

The new rolling technology employs a rolled construction by which the catalyst and holding seal mat are fixed after being rolled into the case. This technology has made it

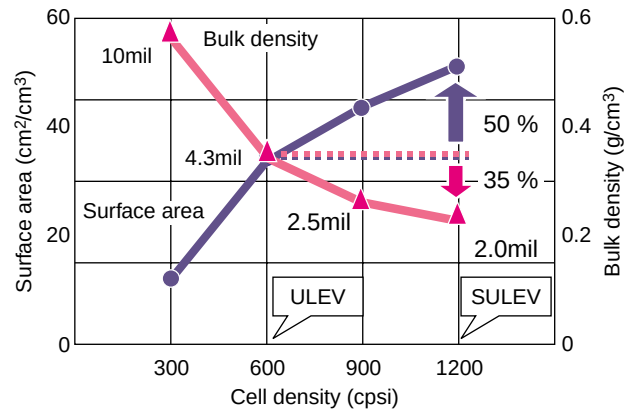


Fig.5 Cell density, bulk density and surface area

Clam shell type canning

Rolling type canning

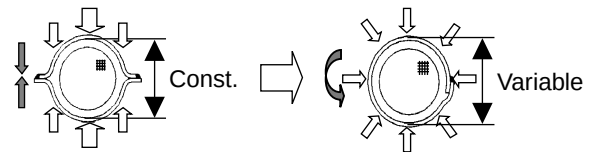


Fig.6 Canning type

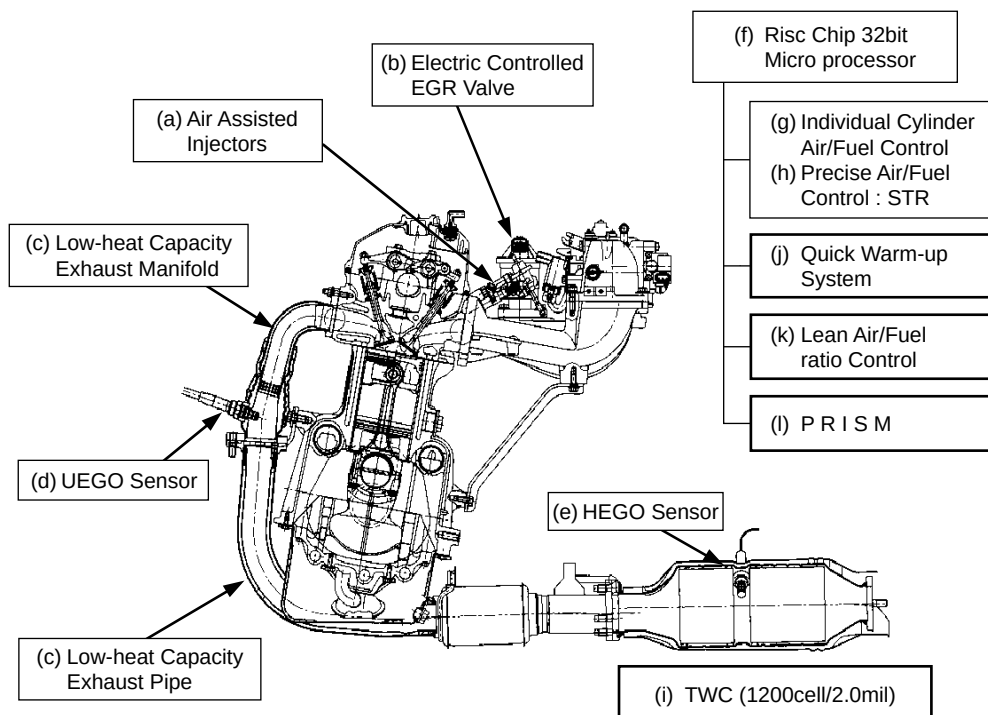


Fig.4 Overview of SULEV System

possible for the surface pressure generated during canning to be evenly distributed around the circumference of the case. It has also made it possible to vary the internal case diameter to match variations in the catalyst diameter. In addition to this rolled-type canning method, selection of the holding seal mat and improvements to the production method have made it possible to reduce the surface pressure generated during canning to less than 5kgm/cm^2 , making canning possible even with a 1200-cell, 2.0-mil catalyst that has low isostatic strength.

While there was some concern about a lower engine output due to greater exhaust pressure loss caused by the increased number of cells, we were able to minimize the decrease in engine output by making the catalyst walls thinner.

Fig. 7 shows the engine output characteristics for the Accord ULEV and the Accord SULEV. The power and torque produced by the SULEV at below 4000rpm are both equal to those produced by the ULEV, indicating no adverse effects from the 1200-cell catalyst, while at maximum output, the decrease in power is only 2kW and the decrease in torque is only 5Nm.

Fig. 8 shows the measured emissions in the LA-4 mode when a 600-cell, 4.3-mil catalyst (aged 100,000 miles) and a 1200-cell, 2.0-mil catalyst (aged 120,000 miles) were installed in the same vehicle. The higher cell density and thinner walls produced a 31% decrease in NMOG emissions and a 48% decrease in NOx emissions. Although a palladium catalyst (Pd, 330g/ft³) is used in the mass-produced version of the ULEV, a tri-metal catalyst (Pt/Pd/Rh = 1:15:1, 330g/ft³) was adopted for the SULEV in order to address the stricter NOx standards.

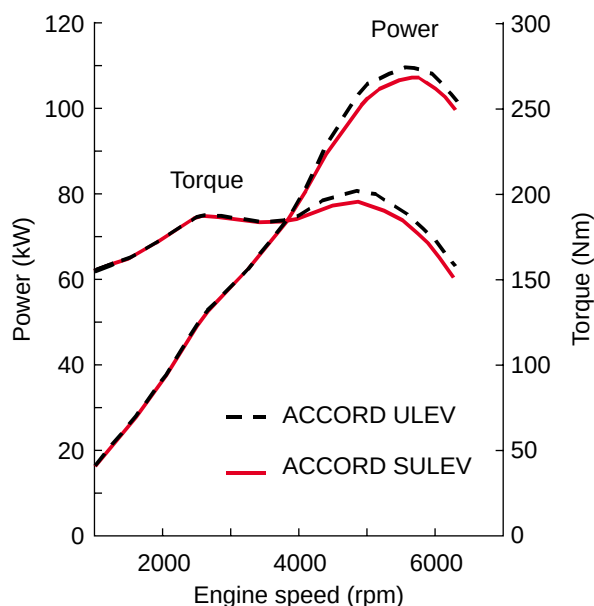


Fig.7 Characteristics of power and torque

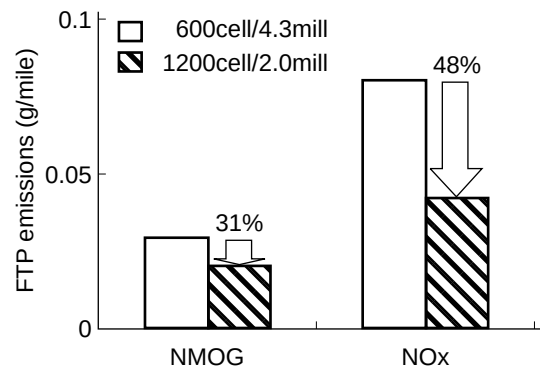


Fig.8 Comparison of emissions between 600 cell and 1200 cell catalysts (FTP cycle)

4. Technologies for Reducing Emissions at Startup and Immediately After Startup

Although a catalyst with a higher cell density and thinner walls made possible a major reduction in emissions, it was not enough by itself to satisfy the SULEV regulations. Therefore, we further reduced the emissions through engine control by achieving earlier catalyst light-off at startup and improving the lean-burn characteristics after startup.

4.1. Quick Warm-up System

The temperature of the exhaust gas begins to rise immediately after the engine is started; a three-way catalyst absorbs energy from the exhaust and thus also heats up becoming more active. It is well-known that retarding the ignition timing is an effective means for increasing the temperature of the catalyst. We created a system (the "Quick Warm-up System"⁽³⁾) that, immediately after a cold start (when the three-way catalyst is not active), uses an RACV (Rotary Air Control Valve) to control the air intake, while simultaneously controlling the ignition timing.

Fig. 9 shows the characteristics of this control system. The horizontal axes represent time, and the vertical axes

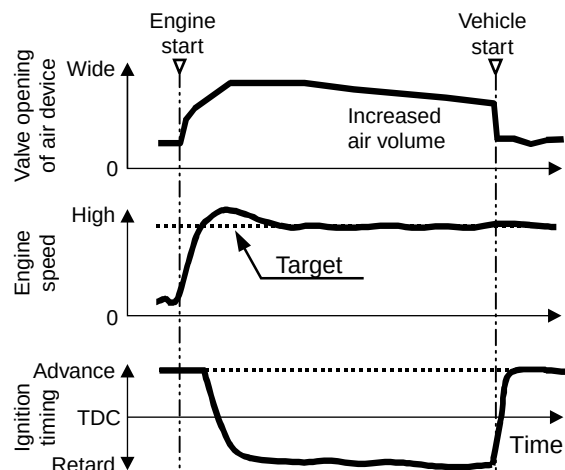


Fig.9 Ignition feedback control for Quick Warm-up System

represent the RACV opening, engine speed, and ignition timing, respectively, from the top. While the engine is idling after startup, the Quick Warm-up System uses the RACV (by means of feed-forward control) to supply a large quantity of intake air. When the engine speed exceeds a preset level, PI (proportional + integral) feedback control of the ignition timing begins. Here, the ignition timing is controlled to maintain the target engine speed. As indicated in the diagram, as the intake air is increased, the ignition timing is retarded in order to maintain the target engine speed. Fig. 10 shows a comparison of the ignition timing and exhaust gas temperature with the Quick Warm-up System used in the SULEV and those with the ULEV system. Twenty seconds after startup, the exhaust gas temperature is 240°C higher with the SULEV system than with the ULEV system. This makes rapid light-off of the catalyst possible.

4.2. Improved Lean-burn Control after Startup

Fig. 11 shows the flow of intake air into this engine with the VTEC-E system. A swirl is generated by opening only one inlet valve, making possible lean control of the air/

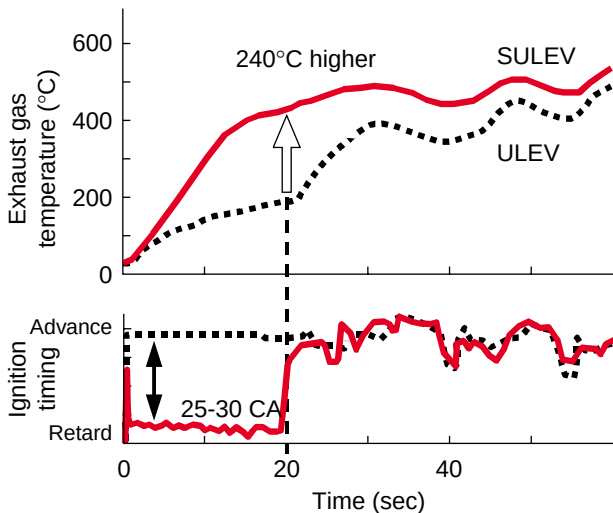


Fig.10 Effect of Quick Warm-up System at start-up period of FTP cycle

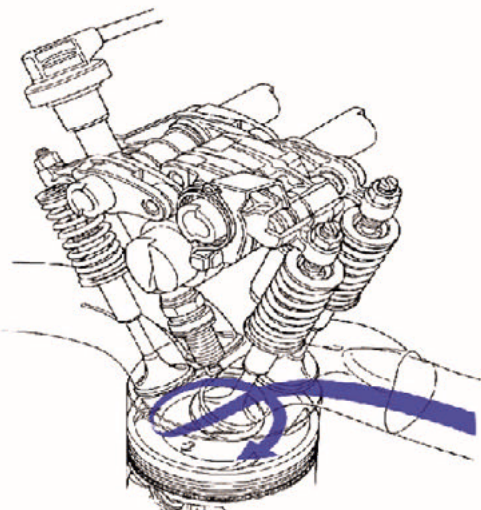


Fig.11 VTEC-E system

fuel ratio after startup. The result of making the fuel spray extremely fine and using the Quick Warm-up System is that it is possible to achieve stable lean-burn control very soon after startup.

Fig. 12 shows the Sauter mean diameter (measured by LDSA) of the fuel sprayed by the air-assisted injectors. The spray is measured at a point 100mm directly above the injector.

Compared with the injectors used in the ULEV system, improvements to assist air pathways have reduced the pressure loss, resulting in a fuel spray that is 35% finer than before.

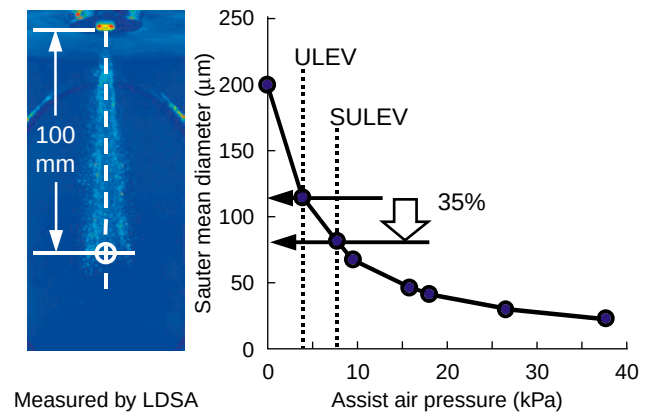


Fig.12 Air assisted injection spray characteristics

The Quick Warm-up System also offers benefits other than earlier catalyst light-off. Fig. 13 depicts the intake manifold pressure and the lean air/fuel control at the start of the FTP cycle for both the ULEV system and the Quick Warm-up System. Since this system delivers more air at startup, the engine can run in a higher load range than before, and combustion is more stable. As a result, the lean-burn range after startup is expanded and combustion is improved, as shown by the lower graph in Fig. 13.

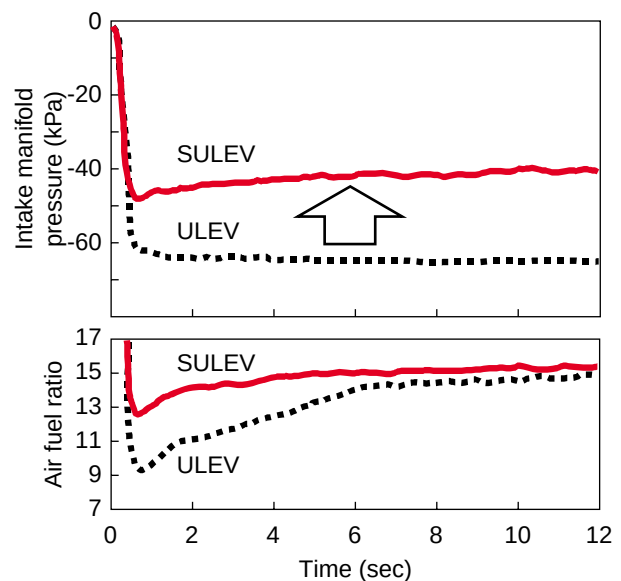


Fig.13 Lean air fuel control at start-up in the FTP cycle

Fig. 14 shows a comparison of the emissions produced by the SULEV system and the ULEV system in the sixty-second period after startup in the FTP cycle. The upper graph shows the total hydrocarbons and the center graph shows the NO_x emissions. Each emission component shows the dramatic decrease achieved by using the Quick Warm-up System and expanding the lean-burn range. This improvement is the result of reducing cold-start emissions and the early light-off of the catalyst. Furthermore, the peak emissions come sooner with the SULEV system than with the ULEV system due to the large volume of air that is fed into the engine after startup by the Quick Warm-up System.

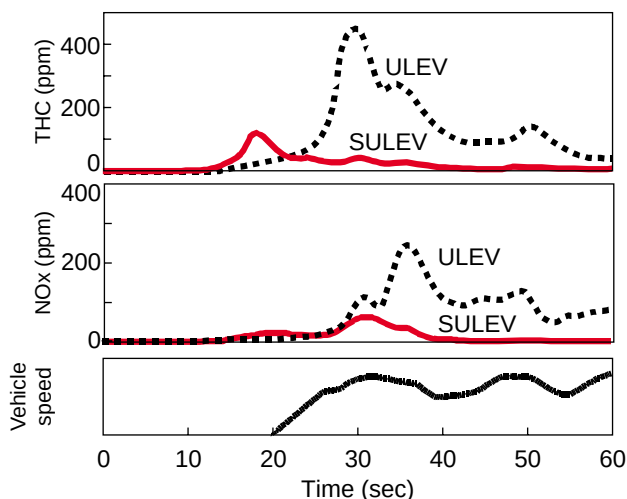


Fig.14 Emissions reduction effects of the SULEV system at startup in the FTP cycle

4.3. Vehicle Test Results

Fig. 15 quantifies the emissions in the CT phase of the FTP cycle. Most of the emissions in the CT phase occur during the first twenty seconds after startup.

As can be seen by the significant reduction in emissions shown in Fig. 14, the SULEV system reduces NMOG emissions by 70% when compared with the ULEV system, and reduces NO_x emissions by 35%. This demonstrates that the SULEV control system utilizes the VTEC-E system and three-way catalyst to their maximum potential.

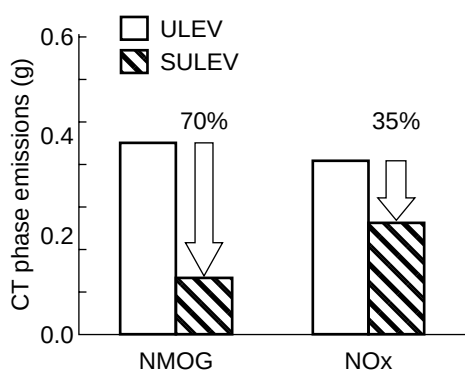


Fig.15 Emissions reduction effects of the SULEV system

5. Technologies for Reducing Emissions after Light-off of the Three-way Catalyst

Although we were able to attain high conversion efficiency by using a high cell density, thin-walled (1200-cell, 2.0-mil) catalyst, a further reduction in emissions was needed after catalyst light-off in order to satisfy the SULEV regulations. A control system based on pre- and post-catalyst air/fuel ratio sensors was previously used to obtain maximum conversion performance from a three-way catalyst. The Accord ULEV similarly used feedback control based on an UEGO sensor (Universal Exhaust Gas Oxygen Sensor), with corrections made to the air/fuel ratio in response to the output from the post-catalyst HEGO sensor (Heated Exhaust Gas Oxygen Sensor). This type of system is typically known as a "Secondary Oxygen Feedback Control System," and its purpose is also to enhance the conversion performance of a three-way catalyst.

We developed for the SULEV system a control technology that would improve the conversion performance of the catalyst beyond that of the Accord ULEV system. Although the purpose of high-precision control is to precisely control the air/fuel ratio for optimal catalyst conversion efficiency, there is an extremely long dead time until the amount of fuel sprayed by the injector to achieve the desired air/fuel ratio is detected by the post-catalyst HEGO sensor. It was essential that we take this dead time into account. Furthermore, we needed to always keep in mind the changing characteristics of the catalyst that occur as a car logs mileage.

Fig. 16 shows an overview of the new feedback control system. A subsystem that calculates the target value for the air/fuel ratio has been added to the STR (Self-Tuning Regulator)⁽⁴⁾⁻⁽⁶⁾ that was adopted in the Accord ULEV. This new subsystem consists of three modules: the Identifier, the Predictor, and the Sliding Mode Controller. The subsystem is called "PRISM" (Prediction and Identification-type Sliding Mode Control).⁽⁷⁾⁽⁸⁾

Identifier: This contains an internal catalyst model (a numerical model comprising the dead time and a low-pass filter), and identifies the dynamic characteristics of the catalyst model in real time based on the pre- and post-catalyst sensor output. As a result, the system can adapt in real time to changes in the catalyst characteristics while the engine is running.

Predictor: This predicts future values of the output from the post-catalyst HEGO sensor on the basis of catalyst model parameters q (a_1 , a_2 , b_1) calculated by the Identifier, and on the basis of pre- and post-catalyst sensor outputs. This module compensates for the long dead time between the actuator and the post-catalyst sensors, resulting in improved performance of the control system.

Sliding Mode Controller: This robust controller uses adaptive sliding mode control⁽⁹⁾.

Fig. 17 illustrates the operating principles of sliding mode control. The Sliding Mode Controller in the PRISM module limits sensor output VO_2 to the switching line in the phase plane with the difference between the HEGO sensor output and target values as the axes, and controls the output by sliding it to the origin. In order to bring VO_2 to the target value, the Sliding Mode Controller calculates the target air/

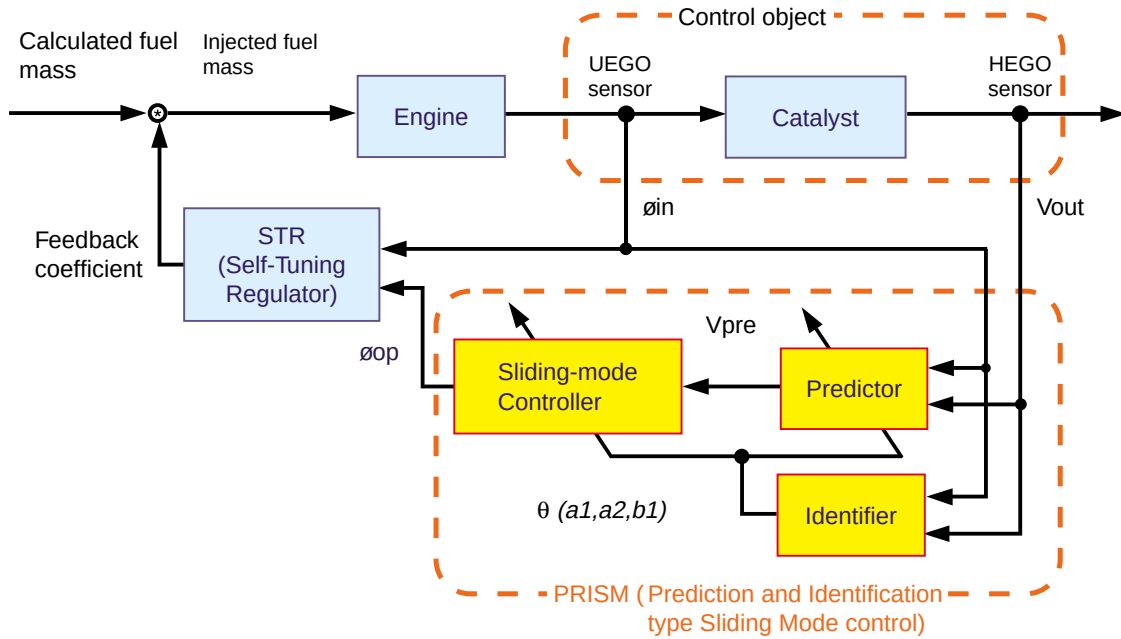


Fig.16 Overview of SULEV Feedback-System

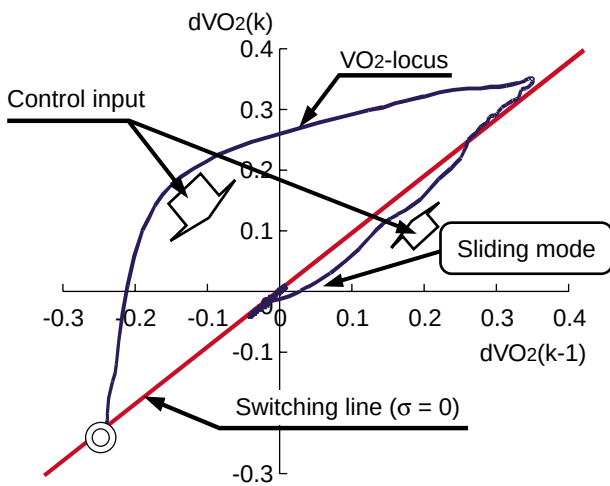
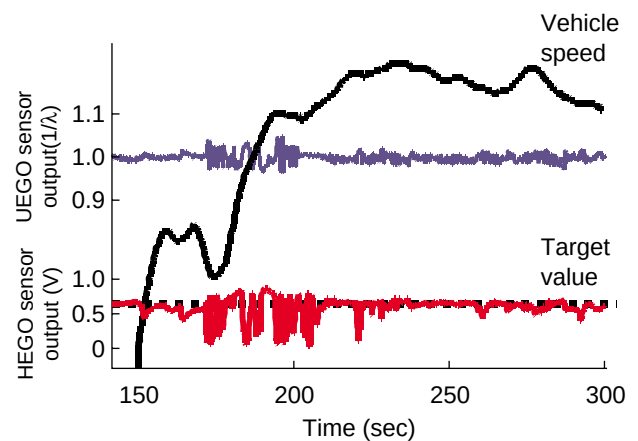

 Fig.17 VO_2 -locus in Phase plane


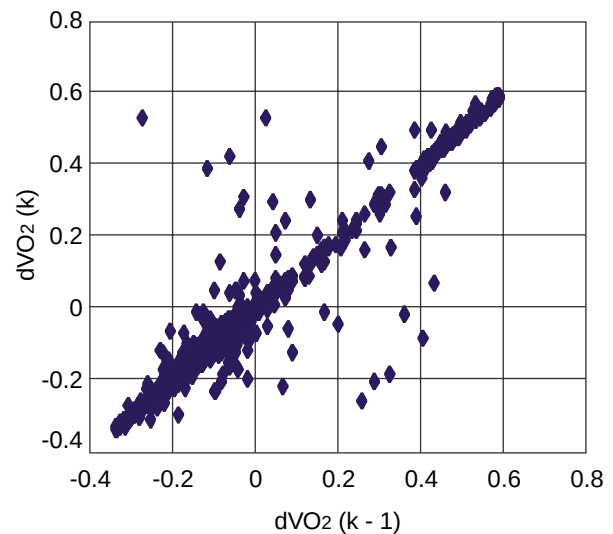
Fig.18 Improved controllability of secondary HEGO sensor output

fuel ratio for the STR (Self-Tuning Regulator), which performs pre-catalyst air/fuel ratio feedback.

Fig. 18 shows the control states of PRISM during the FTP cycle. The output from the post-catalyst HEGO sensor is controlled very well at around the target value without becoming stuck at 0V (a lean atmosphere) or 1V (a rich atmosphere). This indicates that the atmosphere in the three-way catalyst is always in the vicinity of the stoichiometric air/fuel ratio, which clearly demonstrates that PRISM precisely maintains the air/fuel ratio that is best suited for the three-way catalyst.

Fig. 19 shows the sliding mode control states. This figure clearly indicates that HEGO sensor output VO_2 is limited to a switching function in the dVO_2 phase plane.

Fig. 20 shows a comparison between the measured emissions for a ULEV system and PRISM when using the


 Fig.19 Secondary HEGO sensor output in phase plane (LA-4 2ND hill)

1200-cell, 2.0-mil catalyst. The graphs show that PRISM reduces NMOG by 20% and NOx by 48%; PRISM is clearly effective in reducing NOx, which is particularly sensitive to variation in the air/fuel ratio. PRISM has made it possible to further improve the conversion efficiency even after the catalyst has warmed up.

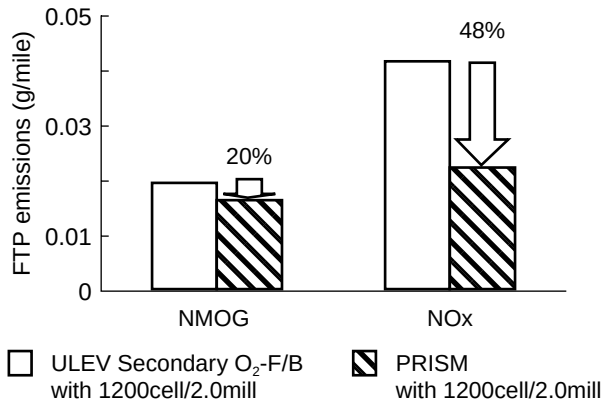


Fig.20 Comparison of ULEV's Secondary O₂ Feedback and PRISM

6. Conclusion

We were able to develop the Accord SULEV in compliance with the LEV-II SULEV regulations by combining the technologies described in this paper. Table 2 shows the emission levels that have been achieved with this system.

In keeping with our original principles, we were able to configure an extremely simple system without adding any special hardware. This system enables us to offer a car that complies with the Super Ultra Low Emission Vehicle regulations while maintaining engine performance that is to that of the Accord ULEV.

Table 2 ACCORD SULEV emissions
In-house data

	NMOG	CO	NOx
STD Values	0.010	1.00	0.020
Achieved Values	0.0084	0.16	0.016

(g/mile)

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Development of New 2.0L Lean-burn Engine

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ABSTRACT

A new lean-burn engine was developed as a 4-cylinder power plant for Honda's next generation 2L-class vehicle. New mechanisms, such as i-VTEC which combines variable valve timing and lift control and continuously variable camshaft phase control, are applied to this power plant. As a result, a 113kW maximum output and a 186Nm maximum torque were achieved while at the same time a 14.2km/L fuel economy in the 10-15 mode and a 50% reduction over the Japanese 2000 Emissions standards were realized. Further, improved assembly conditions were facilitated by decreasing the weight of the engine by 10kg and the length of the engine by 54mm.

1. Introduction

Progress in reducing the fuel consumption and emissions of vehicle engines is an important issue in addressing global concerns about the environment. The requirements of vehicle owners are diversifying, and the demand for high performance and improved noise and vibration performance from engines installed in a medium-class chassis is also increasing.

With this in mind, we developed our next generation 2L engine, aiming specifically at a balance between lower negative impact to the environment and achievement of high performance. The high-output, lightweight and compact DOHC-VTEC engine⁽¹⁾ that was developed for the S2000 was joined with the continuously variable camshaft phase system (VTC) to make a low fuel consumption and emission volume compatible with high torque and power output. We present in this paper an outline of the main components, technological features and structure of the engine.

2. Aims of the Development

It was planned that this engine would be fitted in various chassis types as the main new-generation engine. Therefore, we established the following developmental aims:

- (1) Maximization of efficiency to achieve the best level of fuel economy
- (2) An emission level complying with the 50% reduction in emission volume contained in the 2000 Japanese Emissions Regulations (J-LEV), with the possibility of complying with even stricter regulations
- (3) Improved power output in the most frequently used speed range, i.e., medium to low speeds

- (4) Lightweight and compact design making it suitable for use in small cars

We sought to achieve these aims in a balanced system that would be supplemented by conducting further development of the transmission system.

3. Outline of the Main Features and Specifications

Table 1 compares the specifications of the new engine (K20A) with those of the previous model (F20B), while Fig. 1 presents a graph of the new engine's output characteristics, and Fig. 2 shows a cross-sectional view.

Improved output was achieved by using a high

Table 1 Engine specification

Engine type	K20A New type	F20B Previous type
Cylinder configuration	In-line 4-cylinder	In-line 4-cylinder
Bore × Stroke (mm)	86 × 86	85 × 88
Displacement (cm ³)	1998	1997
Compression ratio	9.8:1	9.0:1
Valve train	DOHC i-VTEC	SOHC VTEC
Valve diameter (mm)		
Inlet	35	34
Exhaust	30	29
Fuel supply system	Programed fuel injection	Programed fuel injection
Ignition system	Programed ignition	Programed ignition
Recommended gas	Regular unleaded	Regular unleaded
Maximum output (kW/rpm)	113/6500	110/6200
Maximum torque (N·m/rpm)	186/4000	186/5000
10-15mode fuel consumption (km/L)	14.2	12.4
Exhaust emission	J-LEV	J-LEV

* Tochigi R&D Center

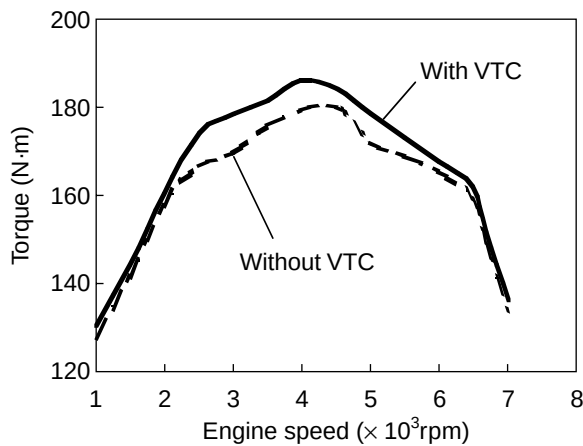


Fig.1 K20A engine torque

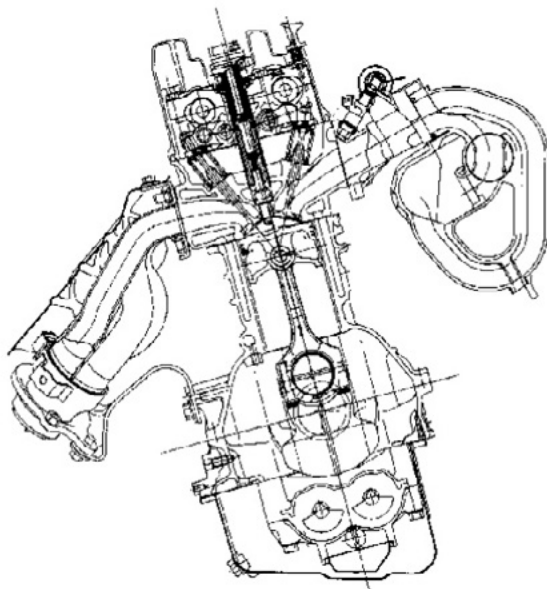


Fig.2 Section view of the K20A engine

compression ratio, larger-diameter intake valves, roller cam followers on the rocker arms and a special surface treatment of the piston skirts. The new i-VTEC valve system combines the proven VTEC system (Variable Valve Timing and Valve Lift Electronic Control) with the continuously variable camshaft phase control system (VTC, Variable Valve Timing Control). VTC was adopted by Honda for the first time, to provide improved performance across the entire driving range. The former VTEC system adjusted the camshaft operating angle and valve lift through two or three stages at low and high engine speeds to provide both low- and medium-speed torque and high power output. By making the valve timing continuously variable in response to the

load with VTC, the valve lift is always at the optimum setting, enabling us to achieve an improvement in volumetric efficiency.

We adopted a silent chain to drive the valve train, enabling the total engine length to be reduced and maintenance-free operation to be achieved. We also utilized a silent chain drive for the balancer shafts which reduce the secondary vibration characteristics of a four cylinder engine and suppress booming noise, and positioned these shafts inside the oil pan. We used a long tubular resin intake manifold to improve the medium- and low-speed torque, and made it compatible with high-speed power output by switching between different lengths with a rotary valve.

We adopted a cross-flow head with front intake and rear exhaust manifolds, making it possible to continue using a dual exhaust system, which reduces exhaust interference, and to place the catalyst close to the engine. In this way, the catalyst is activated almost as soon as the engine is started, enabling improvements in both emissions and output to be attained.

The adoption of this new configuration allowed a reduction in the total engine length of 54mm and in engine weight of 10kg in comparison with the F20B.

4. i-VTEC

Fig. 3 shows the configuration of the i-VTEC system, and Fig. 4 shows the operation map. To achieve a balance between high power output, good fuel economy and low emissions in the K20A engine, we adopted i-VTEC, which controls the valve timing to provide the optimum amount of overlap and operating angle of the cams in response to driving conditions. This system brings together VTEC, in which the rocker arm linked by the hydraulic switching device varies the cam profile, and VTC, in which hydraulic actuation allows continuous variation of the camshaft phase.

Switching between VTEC's low- and high-speed camshaft operation is set at an engine speed of about 2000 rpm. When the low-speed camshaft setting is in operation, one intake valve is almost completely inactive, increasing the airflow speed and leading to an improvement in volumetric efficiency. The high-speed camshaft setting produces higher power output through high valve lift and a wide operating angle. Furthermore, VTC varies the angle for closing the intake valve in the high-load area to achieve the highest volumetric efficiency from the respective camshaft settings. In the low-load area, it controls the amount of valve-opening overlap, to give priority to fuel economy and emission reduction. VTC further reduces the amount of valve overlap to minimize rpm fluctuation in the idling range.

Fig. 5 shows an outline of the VTC configuration. The signal from the ECU governs the oil control valve and provides oil pressure to the actuator. In response to control instructions, the oil pressure acts on the oil chamber for advance or retarding, the vanes change their position to advance or retard, and the camshaft phase is adjusted. As already explained, the optimum control across the entire driving range offered by the combination of VTEC and VTC provides an engine with high power output and good fuel economy.

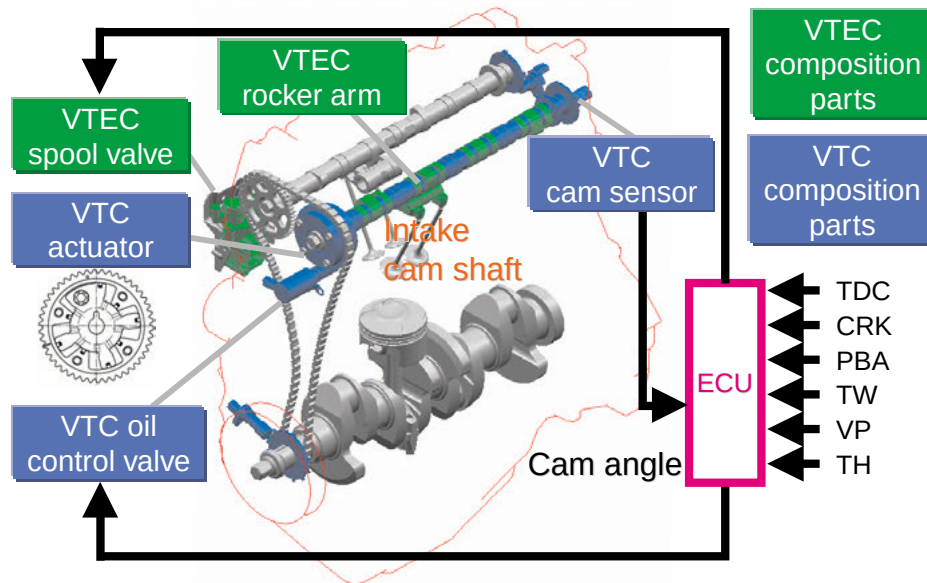


Fig.3 Construction of i-VTEC systems

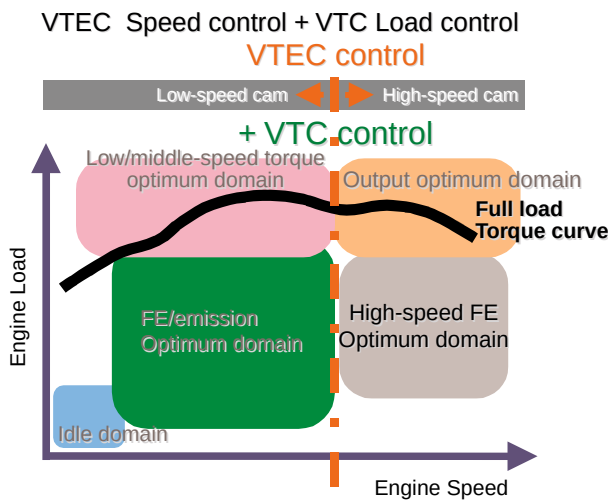


Fig.4 Outline of i-VTEC operation map

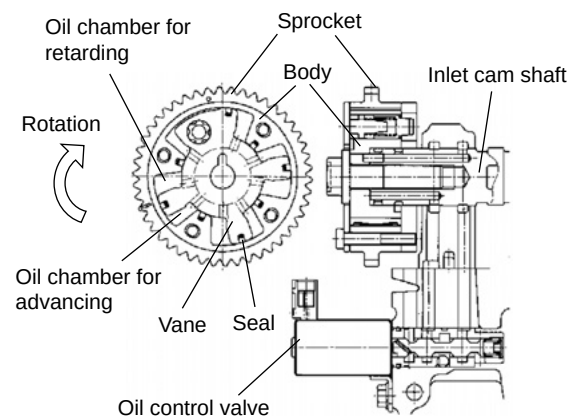


Fig.5 VTC unit and control system

5. Fuel Economy

We achieved a substantial improvement in fuel economy with the K20A by adopting the lean-burn principle, and conducting a thorough review to reduce engine friction. Next, we will discuss the technology applied for this achievement.

5.1. Improved Combustion

5.1.1. Cylinder Head

To improve the lean limit of lean-burn, we adopted a shape for the intake port which induces strong swirl flow. The VTEC system leaves one valve mostly inactive (1mm lift) in the low-speed range. The combination of this minute lift of one valve and the port shape induces strong swirl flow in the combustion chamber. This strong swirl flow facilitates combustion and improves fuel economy. Intake resistance is generally high with a port which induces swirl flow, which limits improvements to the power output. However, our

careful design of the port shape has brought these factors into balance.

Fig. 6 shows the shape of the head port, while Fig. 7 demonstrates the relationship between the mean flow coefficient and the swirl ratio. It can be seen that the intake port of the K20A engine has a higher power output potential than that of the previous design. Looking at the balance between the lean limit and mean flow coefficient with the two-valve system, the optimum point is established with a swirl ratio of 3.4 and a mean flow coefficient of 0.288.

By pre-assembling the supports for the two camshafts, we could make the engine as compact as the SOHC unit, and greatly reduce the amount of machining required.

Furthermore, in addition to improving the bowl-in shape of the piston crown and the squish shape of the combustion chamber, we adopted wide-gap spark plugs with thin iridium center electrodes, thus achieving rapid combustion while improving the lean limit.

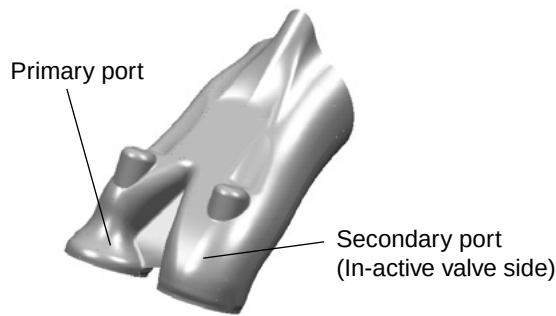


Fig.6 Intake port configuration of K20A

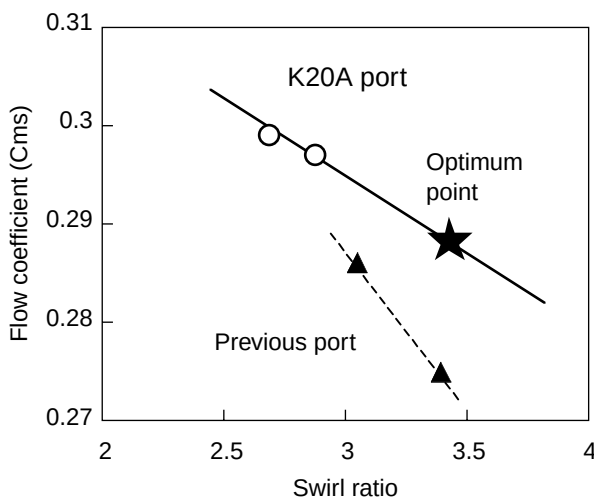


Fig.7 Relationship of swirl ratio and flow coefficient

5.1.2. Optimization of the Valve Timing

In addition to the established benefit of fast combustion with one valve mostly inactive, i-VTEC improves the fuel economy by reducing the pumping loss and raising the lean limit by controlling the amount of valve overlap and turbulence in the cylinder via variable valve timing. We established the i-VTEC system for the K20A engine to maximize these benefits to the fuel economy.

The i-VTEC principle under lean-burn conditions was established by taking into account the fast combustion that can be achieved by making one valve mostly inactive, while enhanced combustion results from the changes in pumping loss and turbulence in the cylinders due to the VTC system. Fig. 8 shows the co-variance of IMEP with the intake valve timing change during lean-mixture combustion. The co-variance of IMEP is an index of the stability of combustion. The co-variance of IMEP is at its lowest when the intake valve open timing is in the region of 20 - 35 degrees after TDC, and hence combustion is most stable in this region. This is the result of the delayed opening of the intake valve after the piston has begun its downward stroke causing the mixture to flow at a relatively high speed therefore increasing turbulence in the cylinder, and of the improved combustion achieved by reducing the residual gas through a smaller valve overlap.

Fig. 9 shows the variation of the indicated specific fuel consumption (ISFC) and co-variance of IMEP with the air-fuel ratio. The effects of the intake port swirl, improved lean limit due to changes in the shape of the combustion chamber and the crown shape of the pistons, and the use of a high compression ratio with optimized variable valve timing enabled an air-fuel ratio of 23 to be set. Therefore the ISFC was improved by 10% over the F20B engine. Further, we solved the issue of NO_x emissions in lean-burn engines during lean-mixture combustion by adopting a lean NO_x catalyst.

There are certain driving conditions under which lean-burn is not applied, and the mixture is regulated to a stoichiometric air-fuel ratio. As a means to improve the fuel economy with a stoichiometric air-fuel ratio, EGR, which recirculates exhaust gases, is effective. The i-VTEC system

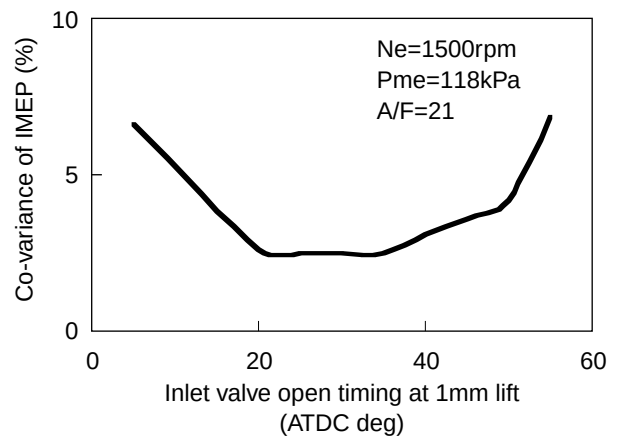


Fig.8 Influence of inlet valve open timing for lean mixture combustion stability

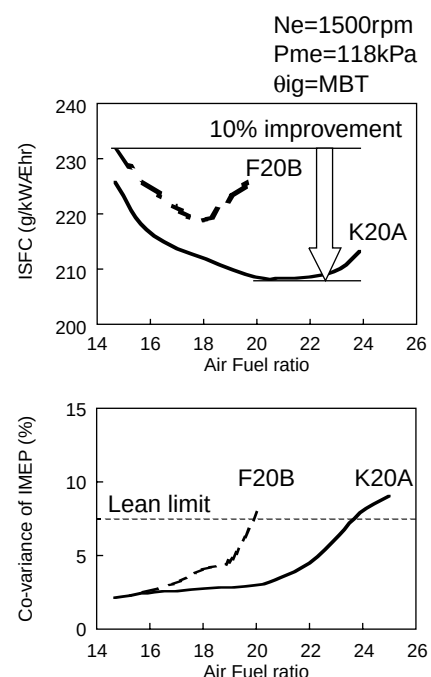


Fig.9 ISFC improvement and co-variance of IMEP

employs internal EGR to reintroduce exhaust gases into the intake manifold by varying the valve overlap.

Fig. 10 shows the relationship between the timing for intake valve opening and the brake specific fuel consumption (BSFC). Fig. 11 compares the fuel economy improvement of this system with that of the external EGR method utilized in the previous engine. Exhaust gas is recirculated into the intake runner when the valve timing is advanced and the amount of overlap is increased; therefore, internal EGR is increased and the fuel economy is improved by reductions in the pumping loss and thermal loss and by the improvement in ratio of specific heat of the working medium. However, if the volume of EGR is too high, combustion becomes unstable and misfires occur. Since i-VTEC combines a high swirl port with one valve being almost completely inactive, combustion is stable even when the EGR volume is increased, enabling fuel economy to be improved by 5%. Furthermore, when we compare internal and external EGR for the same EGR volume, we can see that the rate of fuel economy improvement with internal EGR is higher. This is

because the temperature of the recirculated gas is high, and the intake valve closing angle approaches bottom dead center with internal EGR, thus increasing the effective compression ratio. VTC establishes valve timing which minimizes the fuel consumption at all engine speeds and loads.

By setting the degree of overlap to the minimum in the idling range, there is no loss of market appeal due to rough combustion. Further, since it was possible to reduce the idling speed by 50 rpm, which has a significant effect on fuel economy, the idling mode fuel economy was improved by 2%.

5.2. Friction Reduction Techniques

We made improvements in many areas relating to the reduction of mechanical losses. Table 2 presents an outline of the techniques that were applied to friction reduction. Together with decreasing sliding friction by improving the oil film retention on the rubbing surfaces, we reexamined the valve spring load. In Fig. 12, the friction under partial load is compared with that of mass production models with a reduction of more than 7% being achieved. We explain next how this was achieved.

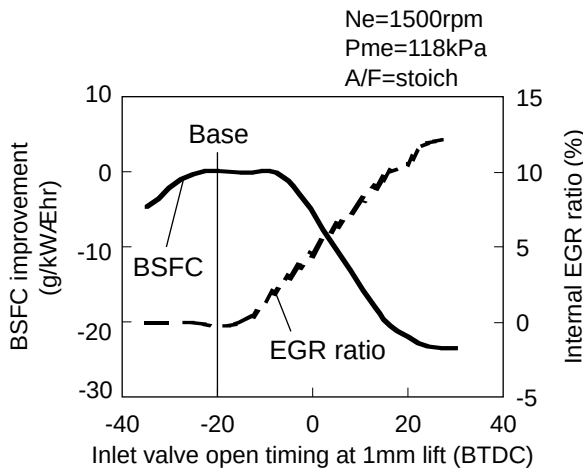


Fig.10 Influence of inlet valve open timing for BSFC improvement

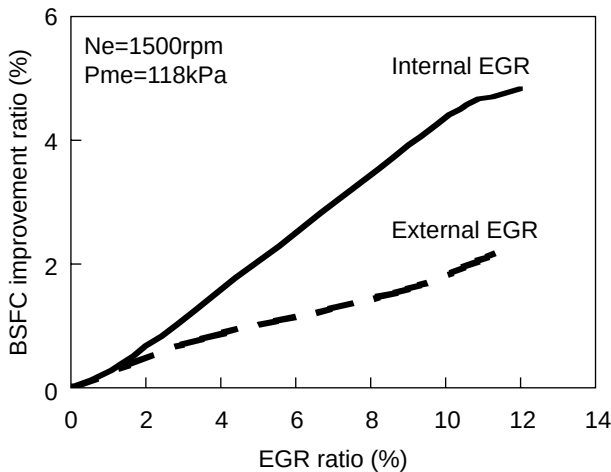


Fig.11 Comparison of BSFC improvement with internal and external EGR

Table 2 Friction reduction technology

Parts	Applied technologies friction reduction
Rocker arms	Roller follower
Valve springs	Spring full lift load reduction
Camshaft	Camshaft bearing clearance optimized
Piston	Skirt contact surface product reduction Skirt surface micro dimple machining Skirt surface molybdenum coating
Piston rings	Tension reduction
Oil pump	Rotor miniaturization Relief setting oil pressure down
Crank journal bearing	Surface single-cut turning machining
Balancer journal bearing	Surface single-cut turning machining
Cam chain tensioner	Tensioner spring load optimized
Engine oil	Low viscosity oil (0W-20)

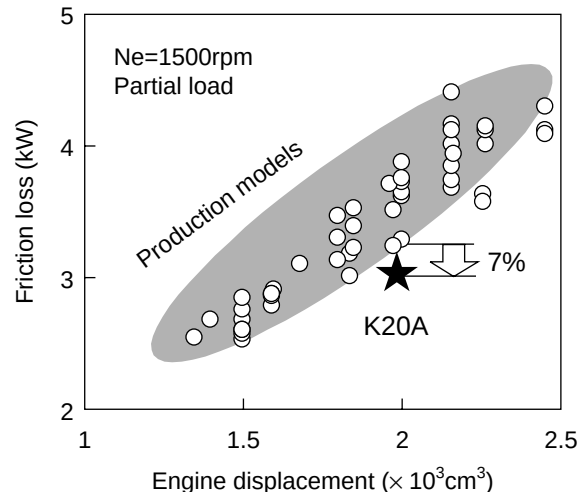


Fig.12 Friction loss and displacement

5.2.1. Valve Train

To reduce friction, we used roller followers throughout the entire valve train. For the intake VTEC rocker arm we adopted a coaxial configuration of three rockers and a roller follower, and for the exhaust we used an aluminum die cast structure featuring two valves in one unit, in order to reduce both weight and costs.

The intake rocker arm is manufactured by lost wax precision casting, making possible a lightweight structure with a high capability to follow high engine speeds. Fig. 13 shows the structure of the rocker arms.

5.2.2. Pistons

To improve fuel economy, we greatly reduced the weight of the pistons and applied friction reduction techniques. As shown in Fig. 14, we shortened the piston skirt and compression height and reduced the width of the skirt, leading to a reduction in weight of 10% compared to the previous engine. To avoid any loss of durability caused by this weight reduction, we evaluated the stress at each point and the pressure distribution on the surface of the skirt by a CAE analysis, and used the results to optimize the structural design. We increased the oil film retention on the skirt surface by a Micro-dimple treatment⁽²⁾ and molybdenum coating. The side force pressure is reduced by the lighter pistons and oil film shearing is reduced by shortening the width of the skirt. Therefore we achieved a friction reduction of about 25% for the pistons.

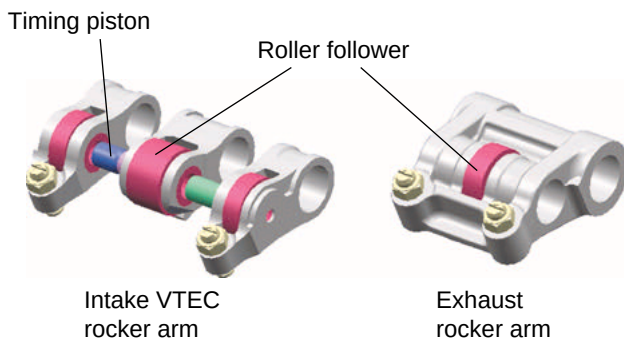


Fig.13 Rocker arm



Fig.14 Piston configuration

6. Power and Torque Output

6.1. Intake System

The lightweight intake system utilizes an intake manifold made of resin. Furthermore, in order to balance the improvement in medium-to-low-speed torque with a high output, we applied a variable-length manifold controlled by a switching valve. We used a rotary type switching valve rather than the previously applied butterfly valve as it has lower flow resistance. Fig. 15 shows a cross-section of the intake manifold. We used resin for the intake manifold rather than the former aluminum alloy to suppress heat transfer from the engine and to reduce the temperature of the air intake, which increased the volumetric efficiency and raised the output. A computer simulation was used to vary the conditions and to establish the basic specifications for the intake manifold length. Fig. 16 shows the effect of varying

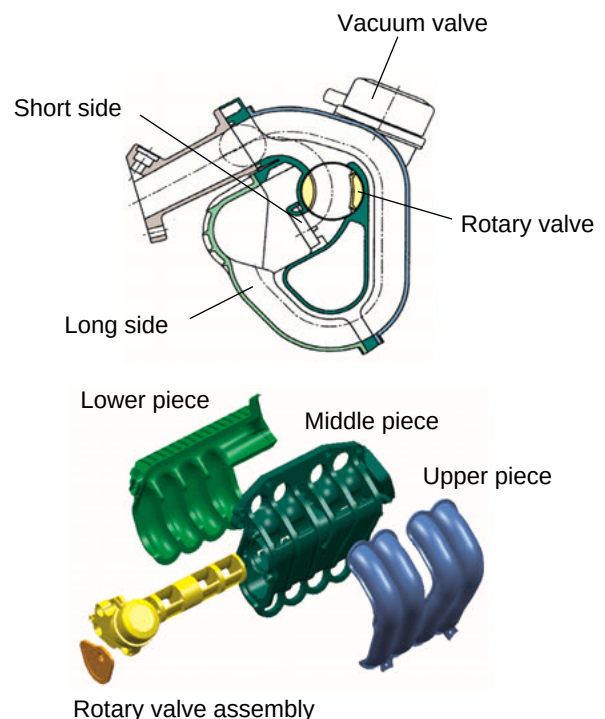


Fig.15 Dual mode resin intake manifold

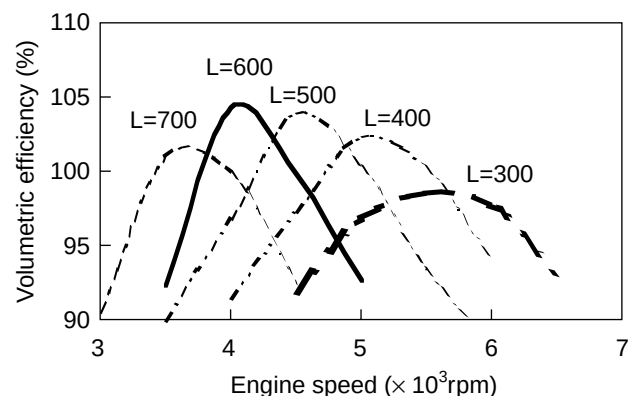


Fig.16 Influence of intake manifold on volumetric efficiency

the pipe length on the volumetric efficiency. The volumetric efficiency of the former model's intake runner length of about 300mm is adequate at high revolutions, but in the medium-to-low-speed range it drops and a so-called torque dip occurs. To avoid this drop, we aimed for a high volumetric efficiency increase throughout the driving range by switching between two tube lengths. We established tube lengths of 540mm and 270mm to achieve the required volumetric efficiency and installation space. VTC establishes the valve timing which will optimize the torque in response to engine speed, and we achieved with this system an improvement of about 13% in the medium-to-low-speed torque compared to the F20B. Fig. 17 demonstrates the torque improvement resulting from the i-VTEC system.

6.2. Exhaust System

A dual system for the exhaust manifold and exhaust pipe is used to reduce exhaust interference and improve engine output. In addition to a low-heat-capacity stainless steel pipe applied to ensure quick heating of the catalyst, the split cross-section produced at the center of the pipe by a partition plate in the confluence area has reduced the heat-radiating area and balanced the output with low emissions. We furthermore achieved a weight reduction of 1.3kg compared to the FC 4-1 exhaust manifold used with the F20B engine.

Fig. 18 shows the construction of the exhaust system, and Fig. 19 demonstrates the improved engine output with the dual inner pipe type exhaust pipe.

6.3. Emissions Performance

We achieved low emissions with the K20A engine by combining a rear exhaust, low-heat-mass exhaust system and three-way catalyst, and a lean NOx catalyst. Fig. 20 shows the temperature rising characteristics of the three-way catalyst directly after starting the engine.

Compared with the F20B, the time taken to reach catalyst activation temperature is reduced, thus reducing emissions at engine start-up.

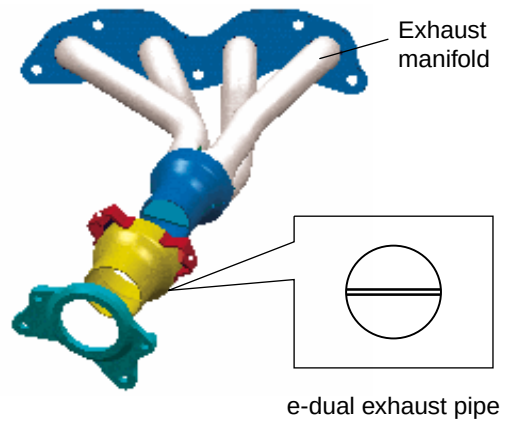


Fig.18 Configuration of e-dual exhaust system

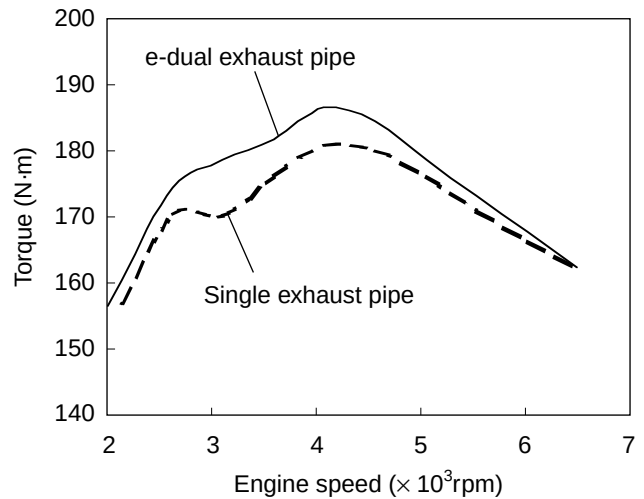


Fig.19 Effect of e-dual exhaust

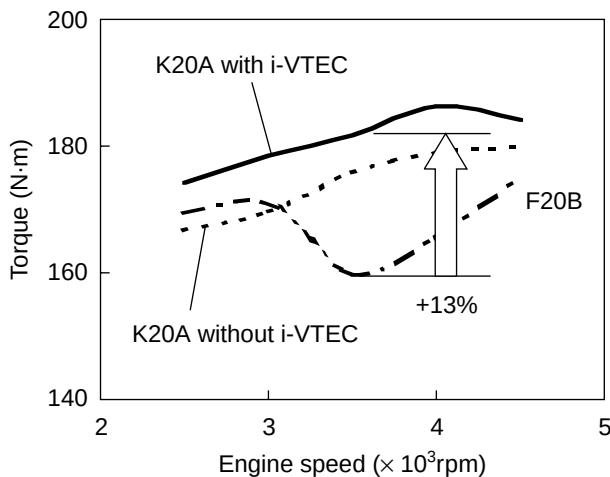


Fig.17 i-VTEC torque improvement effect

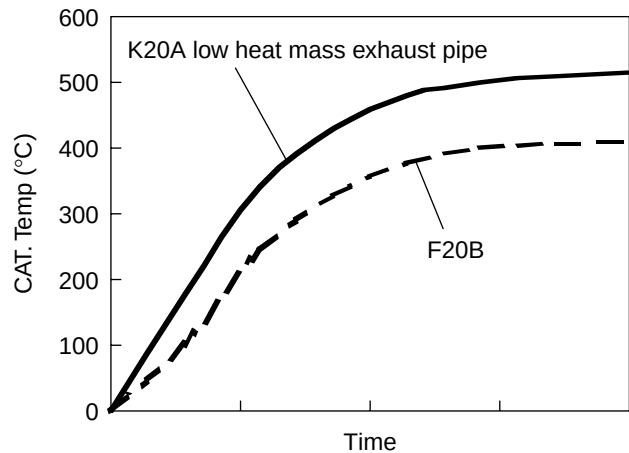


Fig.20 Comparison of catalyst temperature after engine start

7. Engine Construction

The K20A has an improved engine construction with better output performance and fuel economy, as well as improved noise and vibration (NV) properties, decreased weight and size. We discuss next the engine construction used to realize these benefits.

7.1 Improvement in Noise and Vibration (NV)

7.1.1. Cylinder Block Structure

The lower block structure has integrated FC bearing caps in order to reduce the crankshaft hitting noise and the power plant resonance which exacerbate engine noise during acceleration and at high engine speed. Crankshaft hitting noise was reduced by suppressing transmitted vibration by increasing the rigidity of the bearing journals. As for the problem of power plant resonance, we improved the connection rigidity between the engine and transmission, which is weakest point in its flexural rigidity, by setting the connection flange to the transmission on a lower block that has higher rigidity than the previous deep skirt. Fig. 21 shows the cylinder block construction utilizing this lower block. Fig. 22 compares the effectiveness of the improved flexural rigidity in the K20A lower block structure with the former deep skirt and separate bearing caps, as expressed by the vibration response data at the front and rear of the transmission.

7.1.2. Low-mounted Balancers and Oil Pump Structure

To eliminate the second order vibration that is characteristic of an in-line four-cylinder engine and to reduce booming noise in the chassis, we utilized secondary balancers. The trochoid-shaped oil pump is mounted coaxially with the balancer shafts to concentrate their functions and simplify the construction. Fig. 23 shows the configuration of the balancer mechanism with the built-in oil pump. The rear balancer shaft is driven by a chain from the crankshaft, and the front shaft is driven in the opposite direction by helical gears. The split construction of the block and balancer assembly allowed us to optimize the external

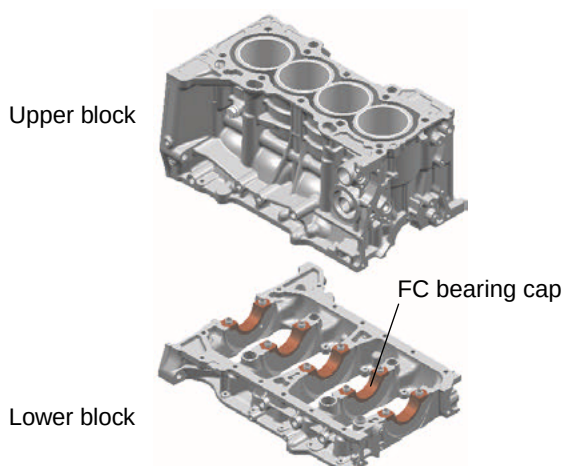


Fig.21 Cylinder block

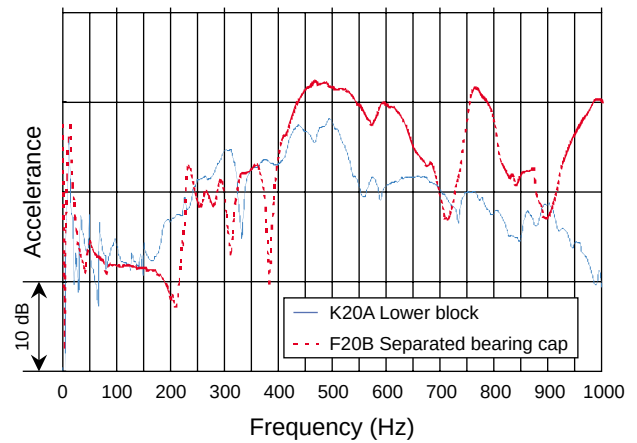


Fig.22 FRF of power plant (Lateral bending direction)

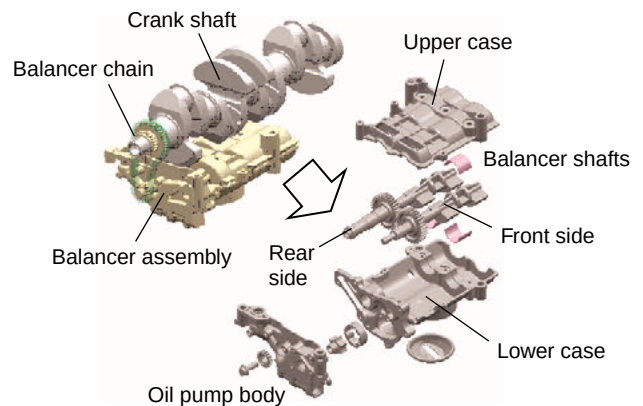


Fig.23 Balancer system

diameter of the balance weights and journals, and to achieve a compact construction with the two journals per balancer shaft. The oil pump rotor is driven by the front balancer shaft at twice the engine speed. Because of this, we were able to halve the external diameter of the rotor and achieve higher efficiency in comparison with the former directly connected crankshaft construction. We were further able to reduce the total length of the engine by 25mm by using a chain drive and locating the balancers below the engine block.

7.2. Lightweight, Compact Design

7.2.1. Variable-Length Resin Intake Manifold

Resin is used instead of the former aluminum alloy for the intake manifold, including the switching valve, to achieve a weight reduction of 6.0kg.

7.2.2. Camshaft Drive System

A silent chain is used to drive the camshafts and balancers, and this has enabled the engine length to be reduced by 13mm in comparison with the former engine employing a timing belt drive. Improvements in the system's durability and reliability have eliminated the necessity for periodic maintenance. Fig. 24 shows the layout of the camshaft drive chain. By adopting an FCD hollow

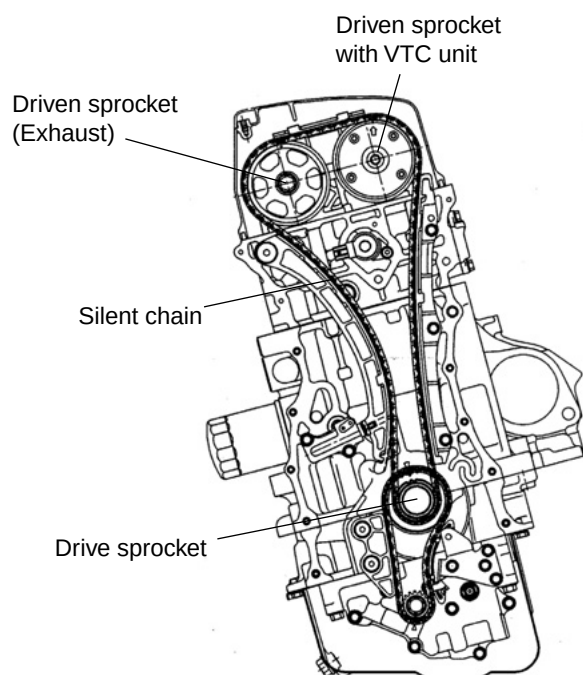


Fig.24 Camshaft chain layout

construction for the camshaft, we achieved a reduction in weight of 0.7kg compared to the former camshafts used with the DOHC engine. We further simplified the construction of the cylinder head by utilizing the hollow camshafts as channels to supply oil to the camshaft journals. We were able to reduce manufacturing costs by utilizing a production process known as Quick Cast with the FCD hollow camshafts for the first time. In this process, sand molds are not required. Rather, the material is directly cast into a die.

7.2.3. Drive System for the Accessories

A serpentine belt drive is used to drive the accessories.

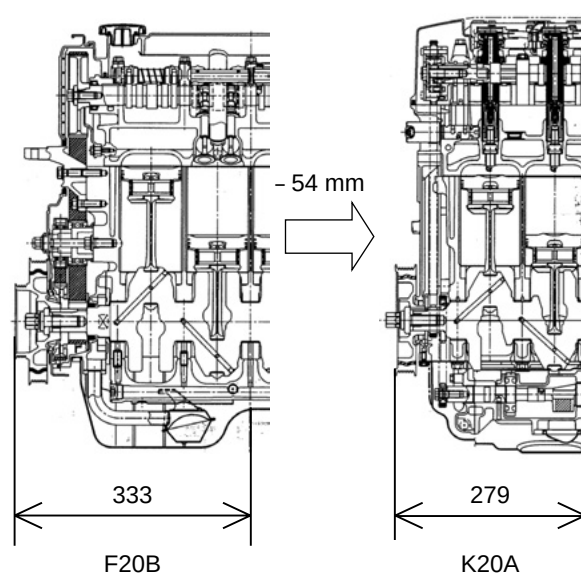


Fig.25 Comparison of engine length

By driving all the accessories with a single multi-groove belt, we achieved a reduction in engine length of 16mm compared to the former two-belt system. As Fig. 25 shows, the change to a camshaft chain drive, lowering the balancers and oil pump and the adoption of a serpentine accessories belt has allowed a total reduction in engine length of 54mm compared to the F20B engine.

A torsion spring auto-tensioner is used to make the accessories belt maintenance-free. Furthermore, the change to a camshaft chain drive allowed us to drive the water pump with the accessories belt, rather than with the former timing belt drive.

Table 3 K20A main technology

Subject	Main technology
<u>Good fuel economy</u>	• i-VTEC • High swirl intake port • Friction reduction technology (see Table 2)
<u>Low emission</u>	• Back exhaust layout • Lean NOx catalyst
<u>High performance</u>	• High compression ratio • e-dual pipe exhaust manifold • Dual mode resin intake manifold
<u>Light weight and compact</u>	• Formation of serpentine accessories belt • Camshaft and balancer chain drive
<u>Silent (NV improvement)</u>	• High rigidity cylinder block • Balancer in oil pan

8. Conclusion

Table 3 summarizes the aims and main techniques used in the development of the K20A engine.

It was possible to realize higher standards for the next-generation 2L engine by improving the output, fuel economy, and emissions utilizing i-VTEC as the key feature. At the same time, the lightweight and compact design and quiet operation make the K20A suitable for installation in various types of chassis.

9. Outlook

The K20A engine has a construction that allows for derivatives such as increased displacement and sport versions as responses to environmental changes and the diversifying needs of users. We are proud of our achievement of this system, however, we plan to put forth great efforts toward enhancing this technology in response to changing demands.

We sincerely thank all those people, both inside and outside Honda R&D, who supported our efforts.

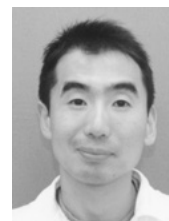
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Development of an Engine Performance Data Acquisition System

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ABSTRACT

In response to the trend of complex engine control, we developed a new generation engine data acquisition system equipped with high precision analyzing capabilities. This system enables measurement data processing using the Windows operating system and is able to perform cylinder pressure analysis. Further, the system can be used with engine test cells and electronic control units (ECUs) with various specs. It can also be used for automatic testing.

This system allows for the effective use of existing equipment. It is a simple system using a personal computer to process the acquired data. Also, the system employs a real-time operating system to control timing and thereby enable simultaneous data acquisition and processing.

As a result, we were able to achieve stable data acquisition, improved analysis capabilities, and a reduction in analysis time.

(Windows is the trademark of Microsoft Corporation.)

1. Introduction

There continues to be an increasing demand for a reduction of automobile fuel consumption and emissions in order to address global environmental issues. The computerized engine control unit (ECU) that controls fuel and intake air amount, ignition timing and so on has improved performance and speed⁽¹⁾. More detailed engine data is required to make engine control more precise. Therefore, the development of data acquisition systems with greater accuracy and higher performance is required.

We have developed an engine performance data acquisition system that can be easily operated to instantaneously confirm and analyze measurement results, and that can make comparisons with other data over a network at any time. The system can also acquire data during automatic testing. Use of this data acquisition system makes it possible to make finer ECU settings and reduce testing time.

This paper will present an introductory overview of the engine performance data acquisition system (Engine Analyzing SYstem, i.e., EASY) devices and the results obtained by application of the system.

2. Engine Performance Data Acquisition System (EASY)

In developing this highly accurate, high-performance data acquisition system, we examined the various tools previously used to test engine performance. We ascertained their problem areas and identified the needed improvements.

The major measurement and analysis tools used conventionally in engine performance testing are as follows:

- Engine performance data acquisition devices
Description: Personal computer-based engine performance data acquisition and storage system.
Functions: Acquisition and storage of performance data.
Generation of graphs and diagrams based on engine performance data.
- ECU monitor (Monitor Mark 5, hereafter MM5)
Description: Monitor ECU data during engine operation.
Functions: Monitor ECU data, data communication (serial communication).

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- Combustion analysis system

Description: Cylinder pressure measurement and analyzer developed in-house.

Functions: Acquire cylinder pressure measurements, calculate IMEP, and perform thermal calculations.

These tools are a group of independent measuring instruments. They do not form a system that would enable simple operation for instantaneous confirmation and analysis of measurement results. Moreover, the separate systems use different data formats, some of them being proprietary file formats that cannot be processed on the widely used Windows operating system. Due to these and other problems, the organizing and analyzing of data imposed high man-hour costs.

Therefore, we developed a system that integrated the function of this group of tools in order to resolve those problems.

3. Overview of the EASY System

3.1. The EASY Concept

The development objective was an integrated engine performance data acquisition system that combined the capability to acquire large amounts of engine test information, compare it with other data during measurement, and analyze the data, while also analyzing the combustion process.

In addition, the above capabilities are integrated into an efficient data acquisition system that operates on a personal computer (hereafter PC) platform to regulate the test engine operating conditions based on ECU data, and that also enables testing without an operator.

3.2. System Configuration

The EASY system was built to meet the following basic requirements:

- (1) Use the same software to perform performance data acquisition, ECU data acquisition, cylinder pressure analysis, and automatic testing.
- (2) Put data into a format that can be processed in Windows.
- (3) Design hardware capable of data acquisition at engine speeds up to 8000 rpm.
- (4) Utilize existing devices (engine test cell data acquisition systems, exhaust gas analyzers, etc.) as much as possible.

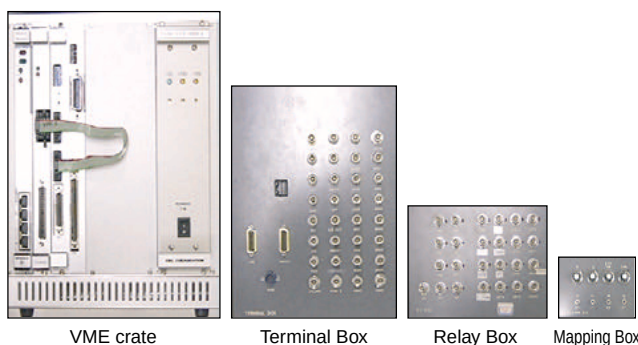


Fig.1 EASY system hardware

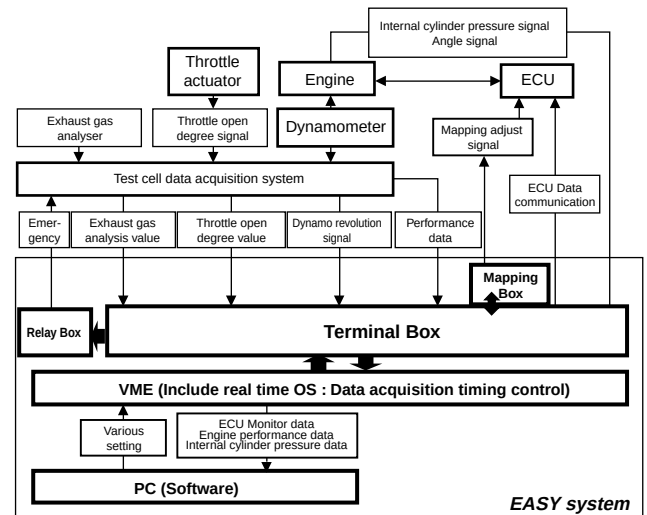


Fig.2 EASY system block diagram

The EASY hardware components are shown in Fig. 1, and the overall system configuration is shown in the block diagram in Fig. 2.

The EASY system adopted the method whereby data acquisition system measurement data (GP-IB communication), cylinder pressure data, and ECU data (Honda original protocol) are input together into the terminal box. There, the data is transferred to a standard bus unit (Versa Module Europe, hereafter VME), which was recommended by Motorola, Inc. The system configuration also includes a relay box and a mapping box. The relay box is linked to the terminal box to provide interlock during automatic testing. The hardware specifications for this configuration are as shown in Table 1.

Next we will give an overview of the various hardware components:

(1) VME Crate

The VME crate contains a CPU board, a high-speed analog-to-digital (A/D) converter board, a GP-IB board, and memory for temporary storage of data. Installation of a real-time operating system allows the VME crate to provide event timing control and communications for the EASY system as a whole. The VME crate is also assigned a portion of the

Table 1 Hardware specifications for the EASY system

High-speed A/D	100kHz/8ch (Simultaneous sampling of each ch)
Low-speed A/D	100Hz/8ch (Simultaneous sampling of each ch)
Memory capacity	4MB (1MB : exclusive for converted A/D data)
ECU monitor communication method	Honda original protocol (50 ~ 375kbps)
GP-IB communication method	Master and slave function (1ch each)
Engine revolution sensor	1pulse/revolution 360pulse/revolution
Relay input/output	Input 8ch/output 16ch
Mapping operation terminal	4ch
Power supply	AC100V

calculations in order to reduce the load on the instrumentation PC⁽²⁾.

The high-speed A/D converter used for acquisition of cylinder pressure signals utilizes operational input to assign a reference potential in order to improve response and cancel out, thereby eliminating the influence of, electrical noise.

The memory is configured for cylinder pressure data acquisition on a maximum of four channels for 400 cycles. (When using eight channels, it is able to operate continuously for 100 cycles.)

(2) Terminal box

The terminal box receives many different kinds of measurement signals, such as engine test cell data acquisition system data, ECU data, and general-purpose analog signals, all together. It integrates these and conducts data interactions with the VME crate.

The terminal box is also connected to a relay box that activates an interlock feature when a problem occurs during engine operation and to a mapping box that performs fine adjustments in operating conditions.

(3) Relay box

The relay box is a terminal box that integrates engine test cell data acquisition system interlock contacts during automatic testing. This relay box is connected to the VME crate by means of the terminal box. When a problem occurs, it activates the data acquisition system interlock contacts to protect both the engine and the device.

(4) Mapping box

When implementing automatic testing, the operating condition set-up file software controls all the operating condition settings. However, the system must switch between manual and automatic modes when automatic testing begins and ends. This mapping box is used to enable appropriate manual control of ignition timing and fuel injection volume at these times in order to allow a smooth transition from manual to automatic and automatic to manual operation.

3.3. Data Acquisition

The EASY system is set up to allow the acquisition of a series of performance data and cylinder pressure data to be handled by a single piece of common software.

On the other hand, it is also possible to acquire data without using the VME crate when cylinder pressure analysis is not being performed. Common software is used regardless of the presence or absence of a VME crate. The processing flow is shown in Fig. 3.

The EASY system allows the effective use of existing engine test equipment by means of selectable set-up software for various types of engine cell data acquisition systems, ECU communication and so on.

(1) Acquisition of performance data and ECU data

The flow of performance data and ECU data acquisition in EASY is shown in Fig. 4. Performance data and ECU data are first combined in the terminal box and then transferred through the VME to the PC, where the processing takes place.

(2) Cylinder pressure analysis

EASY is capable of using the high-speed A/D board incorporated into the VME for cylinder pressure acquisition.

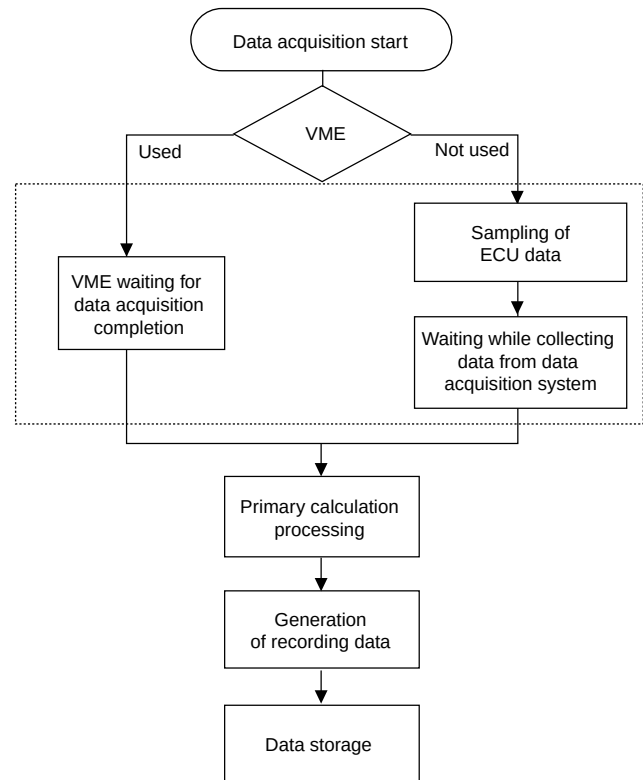


Fig.3 EASY data acquisition flow chart

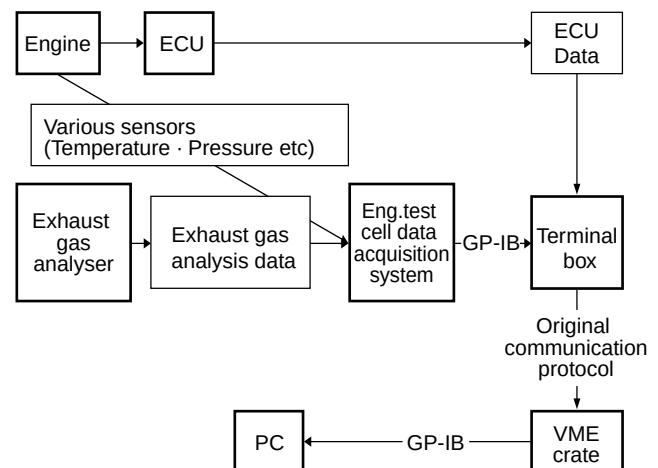


Fig.4 EASY system recording flow chart for engine performance data and ECU data

The data acquisition flow is shown in Fig. 5.

The cylinder pressure data acquisition screen displayed when EASY acquires this data is shown in Fig. 6.

EASY takes into account actual engine development by giving cylinder pressure analysis the following characteristics, which enable the stable measurement of cylinder pressure without variations in the data⁽³⁾⁻⁽⁵⁾:

- Number of acquisition channels: 8 channels maximum
- Continuous acquisition cycles: 400 cycles maximum
- Trigger function using TDC and crank signals
- Selectable cylinder pressure pegging method
- TDC adjustment function (fine adjustment and TDC inversion)
- Stored signal data file name link to performance data

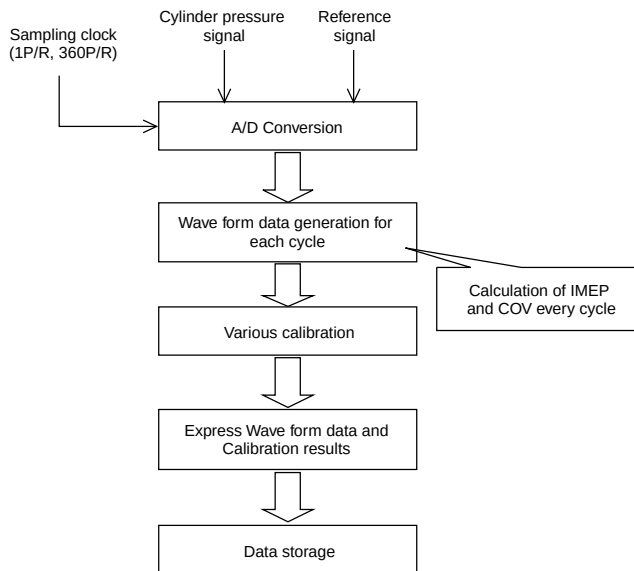


Fig.5 EASY system cylinder pressure data acquisition flow chart

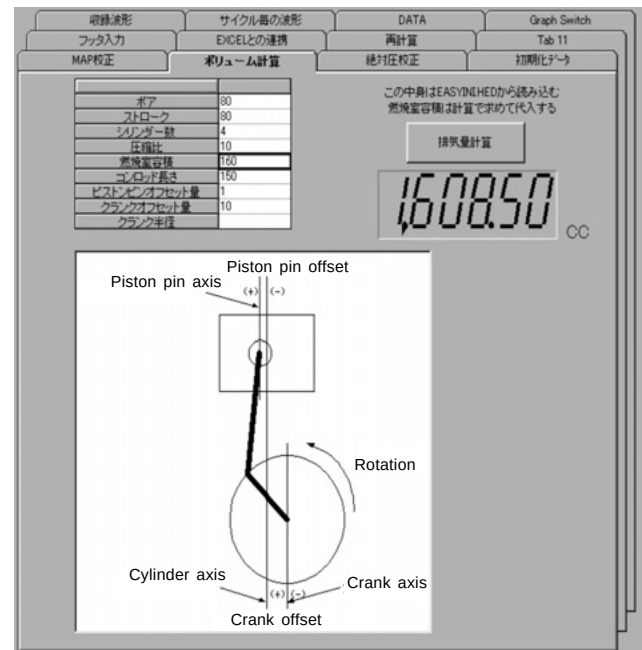


Fig.7 EASY engine specification screen

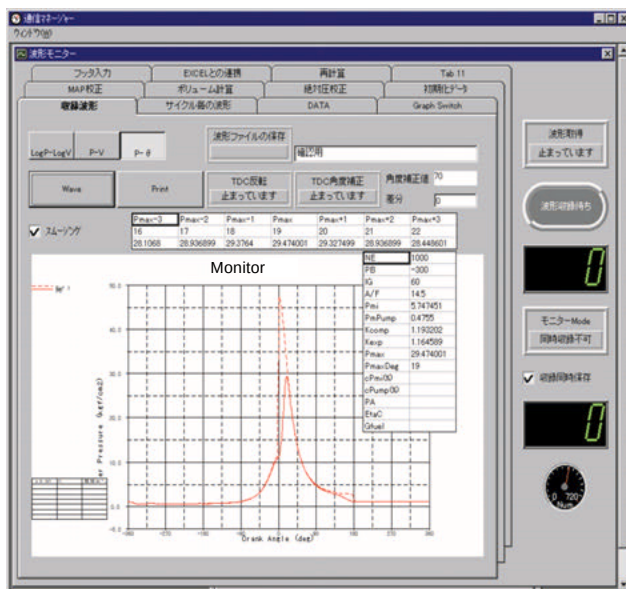


Fig.6 EASY cylinder pressure data screen

- Simplified engine specification set-up screen (positive/negative direction, direction of revolution, input items: see Fig. 7)

3.4. Communications

Apart from communications with engine test cell data acquisition systems, the EASY system also performs a variety of data communications, including ECU communications for the purpose of ECU data acquisition. These are outlined below:

(1) Engine test cell data acquisition system communications

The transfer of various types of measurement data from the engine test cell data acquisition system devices to the PC occurs via GP-IB communications. As shown in Fig. 8, the use of the GP-IB communications protocol enables

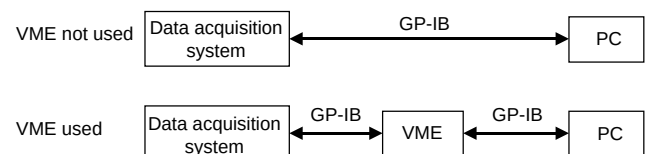


Fig.8 EASY communication between EASY and engine test cell data acquisition system

operation by a common system without any change to the PC communications interface regardless of the presence or absence of a VME⁽⁶⁾.

Reflecting the measurement needs of present-day circumstances, there are many engine test cell data acquisition systems of different types from different manufacturers. EASY is designed as a universal system that can support all of these systems by means of set-up files that enable set-up of GP-IB data definition and other configuration information in an integrated operation.

(2) ECU communications

The ECUs from different manufacturers utilize a variety of protocols for data communications. Therefore, EASY was made universally applicable by giving it the capability to select the communications type, baud rate, and endian status (word data order) with the software.

In addition, EASY uses a method that can comply with either of the two different file formats used for the labels and ECU address lists employed in ECU communications.

On the other hand, as noted above, there are also occasions when data acquisition is conducted without using a VME crate because of the nature of the tests. Therefore, the system also allows ECU communications using RS-232C serial communications by means of an MM5, as shown in Fig. 9.

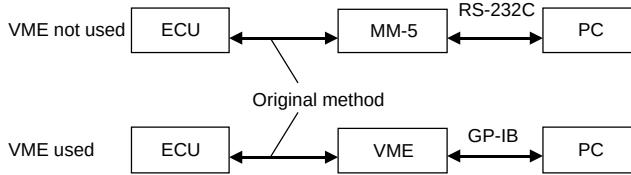


Fig.9 EASY communication between EASY and ECU

3.5. Acquired Data Files

EASY can acquire and analyze the following kinds of data:

- Performance data (rpm, brake horsepower, fuel flow, etc.)
- ECU data
- Cylinder pressure data (IMEP, maximum pressure values, etc.)

In order to reduce the man-hours required for data analysis, the acquired data are stored in file formats that can be processed by commercially available Windows applications. This has a particularly large impact on engine combustion analysis, where large volumes of data are processed.

Acquired data are stored in files with a proprietary format. EASY takes the unified approach of writing ECU data and cylinder pressure calculation results from these stored files into the performance data files together. With regard to cylinder pressure data, we also took into consideration the fact that secondary analysis is performed on the actual acquired waveforms and designed EASY to save additional copies of the waveforms named with the performance data file names plus identification numbers. This makes it possible to easily retrieve the relevant cylinder pressure for the operating conditions when the performance data is searched by operating condition.

4. Automatic Testing

EASY also has an automatic testing function. By setting the engine operating conditions (rpm, intake boost pressure, ignition timing, etc.), the engine can be made to operate

automatically under the target operating conditions while data is being acquired.

This function makes it possible to set the operating condition convergence time, automatically calibrate the exhaust gas analyzer, and set the interlock as required during the test. It can also automatically measure for performance at wide-open throttle, volumetric efficiency characteristics investigations and other engine performance tests.

Under present conditions, however, there is some difficulty in setting the operating conditions and with stability in the control convergence time. These issues will be addressed in the future.

The operating condition items that are set for the EASY automatic testing function are shown in Table 2.

5. Effects of EASY System Introduction

5.1. Improved Data Analyzing Power

The latest engine development efforts have increased the incidences that require cylinder pressure analysis for lean burn limits, combustion speed, and so on. However, the commercially available combustion analysis tools are expensive and they are difficult to use at many test cells at the same time.

The addition of combustion analysis to the EASY system has made it possible to easily implement combustion analysis on many engine test cells at the same time. This has resulted in improved data analysis accuracy.

Development of ordinary mass-production engines also involves combustion analysis. EASY allows on-the-spot comparison with past data and the use of measurement results to substantiate the adjustment of settings, so that combustion analysis can now be used effectively as a development tool for mass-production engines.

5.2. Reduction of Man-hours Required for Data Organization and Analysis

EASY uses files in formats that can be processed in Windows. As a result, it is possible to reduce the time required for data organization and analysis, as shown in Fig. 10.

The use of a company LAN also makes it possible to improve data communications, as shown in Fig. 11.

Table 2 Items of condition for EASY automatic operating functions

Automatic operating functions	Item of condition
Ignition switch	ON/OFF
Exhaust gas analyser switch	ON/OFF
Input/output switch	ON/OFF
Main circuit switch	ON/OFF
Blower switch	ON/OFF
Relay connector	ON/OFF
Engine revolution	Target value
Throttle	Target value
Injected fuel amount	Value of CO or air-fuel ratio
Ignition timing	Control panel or ECU
Quantity of EGR	ECU
Data recording	ON/OFF
Timeout period	Time (s)
Waiting period	Time (s)
Holding period	Time (s)

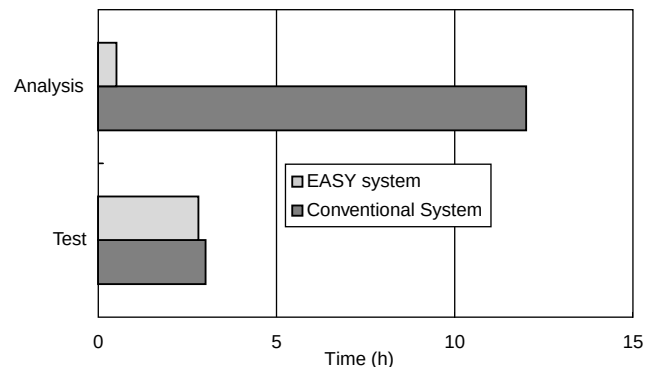


Fig.10 Comparison of time required for cylinder pressure data analysis performed with conventional and EASY systems

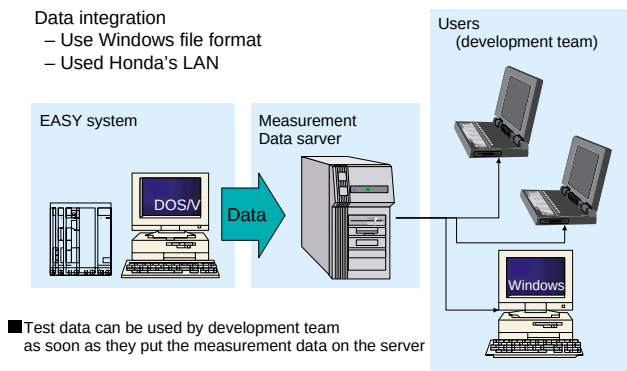


Fig.11 Data communication using Honda's LAN

6. Summary

This EASY engine data acquisition system development project provides universal data communication and interaction among various devices. This makes it possible to utilize a variety of existing equipment, while configuring the system for the PC platform made it relatively painless to install multiple units. The utilization of a real-time operating system to control EASY as a whole made it possible to exercise appropriate control of data acquisition timing and other sequences of actions.

We provided EASY with features such as these in order to provide stable data acquisition, improve the accuracy of data analysis, and enhance the reliability of the acquired data. The following effects were obtained in actual engine development:

(1) Improved accuracy of data analysis

Facilitating combustion analysis also increased the frequency of combustion analysis testing, which brought about advances in the applied analysis of phenomena. This enabled the use of combustion analysis results to substantiate various specification settings and, thus, the use of the system as a development tool.

(2) Improved reliability of acquired data

Tests can now be performed under stable conditions regardless of who is performing the tests. This has eliminated the variations caused when set-up is performed by different individuals, and improved the reliability of acquired data.

(3) Reduced time for testing and data processing

The formats for acquired data have been unified to enable processing on Windows, which reduces the time required for data analysis. Furthermore, it is now possible to perform comparative evaluation of data instantaneously, which means that it is also possible to reduce the time required for the testing itself.

(4) Automatic testing

EASY was given additional functions for performing measurements such as performance tests at wide-open throttle and volumetric efficiency characteristics investigations with the engine operating automatically. This was done to increase the time available for analysis and investigation in order to increase development efficiency.

7. Conclusion

The project succeeded in developing an engine data acquisition system with highly accurate data analysis and reliable acquired data.

Since the quantity of ECU data is likely to increase in the future, it will be necessary to reevaluate the system, especially in terms of enhancing those functions involved in automatic testing, in order to reduce the man-hours required to handle ECU data settings.

As matters now stand with regard to cylinder pressure analysis, the major focus is on engine test cell analysis. However, it is likely that analysis performed on actual vehicles, transient analysis, and simultaneous multiple-cylinder analysis will become increasingly important. Consequently, systems that are capable of data acquisition under those conditions will become necessary.

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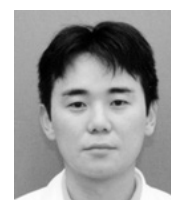
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Development of a Compact CVT for a Hybrid Car – Insight

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ABSTRACT

The continuously variable transmission (CVT) used in the Insight, a hybrid car, was developed using the basic construction of Honda's conventional CVT but with the aim of lighter weight, smaller size, and greater efficiency. This CVT optimizes line hydraulic pressure by utilizing a direct control system for pulley hydraulic pressures and differentiated drive-driven pulley pistons. The mechanism also uses such means as lubricant pressure control to reduce friction in each part, expand the ratio range, and improve the efficiency of engine-motor cooperative control. Further, the utilization of such devices as a forward-reverse select mechanism that operates with a single-pinion planetary gear served to reduce the number of parts.

1. Introduction

The continuously variable transmission (CVT) used in the 1996 model Civic (hereafter the “conventional CVT”) is an automatic transmission which simultaneously achieves both drivability and fuel economy with an original construction which locates the start clutch on the driven pulley axle and by electronically controlling the three factors of starting, shifting and the belt side pressure.⁽¹⁾ (Side pressure is the pressure applied to the V belt in the lateral direction.) The 1999 model further enhanced product appeal by adopting PROSMATEC (Progressive Shift Schedule Management Technology) which performs control to select the optimum gear ratio even on roads with numerous hill climbs and descents, and 4WD was also added as a variation.

The conventional CVT has high fuel economy performance, but research has continued since the Civic's release to further reduce friction, improve efficiency and reduce costs.

When developing the new CVT used in the Insight, the new technologies obtained through this research were combined with the Integrated Motor Assist (IMA) system, and an aerodynamic and lightweight aluminum body to achieve the ultra-low fuel consumption of 32 km/l.⁽²⁾ This CVT has been introduced a number of times thus far^{(3)~(5)}, and thus this paper mainly describes the changes from the conventional CVT.

2. Development Aims

When developing this CVT for use in the Insight together with the IMA system, the aim was to improve fuel economy. Reducing friction and expanding the ratio range are effective means of improving fuel economy.⁽⁶⁾ This CVT reduced the belt friction and the oil pump load, etc., to improve friction and expand the ratio range in order to improve fuel economy, and the weight and size were also optimized for the Insight. An additional aim was to improve product appeal by reducing the number of parts.

3. Basic Specifications and Construction

The basic specifications for this CVT are shown in Table 1. The total length is 23 mm shorter and the weight 7% lighter than the conventional CVT. In addition, the ratio range has

Table 1 CVT specifications

		Insight	Current CIVIC
Transmission ratio	Forward	2.441~0.407	2.466~0.449
	Reverse	4.401~2.441	2.466
	Final	5.688	5.809
Belt axis length (mm)		143	155
Transmission length (mm)		351	374
Transmission weight [wet] (kg)		72.3	79.5

* Tochigi R&D Center

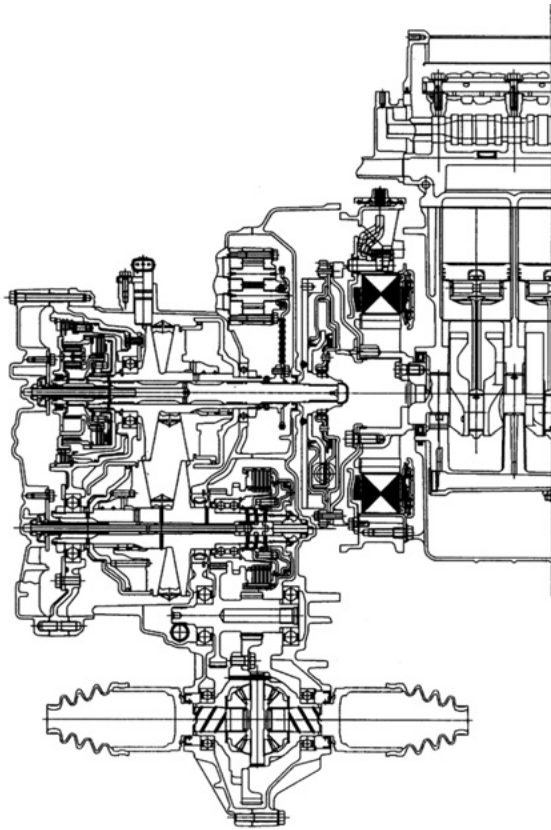


Fig.1 Insight CVT Main section drawing

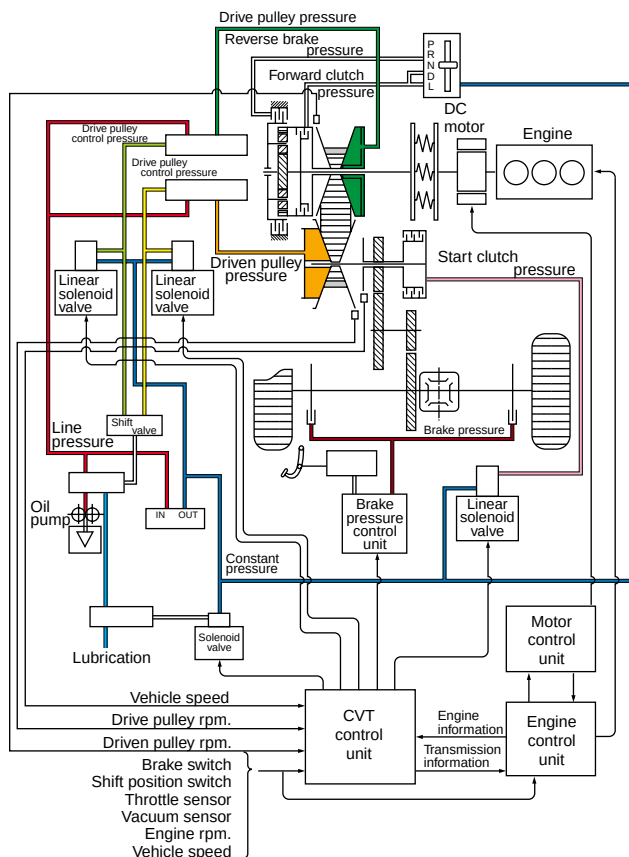


Fig.2 Insight CVT System drawing

been expanded from 5.5 to 6.0.

Fig. 1 shows the main section of this CVT, and Fig. 2 shows the control system.

First, the features of the construction are described below based on Fig. 1.

3.1. Input Block

The engine and motor drive torque are input by a drive plate coupled to the rotor, the torque fluctuation is attenuated by a dual mass flywheel, and the torque is transmitted to the input shaft. A sensor plate for sensing the motor phase is coupled to the drive plate. The input shaft drives the oil pump by a chain like the conventional CVT. In addition, the pitot's flange that was conventionally coupled to the input shaft for start clutch control back-up in the event of electrical system failure has been replaced with a back-up system inside the hydraulic circuit described hereafter.

3.2. Forward-Reverse Select Mechanism

The input shaft is coupled to the forward-reverse select mechanism, and neutral, forward and reverse are selected by the intermittent application of hydraulic pressure to the forward clutch and reverse brake. The forward-reverse select mechanism has been changed from the conventional double-pinion planetary gear to a single-pinion planetary gear. This resulted in the number of gear mesh points being reduced two-third compared to previous, and the working frequency has been halved to reduce high frequency noise and gear noise during reverse, thus improving product appeal. In addition, the single-pinion planetary gear makes the reduction ratio 1.8 times compared to the previous ratio, but the load on the metal belt was reduced and drivability during reverse was improved by the cooperative control with the engine described hereafter and quick shifting. Fig. 3 shows a comparison with the conventional CVT.

3.3. Transmission

After passing through the forward-reverse select mechanism, the drive torque is shifted and transferred from the drive pulley to the driven pulley by a metal belt. The basic CVT specification of the pulley axle center distance was set at 143 mm which is the optimum value in terms of the IMA output characteristics compared to 155 mm for the conventional Civic. This made it possible to reduce the housing and pulley circumference size and the weight.

The metal belt was made by Van Doornes Transmission, and had a new element shape with the ring composition changed from two rows of 12 rings to two rows of 9 rings to reduce friction. The drive and driven piston shapes used different outer diameters to obtain the greatest effect from the direct control system for pulley side pressures, and the oil pump load when cruising was reduced by setting the area of the drive side piston which normally has greater thrust force to 1.3 times that of the driven side piston. (See Fig. 4.)

3.4. Start Clutch

The basic construction is the same as the conventional CVT. However, when restarting from idle stop, the hydraulic pressure must be raised quickly from the condition where hydraulic pressure is not produced to allow starting.

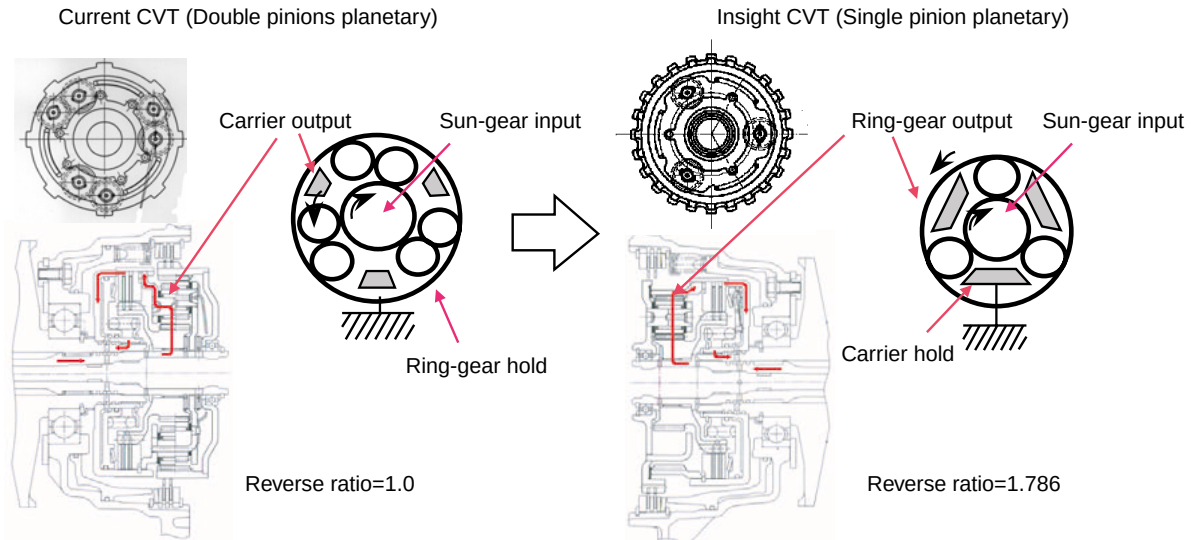


Fig.3 Reverse system comparison

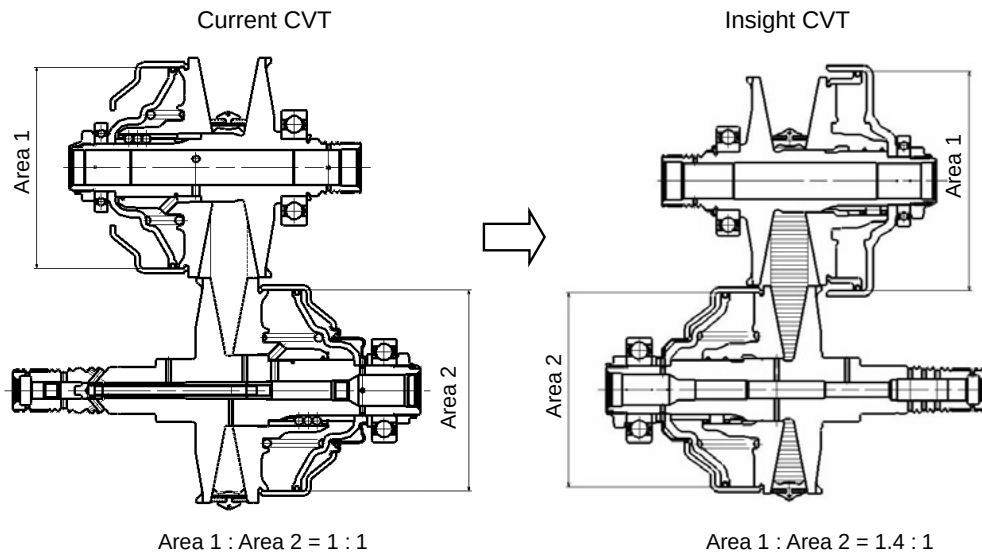


Fig.4 Pulley system comparison

Therefore, the sealing properties of the valve body were improved, dead volume was reduced by shortening the hydraulic circuit, and the control accuracy was improved to allow quick and smooth starting.

3.5. Final Reduction

Changes in gear noise caused by torque fluctuation when decelerating were reduced by setting the twisting direction of the secondary gear with respect to the rotating direction opposite that of the conventional CVT and by decreasing the relative displacement of lead slant when decelerating. The differential gear size was reduced and weight reduction for the differential case was exhaustively pursued by FEM analysis resulting in a weight reduction of approximately 5% compared to other differentials in the same class.

3.6. Hydraulic Pressure Control System

Next, the control system is described. Fig. 5 shows the hydraulic circuit diagram. The conventional CVT is

comprised of a side pressure control system which controls the pulley side pressure and a shift control system which controls the thrust force of the drive and driven pulleys. However, in the new CVT this has been changed to a direct control system for pulley side pressure which controls the drive and driven pulley side pressures independently. This makes it possible to optimize the drive and driven pulley hydraulic pressures in any operating condition, thus greatly reducing the oil pump load. Fig. 6 shows a comparison with the conventional system.

The hydraulic pressure control system has the configuration shown below.

3.6.1. Pulley Side Pressure Control System

The conventional CVT control system used a four-way valve to distribute the control hydraulic pressures of the two high pressure and low pressure systems to the drive and driven pulleys according to the input torque and gear ratio. The newly adopted direct control system for pulley side pressures adjusts the line pressure according to the signal

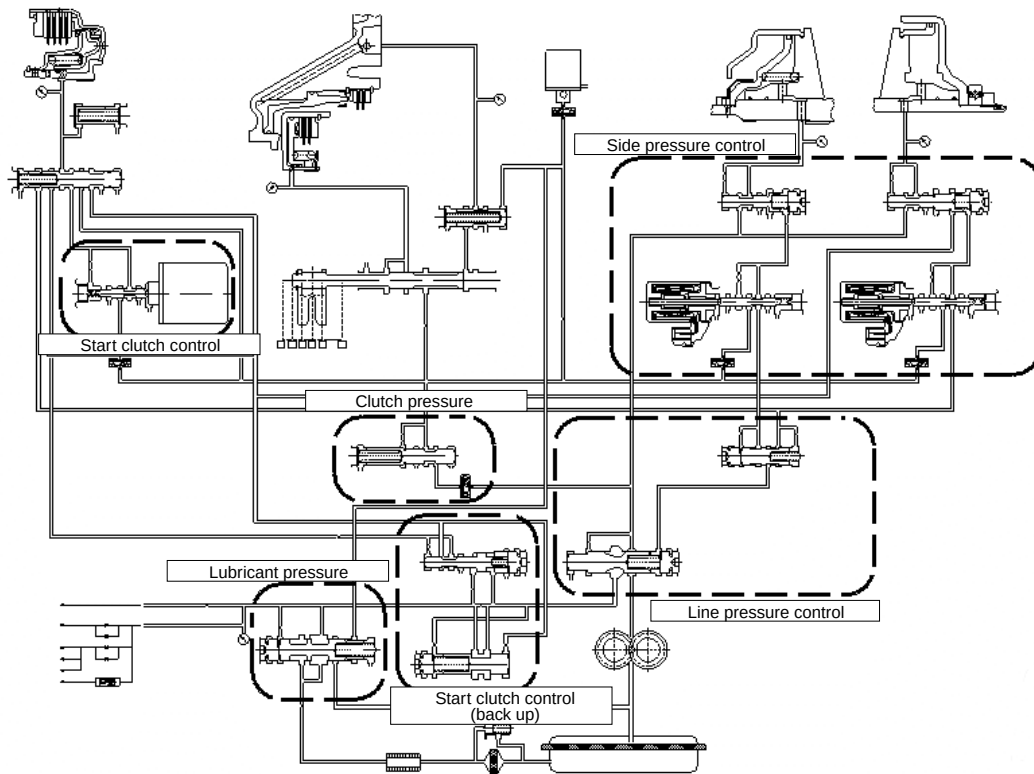


Fig.5 Hydraulic circuit

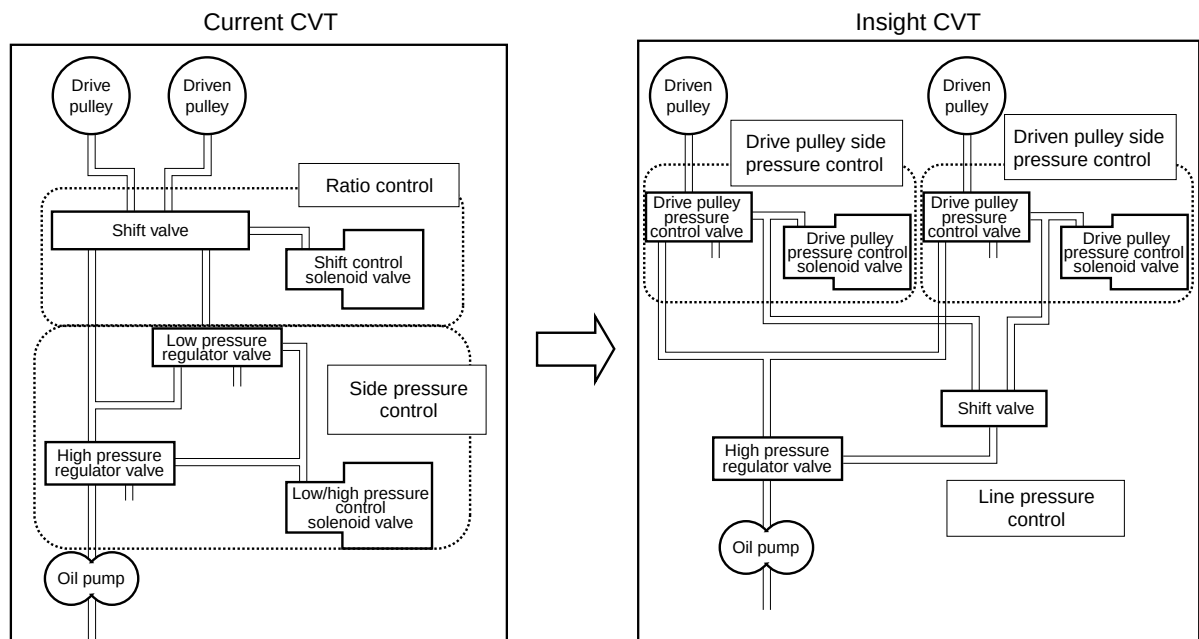


Fig.6 Pressure control system

pressures of independent linear solenoid valves and supplies the drive and driven pulley side pressures to the respective pistons.

3.6.2. Line Pressure Control System

The conventional system has a construction which controls the low pressure, which is the side pressure needed for the pulleys to transmit torque, and the high pressure, which adds to this the hydraulic pressure difference needed

for shifting according to the signal pressure of the linear solenoid valves. This hydraulic pressure difference is the pressure difference needed for quick shifting, and becomes excess hydraulic pressure during normal cruising. In the new CVT, the hydraulic pressure from the oil pump is adjusted using regulator valves and supplied to the drive pulley control valve and driven control valve. The regulator valves select the higher of the linear solenoid valve signal pressures that

control the drive and driven pulley hydraulic pressures with a shift valve, and control the line pressure as the control signal pressure. This means that the line pressure has the minimum margin pressure added to the necessary pulley pressure, thus reducing excess hydraulic pressure. (See Fig. 7.)

3.6.3. Clutch Pressure Control System

The clutch pressure is supplied as hydraulic pressure which has been attenuated by a modulator valve with the line pressure as the hydraulic pressure source. This hydraulic pressure is used as the hydraulic pressure source for the linear solenoid valves which control the forward clutch, reverse brake and start clutch.

3.6.4. Start Clutch Control System

The start clutch is controlled by adjusting the hydraulic pressure with a linear solenoid valve in the same manner as the conventional CVT.

3.6.5. Lubricant Pressure Switching Control

Lubricant pressure switching control was adopted to reduce the oil stirring friction. Lowering the oil surface height inside the transmission as much as possible is an effective means of reducing the stirring friction. The oil surface height is determined so that aeration does not occur at low temperatures, but the oil surface rises causing the stirring

friction to increase at high temperatures. Therefore, the stirring friction was reduced by controlling the lubricant pressure and lubricant flow rate to thereby optimize the oil surface height. The lubricant pressure is set by controlling the solenoid valve according to the oil temperature and vehicle speed conditions and switching the signal pressure to the lubricant pressure control valve. This control is realized without adding devices by the combination of the conventional reverse inhibitor solenoid valve control logic.

3.6.6. Start Clutch Back-up

This control block comprises a system for switching from electronic control to hydraulic pressure control to assure operating performance as a back-up function for the electronic control system. The back-up system for the conventional CVT controlled the start clutch with a governor circuit which used the total pressure corresponding to the engine speed produced by the pitot's flange provided on the input shaft. The pitot's flange constantly rotates and thus it is a source of friction. In this CVT, a system having the same functions as the pitot system was provided inside the control body. Focusing on the fact that the lubricant flow rate changes according to the engine speed, the start clutch pressure is controlled according to the engine speed by applying the difference in hydraulic pressure produced by the lubricant passing through an orifice to both ends of the back-up clutch pressure control valve as shown in Fig. 8. This eliminated the need for a pitot's flange, thus reducing friction and also the number of parts.

3.7. Electronic Control System

The CVT control unit processes a variety of input signals from various sensors and the engine control unit, and then drives the three linear solenoids and one ON/OFF solenoid that control the CVT, and the two ON/OFF solenoids for the auxiliary brake system for starting on hills. In addition, the CVT control unit communicates with the engine control unit to perform cooperative control related to idle stop. The CVT control unit functions are described below.

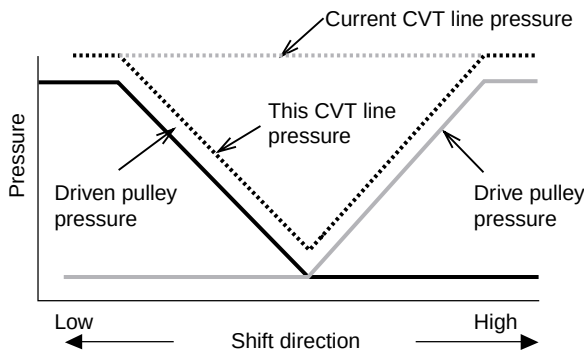


Fig.7 Pulley and line pressure

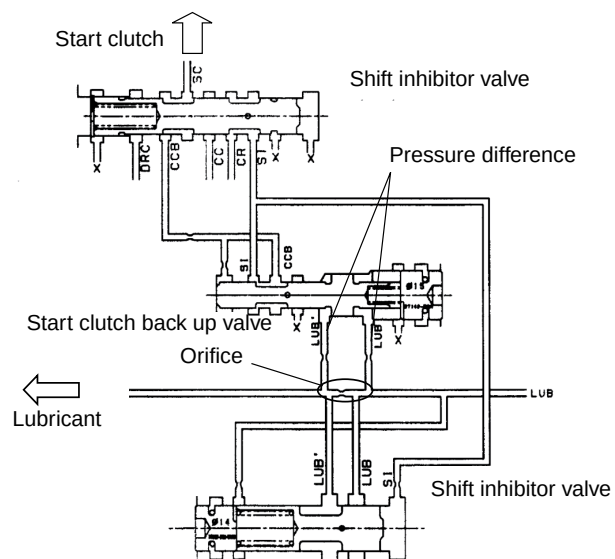
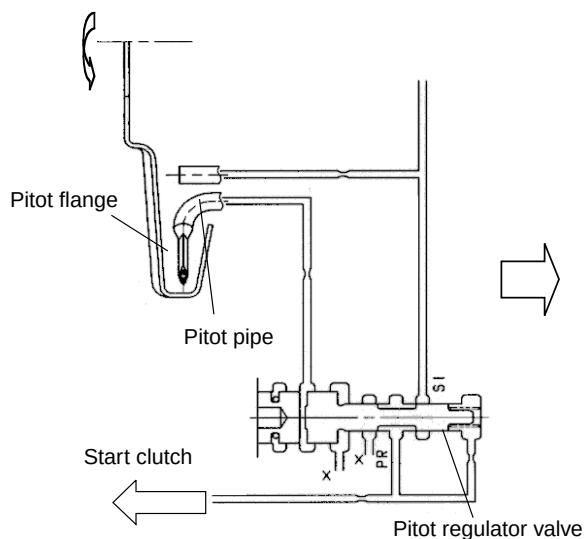


Fig.8 Starting clutch back up system

3.7.1. Input Sensors

A drive pulley rotation sensor, driven pulley rotation sensor, vehicle speed sensor and speedometer sensor (back-up) are used as the rotation sensors for each part. In addition, an engine rotation signal is obtained from the engine. The transmission rotation sensor was changed to a semiconductor type to improve rotation detection accuracy and allow detection at extremely low vehicle speeds.

The operation system sensors detect the shift position signal, mode switch signal, throttle opening angle signal and brake signal. The brake sensor determines whether the brake pedal is depressed, and is used for the auto idle stop judgment described hereafter and the engine restart signal. In addition, the engine intake pipe negative pressure is detected to estimate the engine output torque.

3.7.2. Output Devices

Three linear solenoids for controlling the drive pulley pressure, driven pulley pressure and start clutch pressure, and one ON/OFF solenoid used for both reverse inhibitor control and lubricant pressure switching are provided inside the CVT as output devices. In addition to these devices, the brake hydraulic pressure is maintained and released by two ON/OFF solenoids which control the auxiliary brake system for starting on hills. Creep torque is not produced during auto idle stop which automatically stops the engine while idling, so this system maintains the brake pressure for a certain time to ensure the same operability as the conventional system even when starting on hills from the idle stop condition.

3.7.3. Communication System

The CVT control unit performs cooperative control by exchanging various information with the engine control unit. The exchanged information includes the intake pipe negative pressure, motor torque and other information from the engine control unit, the shift position signal and retard request during

reverse from the CVT control unit, and the idle stop signal from both units.

3.7.4. Pulley Side Pressure Control

Conventional control consisted of side pressure control which calculated the target pressure with the input torque and ratio as parameters and controlled the control valves, and shift control which controlled a four-way valve to achieve the target engine speed. However, the side pressure control system in this CVT uses direct control of the pulley side pressures and has the control configuration shown in Fig. 9 to independently control the drive and driven pulley pressures.

The necessary thrust force is calculated from the pulley ratio and input torque (total of the engine torque and motor torque). The shift schedule is determined by the target engine speed according to the vehicle speed and throttle opening angle, and the thrust force ratio required for this control is set according to a thrust ratio map. The thrust for shifting is fed back to correct the error with the target engine speed. In this manner the target drive and driven pulley thrusts are obtained, the respective control solenoid drive currents are determined and the pulley side pressures are controlled.

3.8. CVT Control during Auto Idle Stop

During auto idle stop, the CVT control unit sends the auto idle stop signal and the regeneration for engine stop signal for stopping the engine smoothly to the engine control unit. When restarting, the clutch engagement pressure corresponding to the start clutch residual pressure is controlled to obtain a smooth starting feeling.

3.9. Oil Filter

Improving durability with respect to contamination of the hydraulic pressure control system is an important issue for transmissions which perform high accuracy, electronic

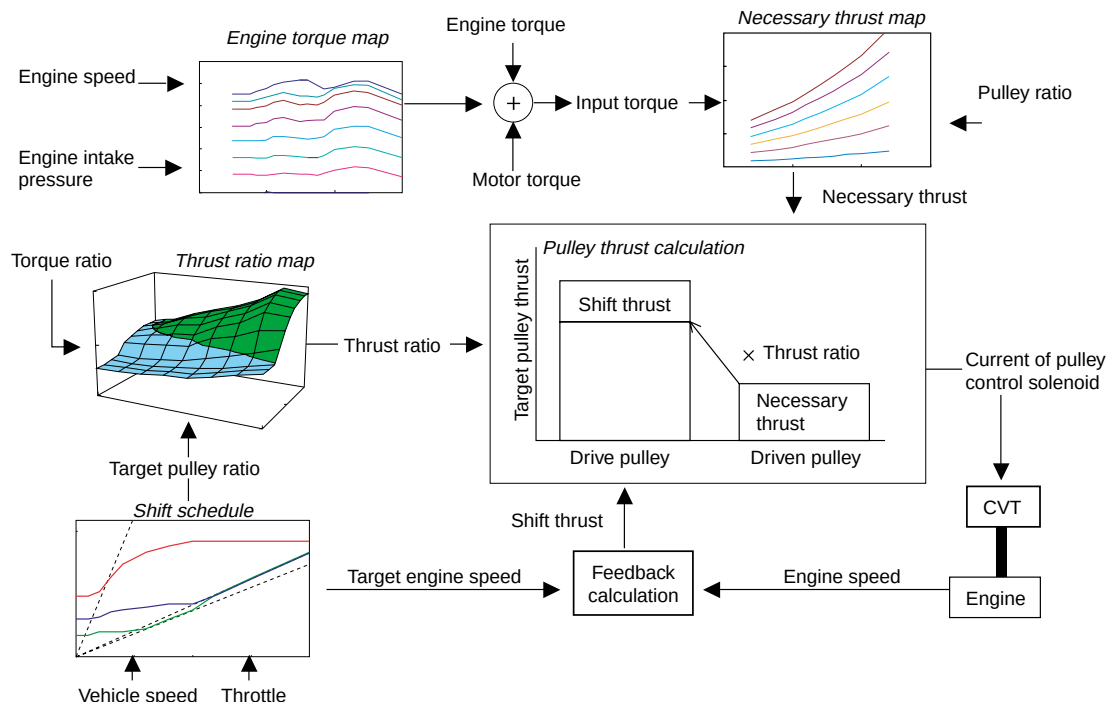


Fig.9 Pulley control block diagram

control using actuators. This CVT modified a non-woven fiber to improve the filtration performance of the oil filter. Increasing the filter material density and other methods of improving the normal filtration performance create such issues as increased pressure loss and shorter life due to early clogging. Therefore, the filtration performance was improved by increasing the types of fiber in the dual-layer non-woven fiber from the conventional one type per layer to multiple types per layer to reduce the gap between fibers, and the housing rib shape was optimized to prevent the non-woven fiber from sticking. These improvements increased the filtration efficiency from 1.5 to 2 times that of the conventional oil filter without increasing pressure loss or shortening the life. Fig. 10 shows a comparison of filter efficiency with the conventional oil filter. In addition, the housing was changed from pressed steel plate to plastic, thus reducing the weight by 50% compared to the conventional oil filter housing.

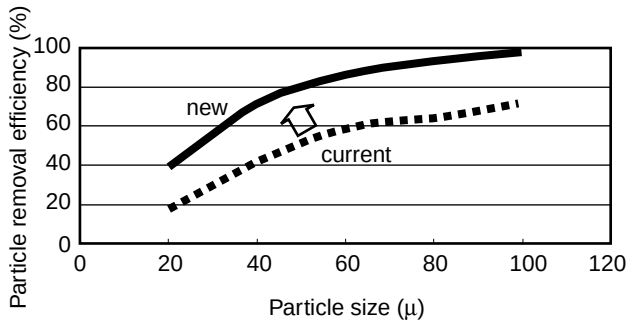


Fig.10 Oil filter efficiency

4. Adopted Technologies

Next, the three technologies adopted in this CVT are described.

4.1. Broad Gear Ratio

Setting a higher ratio allows the engine speed to be reduced when cruising and also during decelerating regeneration for hybrid vehicles, which is extremely advantageous for improving fuel economy. The low ratio is determined according to hill climbing capability which is obtained by the vehicle weight, tire diameter and engine output, and product appeal in terms of starting performance. However, a high top ratio is demanded to improve fuel economy. Therefore, the ratio range must be widened to simultaneously achieve both types of performance.

Expanding the ratio range of standard transmissions causes the step ratio to increase, so this is generally done by increasing the number of steps. However, this increases the number of parts which makes the transmission bigger and heavier. Metal belt CVTs can change the ratio range without increasing the number of parts by reducing the belt winding radius, but they are also limited by the following factors:

- Belt durability
- Shaft strength due to a narrower shaft diameter
- Increased length in the transmission axle direction

The ratio ranges of Honda and other automobile CVTs

sold prior to the Insight were approximately 5.0 to 5.5.

Therefore, the above factors were resolved as follows.

The issue of belt durability was solved by changing the over drive ratio according to the load. A ratio of up to 0.450 was used at high loads, and a high ratio of up to 0.407 was used at low loads, while cruising or during decelerating regeneration. Fig. 11 shows the shift map.

The issue of shaft strength was solved by performing FEM analysis to reduce the shaft diameter. In addition, the slide spline ball was replaced with a roller to shorten the dimension in the transmission axle direction while still ensuring sufficient power transmitting capacity. Furthermore, this greatly improved the power transmitting capacity, so the number of spline grooves was reduced from the conventional three grooves to a single groove.

These measures expanded the ratio range from the 5.5 of the conventional CVT to 6.0, and helped to improve fuel economy while cruising.

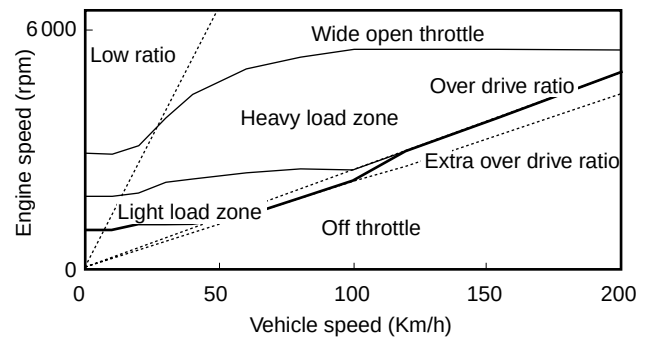


Fig.11 Shift limit lines

4.2. Direct Control for Pulley Side Pressures

The following issues were solved by adopting differentiated pulley pistons and a direct control system for pulley side pressures.

4.2.1. Reducing Hysteresis between Current and Hydraulic Pressure Characteristics

Reducing the pulley side pressure variance relative to the command current value reduces the specified side pressure margin with respect to the actual required side pressure, that is to say the excess hydraulic pressure. This also reduces the belt rotation friction, and is advantageous for improving belt durability. The magnetic materials of the linear solenoids used in this CVT were changed to materials with low hysteresis characteristics, which reduced the pulley pressure variance relative to the command current value by approximately 30% compared to the conventional CVT.

4.2.2. Reducing Hydraulic Pulsation

Low frequency hydraulic pulsation produced by the delay in the pilot pressure response relative to the pulley pressure fluctuation was formerly an issue. This low frequency hydraulic pulsation directly influences the pulley thrust force fluctuation, so the solenoid valve response was improved.

The improvement method consisted of investigating the shape of the pressure adjusting end in terms of the balance between improving the solenoid valve flow rate characteristics and reducing the self-excited vibration of the

valve, and optimizing the spring constants.

4.2.3. Reducing Line Pressure Surge

The direct control system for pulley side pressures has a configuration which controls three hydraulic pressures with two solenoids, and the higher pulley pressure control solenoid also simultaneously controls the line pressure. Therefore, continuous surge tended to occur at low oil temperatures due to the delay in response relative to the high pressure side line pressure fluctuation. To resolve this issue, the damping characteristics of the line pressure control valve were optimized.

As a fail-safe measure, this system uses a reverse theory system which produces the maximum pressure in the condition where the control power supply has been cut off. Therefore, surge produced by the valve operation delay when starting the engine was also an issue. This surge was reduced by optimizing the command hydraulic pressure.

Resolving the above issues made it possible to use direct control and to reduce the excess hydraulic pressure. In addition, the above-mentioned lubricant pressure switching control, elimination of the pitot's flange and other measures greatly reduced friction and improved transfer efficiency. Fig. 12 shows a comparison of friction when cruising.

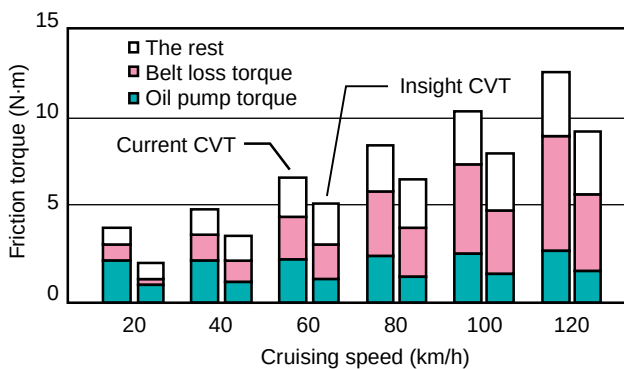


Fig.12 Transmission friction

4.3. Control of the Forward-Reverse Select Mechanism Using a Single-Pinion Planetary Gear

The forward-reverse select mechanism uses a planetary gear in consideration of easy operation and smooth switching. The conventional CVT used a double-pinion planetary gear which allowed the same gear reduction ratio to be set for both the forward and reverse steps to facilitate side pressure control. However, the large number of parts created the issues of increased cost and weight. This CVT focused on reducing weight, and was able to use a single-pinion planetary gear by performing cooperative control between the engine and CVT. The issues involved in using a single-pinion planetary and methods for solving these issues are described below.

The single-pinion planetary gear of this CVT uses a sun-gear input/ring-gear output to form the reverse gear shape. In this case, the reverse gear ratio is 1.786 times the forward gear, and the torque during reverse is amplified, resulting in insufficient side pressure.

The belt ratio is low in the in-gear condition during reverse. When the accelerator pedal is depressed the necessary side pressure is reduced, so the belt is shifted to the target ratio. In this case some time is required to shift the belt from the low ratio to the target ratio, so insufficient side pressure may occur during the shift. In addition, excess drive power may be output with respect to forward, thus worsening drivability. This CVT locates the start clutch on the driven pulley side, so the power transmitting capacity is controlled to the same degree as during forward to ensure sufficient side pressure. In addition, cooperative control is performed to control the engine speed so that there is no feeling of an abnormal rise in rpm and to prevent an unnatural feeling in terms of drivability when starting.

Fig. 13 shows the control logic for engine/CVT cooperative control during reverse starting. When the accelerator pedal is depressed from the reverse gear in-gear

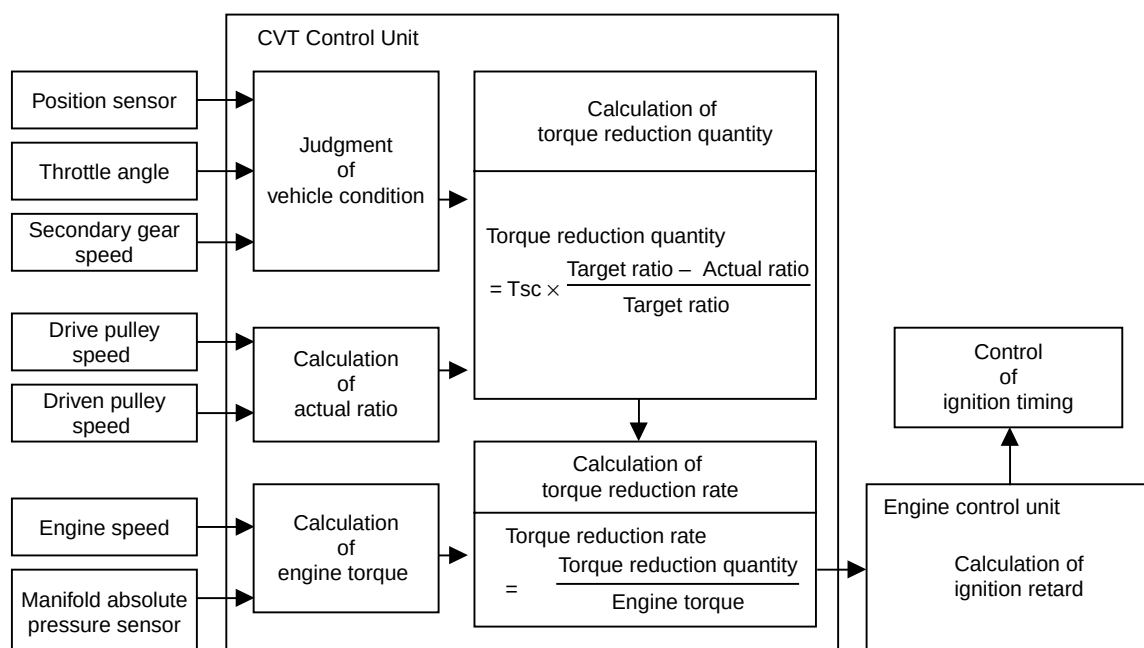


Fig.13 Control logic at reverse starting

condition, the torque reduction quantity is calculated from the actual ratio and the engine speed. This is compared with the conventional engine torque and the results are transmitted to the engine control unit as the torque reduction rate. The engine control unit determines the ignition retard timing from the torque reduction ratio and performs control to reduce the engine torque. This provides the same starting performance as the forward gears. Fig. 14 shows the change in each parameter during this process.

Performing the cooperative control described above made it possible to reduce both the weight and number of parts by adopting a single-pinion planetary gear without significant changes to the conventional system. Additional merits were an increase in creep force during reverse and lower noise.

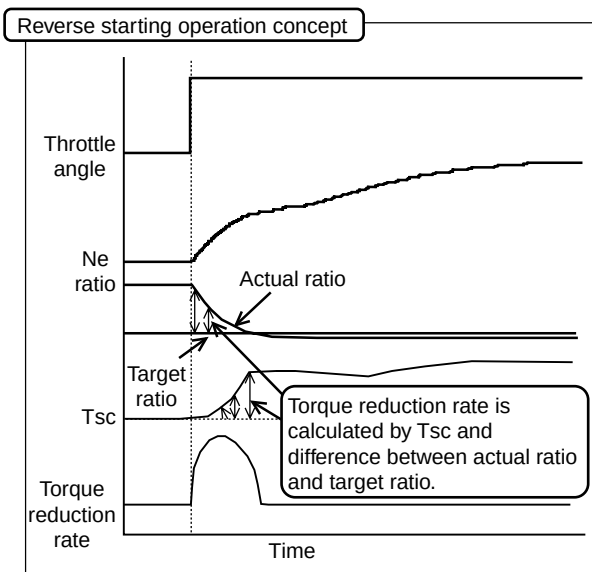


Fig.14 Reverse starting operation concept

5. Conclusion

The line pressure was reduced through a direct control system for pulley side pressures, friction was reduced by lubricant pressure switching control, etc., and shift control with a broad gear ratio was adopted. These measures helped to achieve a fuel economy of 32 km/l in the 10-15 mode, which is the top-level in Japan for automatic vehicles.

In addition, forward-reverse selection with a single-pinion planetary gear and other changes reduced the weight by 7% and the number of parts by 12% compared to the conventional CVT.

The technologies used in this CVT were developed for hybrid vehicles, but can also be used in CVT for conventional vehicles, and we are certain that we have developed a compact new CVT which will serve as the basic framework for CVTs in the 21st century.

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Active Control System for Low Frequency Road Noise Combined with an Audio System

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ABSTRACT

An active control system for low frequency road noise in automobiles combined with an audio system was developed as a commercial application for the first time in the world. The system was installed in a station wagon. The purpose of this paper is to provide an outline of the system and describe the newly developed cost reduction technology used therein. A major factor in the successful application of active noise control technology to a commercial product is reduced system cost. The methods used to reduce costs include utilization of feedback control, implementation by analogue circuits, and common use of audio system speakers. This system reduces low frequency road noise in the front seat by about 10 dB and improves the audio system listening experience while driving.

1. Introduction

Recent years have brought increased emphasis on vehicle weight reduction in order to achieve further fuel economy in automobiles. However, countermeasures for reducing noise in automobiles result in an increase in weight. Active noise control (ANC) technology reduces noise by adding anti-phase noise to potentially solve these contradictory issues. Significant research in ANC application to automobiles has been taking place. However, ANC has not achieved widespread application because of its unfavorable cost performance^{(1), (2)}.

In order to improve cost performance, we concentrated on the cost of the ANC system. As a result, an active control system for low frequency interior road noise (drumming) combined with an audio system was developed and installed in a mass-production station wagon. Although it is limited to the lower frequency ranges, this is the world's first ANC system for road noise and the first combined ANC and car audio system.

The specific methods used to reduce the cost include utilization of feedback control (FB), implementation of analogue circuits, and the common use of car audio speakers. This paper briefly surveys the history of ANC technology, provides an overview of the mass-production system and discusses the newly developed ANC.

2. ANC Technology in Automobiles: A Brief History and Methods for Solving Practical Application Issues

This section briefly reviews the history of ANC technology in automobiles from the perspective of research and development (R&D) and control methods. It also discusses approaches to the solution of issues for practical application of this technology.

2.1. The State of R&D on ANC

ANC R&D became active from the 1980s, in conjunction with the rapid development of hardware such as digital signal processors (DSP). Active R&D on automotive applications for ANC began in the latter half of the 1980s⁽³⁾.

Examples of this technology applied to periodic noise include applications for engine booming noises in the cabin^{(4), (5)}, mufflers, and intake ducts^{(6), (7)}. A paper on practical applications for engine booming noise in the cabin has already been published⁽⁸⁾. Although not related to automobiles, there is an example of practical application of ANC for periodic noise in Shinkansen train cars. Here, an ANC system reduces noise near passenger ears⁽⁹⁾.

A typical example of random noise is generated inside automobiles by input from the road surface while the vehicle is moving. There has been active research on road noise⁽¹⁰⁾⁻

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⁽¹²⁾, but due to unfavorable cost performance, many issues related to practical application still remain.

2.2. Control Methods

The control method mainly used in the ANC field is adaptive feedforward (FF) control. This method is characterized by the use of an error signal for adaptive signal processing and a reference signal for feedforward control. FF control is extremely effective with ANC for engine induced booming noise. Here, the ignition pulse can be used as a reference signal. However, road noise is caused by a variety of input sources. This results in a large number of reference signals. Thus, the high cost of sensors and controllers capable of handling the enormous amount of adaptive signal processing become an issue. In addition, the question of where to place the numerous reference signal sensors is also an issue.

On the other hand, FB control is also available as a control method. However, in ANC applications, the use of FB control has been limited to headphones, where the speakers and microphones are very close together, and to duct ANC utilizing the tight coupled monopole (TCM) method⁽¹³⁾. The controllers used in these applications are only very rarely designed from a control engineering perspective, and the lack of knowledge about just how these controllers are designed can present issues. In this paper, designing a controller from a control engineering perspective means modeling the control object using various control theories (classic control theory, modern control theory, and post-modern control theory as represented typically by robust control theory) and designing a controller using the model obtained.

The only reported research⁽¹³⁾ on FB-ANC in automobiles appears to be on the use of adaptive signal processing for wind throb noise, which can be heard, for example, when the sunroof is open. There are no previous examples of ANC application to a three-dimensional closed sound field such as an automobile interior using controllers designed using control engineering methods. The status of practical application of ANC technology is summarized in Table 1.

Table 1 Examples of practical use of ANC

Acoustic condition	Feedforward control	Feedback control
1-dimension	• Headphone • Duct of air-conditioner • Refrigerator	Headphone
3-dimension in closed space	• Engine booming noise in a car • Motor booming noise in a train	Nothing

2.3. Approaches to Solving Issues in Practical Application

One of the factors that determine the value of a technology is cost performance. In other words, practical application of ANC can be achieved either by reducing cost or by significantly improving performance. Since the strength of ANC lies in its noise reduction capabilities below about 500 Hz, the use of ANC to achieve quietness that extends to higher frequency ranges would be very problematic. Even if this were achieved, the cost would be unrealistic. Therefore, the authors have directed their efforts toward cost reduction.

First, the research is focused on control of drumming, the low frequency component of road noise. The use of passive techniques to reduce drumming entails vibration control by such means as reinforcement of the car body frame⁽¹⁵⁾. This involves increasing weight. The development objective is, therefore, to use ANC technology to realize cost and weight constraints on par with passive countermeasures. The next section will present an overview of the system and the specific methods used to reduce costs.

3. Overview of Mass-Production System and Cost-Reduction Methods

3.1. System Overview

The acoustic first-order mode shown in Fig.1, and is calculated by eigen value analysis. It shows that a drumming noise tends to be generated at around 40 Hz in the front seat of a station wagon. This is due to the sound field determined by the cabin configuration. The yellow and red portions in the figure show the loop while the green and blue portions show the node.

The drumming noise of around 40 Hz heard in the front seat is the noise to be controlled. Development was initially directed toward cost reduction of the adaptive FF control. However, even with the focus narrowed to drumming, ANC implementation using adaptive FF control required multiple reference signal sensors, error sensors, and a digital controller employing a DSP. Elimination of noise at around 40 Hz further presupposed the use of woofer speakers. The resulting total cost showed that practical application would be difficult because system cost would be much higher than passive countermeasure technology.

This development project utilizes a number of different methods to achieve reduced cost. The control method of the system adopted for mass production is shown in Fig.2. The configuration of the system as installed in a car is shown

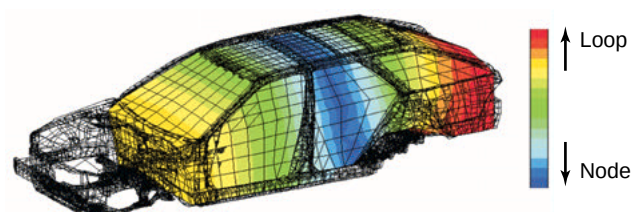


Fig.1 An acoustic mode of a station wagon at 40Hz

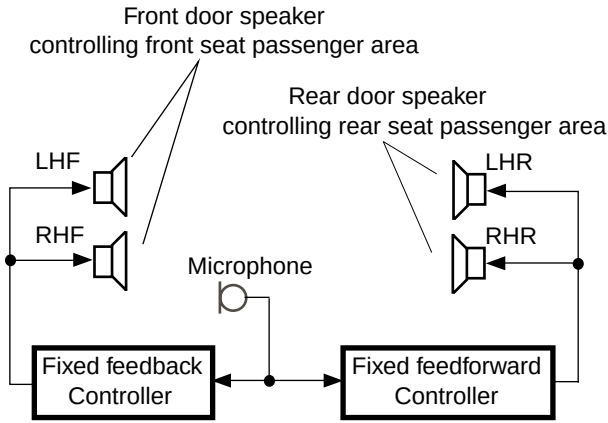


Fig.2 Control strategy in the car

in Fig.3. The control method will be outlined using these figures. Further details of the control technology developed will be provided in section 4.

The electronic control unit (ECU) is placed under the driver's seat. Results of the 40Hz sound field analysis in Fig.1 show that the area from the floor to the roof in the front passenger area is generally in phase. It is clear, therefore, that placing the microphone under the driver's seat would allow reduction of the drumming noise for the front seats. Consequently, the microphone is installed on the ECU circuit board. The fixed FB controller and the front door speakers are then used to reduce drumming in the front seats. Moreover, fixed FF control and the rear door speakers are used to control the resulting amplification in noise in the rear seats. In other words, the microphone installed under the driver's seat is used by FB control as an error microphone (error sensor) and by FF control as a reference signal microphone (reference signal sensor).

The results of sound field analysis also show that there is virtually no phase difference between left and right. Therefore, the FB controller for the front seats simultaneously drives the left and right front door speakers. The FF controller for the rear seats also drives the rear door speakers simultaneously.

The next topic is approaches and methods for cost reduction.

3.2. Approaches and Methods for Cost Reduction

Cost reduction is carried out for the key components for control: sensors, controllers, and actuators (speakers). The cost reduction methods implemented for this system can be summarized as follows:

- (1) Cost reduction methods for sensors and controllers
 - Use of FB control to reduce noise in the front seats and FF control to limit the increase of noise in the rear seats
 - Adoption of fixed control rather than adaptive control
 - Implementation of analogue circuits
 - Use of a microphone as a sensor and mounting this microphone on a circuit board
- (2) Cost reduction methods for speakers
 - Common use of standard car audio system speakers

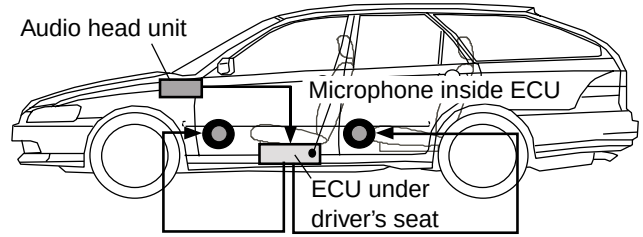


Fig.3 System installation configuration

- Insertion of a limiter circuit to limit output level when sound pressure is excessive

These cost reduction methods are explained below.

3.2.1. Cost Reduction for Sensors and Controllers

We adopted FB control as the front seat control method in order to significantly reduce the number of sensors. FB control allows significant reduction of the number of sensors because it does not require reference signals and only uses an error sensor. Since FB-ANC is a form of sound field control, it uses microphones as error sensors (error detection sensors). Microphones are less expensive than acceleration and force sensors. However, much less research has been performed on FB-ANC than on FF-ANC. There are two reasons for this, as follows:

- (1) The non-collocation of a speaker and a microphone means that the control object (the plant) becomes a non-minimum phase system. This makes it difficult to reduce noise over a broad range of frequencies, and further, it involves an inevitable waterbed effect^{(16), (17)}, whereby sounds at frequencies above the noise reduction range are increased.
- (2) Since a sound field of a three-dimensional closed space such as a car cabin has reverberation, the plant is complex. This increases the difficulty of modeling the plant and designing a controller.

Despite these conditions, research to investigate the possibilities of FB-ANC has been carried out by the authors⁽¹⁸⁾⁻⁽²²⁾ and the Ohki team⁽²³⁾. It was determined that by restricting the control range, the noise amplification range caused by the waterbed effect can also be limited. Thus, FB-ANC proves to be effective and consequently, the authors adopted FB control for reduction of drumming in the front seats⁽²⁴⁾.

Both the FB and FF controllers are implemented using analogue circuits. Currently, analogue circuits still have a significant cost advantage over digital circuits. In addition to this cost advantage, analogue controllers are continuous time controllers so they are not affected by the sampling delay found in digital controllers nor by the delays of anti-aliasing filters, reconstruction filters, and other such analogue circuits. Therefore, analogue controllers have the advantage that FB speed control is extremely fast.

On the other hand, since the voltage of the microphone signal is very low, the wire from the microphone to the ECU must be shielded when the microphone is installed remotely. However, shielded wire is more expensive than ordinary wire. Therefore, in this development, the sound field analysis

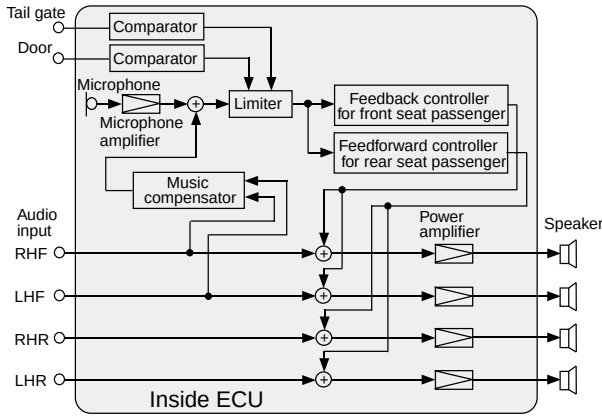


Fig.4 The system configuration in ECU

shown in Fig.1 is performed, demonstrating that placement of a microphone under the driver's seat would allow reduction of drumming in the front seat area. The microphone is mounted on the ECU circuit board under the driver's seat to eliminate the need for a microphone wire, to reduce microphone-related costs, including the cost of wiring, and additionally to improve the controller's robustness with regard to electrical noise.

3.2.2. Cost Reduction for Speakers

Implementing noise reduction at around 40 Hz on all types of road surfaces requires speakers with high-performance woofer characteristics. This is disadvantageous in terms of cost. However, standard car audio system speakers (standard speakers) are now provided with good performance in order to improve audio system quality. Further, these speakers are capable of reproducing sounds in the vicinity of 40 Hz. Noise reduction tests by ANC are conducted with this in mind. The results show that although noise reduction is difficult for standard speakers when driving on surfaces that generate large inputs, such as a Belgian road or a road with large bumps, standard speakers function well on roads that tend to generate a bothersome drumming noise. In this development project, the limiter function described below is introduced to an ANC circuit. This enables the ANC to share standard audio system speakers.

Fig.4 shows the system configuration of the ECU. The limiter circuit is inserted after microphone output. When sound pressure measured by the microphone exceeds a preset specified value, the circuit significantly limits the amplitude of that signal, so that ANC operation is minimal. As a result, when the car is driving over an extremely rough surface, when the audio system volume is high, or when the microphone input is at very high pressure, the limiter circuit both prevents speaker distortion and improves speaker durability.

3.3. Benefits for the Car Audio System

Since this system utilizes FB control, it is necessary to include a music compensation circuit as described next. If this circuit is not included the system will reduce not only

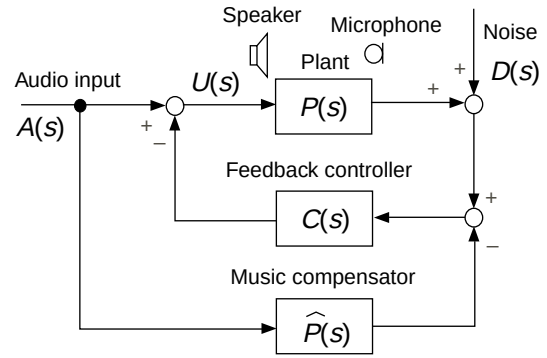


Fig.5 A music compensator

the noise at around 40 Hz but also the components of music in that frequency range as well. To prevent this from occurring, the music compensator shown in Figs.4 and 5 is added. This circuit identifies the transfer characteristics of the acoustic transfer system between the speakers and the microphone in advance. It then subtracts the music component generated by the audio head unit from the microphone output signal. In this way, the system operates without canceling out the music. The basic principle is feedforward control in order not to erase the music signal. This is equivalent to the internal model control (IMC)⁽²⁵⁾ in control engineering and the echo canceller in telecommunications⁽²⁶⁾.

In Fig.5, $P(s)$ is the plant. It represents the transfer function of the acoustic path between the front door speakers and the microphone. The $\hat{P}(s)$ estimating transfer function of $P(s)$ is now sought. If $P(s) = \hat{P}(s)$, then

$$U(s) = A(s) - \frac{C(s)}{1 + P(s)C(s)} D(s) \quad (1)$$

It is apparent that the audio signal $A(s)$ is not influenced by the controller characteristic $C(s)$.

The music compensator and ANC effect generate the following benefits for the car audio system:

- Clarity of bass sounds
- Improved music listening quality provided by the masking effect

These benefits add value for the user as well as reducing drumming noise.

3.4. Replacing the Audio Head Unit

A number of different audio head units are offered as options for vehicles that apply this system. Therefore, the ANC system must be able to operate with any of these units. A power amplifier is consequently added in the ECU, as shown in Fig.4, in order to make it possible for the user to exchange audio head units. Unfortunately, this feature leads to increased ANC costs. Cost reduction can be sought in vehicles that are not likely to have the audio system exchanged, such as high-end cars, by sharing the internal power amplifier in the audio head unit.

3.5. Howling Prevention

The use of FB control means that howling can occur when the control system becomes unstable. Major factors influencing stability are the plant transfer characteristics. Here, since the plant is the transfer characteristics from the speakers to the microphone, the plant transfer characteristics are most severely affected when the doors and tailgate are open. This system has sufficient enough stability that howling will not occur even when a door or tailgate is open. In order to avoid any risk, however, the limiter shown in Fig.4 operates when a door or tailgate is open.

4. New ANC Technology

This section describes the ANC technology developed for the drumming reduction system. This includes the adaptive FF control system that was examined during the initial development effort, the fixed FB control system, the fixed FF control system that was designed to control noise amplification in the rear seat, the ANC performance in the actual vehicle and an observation of FB control.

4.1. Adaptive FF Control System

During initial development, the system was constructed with adaptive FF control. The algorithm used for adaptive signal processing is the filtered-reference LMS algorithm⁽³⁾,⁽⁹⁾ commonly used for FF-ANC. The developed system is characterized by the use of a microphone rather than an accelerometer as the reference signal sensor. Conventional ANC for road noise using adaptive FF control uses an accelerometer that detects vibration acceleration in the suspension or body. This is used as the reference signal sensor⁽⁶⁾,⁽¹⁰⁾⁻⁽¹²⁾. However, the use of an accelerometer involves the following issues:

- (1) Accelerometers are extremely expensive compared to microphones.
- (2) Installing accelerometers in the suspension is more expensive than installing them in the cabin as the environment in the suspension is more severe than that of a cabin.

In order to resolve these issues, a method was devised to install a microphone inside the cabin and use it as the reference signal sensor. However, using a microphone as the reference signal sensor may cause howling. This is because the canceling sound from the speakers detected by the reference microphone generates acoustic FB. The developed system prevents this by adopting a howling canceller (based on the same principle as the IMC described above).

The application of a microphone as the reference signal sensor allows a major cost reduction over the use of an accelerometer. However, this system requires the use of a digital controller employing DSP in order to handle multiple microphones and adaptive signal processing. This makes the cost still much higher than passive technology. Therefore, unfortunately, it has not been matured for use in mass-produced vehicles.

Still, this method enables good control even when the reference signal microphone and the error microphone are close together. This result provides a powerful impetus to

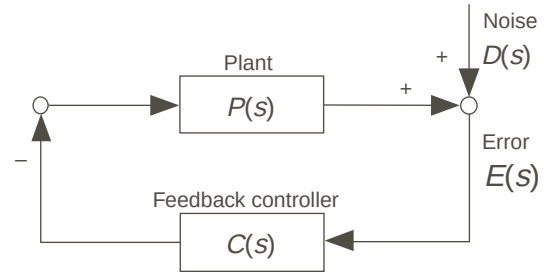


Fig.6 Block diagram of feedback ANC

accelerate development of the FB control system, described in the following subsection.

4.2. Fixed FB Control System

In regard to FB control within the field of ANC, there is little precedent for designing controllers based on control engineering methodology. Therefore, the authors have sought to design control systems on the basis of modeling. This subsection describes the method used to model the plant, and the method used to design a controller based on the model thus obtained.

4.2.1. Design Specifications for the Control System

Fig.6 shows a block diagram of the FB-ANC. It is apparent, from the perspective of control engineering, that FB-ANC is equivalent to disturbance rejection control. Seen in this way, it is a simple matter to apply various theories systematized by control engineering.

The first and most important point for the control specifications is to stabilize the closed loop system. In other words, it is necessary to assure control performance with the condition that feedback will not cause howling.

Next, the performance specifications are described as follows: The aim of control is for the FB controller $C(s)$ to reduce the disturbance, noise $D(s)$, added to the output of the plant $P(s)$ (here, the input to the microphone). The transfer function $S(s)$ from $D(s)$ to error $E(s)$ shown in Fig.6 is called the sensitivity function, and it is expressed as

$$S(s) = \frac{1}{1 + P(s)C(s)} \quad (2)$$

Consequently, it will be possible to reduce noise if $|S(j\omega)|$ can be kept lower than 1. If a model of $P(s)$ is obtained, then $S(s)$ can be derived as soon as $C(s)$ has been designed. Therefore, it is possible to predict noise reduction performance in advance of implementation.

Based on the above, the FB-ANC design specifications are summarized as follows:

- (1) Stabilization of the closed-loop system
- (2) Reduction of the sensitivity function for the frequency range of the noise to be controlled

4.2.2. Fixed FB Control Design Based on Classic Control Theory

The point of this method is that the controller is designed so that the gain of the open-loop transfer function

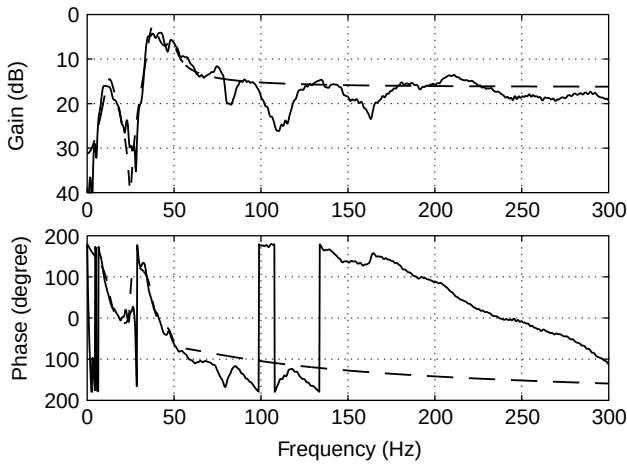


Fig.7 An identified frequency response function of the plant (solid line) and a reduced order model of the plant (dashed line)

$L(j\omega) = P(j\omega)C(j\omega)$ for the frequency range of the sounds targeted for control is set to be greater than 1 and the phase is set at zero. This approach is similar to the design method for loop shaping using a Bode plot in classic control theory. For example, when $|L(j\omega_1)| = 1$ and $\angle L(j\omega_1) = 0^\circ$ in the frequency ω_1 to be controlled, then the sensitivity function becomes

$$|S(j\omega_1)| = \left| \frac{1}{1 + L(j\omega_1)} \right| = \frac{1}{2} \quad (3)$$

This means a potential 6 dB reduction of noise.

As preparation for application of this design method, the frequency response function (FRF)⁽²⁷⁾, which is the non-parametric model of the plant, is obtained by using a sine sweep signal. The Bode plot is shown in Fig.7.

The reason a sine sweep signal is used is to assure a signal-to-noise ratio that accounts for the low-frequency characteristics of the speakers.

Next, the open loop transfer function is shaped by adjusting the controller parameters so that the gain in the frequency range of the targeted noise is greater than one and the phase difference is zero, while the gain rolls off in the high frequency range. Although this design method involves some trial and error in setting the parameters, it is effective for the control of narrow band noise such as drumming. In addition, since a low order controller can be designed by this method, the method is not only advantageous in terms of cost, but further it has the significant advantage of allowing easy on-site adjustment when implemented by an analogue circuit.

The noise reduction performance will be discussed in subsection 4.4.

By comparing this design method and the results thereby obtained with the H_∞ control method and the results thus obtained as described in the next subsection, it is clear that this method is superior. Therefore, this design method was selected for mass-production application.

4.2.3. Fixed FB Controller Design by H_∞ Control

The advantages of using H_∞ control⁽²⁸⁾ for FB-ANC are that noise reduction performance is defined in the frequency domain by the sensitivity function and the robustness to uncertainty of the plant can be taken into account.

It is necessary first to obtain a parametric model of the plant. Obtaining a low order model not only makes controller design easier, but it also contributes to the designing of a reduced order controller, which is also important in terms of cost reduction. The FRF obtained in the previous subsection is used to model the plant using the following procedure:

- (1) Calculating a parametric model using an FRF whose frequency characteristic is close to the FRF in the frequency range targeted for control⁽²⁹⁾.
- (2) Consider the parametric model obtained in (1) to be the nominal plant $P(s)$ for the purpose of designing a controller, and utilize the difference between the gain of the original FRF in (1) and that of the nominal plant FRF as the plant uncertainty.

The method in (1) is a method known for calculating the transfer function from the FRF by the least square method. A low order parametric model is obtained using this method, and the FRF in the obtained parametric model is shown in Fig.7. It is apparent from this figure that the frequency characteristics of the nominal plant is close to that of the plant in the frequency range targeted for control.

The design specifications for FB-ANC with respect to nominal performance is expressed as

$$\|W_s(s)S(s)\|_\infty < 1, \quad (4)$$

where $W_s(s)$ is the weighting function of control performance. The robust stability condition for the plant uncertainty is expressed by

$$\|W_T(s)T(s)\|_\infty < 1, \quad (5)$$

where $W_T(s)$ is the weighting function covering plant uncertainty, and $T(s)$ is the complementary sensitivity function expressed by

$$T(s) = \frac{P(s)C(s)}{1 + P(s)C(s)}. \quad (6)$$

Then, $C(s)$ is calculated by solving the mixed sensitivity problem^{(28), (31)}.

Fig.8 shows $1/W_s(s)$. As shown, the weighting function is defined to reduce noise only in the 40 Hz region. The $S(s)$ calculated using the $C(s)$ is also shown in Fig.8. Based on these results, it is predicted that noise around 40 Hz is reduced by about 10 dB. This has been confirmed by results obtained from implementation in an actual digital circuit, which had roughly equivalent performance. However, the controller is of a high order and on-site adjustment is difficult when the controller is implemented by an analogue circuit.

Also, the weighting function setting still involves trial and error in H_∞ control. However, methods for obtaining a low order controller have been developed⁽³²⁾, and if it becomes possible to implement these in digital circuits at a low cost, then it can be expected to simplify on-site adjustment. Therefore, this avenue will continue to be explored in the future.

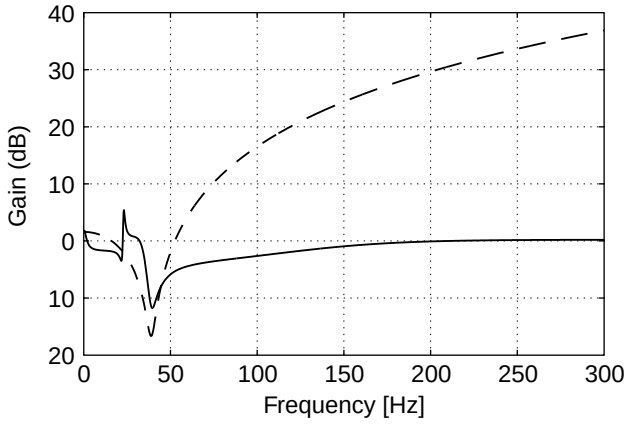


Fig.8 Sensitivity function (solid line) and $1/W_s(s)$ (dashed line)

4.3. Design of the Fixed FF Control System

The method used to design the fixed FB controllers discussed in subsections 4.2 and 4.3 has the problem that when the drumming at about 40 Hz is reduced in the front seats, the noise at the same frequency in the rear seat area is amplified by about 3 dB. This problem must be solved.

The method devised here is to use the output from the microphone utilized in FB control as a reference signal to suppress the amplified noise in the rear seat by FF control. As shown in Fig.9, error microphones are placed in the center of both the front and rear seats while FB control is applied to the front seat. Adaptive FF control is then used to suppress the amplification in the rear seat so that it would not adversely influence the front seats. This adaptive FF control is carried out using digital control. In Fig.9, $Q_1(z)$ and $Q_2(z)$ are the pulse transfer functions from the rear door speakers to the rear seat error microphone and from the rear door speakers to the front seat error microphone, respectively. $\hat{Q}_1(z)$ and $\hat{Q}_2(z)$ express the related estimated pulse transfer functions. The filtered-reference LMS algorithm is used as the adaptive signal processing algorithm. The value defined by the adaptive controller is then used as a fixed FF controller. This makes it unnecessary to use microphones in the rear seats for error sensors in adaptive FF control.

The discrete time controller (digital filter) obtained using adaptive FF control has the characteristics of a 40Hz bandpass filter. Therefore, an analogue filter with very close characteristics to the digital filter was designed and implemented.

It is conceivable that FB control could also be used here to suppress the amplification that occurs in the rear seat. However, it is difficult to design a multiple input multiple output (MIMO) controller. On the other hand, it is extremely simple to apply adaptive FF control to an MIMO system, and the issue of noise amplification in the rear seat is solved using the method described above.

4.4. Noise Reduction in an Actual Vehicle

The controllers derived as described above for use in front seat noise reduction and rear seat noise suppression

- Dashed line parts are only used in development stage.
- The filtered-reference LMS algorithm is utilized.
- After the adaptation, a feedforward controller is fixed.

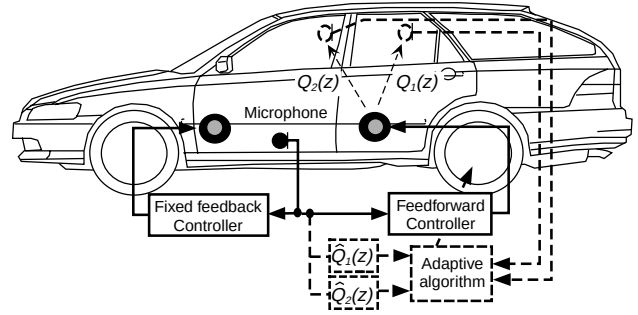


Fig.9 Rear seat control for suppressing amplification

are implemented in analogue circuits. Figs.10 and 11 show the drumming noise reduction performance achieved in the front and rear seats in a car driven at 50 km/h over a rough road surface. These figures show that drumming in the front seat is reduced about 10 dB, and the amplification of noise in the rear seat is suppressed to the level it is at before noise control is used.

4.5. Observations of FB Control

It used to be thought that ANC utilizing FB control was difficult to achieve when the distances between the speakers and the microphones were increased. It is natural to expect FB control to become difficult in terms of stability when there is a phase lag in the transfer function due to a distance delay. However, the present FB-ANC system achieved its success due to consideration given to the following three points:

- (1) Since the speaker has the characteristics of a high pass filter, the phase leads at frequencies below the resonance frequency of the speaker.
- (2) When the noise is at about 40 Hz, the phase lag will be slight, reaching only about 42° , even when there is a distance delay equivalent to 1 m.
- (3) The range of frequencies to be controlled is limited.

Point (1) is realized when physical modeling of the plant is performed^{(20), (21)}. As shown in Fig.7, phase lag in the plant is small because of (1) and (2) even when there is a distance delay. The small size of the phase lag is advantageous for controller design.

Another consideration is that not only does the persistent attempt to broaden the range of frequencies to as great a degree as possible make controller design difficult, it also results in amplification due to the waterbed effect in the same area as noise reduction. On the other hand, the noise in a narrow range of frequencies caused by resonance frequently becomes a concern in actual vehicle development. Therefore, it was decided to limit control to the phenomena within a narrow range of frequencies, as in point (3). Not only does this simplify controller design, but it also makes it possible to keep the range of frequencies amplified by the waterbed effect to a minimum.

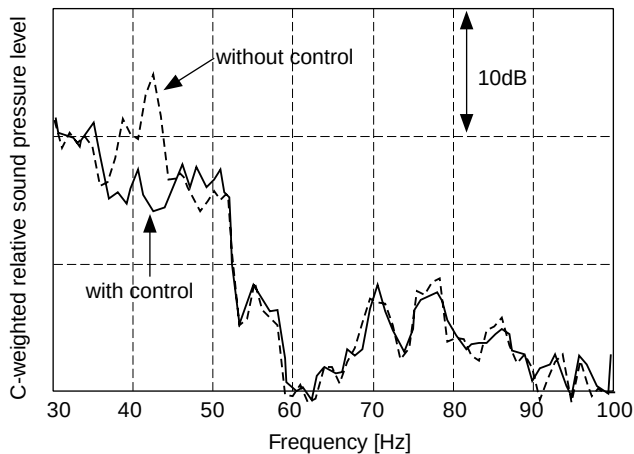


Fig.10 Sound pressure level at front passenger

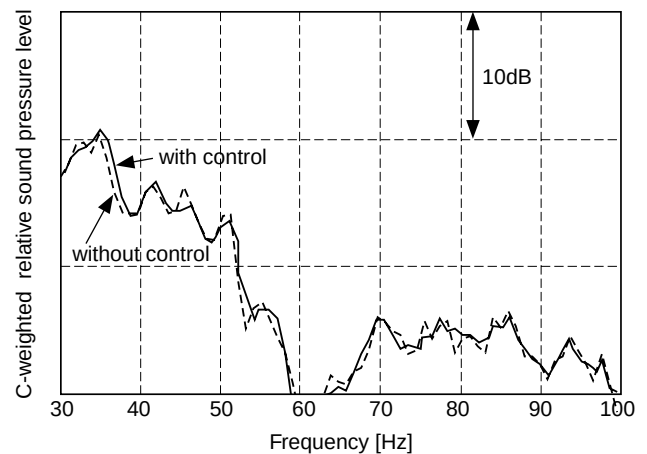


Fig.11 Sound pressure level at rear passenger

5. Conclusions

A product was developed that combines an ANC system that reduces drumming and a car audio system. This product was installed on a mass production model station wagon. This is the world's first ANC system targeting low frequency road noise and the world's first combined car audio and ANC system. Mass production is made possible by cost reduction. The technologies developed with the goal of cost reduction are summarized as follows:

- (1) FB-ANC for the closed sound field within the space of a car cabin
- (2) Implementation by analogue circuits
- (3) Achieving shared use with car audio system speakers by introducing a limiter circuit to prevent speaker distortion
- (4) Music compensator to prevent the canceling out of music
- (5) Fixed FB controllers designed from a control engineering perspective and fixed FF controllers using adaptive FF control

This development project, carried out in the digital age, leads to a renewed recognition of the positive qualities of analogue circuits. The authors intend to conduct a constant comparison of analogue and digital systems as they proceed into the next development project.

In conclusion, a review of the past decade or so of ANC technology development shows that ANC has been largely misunderstood to be a technology that provides revolutionary noise reduction methods to cancel all kinds of noises. Therefore, the system cost is high for the noise reduction obtained by ANC. This seems likely to prevent the automobile industry from pursuing this technology into the active application stage for many years. However, when we consider that it will be necessary to further reduce vehicle weight in the future, it appears that the importance of ANC as a noise control technology will continue to increase. It is hoped that this effort in practical application will serve to stimulate further work on the commercial applications of ANC not only in the automobile industry but also in other fields involved in acoustic engineering.

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Development of LCA Software for Automotive Parts

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ABSTRACT

Conventional life cycle assessment (LCA) software was modified for application to automotive parts by adding an algorithm to determine end-of-life recycling paths identified and to calculate the environmental load for each of these paths. The software was tested in terms of its ability to calculate the environmental load in a bumper recycling system and results were confirmed to be equivalent to those calculated using a process analysis method. The software was applied to the LCA of rubber parts to compare the environmental load of the disposal of waste rubber in the factory process with that of recycling the rubber. Results showed that recycling waste rubber is effective in reducing the environmental load, but that it is very important to improve the quality of the recycled material and reduce the environmental load of the recycling process itself.

1. Introduction

There are calls for improvement in recycling rates with respect to scrap automobiles⁽¹⁾, making it necessary to take into account the recyclability of parts from the design stage. There are also demands for the reduction of emissions of substances that generate an environmental load {carbon dioxide (CO₂), sulfur oxides (SOx), nitrogen oxides (NOx), etc.} over the life cycle of the automobile⁽²⁾. Therefore, efforts to improve the recycling rate must also simultaneously consider methods for optimizing the environmental load. This requires the repeated implementation of life cycle assessments (LCA), which are examinations of the materials specifications, manufacturing methods, and so on of the items in question in order to assess the environmental load generated during their life cycle. Software that makes it possible to reduce the man-hours required to examine, manage, and conduct calculations using the data involved is indispensable when conducting repeated LCAs. However, most of the conventional LCA calculation software that is widely available (hereafter referred to as conventional LCA software) is designed for use with household electrical appliances. The life cycle stages of disposal and recycling are simplified in this software, making it unsuitable for use with automotive parts. Therefore, this project modified such conventional LCA software to enable calculation of the environmental load for the various processing paths at the disposal and recycling stages that are specific to automotive parts. This article reports on the development of this LCA software for automotive parts (hereafter referred to as the Software).

2. Software Aims

2.1. Current Status and Problem Areas of Conventional LCA Software

The basic algorithm for conventional software allows for the entry of the kinds and quantities of the materials and energy that are input in the various stages of the product's life cycle (manufacture of raw materials and materials, manufacture of the product, use and maintenance of the product, transportation, disposal and recycling, etc.). Then, in order to determine the unit quantities (hereafter referred to as base unit data) for each of the related materials and energies, the quantities of substances producing an environmental load that are emitted in each stage are totaled using data for the amounts of environmental load-generating substances emitted. One type of software handles base unit data through proprietary means of survey data entry, while another type is equipped with a database. The latter approach is advantageous for easily obtaining results at a certain level of accuracy and for conducting comparisons of specifications focused on a design, a process, and so on.

The problem with conventional software when applied to LCA calculations for automotive parts is that it allows only for limited processing paths for the disposal and recycling stages. Therefore, it cannot be used to assess the diverse processing paths found with automotive parts. The conventional LCA software disposal and recycling stages allow for shredding of the product after disposal, followed by recycling of only a limited proportion of certain parts of the shredded material for application in new products. The remainder is to be processed as landfill or incinerated. In

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the case of automotive parts, however, materials may have distinctive, specific compositions or functions, so that compound materials require their own specific processes during disposal and recycling, and other items may be recovered as parts and refurbished for reuse. Thus assessment must account for a variety of processing paths.

2.2. Software Objectives

The development objectives for this Software were defined as follows:

- (1) To be capable of assessment that accounts for the characteristic processing paths in the disposal and recycling stages of automotive parts; and
- (2) To be capable of assessing the recyclability of all materials rather than limiting the assessment to specific metal materials.

3. Methods for Achieving Objectives

3.1. Selection of Base Software

The conventional software used as a base for modifications was equipped with a base unit database derived from input-output analysis. This was selected because it allowed easy compilation of data lists showing the amounts of environmental load-generating substances for each life cycle stage of the parts in question. Therefore, this suited an LCA-based comparison for selecting automotive parts specifications.

Input-output analysis uses input-output tables⁽³⁾ to derive the production ripple effect coefficients by which the existence of demand in a particular industry in Japan for production of a particular product causes an impact with regard to the supply of other products or materials, and then successively causes impacts on other industries. The base software determines the base unit data. It does so by first determining the per unit demand value for each type of material (materials, energy, etc.) and part based on the production ripple effect coefficient. Next, this is combined with, for example, various statistical data from the industry⁽⁴⁾⁻⁽⁷⁾ to estimate the fossil fuel supply value corresponding to the demand and, further, estimates the amount of substances generating an environmental load emitted in the production and consumption of that fossil fuel, etc.

3.2. Identifying Processing Paths at the Disposal and Recycling Stages

The processing paths and conditions at the automotive parts disposal and recycling stages were identified as follows:

- (1) After shredding, only specified materials are recovered and recycled as low-grade materials (steel being used to make shaped-beams, etc.). The remainder is transported to a disposal center and separated for processing as landfill and for incineration in set proportions.
- (2) Portions that have been disassembled and dismantled are recycled according to the paths below and the remainder is processed by the path in (1) above:
 - a. Compound materials and other items requiring distinctive recycling processes are processed for recycling accordingly (recycled material manufacture: bumper material recycling, etc.).

- b. Items are reused as parts after the application of simple processing (reuse: aluminum wheels, etc.).
 - c. Items are reused as parts after refurbishing by replacing small parts, etc. (rebuilt: air conditioner compressors, etc.).
 - d. Items are sold with functional compensation after refurbishing of functional parts, including the replacement of small parts (remanufactured: drive shafts, etc.).
 - e. Items are recovered and kept in storage (stocked).
- (3) The recycled materials in item "a" above are used as raw materials in the same kinds of parts as those they were recovered from.
 - (4) The recycled materials in item "a" above are used as raw materials for other parts.
 - (5) Further recycling takes place after shredding, with the recovery and reuse of steel, copper, and aluminum, as well as magnesium, plastic, glass, rubber, and so on.

3.3. Improvement of Calculation Flow Algorithm and Conditions

The calculation flow algorithms that work with the identified automobile parts processing paths and conditions were improved and the calculation conditions were set. Fig. 1 shows the basic algorithm used in the software. The points that were improved are explained below. The right-hand portion of Fig. 1 shows the portions that were added in order to perform assessment of the disposal and recycling stages

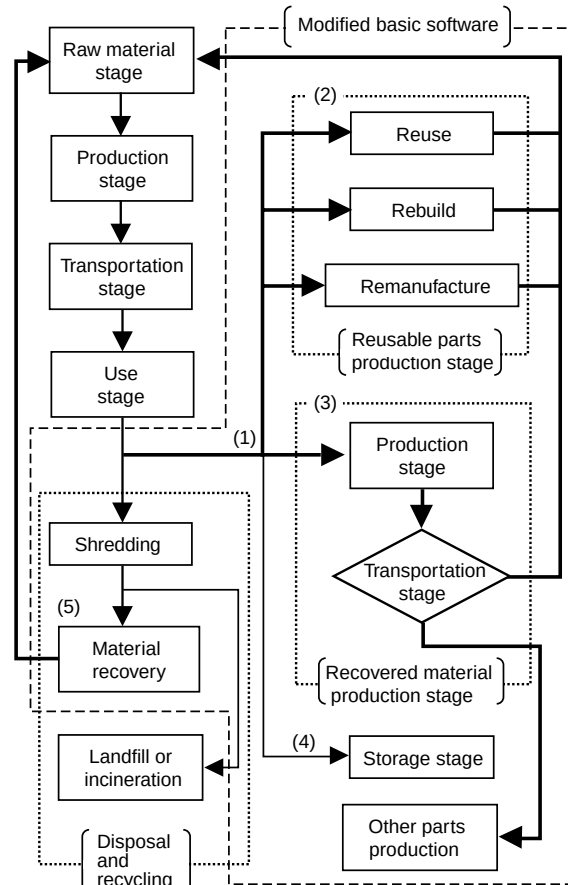


Fig.1 Main algorithm of the developed software

for automotive parts. Arrow symbols indicate the flow of the calculation procedures, and branching lines indicate where the percentage of a material that is to be moved on to the next stage is set. The numbers in the figure correspond to the numbers of the following items:

- (1) A path was established for the recycling of items that are collected from the market and not shredded before recycling. A routine was established to take that percentage and use it as the recovery rate.
- (2) A routine was established for parts that are reused in the same form as they were previously.
- (3) A routine was established for the recycling of compound materials and other such items that undergo distinctive recycling processes and then are recycled as raw materials for use in the same parts as before or in different parts.
- (4) A path was established for items that are held in storage after being recovered.
- (5) Steel, copper, aluminum, and additionally magnesium, thermosetting resin, thermoplastic resin, glass, rubber, paper, and polyvinyl chloride were classified as materials that are recycled as raw materials after shredding.

4. Software Features and Operating Environment

4.1. Features

- (1) The Software is capable of assessing environmental loads (CO_2 , NO_x , SO_x) for all automotive parts recycling paths.
- (2) The Software is capable of assessing the recycling effect for environmental loads (CO_2 , NO_x , SO_x) of all materials including compound materials. (Shredding recycling is excluded.)
- (3) The software is capable of assessing the recycling effect for environmental loads (CO_2 , NO_x , SO_x) in cases when raw materials recycled from parts are reused for different parts.

4.2 Software Operating Environment

- Hardware: Pentium (75 MHz) or better CPU, 50 MB or larger hard drive, 16 MB or more RAM
- Software: Microsoft Windows 95, Excel 97, Access 97

5. Software Verification

A report has been published regarding an automotive parts LCA of a bumper recycling system using the process analysis method, which is said to have superior data accuracy⁽⁸⁾. This report compared the environmental loads in two cases. In one, 30% of the recovered and recycled bumper material was used as raw material for bumpers (hereafter referred to as part-to-part recycling). In the other case, 100% of the recycled bumper material was used as raw material for splash shields (hereafter referred to as cascade recycling, meaning recycling into lower-grade materials or into parts with lower-order functions). Here, the Software was used to perform LCA calculations for similar systems, and the CO_2 emission quantities for the two were compared. Fig. 2 shows the processes and quantities of input materials used in calculations by the Software for the part-to-part system, and Fig. 3 shows the same for the

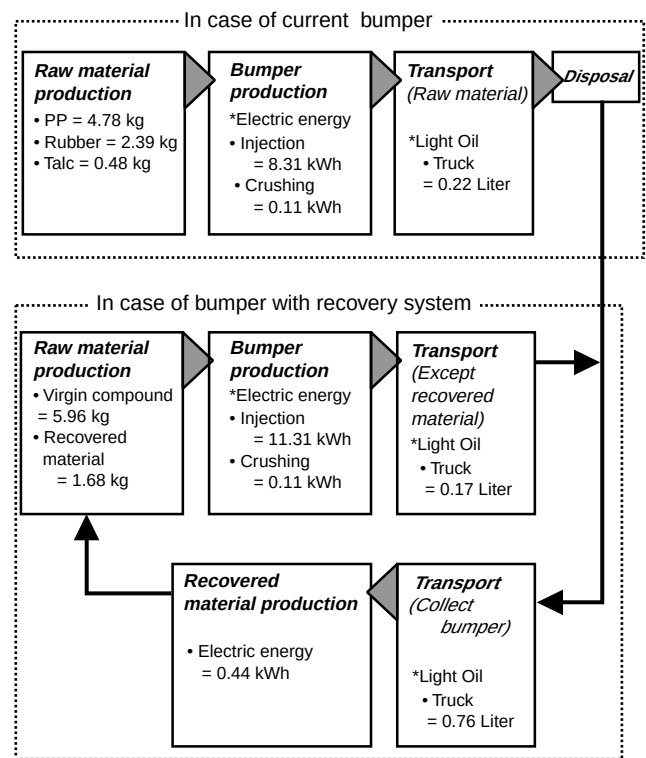


Fig.2 Process and material data for bumper part-to-part recycling

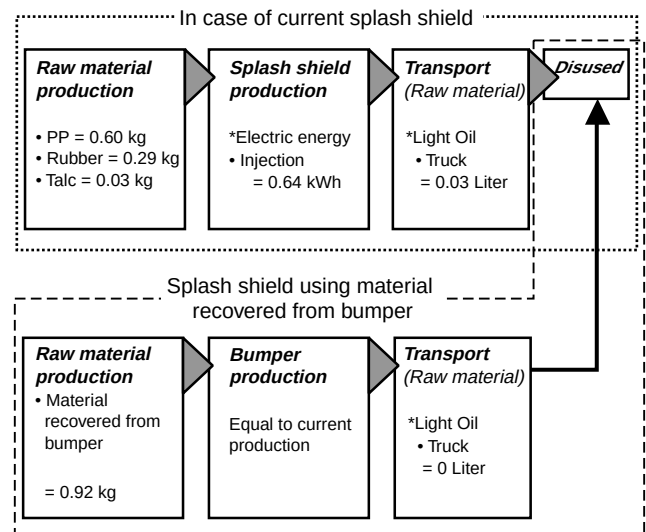


Fig.3 Process and material data for bumper cascade recycling for use in splash shields

cascade system.

The results of LCA calculation by the Software came out nearly equal to the results from the process analysis method. Table 1 shows a comparison of CO_2 emission quantities for both methods. Fig. 4 shows details of the Software calculation results for the part-to-part and cascade cases.

Part-to-part recycling shows a reduction of the environmental load generated by raw materials. However, the environmental load from the manufacturing stage was large, so the overall effect was slight. The reason for this is

that the reduced physical properties of the raw materials from recycled materials resulted in the recycled materials having a low rate of application as raw materials and further greater energy was required in the forming process, as two materials must be sandwiched together. On the other hand, cascade recycling imposes a larger load in the transportation and manufacturing stages for both the bumper recovery and the recycled material manufacturing portions of the process. It was demonstrated, however, that the overall recycling effect was greater because the raw material input portion is zero. This is because the physical properties of the raw material for splash shields are of a lower order than that for bumpers,

Table 1 Life cycle environmental load (CO₂) of bumper and splash shields according to LCA method

		Software	Process analysis method
Bumper using	Current production	19.13 kg	19.21 kg
	Recovered material from bumper 30 %	17.58 kg	18.74 kg
Splash shield	Current production	1.95 kg	2.09 kg
	Using recovered material from bumper 100 %	0.59 kg	0.67 kg

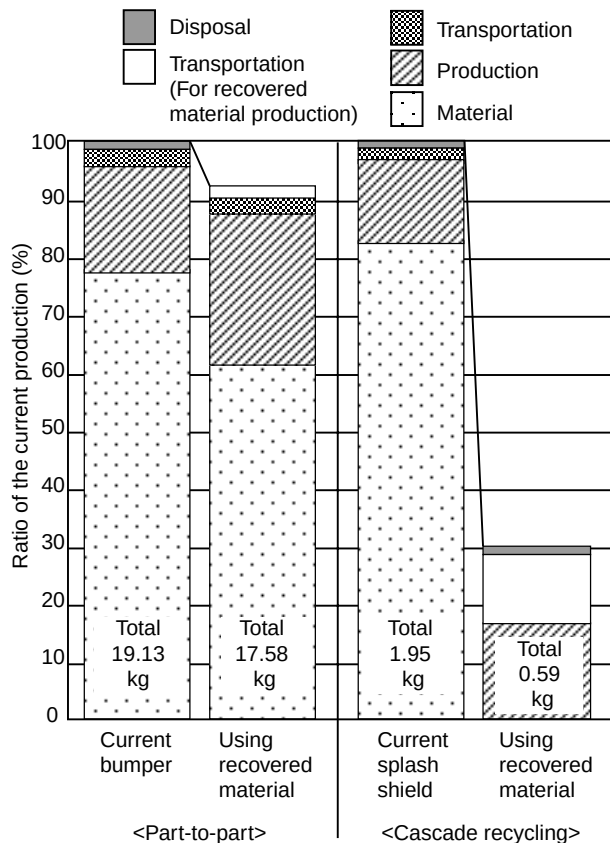


Fig.4 Environmental load (CO₂) of bumper recycling system calculated by the Software

so the recycled material from bumpers can be applied as raw material for splash shields. As a result, this approach leads to the same conclusion as the process analysis method, which is that the cascade method of bumper recycling has a greater effect on environmental load.

6. Software Application Example

The software was used to conduct an LCA on rubber parts, and simulations of CO₂ emission quantities were performed for the recycling of manufacturing process waste materials. The rubber part selected for study was the water hose. Working from the assumption that the shearing method would be used for recycling, the energy required for manufacturing recycled material was set at 1 kWh per 1 kg of waste material. Product weight was set at 1 kg per vehicle. Fig. 5 shows the processes and the quantities of input resources used for the calculations.

The calculation results show that, compared to the 2.02 kg of CO₂ emitted with the ordinary product, the amount emitted when 30% recycled raw materials were used was 1.68 kg, and when 60% recycled raw materials were used it was 1.33 kg. This demonstrated the environmental load reduction effect. Fig. 6 shows the breakdown of these three cases. The use of recycled material as raw material increases CO₂ emissions at the stage of recycled material manufacturing. However, the reduction in CO₂ emissions at the stage of raw material manufacturing is even greater than this increase. Therefore, the overall environmental load

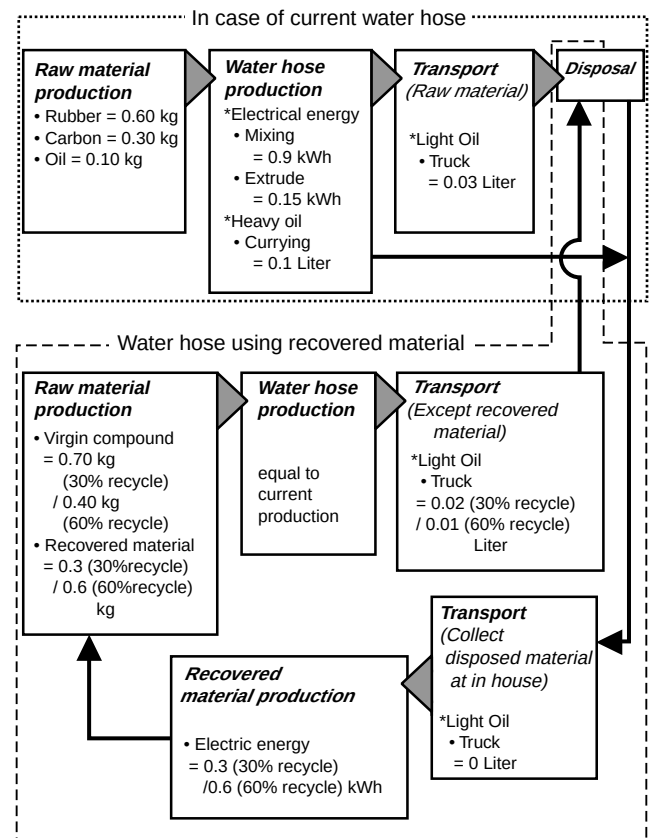


Fig.5 Process data for cascade recycling for water hoses

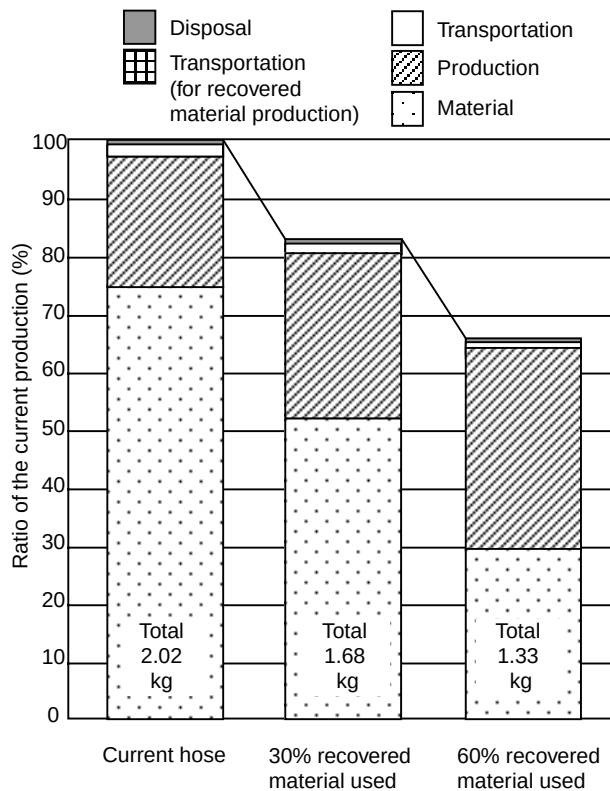


Fig.6 Environmental load (CO₂) of water hoses

decreases in proportion to the higher rate of utilization of recycled material as raw material for the product. This leads to the observation that improvements in the quality of recycled material that link to increased utilization of recycled material as product raw material, and the reduction of the environmental load of the manufacturing stage of the recycled material, are vital in the recycling of waste rubber. It is apparent from the above, therefore, that the Software is effective in the assessment of recycling by LCA.

7. Conclusions

The following results were obtained by improving conventional LCA software through the addition of algorithms thereby enabling it to assess the processing paths in the automotive parts disposal and recycling stages that are specific to such parts:

- (1) The development of Software that is capable of assessing environmental loads, including those from the recycling of parts made of compound materials that involve distinctive, specific processes for recycling, and that provides assessment accuracy nearly equal to the process analysis method.
- (2) Assessment of the effects of recycling process waste material for utilization in rubber parts yielded confirmation of the recycling effect and the issues involved.

In the future, the project will examine the application of this Software in LCAs of larger modules and of entire vehicles.

8. Acknowledgements

In conclusion, grateful acknowledgement is given to Mr. Haruo Suzuki and Mr. Hitoshi Takiguchi of Toshiba Engineering Corporation for their significant cooperation in this development project.

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Optimization of Endmill Cutting Conditions for Nickel-Based Super Alloys

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ABSTRACT

Parameter design based on the Taguchi Method was applied to the endmill cutting process for nickel-based super alloys rather than the conventional sequential methods or experimental design methods in order to optimize the cutting conditions.

“Machining transferability”, which compares the preset removal volume to the actual removed volume, was selected as the evaluation characteristic used in this method.

As a result, if an alloy is cut with the optimized conditions obtained with parameter design, an approximately 60% reduction in variations in machining transferability, in terms of the standard deviation, can be achieved as compared to the conventional conditions.

1. Introduction

Conventional machining evaluations are typically performed using a method based on such quality characteristics as the machined surface roughness and dimensions, tool flank wear, and device cutting resistance, for which experiments are conducted by sequentially combining various parameters or by cross multiplication to optimize the machining conditions. When employing such methods, even if the level of a certain parameter is changed in a direction that is thought would yield favorable results, often the results are not as good as had been expected due to the interaction between parameters. In addition, while the experimental design method may employ statistical methods such that the contribution ratio of each parameter relative to the quality characteristics is expressed quantitatively, it is sometimes difficult to obtain the optimum machining conditions as a reduced dispersion.

In recent years, parameter design using the Taguchi method has been proposed. This method differs from the conventional sequential experimental method and the experimental design method. Examples of using this method to optimize the high speed milling conditions for titanium materials and the endmill cutting conditions for quenched steel have been previously reported⁽¹⁾⁽²⁾. However, there are no reported examples of this method being applied to the cutting of nickel-based super alloys (difficult-to-machine materials), which are generally processed by casting and grinding.

We therefore attempted to optimize the machining conditions for cutting nickel-based super alloys (difficult-to-machine materials) with a milling machine by applying the Taguchi parameter design method.

2. Experimental Design

2.1. Fundamental Function

Evaluating machining according to the Taguchi method consists primarily of two methods; namely, a method in which the evaluation is based on transcription of the machining dimensions, and a method in which the evaluation is based on the machining energy. There has been considerable research conducted recently on the latter method involving evaluations based on the energy consumed in machining. This method uses the energy input and output as determined by the amount of electrical power supplied to the machine and the actual amount of electrical power used by the machine for machining⁽³⁾⁽⁴⁾. However, it has been reported that when the energy consumption of the cutting machine was actually measured, as the electrical power during machining is relatively small in comparison with the power consumed during idling, the difference between the two could not be measured⁽⁵⁾.

Therefore, in this study, an evaluation method was applied based on transcription which works on the principle that the function of cutting is to cut a preset removal volume.

2.2. Signal Factor and Evaluation Characteristic

The fundamental function of cutting is to cut and remove a preset removal volume according to the settings mode. Ideally, the settings and amount removed are in a proportional relationship such that, if the settings are doubled, then the amount removed is doubled, and if the settings are tripled, then the amount removed is tripled. If the input (signal factor) is taken as M (preset removal volume), and the output (evaluation characteristic) is taken as y (the actual material removed, or “machined mass”), then the ideal

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relationship between the two is the following proportional expression:

$$y = \beta M \quad (1)$$

However, in actual cases of metal cutting, the following complex equation results due to the effects of various errors that cannot be controlled ($X_1, X_2, \dots, X_n$):

$$y = f(M, X_1, X_2, \dots, X_n) \quad (2)$$

However, even if the correlation between each effect and characteristic value can be determined by resolving this complex relationship, it is not actually possible to obtain an optimum solution, including a reduction of the dispersion.

The Taguchi method is a form of optimization for bringing the functional relationship expressed in equation (2) closer to the proportional relationship of equation (1). With the conventional experimental design method, quality characteristics are used directly as the characteristic values for analysis. However, with the Taguchi method, experimental analysis is performed using the characteristic value of the SN ratio that is converted to the ratio of the signal and dispersion from signal factor M and evaluation characteristic y .

In this experiment, the relationship between signal factor M and evaluation characteristic y is expressed by the following proportional expression:

$$y^{1/2} = \beta M^{1/2} \quad (3)$$

Three levels of signal factors were set (preset removal volume). Although the actual material removed increases in proportion to the increase in preset removal volume, there are some conditions that prevent machining from continuing due to breakage of the tool and so forth. In such cases, the value for the final machined surface was substituted for the actual material removed to facilitate data processing (Fig. 1). This was done to identify those conditions that would result in longer tool life and higher machining accuracy.

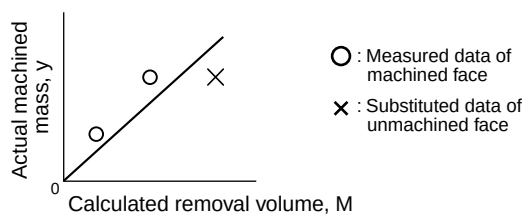


Fig.1 Schematic of ideal function

2.3. Noise Factors

Although the conditions can be selected, including dispersion reduction, to optimize the cutting conditions as well as the machining and production conditions, at the experimental design stage, there are many cases in which optimization is not possible due to unpredictable and not completely understood error factors. With the Taguchi method, consideration is given to which realistic conditions need to be selected, including these error factors. These unknown elements are incorporated into the experiment as noise factors.

In this experiment, it was decided to find realistic

conditions with respect to the following two points from the viewpoint of machining efficiency:

- (1) Being able to cut accurately, regardless of the cutting direction (up-cutting or down-cutting)
- (2) Being able to cut accurately regardless of the overhang (short or long).

Integration of these two points yielded the following noise factors:

N1 (easy cutting): short overhang - down-cutting, the error acts in the positive direction of cutting

N2 (difficult cutting): Long overhang - up-cutting, the error acts in the negative direction of cutting

2.4. Control Factors

Control factors refer to each of the parameters that can be controlled during cutting. In the case of milling, there are numerous parameters such as the tool specifications (material, coating, clearance angle, included angle, tip radius, tool diameter, etc.), feed per tooth, cutting speed, rotational speed and cutting direction. With the Taguchi method, an L18 orthogonal array, which is less susceptible to the effects of parameter interaction, is typically used when designing the experiment. In this case, although a maximum of eight parameters can be assigned, each parameter must be able to be independently controlled. Five parameters were selected for this experiment consisting of the tool manufacturers, feed

Table 1 Control factors and levels

Factor		Level		
		1	2	3
A	Tool maker	1	2	
B	Error			
C	Feed per tooth	Low	Middle	High
D	Cutting speed	Low	Middle	High
E	Width of cut	Low	Middle	High
F	Depth of cut	Low	Middle	High
G	Error			
H	Error			

Table 2 L18

Exp. No.	Control factors								Noise	
	A	B	C	D	E	F	G	H	N1	N2
1	1		Low	Low	Low	Low				
2	1		Middle	Middle	Middle	Middle				
3	1		High	High	High	High				
4	1		Low	Low	Middle	Middle				
5	1		Middle	Middle	High	High				
6	1		High	High	Low	Low				
7	1		Low	Middle	Low	High				
8	1		Middle	High	Middle	Low				
9	1		High	Low	High	Middle				
10	2		Low	High	High	Middle				
11	2		Middle	Low	Low	High				
12	2		High	Middle	Middle	Low				
13	2		Low	Middle	High	Low				
14	2		Middle	High	Low	Middle				
15	2		High	Low	Middle	High				
16	2		Low	High	Middle	High				
17	2		Middle	Low	High	Low				
18	2		High	Middle	Low	Middle				

per tooth, cutting speed, width of cut and depth of cut. With respect to the level values for each parameter, the tool manufacturers were assigned the two levels of 1 and 2, while the other four parameters were set the three levels of low, medium and high. These were allocated as shown in Table 1. An array (L18 orthogonal array) that arranges the levels of each parameter in the form of 18 sets of experimental conditions is shown in Table 2.

2.5. Fixed Conditions

The following were used as fixed conditions:

- tool holder
- tool type (diameter, number of teeth)
- cutting fluid (fluid type, amount of fluid, etc.).

3. Experimental Results

3.1. Examination of the Data

Cutting was performed according to the combinations in the L18 orthogonal array. According to the Taguchi method, the larger the difference in the data for each experiment (difference in SN ratio), the higher the accuracy that can be evaluated. As an example of this, photographs of the test pieces after machining in experiments No. 5 and 13 are shown in Figs. 2 and 3. In addition, the data are shown in Tables 3 and 4, while graphic representations of those data are shown in Figs. 4 and 5.

It is clear from the photographs that, although machining could be performed to the third level (the third machined

Table 3 Data of No. 5

M (mm ³)		y (g)	
		N1	N2
1	1289.217	11.10	10.49
2	2578.434	22.10	20.97
3	3867.651	33.08	20.97

Table 4 Data of No. 13

M (mm ³)		y (g)	
		N1	N2
1	1285.806	10.97	10.89
2	2571.612	22.01	21.84
3	3857.419	32.95	32.79

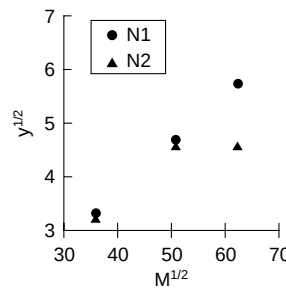


Fig.4 Exp. result (No.5)

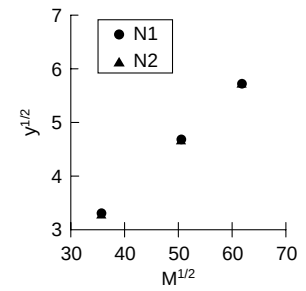


Fig.5 Exp. result (No.13)

surface) of the signal factor at noise factor N1 (easy cutting) in experiment No. 5, in the case of N2 (difficult cutting), further machining was not possible from part of the way through the second level (the second surface). In experiment No. 13, although tool marks remained on the third surface at N2, machining could be performed to the third surface for both N1 and N2.

3.2. Calculation of the SN Ratio

An example of calculating the SN ratio is indicated for the results of experiment No. 5. In addition to showing the data obtained by taking the square root ($\sqrt{\quad}$) of the signal factor and evaluation characteristic, the sum of the squares was analyzed using those results.

Table 5 Data of No. 5

M ^{1/2}		y ^{1/2}	
		N1	N2
1	35.905 7	3.331 7	3.238 9
2	50.778 3	4.701 1	4.579 5
3	62.190 4	5.751 5	4.579 5

- Total sum of squares

$$S_T = 3.3317^2 + 4.7011^2 + \dots + 4.5795^2 + 4.5795^2 = 118.7138 \quad (f=6) \quad (4)$$

- Linear expressions

$$L1 = 35.9057 \times 3.3317 + 50.7783 \times 4.7011 + 62.1904 \times 5.7515 = 716.0273 \quad (5)$$

$$L2 = 633.6328 \quad (6)$$

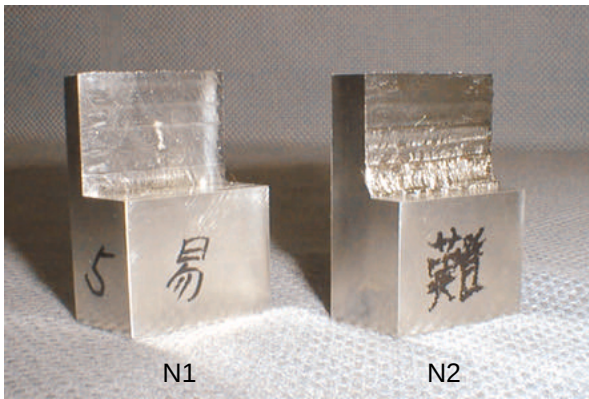


Fig.2 Machined test pieces (No. 5)

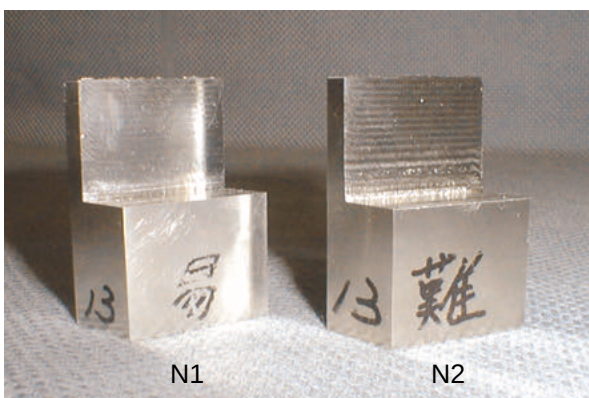


Fig.3 Machined test pieces (No. 13)

- Sum of squares of the primary effect of the signal factor

$$S_{\beta} = (L1 + L2)^2 / 2r \\ = 117.744 \ 8 \ (f=1) \quad (7)$$

where,

$$r = 35.905 \ 7^2 + 50.778 \ 3^2 + 62.190 \ 4^2 \\ = 7 \ 735.301 \ 3 \quad (8)$$

- Sum of squares of the noise factor

$$S_N = S_{N \times \beta} \\ = (716.027 \ 3^2 + 633.632 \ 8^2) / r - 117.744 \ 8 \\ = 0.438 \ 8 \ (f=1) \quad (9)$$

- Sum of squares of the error

$$S_e = S_T - S_{\beta} - S_N \\ = 0.530 \ 2 \ (f=4) \quad (10)$$

- Sum of squares of the error, including the effect of the noise factor

$$S_{N'} = S_{N+e} = S_T - S_{\beta} \\ = 0.969 \ 0 \ (f=5) \quad (11)$$

The results of the foregoing calculations are summarized in the variance results analysis shown in Table 6. Based on the variance analysis, SN ratio η and sensitivity S could be obtained from the following equations.

$$\text{SN ratio: } \eta = 10 \log \frac{\frac{1}{2r} (S_{\beta} - V_e)}{V_{N'}} = -14.06 \text{db} \quad (12)$$

$$\text{Sensitivity: } S = 10 \log \frac{1}{2r} (S_{\beta} - V_e) = -21.19 \text{db} \quad (13)$$

The results from a similar calculation of the SN ratio and the sensitivity from the data for each of the experiments

Table 6 Analysis of variance (No.5)

	Factor	F	Sum of squares	Variance
Primary effect	β	1	117.744 8	117.744 8
Noise factor	$N \times \beta$	1	0.438 8	0.438 8
Error	e	4	0.530 2	0.132 5
	N'	5	0.969 0	0.193 8
Total	T	6	118.713 8	

are shown in Table 7.

Next, the variance analysis (SN ratio) for each control factor according to the calculated results for the SN ratio and sensitivity is shown in Table 8. In addition, the factor effect diagram between the SN ration and sensitivity is shown in Fig. 6 and 7.

Table 7 SN ratio & S (sensitivity) for each experiment

Exp. No.	SN ratio (db)	Sensitivity (db)
1	11.52	- 20.73
2	16.83	- 20.70
3	- 22.19	- 22.28
4	9.85	- 20.70
5	- 14.06	- 21.19
6	19.86	- 20.70
7	13.38	- 20.73
8	17.86	- 20.69
9	- 13.98	- 21.16
10	- 13.41	- 21.12
11	22.78	- 20.67
12	- 13.71	- 21.13
13	20.74	- 20.70
14	17.11	- 20.68
15	14.58	- 20.68
16	7.85	- 20.72
17	12.46	- 20.71
18	20.96	- 20.68
Average	7.13	- 20.89

Table 8 Analysis of variance – SN ratio

Factor	F	S	V	ρ
A Tool maker	1	140.546 3	140.546 3	3.6%
C Feed per tooth	2	391.876 0	195.928 0	10.2%
D Cutting speed	2	76.083 6	38.041 8	2.0%
E Width of cut	2	1570.217 3	785.108 7	40.7%
F Depth of cut	2	186.696 2	93.348 1	4.8%
e Error	8	1488.005 9	186.000 7	38.6%
Total	17	3853.425 3	226.672 1	

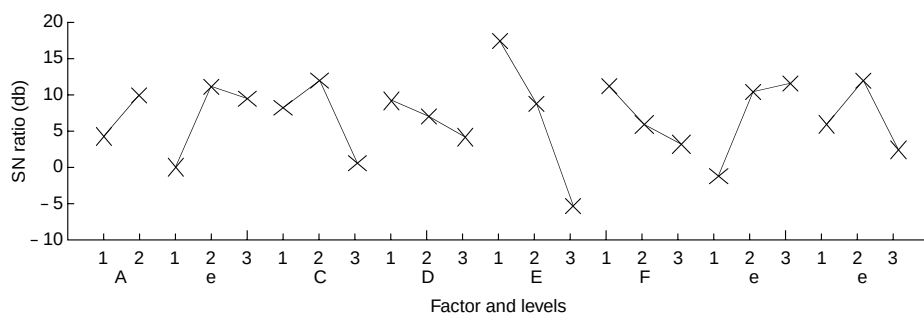


Fig.6 Factor effect diagram of the SN ratio

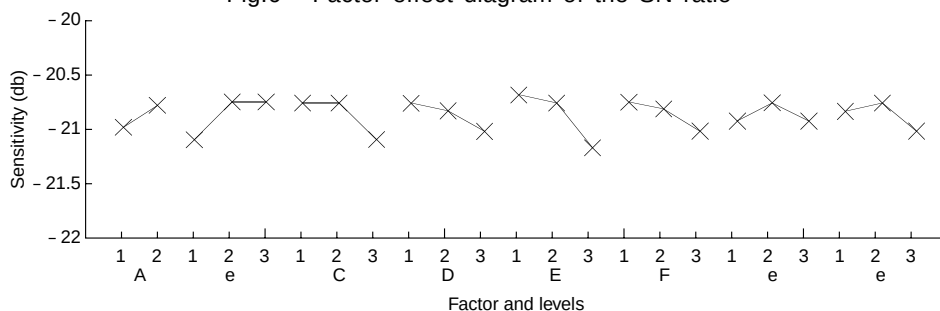


Fig.7 Factor effect diagram of the Sensitivity

According to the Taguchi method, in a factor effect diagram of the SN ratio, the larger the difference between the levels of a factor, the higher the contribution ratio relative to the evaluated characteristic of that factor. In addition, the larger the SN ratio of a level, the greater the degree of agreement with the goal (the improvement of transcription) of that level.

Sensitivity S calculated from equation (13) can be expressed by equation (14) using linear slope b from the fundamental function (Fig. 1). In this experiment, β theoretically expresses the density of the portion removed by cutting.

$$\text{Sensitivity: } S = 10 \log \beta^2 \quad (14)$$

Since the density of the nickel-based super alloy used in this study was $8.53 \times 10^{-3} \text{ g/mm}^3$, ideal sensitivity S calculated by equation (14) becomes -20.69 db. If the value of sensitivity S is small, the density of the portion removed by cutting is also small. In other words, this means that the preset cutting volume was not removed.

Since the trends for each level was the same for both the SN ratio and sensitivity based on Figs. 6 and 7, it is believed that the sensitivity could be improved by selecting a level having a high SN ratio.

In this experiment, as is clear from Table 8, the contribution ratio of the width of cut demonstrated the highest value at 40.7%. Although the large effect of the width of cut demonstrated a similar trend in previous machining experiments, the contribution ratio in this present experiment could be quantitatively determined with the other factors. On the other hand, since the contribution ratio of the error was also 38.6%, this indicated that it is at a level that could not be ignored in comparison with the control factor.

4. Estimation of the Optimal Conditions

4.1. Estimation of the Optimal Conditions

In order to estimate the optimal conditions, a level was selected at which the SN ratio reached a maximum for each control factor based on the factor effect diagram of the SN ratio shown in Fig. 6, and this was compared with the current conditions as shown in Table 9. The current conditions were set using the results of sequential combinations based on each type of machining experiment, and coincidentally agreed with those of current experiment No. 2.

Table 9 Optimal and current conditions

Factor		Optimal condition	Current condition
A	Tool maker	2	1
C	Feed per tooth	Middle	Middle
D	Cutting speed	Low	Middle
E	Width of cut	Low	Middle
F	Depth of cut	Low	Middle

Since the only control factor that exhibited the same level for the optimal conditions and current conditions was the feed per tooth, and the levels were different for the other four factors, the extent to which an improvement in transcription could be expected was quantitatively estimated.

4.2. Estimation of the SN Ratio under Optimal Conditions

The SN ratio was estimated under optimal and current conditions from the SN ratios at each level of all the control factors to determine the gain. The comparative results are shown in Table 10.

Table 10 Estimation of average value (db)

	Optimal condition	Current condition	Gain
SN ratio	32.15	12.33	19.82
Sensitivity	- 20.25	- 20.67	0.42

According to the estimated results for machining performed under optimal conditions, the SN ratio was 32.15 db, indicating that a gain of approximately 20 db could be expected in comparison with the 12.33 db figure under current conditions. On the other hand, the SN ratio based on the results of experiment No. 2, which was equivalent to the current conditions, was 16.83 db, resulting in a difference of 4.50 db between this value and the estimated value of 12.33 db. This is believed to have been due to error factors not grasped conventionally, as was shown in the previously mentioned table for the control factor analysis of variance (Table 8). A confirmation experiment was then conducted to clarify the manner in which the gain would be affected in the case of machining under the estimated optimal conditions.

5. Confirmation Experiment Results and Discussion

The confirmation experiment was conducted under the optimal and current machining conditions (Table 9) as determined by the Taguchi parameter design method as previously described. The results of calculating the SN ratio and sensitivity are shown in Table 11.

Table 11 Confirmed result (db)

	Optimal condition	Current condition	Gain
SN ratio	18.48	10.50	7.98
Sensitivity	- 20.67	- 20.70	0.03

In the case of machining under the optimal conditions, the gain of the SN ratio in comparison with that under the current conditions was 7.98 db. As is shown in the following calculations, this indicates that dispersion of the transcription of machining can be reduced by roughly 60% in terms of the standard deviation.

When the slope is represented by β and the dispersion (standard deviation) is represented by σ , the SN ratio calculated by equation (12) can be represented by equation (15).

$$\text{SN ratio: } \eta = 10 \log \left(\frac{\beta}{\sigma} \right)^2 \quad (15)$$

Accordingly, the following results were determined when the SN ratio under optimum conditions is represented by $\eta_{(opt)}$, the slope by $\beta_{(opt)}$, and the dispersion by $\sigma_{(opt)}$, while the SN ratio under current conditions is represented by $\eta_{(cur)}$, the slope by $\beta_{(cur)}$ and the dispersion by $\sigma_{(cur)}$.

$$h_{(opt)} - h_{(cur)} = 10 \log \left(\frac{\beta_{(opt)}}{\sigma_{(opt)}} \right)^2 - 10 \log \left(\frac{\beta_{(cur)}}{\sigma_{(cur)}} \right)^2 = 7.98 \text{ db} \quad (16)$$

In addition, the following results hold true since $\beta_{(opt)} \doteq \beta_{(cur)}$ from Table 11.

$$\eta_{(opt)} - \eta_{(cur)} = 10 \log \left(\frac{\sigma_{(cur)}}{\sigma_{(opt)}} \right)^2 = 7.98 \quad (17)$$

$$\therefore \left(\frac{\sigma_{(cur)}}{\sigma_{(opt)}} \right)^2 = 10^{0.798} = 6.281 \quad (18)$$

$$\therefore \frac{\sigma_{(cur)}}{\sigma_{(opt)}} = 2.506 \quad (19)$$

Accordingly,

$$\sigma_{(opt)} \doteq 0.4 \sigma_{(cur)} \quad (20)$$

On the other hand, a gain of only 7.98 db could be obtained in the confirmation experiment, corresponding to roughly half that of the 19.82 db gain for the SN ratio indicated by the estimated results. In addition, according to the results of the confirmation experiment, the SN ratio under current conditions was 10.50 db, indicating a considerable difference from the 16.83 db figure for the SN ratio in the case of the preliminary experiment. Thus, problems remain with respect to the reproducibility of the experiment itself.

In this study, the machining transcription, which compares the preset removal volume with the actual removed volume, was selected for the evaluation characteristic. In this case, although the SN ratios are equal if the actual machined mass as the evaluation characteristic is the same, even under different machining of conditions, when considering an evaluation of the entire machining system, there is the possibility that the effects of the parameters, including the error, were not fully understood.

For example, in the case of a machine whose spindle rotational speed is set by feedback control, the rotational speed is kept constant by increasing the electrical power when the cutting resistance increases and the spindle rotational speed momentarily decreases. As a result, even if the evaluation characteristic in the form of transcription were to be the same, the amount of energy used for machining (the amount of power consumed) would increase. This surplus energy would then end up being converted into heat, vibration or noise, and this could generate dispersion.

Thus, instead of just considering the machined mass, it is believed to be necessary in the future to further add the amount of power consumed and other factors relevant to the entire machining system to the evaluation characteristics in order to improve the optimization of machining conditions and the accuracy of gain estimation.

6. Conclusions

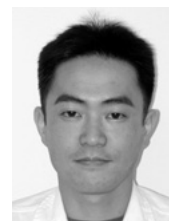
The Taguchi parameter design method was used to optimize the endmill cutting conditions for nickel-based super alloys with the following results:

- (1) The dispersion of machining transcription could be reduced by about 60% when machining was performed under the optimal conditions.
- (2) It is necessary to review the evaluation characteristic based on the complete machining system in order to further improve the accuracy of conditions selection.

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Technology for Reducing Engine Rubbing Resistance by Means of Surface Improvement

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ABSTRACT

A surface modification method has been developed that uses shot peening to reduce the rubbing resistance on piston wear surfaces. This technique shoots roughly spherical ceramic bead having a central particle diameter of 40 μm with compressed air at a speed of 100 m/sec or more against the surface of the object to be processed. This peening treatment produces countless microscopic dimples that improve the oil retention capability of the treated surface and results in a 2–2.6% reduction of the lost output from an engine unit due to friction. Engine units with this treatment were subjected to an extended operation test to verify the sustainability of the effects from this treatment. The results showed that dimples remained on the parts even after an extended period of operation. The treatment was also proven to be effective in increasing the anti-seizure characteristics of the pistons.

1. Introduction

Amid the ever-growing concern for the global environment and for the conservation of the planet's resources, there is an increasing demand for automobiles that consume less fuel. In order to improve heat efficiency and reduce mechanical loss to thereby improve the efficiency of automobile engines, research ⁽¹⁾⁽²⁾ has been undertaken from various approaches. Particularly worthy of note in terms of mechanical loss are the improvements that focus on even minute aspects of engine parts such as the improvements in the roller follower valve motion systems and the reduction of the weight and tensile strength of piston rings. The authors focused their current studies on reducing rubbing resistance by improving the properties of the rubbing area surfaces. As a result of their efforts, they managed to reduce piston drive loss, which accounts for 40% of mechanical loss, by 30% by improving the surface properties. This paper reports on the friction loss reduction results, anti-seizure characteristics, and sustainability of the effects accomplished by the authors. Further, it provides mass production application examples.

2. Technology for reducing mechanical loss

As shown in Fig. 1, 60% of the total loss sustained during partial load operation is accounted for by mechanical loss. A breakdown of the mechanical loss shows that piston driven loss (rubbing resistance of the pistons) is responsible for 40% of this loss, a figure that clearly shows that a reduction

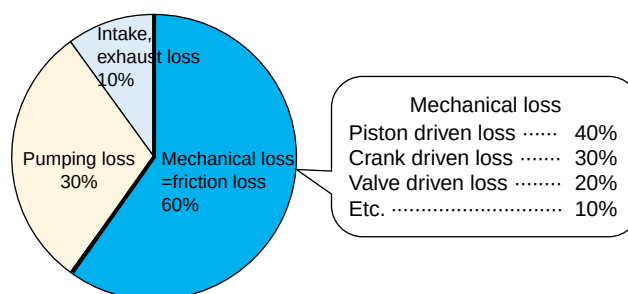


Fig.1 Engine component loss division at Ne =1500rpm, P intake =-53kPa

of the rubbing resistance of the pistons will be an effective way of improving fuel economy. In the past, a number of techniques have been used to reduce this rubbing resistance. They include coating solid lubricants typified by molybdenum and applying coatings or plating tin onto the walls of the pistons. However, these techniques were only marginally effective in reducing rubbing resistance, and the sustainability of their effects was called into question by peeling and wear. Marking a departure from these techniques, the authors tackled the issue by improving the surface properties of the rubbing areas to achieve sustainable rubbing resistance reductions.

2.1. Lubrication condition of piston skirts

In order for the rubbing resistance to be reduced by engineering improvements in surface properties, it is necessary to effectively increase the retention of the

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lubricating oil that comes between the mechanical parts and rubbing areas. By doing this, it then becomes possible to continuously maintain the fluid lubrication status under any engine operating conditions, facilitate smooth rubbing actions, and reduce rubbing resistance.

First, the rubbing surface characteristics of the currently available pistons and cylinders were examined (see Fig. 2). Spirally shaped tool marks around the piston skirt sides known as skirt streaks whose main purpose is to improve the anti-seizure characteristics were made by machining in the circumferential direction.

Furthermore, when the piston sides were observed under a microscope, it was found that countless microscopic tool marks had been made during the machining also in the circumferential direction.

These surface machining characteristics affect the surface lubrication characteristics during operation. When a piston is subjected by the combustion gases to an explosive load so that it moves in the thrust direction and is pushed forcefully against the cylinder, the capillary phenomenon occurs. According to this phenomenon, the lubricating oil between the cylinder and piston flows along these microscopic machine tool marks in the circumferential direction. For this reason, it is difficult for the oil film to be maintained between the rubbing sections of the cylinder and piston, which leads to an increase in the rubbing resistance and, in some cases, promotes galling and seizure.

3. Reduction in rubbing resistance achieved by micro dimple processing

The machine tool marks must be removed if the capillary phenomenon corresponding with these tool marks is to be prevented.

The authors used micro dimple (or “MD” for short) processing as the means to achieve this.

MD processing is a type of shot peening in which roughly spherical ceramic beads with a central particle diameter of 40 μm are shot with compressed air at high speeds of 100 m/sec or more against the surface of the object to be processed. It adds countless microscopic dimples (MD) to this surface and eliminates the machine tool marks without changing the dimensions of the object to be processed, and it forms oil pits which are effective in reducing rubbing resistance.

3.1. Verification of rubbing resistance reduction by a reciprocating friction test using test pieces

In order to verify the effectiveness of the MD processing in reducing rubbing resistance, the authors measured the coefficient of friction and seizure load by conducting a reciprocating friction test using test pieces (or “TPs”). An overview of this test is shown in Fig. 3.

While allowing lubricating oil to drip onto the rubbing surface of a plate that simulates a cylinder, the TP was rubbed back and forth at a stroke of 50 mm over the plate surface,

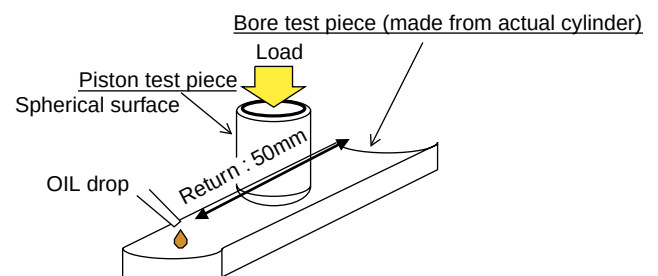


Fig.3 Two way friction test machine (simple drawing)

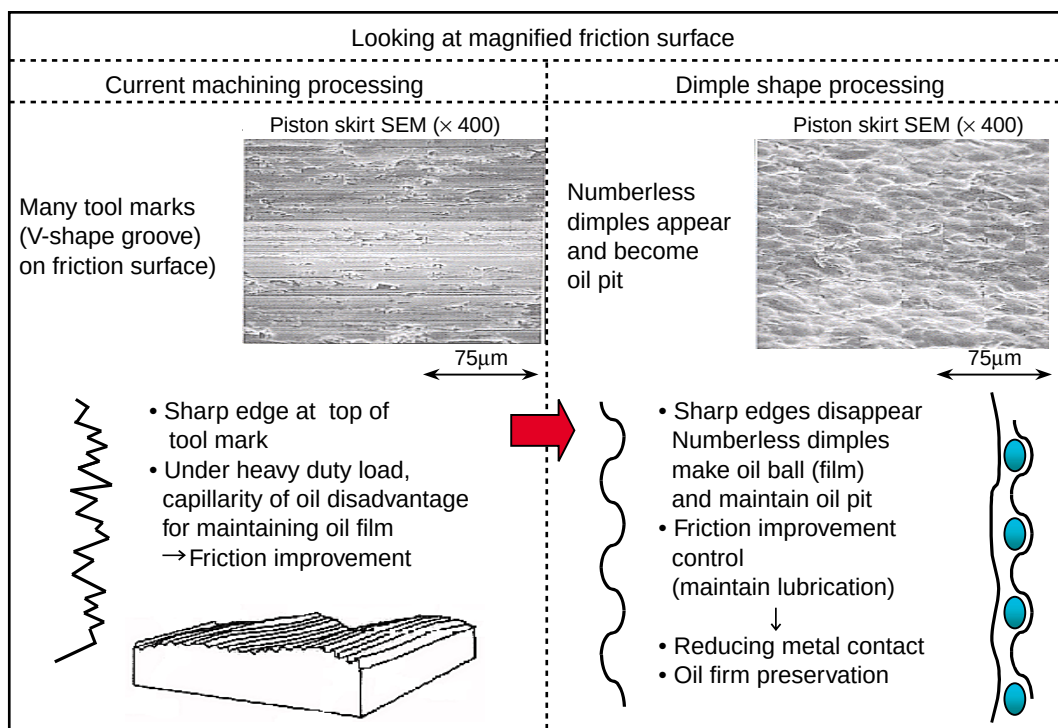


Fig.2 Dimple effect (expectation)

the pressing load was incrementally increased, the coefficient of friction (μ) was measured in real time and integrated with the time to calculate the average friction resistance per unit time. The results of this are shown in Fig. 4.

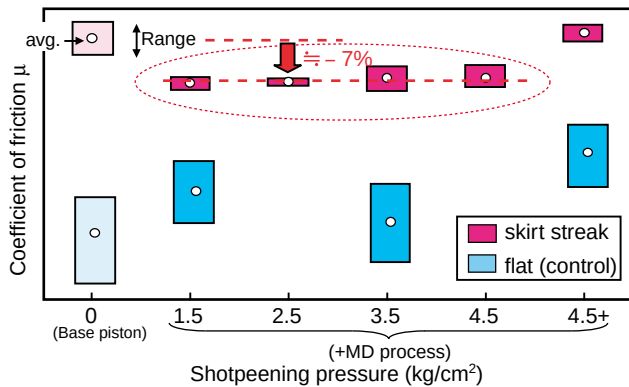


Fig. 4 Two way friction test result

The TP which was subjected to MD processing on its skirt streaked rubbing surface showed a reduction in the coefficient of friction (μ) of about 7% across the shot peening pressure range of 1.5 to 4.5 atmospheres (gauge pressure; hereafter all pressures will be given as gauge pressures) compared with a surface which had merely been skirt streaked.

For the purposes of comparison, the results for a TP which had been MD processed on a flat surface devoid of skirt streaking are also given. The TP with a flat surface which had not been MD processed yielded the lowest coefficient of friction (μ). Nevertheless, as shown in Fig. 5, the seizure load is higher for the skirt-streaked surface than for the flat surface, and it was clearly established that this load is further improved by conducting MD processing for the skirt streaking.

The authors were able to verify from the results of this test that performing MD processing on the skirt streaking is an effective way of improving both the coefficient of friction and seizure load.

3.2. Reduction in rubbing resistance by MD processing conditions

Based on the results obtained from the above TP friction test, the authors fabricated pistons whereon the entire rubbing

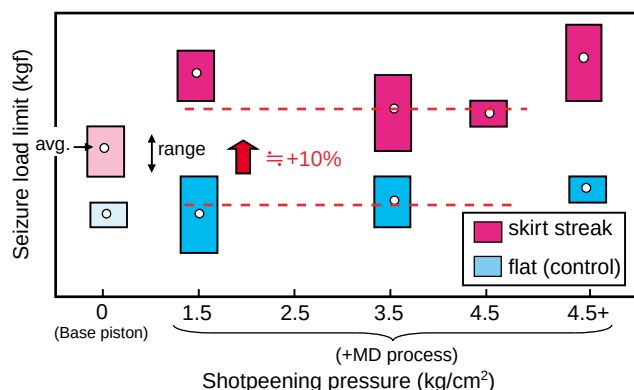


Fig. 5 Two way friction test result

surface of the skirt was streaked and MD processing was implemented at a shot peening pressure of 1.5 to 4.5 atmospheres as shown in Fig. 6. The mechanical loss for engine units was measured using the motoring method.

Fig. 7 shows the characteristics of the mechanical loss decrease versus shot peening pressure. Three engines with a displacement of 1.0, 1.5 and 2.2 liters were tested. A mechanical loss decrease rate of more than 1% was obtained with a shot peening pressure ranging from 1.7 to 4.5 atmospheres. This rate was at its highest at a shot peening pressure of 3.5 atmospheres for all the engines and a friction reduction rate of 2% to 2.6% was achieved.

On the basis of the above results, a shot peening pressure of 3.5 atmospheres was set as the MD processing condition, and a central particle diameter of 40 μ for the ceramic beads shot material was selected. The average MD depth obtained under these conditions turned out to be 0.6 to 1.8 μ . To determine this depth, surface roughness Ra was measured, and the resulting value was used.

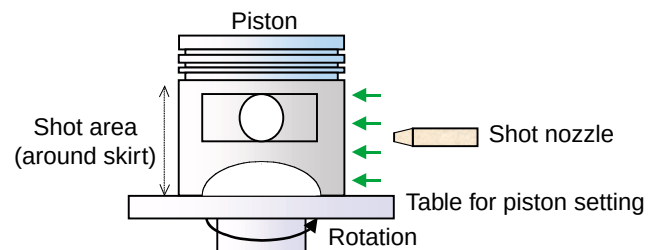


Fig. 6 MD process piston setting layout for machine

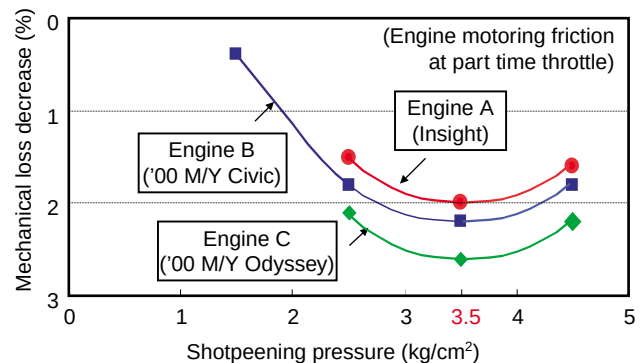


Fig. 7 Shotpeening pressure and mechanical friction loss

3.3. Verification of the sustainability of the MD processing effect

In order to understand the sustainability of the MD effect, the engine units were run on a dynamometer under a high load for a prolonged period. After a prescribed period of time had elapsed, the motoring method was used to measure the mechanical loss.

Fig. 8 shows that, compared with the unprocessed piston, the MD processed piston had a lower mechanical loss starting from the initial state. Even after 120 hours, it showed a mechanical loss reduction effect of at least 2%. The changes in the wear level of the skirt streaks occurring during this period are shown in Fig. 9.

The initial skirt streak height was approximately 10 μ

and, after a prolonged period of engine operation, it dropped to about 8 or 9 μ , signifying a wear level of 1 to 2 μ .

This wear is not progressive in nature but remains steady after approximately 100 hours of engine operation. As mentioned above, a mechanical loss reduction effect of at

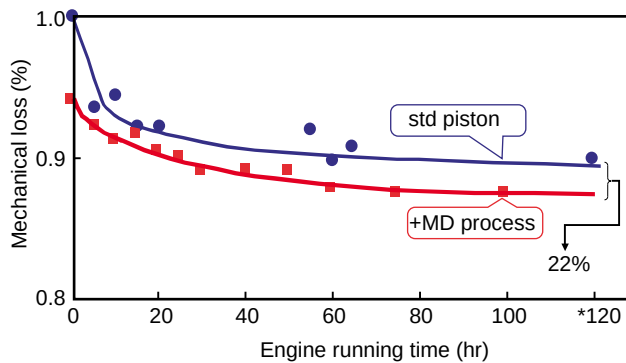


Fig.8 Mechanical loss decrease & eng running time

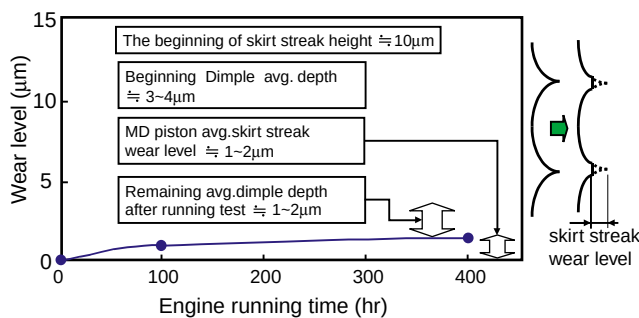


Fig.9 Wear level of MD process piston skirt streak

least 2% compared with the unprocessed piston was achieved in this state.

Fig. 10 shows the status of the rubbing surfaces before and after the prolonged period of engine operation.

It is possible to verify skirt streaking on the unprocessed piston prior to rubbing (see Fig. 10-1). In contrast, on the unprocessed piston after rubbing, the edges of the skirt streaks have been worn away to result in a roughly trapezoidal shape (see Fig. 10-2).

The edges are formed at the borders of the roughly trapezoidal areas by the wearing away of the skirt streak edges accompanying the rubbing of the piston. These edges then act as wipers for the lubricating oil with the reciprocating movement of the piston, and under heavy-duty loads, it is believed that this action, together with the capillary phenomenon, makes it difficult for the oil to be supplied to the actual rubbing parts.

In contrast, countless MDs with no gaps between them are visible on the MD processed piston prior to rubbing (see Fig. 10-3).

Furthermore, even after rubbing, the MDs still remain although the skirt streak edges which constitute the actual rubbing surface have been worn away (see Figs. 10-4 and 11).

The shape of the border areas on the actual rubbing surface is changed from a roughly trapezoidal one to that with many inlets formed by the subsequent MD processing. For this reason, it is believed that the lubricating oil present in the large valleys between the skirt streaks is supplied to the actual rubbing areas from these inlets while the piston reciprocates as well as being supplied to the MDs still remaining on the actual rubbing surface. This process results

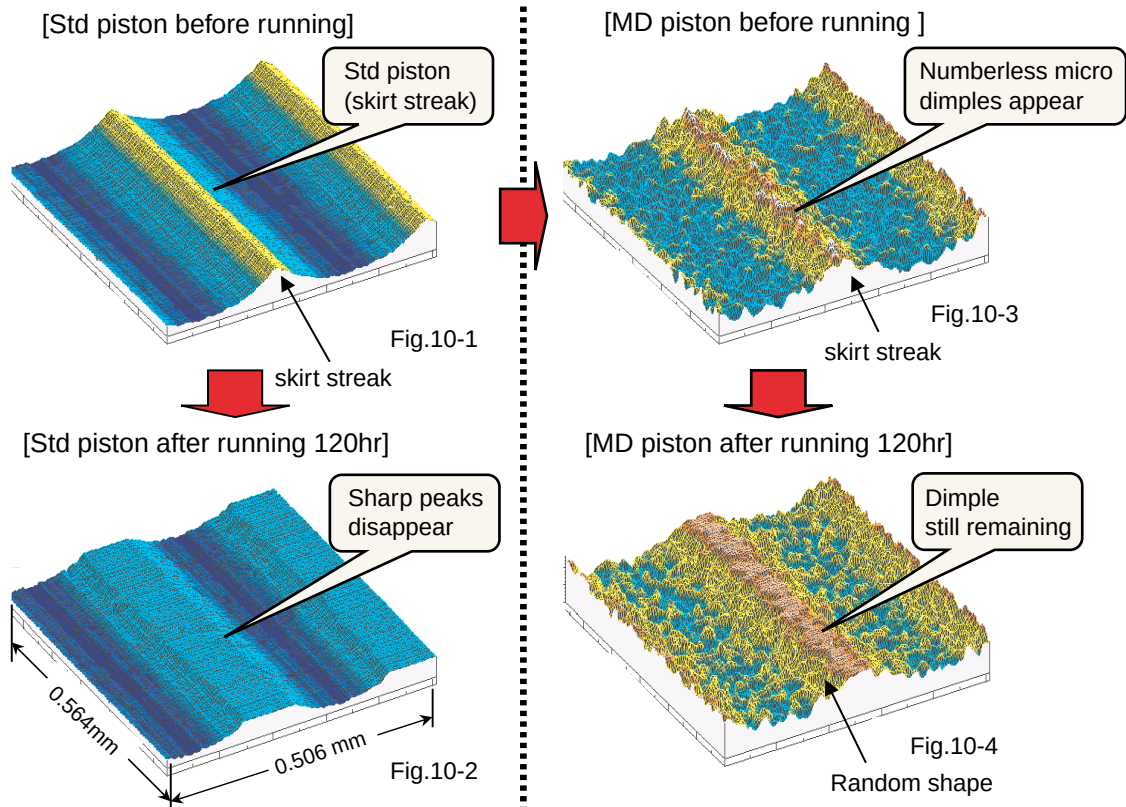


Fig.10 Characteristic of piston wear

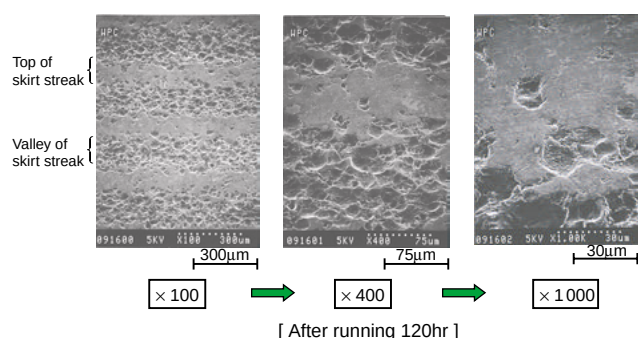


Fig.11 Looking at magnified MD piston skirt surface

in improved rubbing characteristics.

Figs. 12 and 13 show the results obtained by measuring the 2-dimensional surface shapes of piston skirt cross-sections with and without MD processing before and after prolonged engine operation. Countless microscopic dimples are formed by the MD processing, and the existence of these dimples can be verified even after the prolonged engine operation.

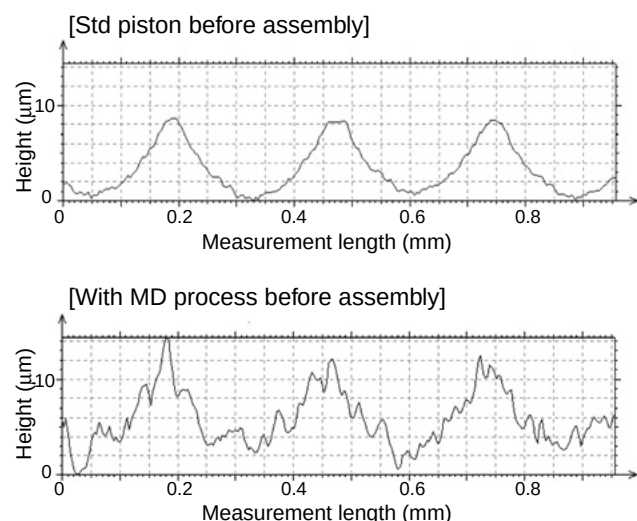


Fig.12 Surface curve of piston skirt-1

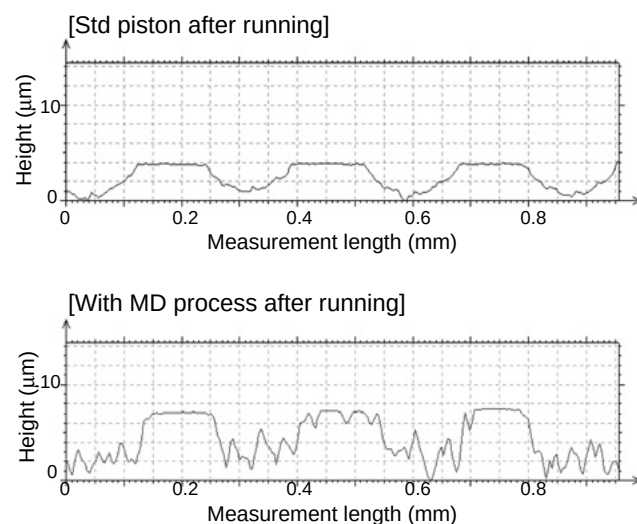


Fig.13 Surface curve of piston skirt-2

4. Summary

The following results were obtained by refining (MD processing) the surfaces of piston skirts with a shot peening process:

- (1) MD processing reduced the piston drive loss and cut the mechanical loss of the engine by 2% to 2.6%.
- (2) In terms of the MD processing conditions conducive to producing the most significant effect in reducing rubbing resistance, roughly spherical ceramic beads with a central particle diameter of 40 μ are shot at a shot peening pressure of 3.5 atmospheres with compressed air at speeds of 100 m/sec or more around the entire circumference of the piston skirts.

5. Conclusions

Application of the technology achieved through the authors' research has expanded since its initial application to Honda's first hybrid car, the Insight.

The authors believe that research to identify, improve and refine the kinds of local lubrication conditions which were explored in their own research will become more and more important with time.

Even in terms of surface properties, there is still a great deal of potential for improvement, development and application in terms of adoption with other parts, welding of different kinds of materials and improvements in surface roughness.

The authors intend to continue their efforts to develop the technology that will enable the creation at will of surface properties that correspond to the application.

Last but not least, the authors wish to acknowledge all the support they received in conducting their research and also express their sincere gratitude for all the valuable technological guidance and advice they received from Mr. Mase, Mr. Sugawara, Mr. Kumagaya and Mr. Ishiwata of Fuji Seisakusho and from Mr. Miyasaka of Fuji Kihan.

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Study of a Selective Catalytic Reduction Type Lean NOx Catalyst System

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ABSTRACT

The focus of this study on NOx reduction during lean-burn conditions was a Selective Catalytic Reduction (SCR)-type lean NOx catalyst. The SCR-type lean NOx catalyst exhibits stable NOx reduction performance under excess oxygen conditions.

We selected iridium (Ir) as the active species and an improved support material for enhanced selectivity of NOx reduction in competitive HC oxidation reaction under excessive oxygen conditions. This resulted in advanced NOx conversion.

Furthermore, catalyst durability, always a problem in SCR-type lean NOx catalysts, was improved by coating the catalyst with the material LaCoO_3 , which can convert NO to NO_2 .

Using this improved SCR-type Ir catalyst and performing engine control correlated with the conversion characteristics, a 63% NOx conversion rate was achieved under all lean-burn conditions in the 10-15 mode.

1. Introduction

Concern for global warming has increased in recent years, and, thus, improvements in automobile fuel economy are being demanded around the world to reduce CO_2 emissions.

Lean-burn combustion is an effective means of improving the fuel economy of gasoline engines. However, lean-burn exhaust gas contains a high oxygen concentration, making it difficult for conventional automobile three-way catalysts to reduce NOx. Thus, it is necessary to develop a catalytic system capable of converting NOx under excess oxygen conditions in order to improve the fuel economy of gasoline engines.

This study focused on selective-type catalysts that exhibit NOx conversion performance under excess oxygen conditions, and developed a catalytic system functional under all lean operating conditions in the Japanese 10-15 mode. The details are described below.

2. Focus and Approach

The dry adsorption method⁽¹⁾ has been proposed as a method of converting NOx under excess oxygen conditions using a catalyst. This method adsorbs and desorbs NOx using various precious metals, and performs catalytic reduction at the theoretical air-fuel ratio and reduction area. In addition, selective reduction methods using Zeolite catalysts⁽²⁾, compound oxide catalysts⁽³⁾ and precious metal catalysts⁽⁴⁾⁽⁵⁾ have also been proposed.

In order to achieve high fuel economy effects, this study promoted development focusing on methods using SCR

(Selective Catalytic Reduction)-type precious metal catalysts which are capable of stable NOx conversion even under excess oxygen conditions.

Development also aimed to improve heat resistance⁽¹⁾⁽⁶⁾⁽⁷⁾, which had been a problem for selective-type catalysts thus far.

3. Study Targets

This study focused on developing a catalytic system that is capable of stable NOx conversion even under excess oxygen conditions.

The development targets were NOx conversion ratios of 60% or more under all lean-burn conditions in the 10-15 mode when applied in an actual vehicle, and 60% or more for the discrete catalyst with a model gas which simulated exhaust gas.

4. Catalyst Development

4.1. Selecting the Active Species

Selective-type catalysts exhibit a competitive HC oxidizing reaction with O_2 and NOx under excess oxygen conditions. Thus, there tends to be insufficient HC and CO oxidization as NOx reduction selectivity is increased. Therefore, when developing the catalyst, the active species was selected from the following perspectives and assuming the application of a downstream oxidizing catalyst.

(1) The NOx conversion temperature range should be a temperature range that allows conversion by a downstream oxidizing catalyst.

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- (2) The NO_x conversion ratio should be high, and the conversion temperature range should be wide.

Fig. 1 shows the NO_x conversion characteristics by temperature for various catalyst active species.

Based on these results, iridium (Ir) was selected as the active species having a performance that most satisfies the conditions (1) and (2) above.

4.2. Setting the Model Gas

Ir catalysts have been reported to possess the characteristic of specifically reducing NO_x depending on the low boiling point unsaturated HC species. Therefore, the authors investigated NO_x conversion characteristics according to HC species. Fig. 2 shows the effects of HC species on NO_x conversion characteristics according to temperature. This shows that Ir catalysts convert NO_x for olefins and aromatics, which have unsaturated bonds, and that the conversion temperature range also changes according to the HC species. Therefore, this team set a model gas with a paraffin, olefin and aromatic component ratio matching

that of exhaust gas for evaluation. Table 1 shows the formulation of the set model gas.

Fig. 3 shows a comparison of NO_x conversion characteristics according to temperature for the model gas and the exhaust gas. Note that the temperature in this figure is the catalyst inlet gas temperature in the model gas tests, and the catalyst bed temperature in the actual vehicle tests. These results confirmed that the model gas could accurately simulate both the NO_x conversion temperature range and the NO_x conversion ratio for exhaust gas.

4.3. Improving NO_x Conversion Performance

SCR-type lean NO_x catalysts have a reaction mode that converts NO_x using HC as the reducing agent under excess oxygen conditions. This HC oxidizing reaction is a competitive reaction by the oxidizing agents O₂ and NO_x.

For that reason, an improved NO_x conversion ratio and expanded NO_x conversion temperature range can be expected by inhibiting HC oxidation by O₂ and increasing the selectivity of the NO_x reduction reaction. Therefore, the

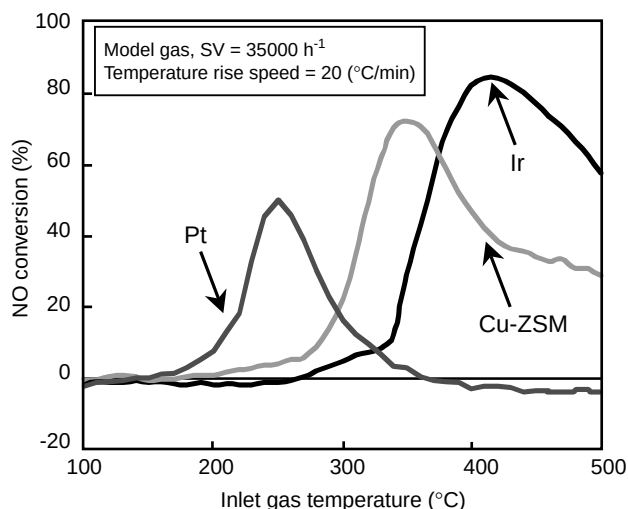


Fig.1 NO conversion characteristics of Ir, Pt and Cu catalysts

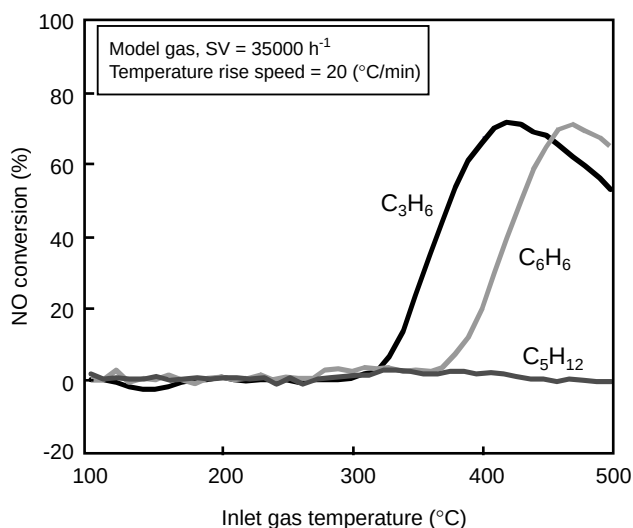


Fig.2 NO conversion characteristics of kinds of HC

Table 1 Formulation of model gas

Gas	Concentration
O ₂	7.4%
CO ₂	9.5%
T-HC	2800 ppmC
CO	800 ppm
NO	130 ppm
H ₂	300 ppm
N ₂	Bal.
H ₂ O	10%

HC: C₃H₆, C₆H₆, i-C₅H₁₂

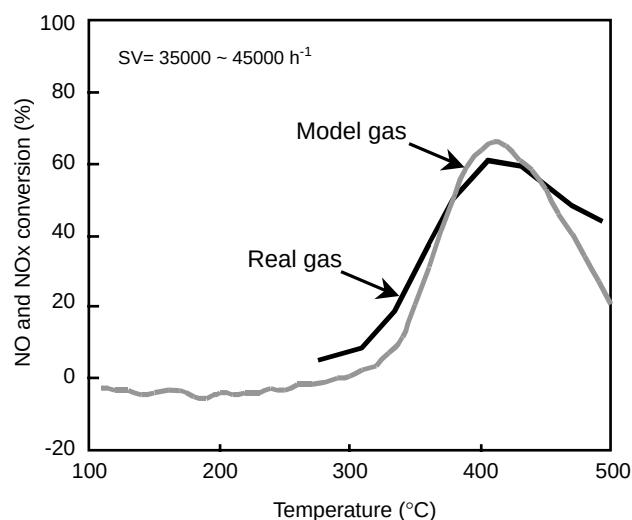


Fig.3 NO and NO_x conversion characteristics of experimental gases

Ir support materials were investigated as a means of inhibiting HC oxidation.

Fig. 4 shows the effects of different support materials on NO_x conversion characteristics using a flow method catalytic reactor in a fixed bed. These results show that when the support material has relatively little HC oxidation inhibition effect (Support C), the maximum NO_x conversion ratio is lower and the conversion temperature range is higher. In contrast, strengthening the inhibition effect of HC oxidation improved both the maximum NO_x conversion ratio and the conversion temperature range, thus improving the NO_x conversion performance.

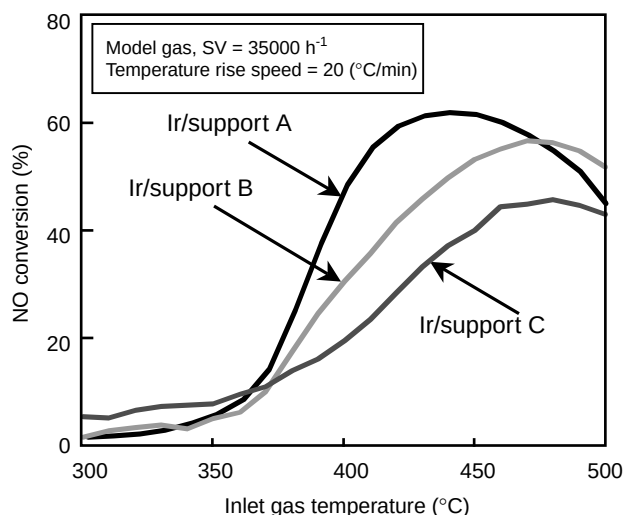


Fig.4 NO conversion characteristics of support materials

4.4. Improving Durability

4.4.1. Problems after Aging

Heat resistance is known to be an issue for selective-type catalysts⁽¹⁾⁽⁶⁾⁽⁷⁾. Therefore, durability was confirmed using a catalyst obtained by optimizing the support materials.

Fig. 5 shows the changes in NO_x and HC conversion characteristics using a flow method catalytic reactor in a fixed bed in a new condition and after aging equivalent to

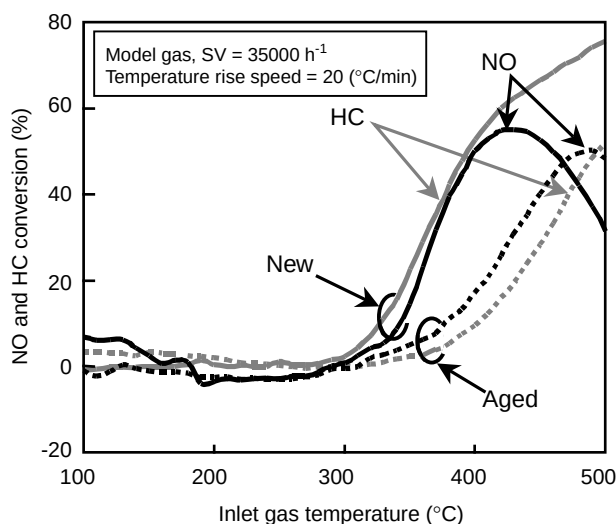


Fig.5 NO and HC conversion characteristics of aging

80,000 km. This shows that in addition to the oxidation capability for the HC that acts as the NO_x reducing agent, NO_x conversion also takes place at higher temperatures. Thus, under the aging conditions used in this study, the drop in NO_x conversion ratio and the higher NO_x conversion temperature range after aging are problems, and improvement is required in these areas.

4.4.2. Methods for Improving Durability

The NO_x reaction mechanism of the Ir catalyst was inferred, and the methods for improving this inferred reaction mechanism were investigated to solve the problems clarified in 4.4.1.

The NO_x reaction mechanism of the Ir catalyst under excess oxygen conditions is hypothesized to be as shown in Fig. 6. Here, the NO that accounts for the majority of the exhausted NO_x reacts with O₂ at the Ir surface to become the highly reactive NO₂, which is reduced by HC with unsaturated bonds⁽⁸⁾. Therefore, it is thought that if the NO has already been converted to NO₂ when it reaches the Ir surface, the reaction to convert NO to NO₂ at the Ir surface can be omitted, thus improving the NO_x conversion ratio and preventing a rise in the conversion temperature range. Coating the surface of the Ir catalyst with a material that converts NO to NO₂ was presumed to be an effective means for realizing this.

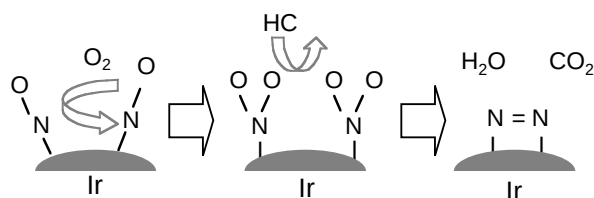
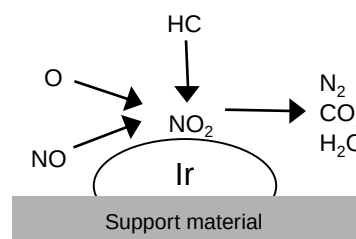


Fig.6 X Hypothesis of exhaust gas conversion mechanism

4.4.3. Investigation of NO₂ Conversion Materials

NO₂ conversion materials were investigated based on the hypotheses described in 4.4.2. The functions required of the NO₂ conversion material are as follows.

- (1) The material should have a high NO to NO₂ conversion efficiency.
- (2) The NO₂ conversion temperature range should match the NO_x conversion temperature range of the Ir catalyst.

Therefore, the authors focused on ABO₃ type compound oxides and base metal oxides as NO₂ conversion materials. When ABO₃ type compound oxides have an oxygen vacancy in their structure, this oxygen vacancy becomes a NO adsorption site as shown in Fig. 7, and at the same time the B site in the structure acts as an oxidation site. Therefore, a

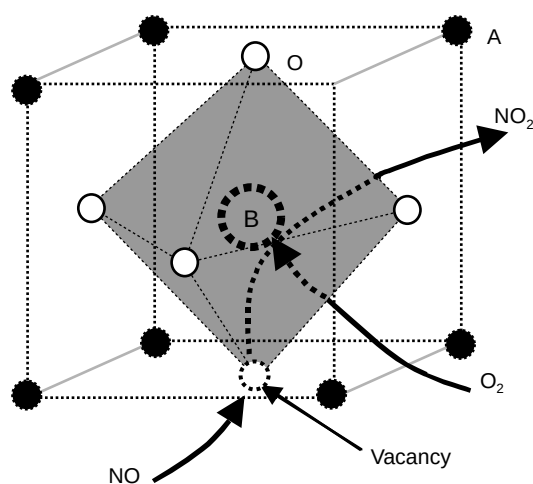


Fig.7 X Hypothesis of NO conversion to NO₂ mechanism

high NO₂ conversion efficiency can be expected due to the functions of both of these sites.

On the other hand, base metal oxides and particularly lanthanoid oxides typified by CeO₂ have NO_x adsorption capabilities and also exhibit oxidizing functions at temperatures of 300°C or higher, so a high NO₂ conversion efficiency can be expected.

Fig. 8 shows the NO₂ conversion characteristics of NO₂ conversion materials in a screening test using a flow method catalytic reactor in a fixed bed. These results show that in terms of NO₂ conversion characteristics, the LaCoO₃ compound oxide has a maximum conversion ratio of 70% in a temperature range which is near the NO_x conversion temperature range of the Ir catalyst, making it superior to the other materials. In addition, the LaCoO₃ compound oxide also has stronger oxidizing capability than the other materials, and can be expected to inhibit rises in the NO_x conversion temperature range. Based on these viewpoints, the LaCoO₃ compound oxide was selected as the NO₂ conversion material for coating the Ir catalyst surface.

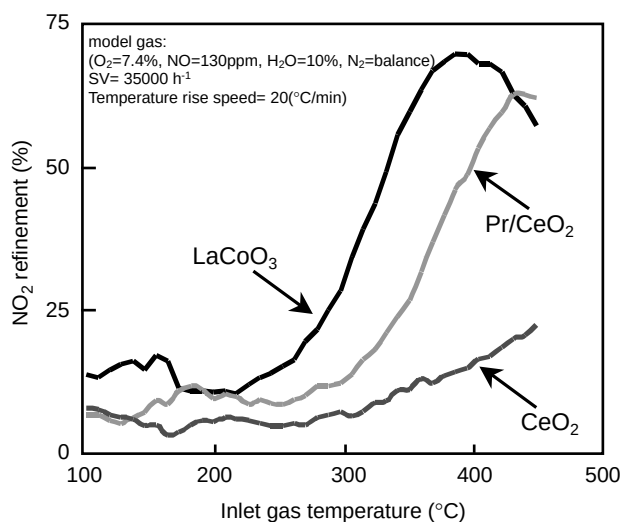


Fig.8 NO₂ conversion characteristics of materials

4.4.4. Performance Confirmation Results

Ir catalysts were coated with the LaCoO₃ obtained as described in 4.4.3. and the performance of catalysts aged to a mileage equivalent to 80,000 km was confirmed.

Fig. 9 shows the effects of the coated NO₂ conversion material on HC conversion characteristics, and Fig. 10 shows the effects on NO_x conversion characteristics in the new and aged conditions. The new material performance was improved by the high NO₂ conversion capability of the LaCoO₃. In addition, whereas oxidation occurred at higher temperatures without the coat, oxidizing performance was maintained by the NO₂ conversion capability which was preserved even after aging and also by the oxidizing power of the LaCoO₃ itself. As a result, the conversion temperature shift toward higher temperatures was inhibited, and the aged performance exhibited almost the same NO_x conversion characteristics as the new performance.

Thus, the NO₂ conversion material coat was able to solve the problems of a reduced NO_x conversion ratio and higher NO_x conversion temperature range in the aged condition.

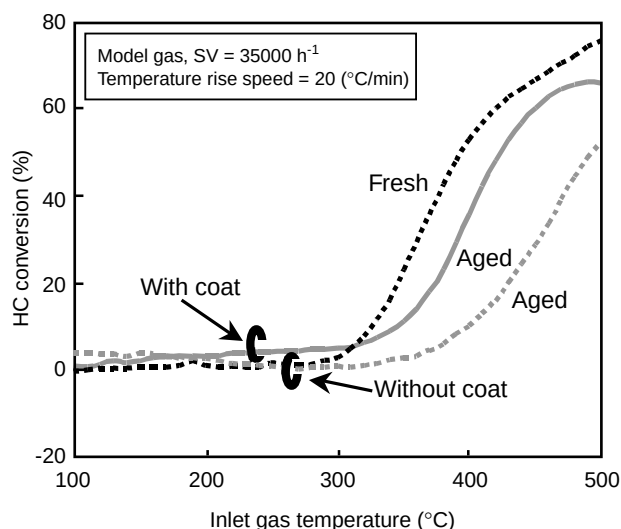


Fig.9 HC conversion improvements of the coating material

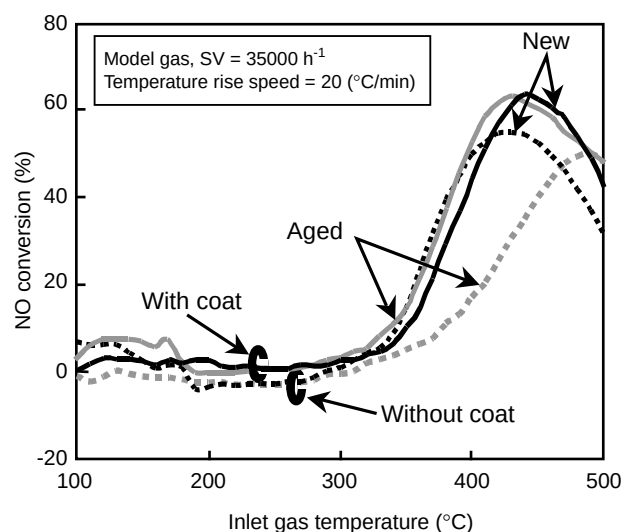


Fig.10 NO conversion improvements of the coating material

5. Evaluation of Actual Vehicle Emissions

5.1. Understanding Control Parameters

When constructing an actual vehicle system, it is important to obtain the maximum performance from the Ir catalyst. For this reason, the parameters governing NO_x conversion were investigated.

Fig. 11 shows the effects of O₂ concentration and HC/NO_x ratio on the NO_x conversion ratio using a flow method catalytic reactor in a fixed bed at 400°C. Note that the HC/NO_x ratio is the mass ratio. These results confirmed that lowering the O₂ concentration and increasing the HC/NO_x ratio are effective for improving the NO_x conversion performance of the Ir catalyst.

Based on these characteristics, HC/NO_x ratio and O₂ concentration control are required as engine control, so EGR control and air-fuel ratio feedback control using a UEGO sensor were adopted.

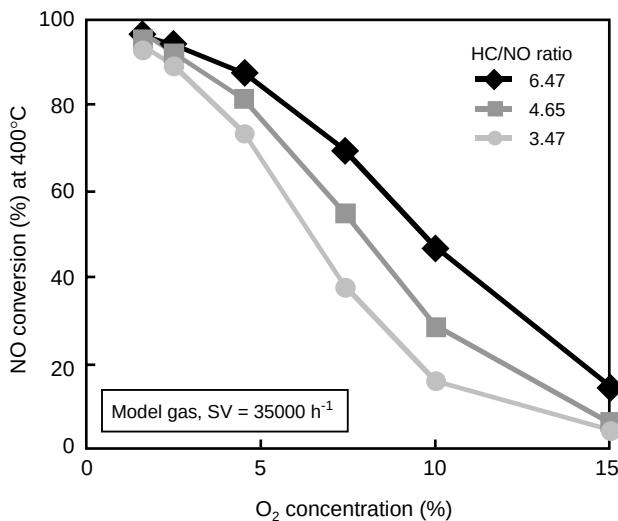


Fig.11 NO conversion characteristics of HC/NO ratios and O₂ concentrations

5.2. Conditions

The test vehicle was equipped with an in-line four cylinder 1.5 l prototype engine.

Based on the Ir catalyst characteristics, an EGR control system and an air-fuel ratio feedback control system using a UEGO sensor were adopted. In addition, emissions tests were performed in the 10-15 mode using a catalyst aged to an equivalent mileage of 80,000 km.

5.3. Results of 10-15 Mode Measurement

Figs. 12 and 13 show the instantaneous NO_x exhaust level, NO_x conversion ratio, and HC/NO_x ratio as defined by the mass ratio in the 10-15 mode using modal measurement.

These figures confirmed that NO_x conversion was obtained under all lean-burn conditions by controlling the HC/NO_x ratio which governs the NO_x conversion performance of Ir catalysts. Note that a conversion ratio of 63% was obtained in the 10-15 mode.

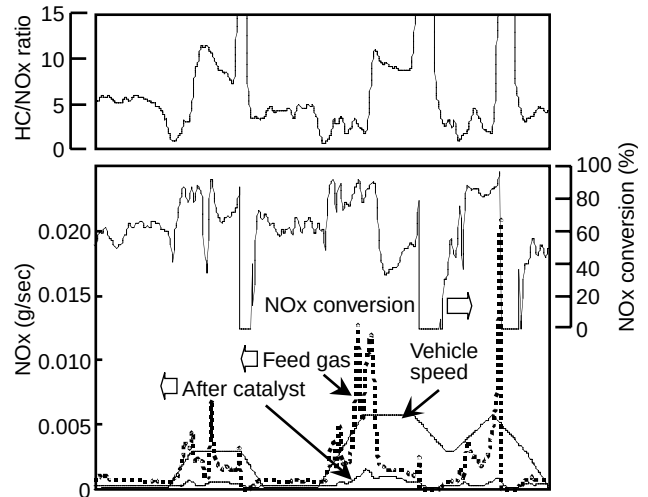


Fig.12 MODE test results (10 MODE)

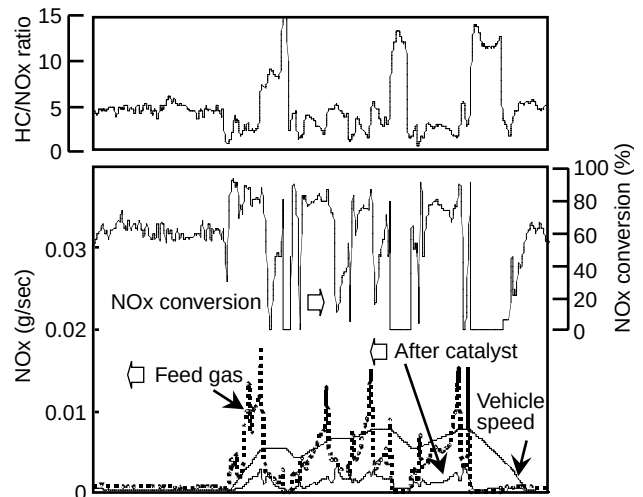


Fig.13 MODE test results (15 MODE)

6. Conclusion

Catalytic systems were studied focusing on SCR-type lean NO_x catalysts which exhibit stable conversion performance under excess oxygen conditions for NO_x conversion in lean-burn exhaust gas, and the following results were obtained.

- (1) Ir was selected as the active species, and it was confirmed that improving the support materials improved the NO_x selectivity in the HC and NO_x competitive reaction under excess oxygen conditions.
- (2) The problems of performance drops and higher conversion temperature ranges after aging were solved by coating the Ir catalyst surface with a LaCoO₃ compound oxide which has superior NO to NO₂ conversion capability in the NO_x conversion temperature range of the Ir catalyst.
- (3) Engine control was performed in accordance with the catalyst conversion characteristics, and a NO_x conversion ratio of 63% was obtained under all lean-burn conditions in the 10-15 mode.

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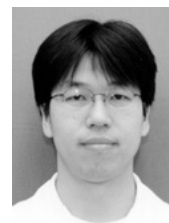
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Development of a Compact, Highly Efficient Traction Motor for Fuel Cell and Electric Vehicles

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Tomoyuki ITOH*

Hiroyuki SATOH*

ABSTRACT

This paper describes the development of a high efficiency, compact, high power motor applied as a traction motor for an electric vehicle.

Both magnetic torque and reluctance torque are utilized. In order to reduce the amount of magnetic material, the magnets are arranged so as to be exposed on the rotor surface. The rotor is constructed so that a magnetic salient pole is set between adjacent magnets and a portion of this salient pole is used as a hook so that both edges of the magnets are secured. This structure enabled a 35% reduction in the amount of magnetic material used in comparison to the interior permanent magnet (IPM) type rotor without sacrificing output torque.

The magnetic flux changes in a motor utilizing reluctance torque are great. One obvious problem caused by these changes is the heat generated by the rotor's eddy current. This motor insulates the magnet surfaces to cut off the eddy flow passage between the magnet and rotor yoke Si-Fe sheets. This leads to the reduction of the eddy current loss by up to 1 kW compared to a non-insulated type motor.

As a result, a maximum torque of 238 Nm, a maximum power of 60 kW and a maximum rpm of 11,000 rpm were obtained during testing with this motor. The motor output density was 1.8 kW/kg and torque density was 7 Nm/kg. Further, the total energy conversion efficiency of the motor inverter reached 93% average drive efficiency during Urban Dynamometer Driving Schedule (UDDS) mode operation.

1. Introduction

Today, there are ever-increasing demands for automobiles with reduced environmental impact. One method to address these demands is the development and market penetration of electric and fuel cell vehicles. The traction motors in such automobiles must be high efficiency, high power, lightweight motors.

Honda began leasing its mass-produced electric vehicle, the EV PLUS, in 1997. The motor installed in the EV PLUS is a uniquely developed permanent magnet motor called the "MCF11". This motor produced a power density of 1 kW/kg, a torque density of 6 Nm/kg, and an average drive efficiency of 90% in the Urban Dynamometer Driving Schedule (UDDS) mode.

At the same time, further motor evolution is necessary to facilitate an extended cruising range, lower energy consumption, and lower cost for future electric vehicles.

This paper discusses not only the aims and features of the newly developed next-generation traction motor (Gen II Motor) but also the knowledge gained during development, and the test performance results achieved with the experimental prototype.

2. Targets of Development

The development of the Gen II motor aimed at the following three targets:

- High efficiency and high power at a reduced weight
- Cost reduction
- Noise reduction

With an electric vehicle, highly efficient conversion of electricity to kinetic energy contributes to an extended driving range on a single charge. It is likewise important for reducing fuel consumption when electrical energy is supplied by a fuel cell.

An analysis of energy management simulation⁽¹⁾ has shown that a 1% increase in motor efficiency provides the same effect on reducing the energy consumption of an electric vehicle as when the vehicle weight is reduced by 50 kg (Fig. 1). It is necessary to improve motor efficiency in order to reduce energy consumption effectively.

On the other hand, it is necessary to reduce the motor size since it must be installed in the vehicle together with other components such as batteries and fuel cells. Further, a reduction in the amount of materials used, that is, a reduction

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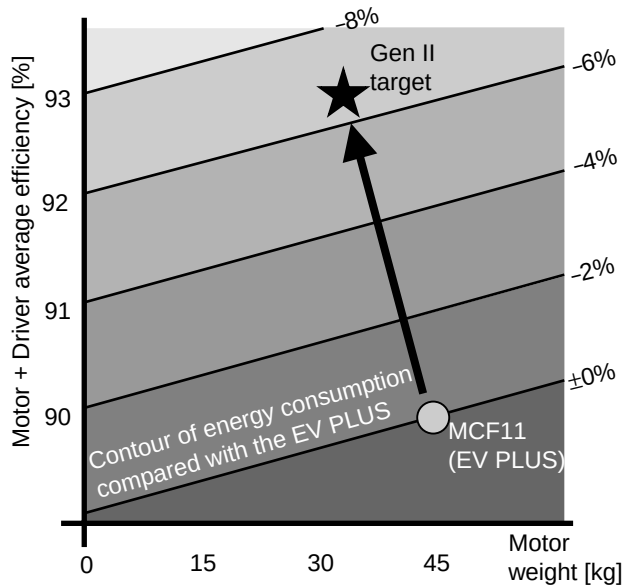


Fig.1 The relation of motor weight and efficiency to vehicle energy consumption during the UDDS mode

in motor size, is required in terms of cost reduction.

As it was necessary to achieve high efficiency and high output with significant weight reductions with the Gen II motor, we aimed for a 3% increase in efficiency and a 25% reduction in weight over the MCF11.

Further, we aimed to decrease motor noise in consideration of future mass production motors.

3. Features of the Gen II Motor

The appearance of the Gen II motor and its main specifications are shown in Figs. 2 and 3, and Table 1. Compared to the MCF11, the external diameter of the new motor has been reduced by 20%, and its weight has been reduced by 25%.

The Gen II motor is classified as a permanent magnet (PM) motor, the same as the MCF11. A PM motor generates a magnetic field through the arrangement of magnets on its

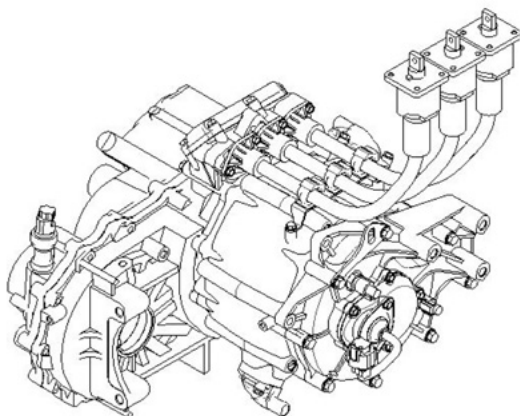


Fig.2 Gen II motor (with gearbox)

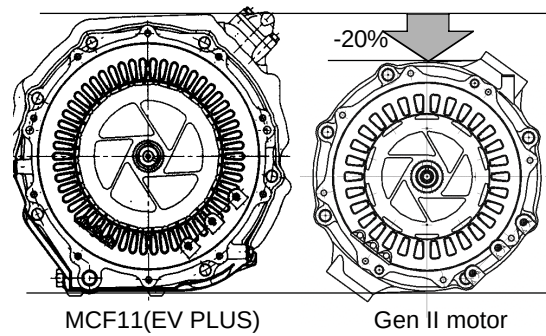


Fig.3 Section of motor

Table 1 Motor Specifications

	MCF11(EV PLUS)	Gen II motor
Type	PM synchronous motor	PM synchronous motor
Phase	3	3
Mass	45kg (without gearbox)	34kg (without gearbox)
Pole	18	10
Magnet	Nd-Fe-B (iHc>2.4MA/m)	Nd-Fe-B (iHc>2.8MA/m)
Rotor type	SPM	Salient SPM
Rotor cover	GFRP	None
Cooling system	Water cooled	Water cooled

rotor. The existence of this magnetic field renders the input of energy unnecessary and thus a PM motor is generally more efficient than other motor designs⁽²⁾. In addition, the Gen II motor uses the new technologies outlined below so as to achieve the aims noted in the preceding section.

3.1. Increasing Maximum Motor Revolution Speed

When the motor was mounted in a test vehicle in combination with a single-speed fixed gear, the relationship among maximum motor torque Q_{max} , maximum motor revolution speed N_{max} , maximum vehicle driving force T_{max} , and maximum vehicle speed V_{max} is described as follows:

$$T_{max} V_{max} = 2 \pi Q_{max} N_{max} \quad (1)$$

This relationship does not depend on gear ratio nor tire radius.

If the N_{max} is set higher, Q_{max} , which is a main factor in determining motor size and weight, can be reduced without sacrificing the vehicle T_{max} and V_{max} .

The N_{max} of the Gen II motor was set at 11,000rpm, 25% higher than that of the MCF11. The N_{max} was set after careful consideration of the balance between the N_{max} and the increased gearbox friction loss resulting from the increase in the revolution speed.

3.2. Use of Reluctance Torque

Below we have shown the basic formulas expressing the torque Tq and back electro-motive force (EMF) voltage V_a for a motor using only the rotational torque produced by the attraction and repulsion of the magnet and the stator pole (hereinafter referred to as "magnetic torque"). When the magnetic flux linkage is ϕ_a , the number of pole pairs is P_n , the electrical angular velocity is ω , the mechanical angular

velocity (revolution speed) is N , and the d- and q-axis currents are i_d and i_q , the following relationships result:

$$T_q = P_n \phi_a i_q \quad (2)$$

$$V_a = \phi_a \omega = P_n \phi_a N \quad (3)$$

The torque/current ratio is the torque constant K_t , while the back EMF voltage/revolution speed ratio is the back EMF constant K_e . Also, when the units K_t and K_e are expressed in a consistent form, we can say, based on formulas (2) and (3), that they take the relationship

$$K_e = K_t \quad (4)$$

When operated at a relatively high revolution speed, this kind of motor produces a V_a greater than the power voltage V_b , thus limiting operation. In the MCF11 motor, however, flux weakening control is used to equivalently suppress the magnetic field, to thereby expand the operating revolution speed range. This flux weakening control⁽³⁾, however, needs a current, which does not contribute to the production of torque, thus causing a drop in efficiency. Fig. 4 indicates the measured current necessary for flux weakening on the MCF11.

The decrease in efficiency resulting from the factors noted above can be suppressed by decreasing the K_e through the modification of the number of coil turns or the magnetic circuit to thereby reduce the flux weakening circuit. In such cases, however, it is impossible to avoid lowering K_t , which has the same value as K_e . These measures make it possible to reduce the loss at high revolution speeds, but the loss is increased at low revolution speeds, because the current for torque production increased. For these reasons, it is difficult to increase efficiency over a wide range of revolution speeds with motors if only magnetic torque is used.

As a result, with the Gen II motor, we decided to use not only magnetic torque, but also to give magnetic salience to the rotor, thus utilizing the rotational torque produced by the attraction between the salient and stator pole. In other words, the motor would make combined use of reluctance and magnetic torque.

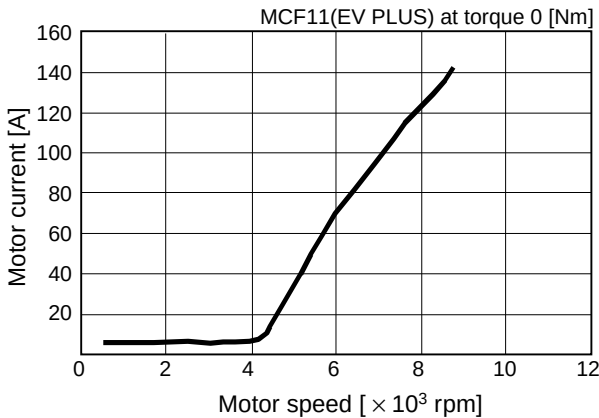


Fig.4 Motor input current for flux weakening

With a motor relying exclusively upon magnetic torque, it is impossible to avoid the principle whereby the torque/current ratio is equivalent to the back EMF voltage/revolution speed ratio. When the motor is designed to utilize reluctance torque such as this motor is, it is possible to keep the back EMF voltage/ revolution speed ratio lower than the torque/current ratio. It is also possible to expand the range of operation revolution speed, as it is not dependent on the flux weakening control, without sacrificing low-revolution number efficiency.

The torque T_q produced by the PM motor utilizing reluctance torque is expressed as

$$T_q = P_n \phi_a i_q + P_n (L_d - L_q) i_d i_q \quad (5)$$

where the d- and q-axis inductance values are L_d and L_q ⁽⁴⁾.

The generation of reluctance torque is dependent upon L_d and L_q having different values, with a greater difference between L_d and L_q resulting in increased generation of reluctance torque. With the Gen II motor, the number of coil turns was increased in order to achieve a high inductance and thereby increase reluctance torque.

The current required for flux weakening, i_d , is shown in formula (6). A high motor inductance increases the L_d , and also has the effect of lowering the flux weakening current.

$$i_d = -\frac{\phi_a}{L_d} + \frac{1}{L_d} \sqrt{\frac{V_b^2}{\omega^2} - (L_q i_q)^2} \quad (6)$$

Fig. 5 depicts a schematic of the foregoing measures.

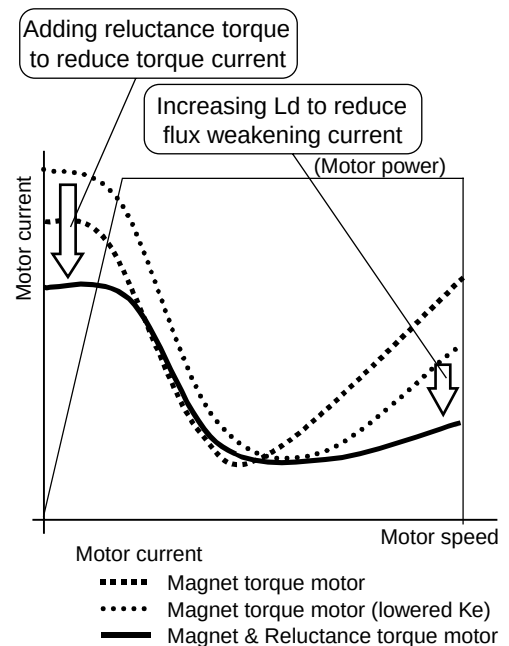


Fig.5 Motor current reduction concept

3.3. Salient Surface Permanent Magnet Rotor

A variety of configurations were considered for the rotor utilizing reluctance torque⁽²⁾. A typical design is the Interior Permanent Magnet (IPM) rotor configuration, in which magnets are embedded within the rotor yoke. Based on the fundamental principle that reluctance torque will be generated if the rotor possesses magnetic saliency, the Surface Permanent Magnet (SPM) configuration, in which magnets are exposed on the surface, can also be used to generate reluctance torque if it is constructed so as to provide magnetic saliency between the adjacent magnets.

The characteristics of the SPM, IPM and Salient SPM are shown in Table 2.

The IPM structure itself allows for inherently strong support of the magnets. However as magnetic bodies exist between the magnets and stator pole, there is the concern that magnetic flux shorting will occur within the rotor. This would result in the torque generated being relatively small for the amount of magnetic material used.

Using magnetic field analysis, the relationship of the torque constant K_t and magnet weight was compared for the IPM and the Salient SPM. The results of that analysis are shown in Fig. 6. Although it should be noted that the hook structure used to support the magnets in the Salient SPM was not modeled, the results of the analysis indicated that compared to the IPM, the Salient SPM would allow at least a 40% reduction in the amount of magnetic material used.

We also constructed IPM and Salient SPM prototypes and compared their actual operation. Results of the comparison confirmed that the Salient SPM required only 65% of the magnetic material used in an IPM to generate the same amount of torque (Fig. 7). Further, the Salient SPM rotor used in measurements was provided with the hook-shaped magnet support depicted in the structural diagram of Table 2.

Based on the above, we decided to use the Salient SPM in the Gen II motor as it would allow for a reduction in the amount of magnetic material used.

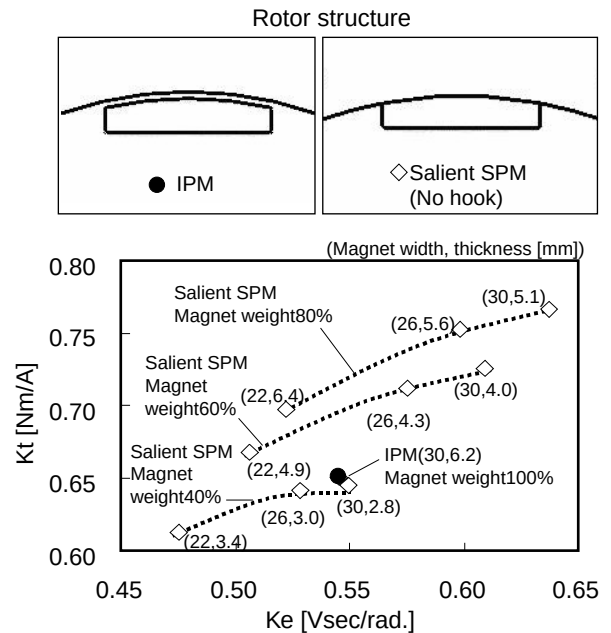


Fig.6 Calculation results of K_t , K_e by rotor structure and magnet shape

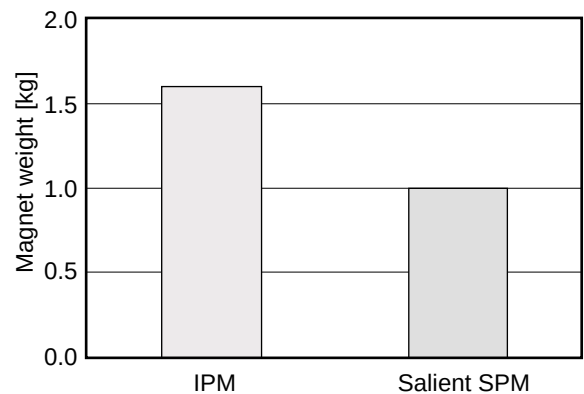


Fig.7 Amount of magnetic material used to reach the same motor torque

Table 2 Characteristics due to rotor structure

Type	SPM with non-magnetic cover	IPM	Salient SPM
Application	MCF11 (EV PLUS)		Gen II motor
Structure			
Reluctance torque use	Impossible to use	Possible to use	Possible to use
Magnetic air gap	Big air gap	Small air gap	Small air gap
Short magnet flux	Little	Short circuit due to magnetic flux in rotor	Little
Magnet torque/Magnet weight	Not so good	Not so good	Good
Rotor-stator magnetic linkage	Low inductance	High inductance	High inductance
Rotor eddy current	Small	Big	Big

3.3.1. Rotor Strength

The rotor in the Gen II motor is not furnished with a cover to prevent the magnets from separating. Instead, both edges of each magnet are held in place by hook-shaped protrusions constructed in the salient portion of the yoke. The shapes of the magnets and hooks were optimized by non-linear analysis as shown in Fig. 8. Destructive prototype testing showed the designs were capable of withstanding revolution speeds above 20,000 rpm.

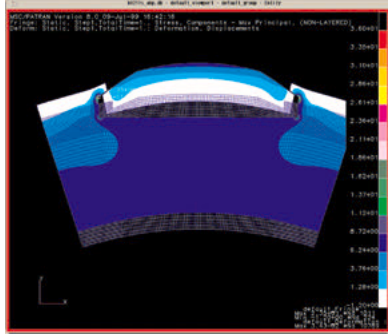


Fig.8 Nonlinear analysis used to optimize hook and magnet shape

3.3.2. Resistance to Irreversible Demagnetization

Significantly reducing the weight of magnets, increasing inductance, and exposing magnets on the rotor surface all contribute to increasing demagnetizing fields for the magnets. In order to maintain the magnets' resistance to irreversible demagnetization in the presence of large demagnetizing fields, we decided that the Gen II motor would be equipped with Nd-Fe-B magnets, which possess a coercive force of more than 2.8MA/m at room temperature. These magnets also have a coercive force of more than 1.0MA/m at 180 degrees Celsius (Fig. 9).

Magnetic field analysis also confirmed that the hook shapes used to support the magnets had the effect of enhancing the local permeance of the corners of the magnets (Fig. 10).

As a result of these combined effects, the Gen II motor

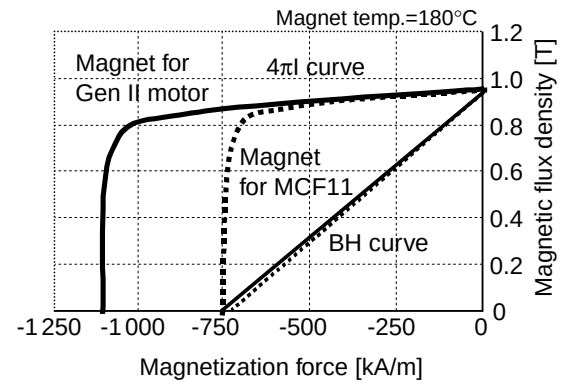


Fig.9 Properties of the magnetic material

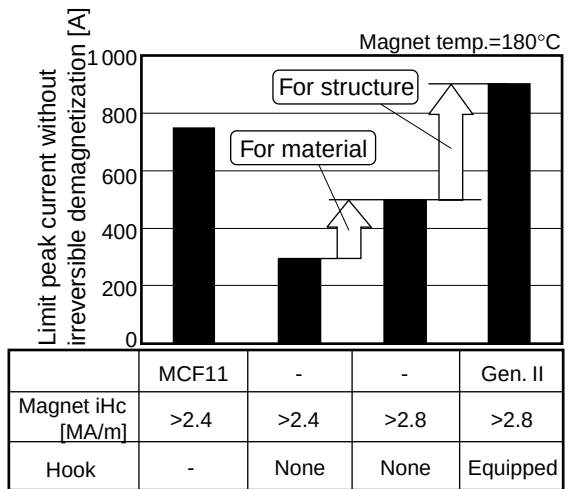


Fig.11 Simulation results of resistance to irreversible demagnetization

was provided with resistance to irreversible demagnetization equal to or greater than that of the MCF11 (Fig. 11).

3.4. Reducing Rotor Eddy-Current Loss

A PM motor utilizing reluctance torque has a tendency to experience higher iron loss in the rotor. In order to make the motor generate high reluctance torque, the salient

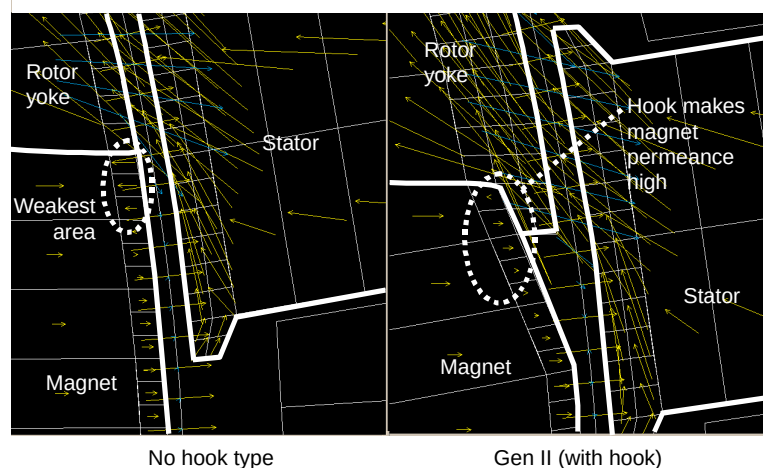


Fig.10 Motor flux density vector under high demagnetizing field conditions

magnetic material must be arranged near the stator and a higher inductance must be set. This results in a strong magnetic connection between the stator and rotor and magnetic flux fluctuations inside the rotor that are higher than those experienced within the conventional motor which uses only magnetic torque. It is believed that these flux fluctuations are responsible for generating the eddy current inside the rotor conductor.

Initially, the Gen II motor was designed to reduce eddy current loss by utilizing a rotor yoke made of the same laminated Si-Fe sheet steel as the MCF11. It was discovered, however, that greater-than-anticipated rotor heating would occur. When the temperature distribution in the rotor was measured, the results shown in Fig. 12 were produced, indicating variations in temperature distribution depending on the part of the rotor. Particularly high temperatures were measured at areas where the salient parts contacted the magnets.

When we considered the reason for this phenomenon, we discovered that since the punched surface of the laminated steel plate was in contact with the conductive magnet, an

eddy current loop composed of the magnets and Si-Fe sheets was formed as indicated in Fig. 13.

The eddy current in this loop can be suppressed by maintaining an insulating coating between the magnets and the Si-Fe sheet (Fig. 14). To this end, we tried coating the magnets with a resin, and performed the experiments again. The difference in motor loss with and without the insulation coating is shown in Fig. 15. A reduction in loss was detected during high-revolution speed, high-current operation, reaching a maximum 1 kW effect.

It is believed that eddy current loss depends on frequency and size of flux change, and that the loss is particularly generated under high revolution speed and high current conditions. Since this trend was evident in our results as well, we theorized that the insulation had the effect of reducing eddy current within the rotor.

3.5. Elimination of Resonance Noise

In the case of the EV PLUS, motor noise became quite noticeable at vehicle speeds around 80 km/h, when the motor reached a revolution speed of approximately 5,000 rpm (see

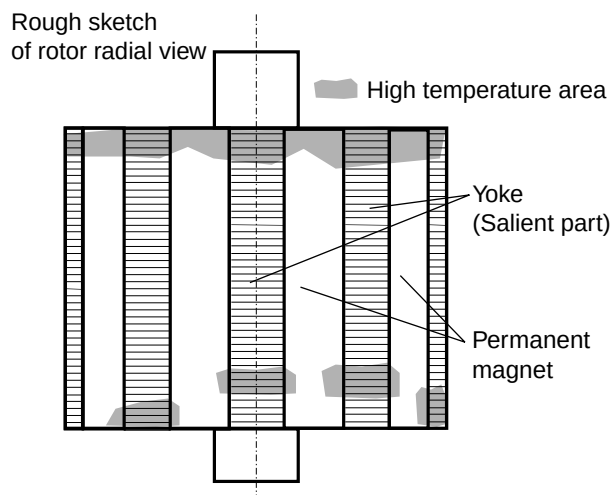


Fig.12 Rotor surface temperature distribution

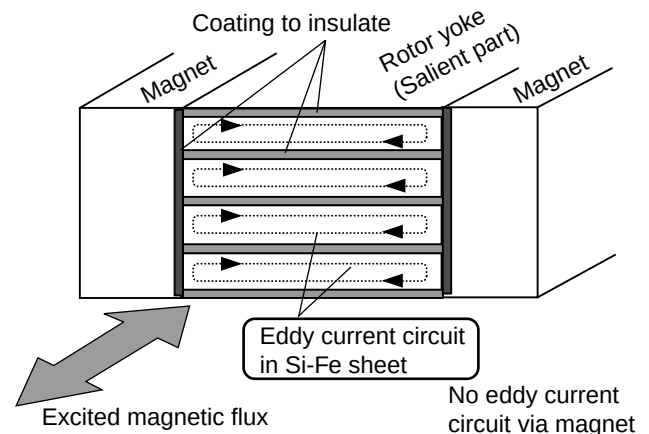


Fig.14 Eddy current behavior with insulation between magnets and Si-Fe sheets

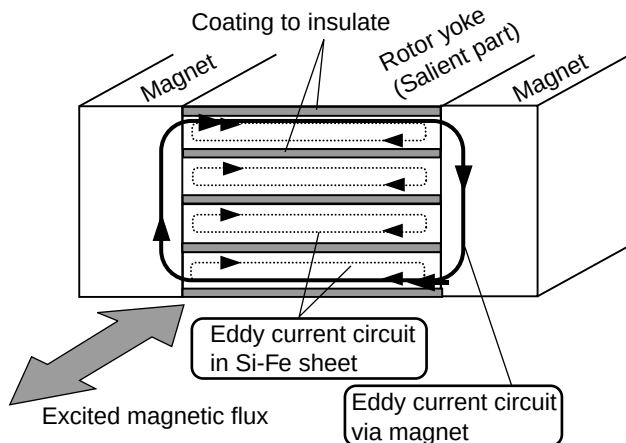


Fig.13 Eddy current behavior with no insulation between magnets and Si-Fe sheets

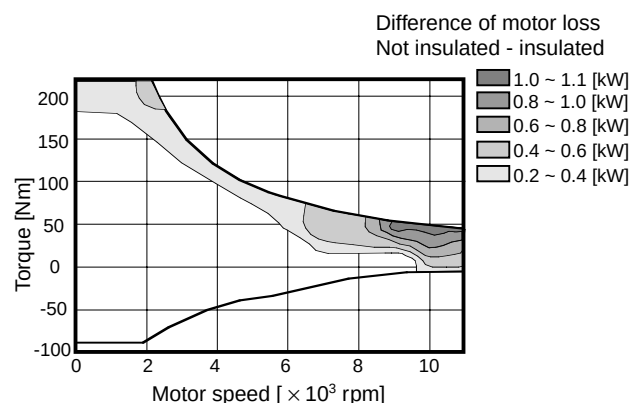


Fig.15 Effect of insulation between magnets and Si-Fe sheets

Fig. 16). When this noise was analyzed in detail, it was discovered that the peak of a very obvious noise, perhaps a resonance noise, occurred at this point (Fig.17).

The factors involved in this resonating noise were studied using eigenvalue analysis, frequency response analysis, and hammering tests, whereupon it was discovered that the resonance was occurring as a result of expansion and shrinkage of the stator cylinder. The expansion and shrinkage mode of a cylindrical shape can be expressed by the formula for the equivalent mode, which is the mode for the elongation of a plane such as that formed by cutting and flattening the cylinder in the axial direction. As a result, the resonating frequency fr can be expressed as

$$fr = \sqrt{\frac{E Trp Wrp}{Lrp Mst}} \quad (7)$$

where the Young's modulus of the stator is E , the return path height is Trp , width is Wrp , length is Lrp , and stator mass is Mst .

The three-phase current generates the attraction-and-repulsion force between the rotor and stator. This force becomes the excitation force whose primary excitation frequency fe is expressed as

$$fe = 6N Pa \quad (8)$$

A noticeable resonance noise is generated when fe and fr coincide at motor revolution speed Nr .

$$Nr = \frac{1}{6Pn} \sqrt{\frac{E Trp Wrp}{Lrp Mst}} \quad (9)$$

The resonance revolution speed Nr can be shifted from the range of actual operating revolution speeds by changing the parameters on the right side of formula (9). However, a

Table 3 Motor performance

	MCF11 (EV PLUS)	Gen II motor
Max power [kW]	49	60
Max torque [Nm]	275	238
Max speed [rpm]	8750	11000
Max torque × Max speed [k Nm rpm]	2406	2618
UDDS ave. efficiency [%]	90	93
Resonant speed [rpm]	4800	11600

particularly effective means of substantially shifting Nr is to change the number of pole pairs.

To fulfill these conditions, the Gen II motor was designed with five pole pairs so that the resonance noise occurred at 11,600 rpm, a point that exceeds the maximum normal operating revolution speed of 11,000 rpm.

4. Motor Performance

The measured performance of the Gen II motor is shown in Table 3.

4.1. Power Performance

Experimental results produced with the Gen II motor include a maximum torque of 238 Nm, a maximum power of 60 kW, and a maximum motor revolution speed of 11,000 rpm. The power density of the motor is 1.8 kW/kg, and torque density is 7 Nm/kg.

The sum of the multiplication of the maximum motor torque and the maximum motor revolution speed for the Gen II exceeds that for the MCF11, which means maximum revolution speed or maximum driving force can be improved over a vehicle equipped with the MCF11 (see formula (1)).

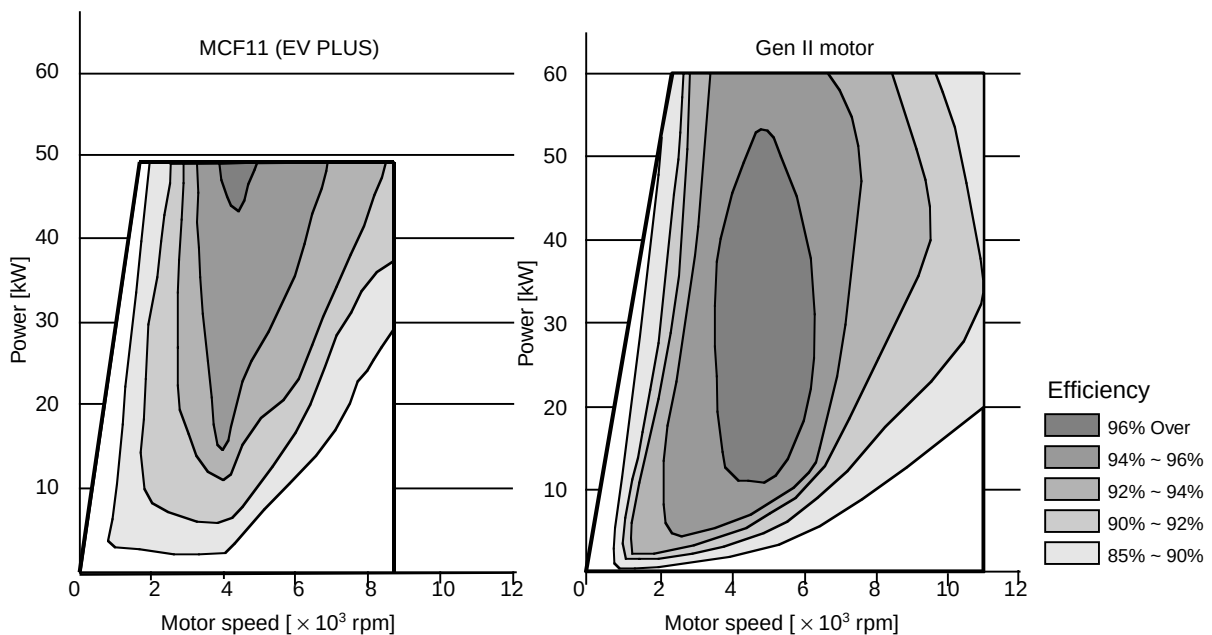


Fig.16 Motor efficiency and power performance (Inverter efficiency included)

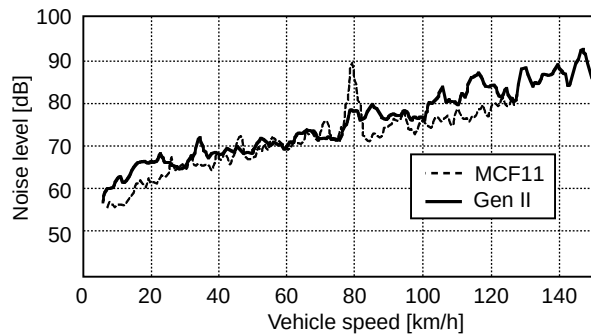


Fig.17 Acoustic noise of the vehicle

4.2. Efficiency

Fig. 16 shows the measured results of the overall motor and drive inverter efficiency. Compared to the MCF11, the Gen II motor achieves high efficiency over a wide range of revolution speeds.

The average efficiency when operating in the UDDS mode is 93%, indicating an improvement in efficiency of 3% over the MCF11's 90%. This increase is equivalent to a 30% reduction in power plant loss within the mode.

4.3. Noise

Fig. 17 shows interior noise measurements during gradual acceleration when the Gen II motor was installed in a test vehicle. The graph illustrates that the peak noise generated from resonance in the MCF11 is not produced by the Gen II motor anywhere within its range of actual operating revolution speeds. Further, these measurements were achieved when the MCF11 model was equipped with a plastic sound-insulation cover placed around the stator housing, while the Gen II sound measurements were achieved without such a device.

It should be noted that when measurements were made of resonance-acceleration in the motor housing, resonance generation was confirmed at the design revolution speed of 11,600 rpm.

5. Conclusion

This paper has discussed the development of a new traction motor for use in future electric and fuel cell vehicles. By combining the use of reluctance torque in the motor, the operating range was expanded and the efficiency was increased so that an average UDDS efficiency of 93% was obtained. The new motor also demonstrated high performance with a compact size, achieving a torque density of 7 Nm/kg.

This research also produced the following new knowledge:

- (1) The shape of the Salient SPM rotor has a large effect on reducing the amount of magnetic material required compared to the IPM rotor configuration.
- (2) The eddy current loss in the rotor is reduced by using Si-Fe sheets in the rotor and by providing insulation between the magnets and Si-Fe sheets.
- (3) By predicting the operating range wherein cylindrical

expansion and contraction resonance is generated and by changing the fundamental motor design characteristics, it is possible to suppress the generation of resonance.

Acknowledgment

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Quantification of Static Seating Comfort of Motorcycle Seats

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ABSTRACT

We established a posterior characteristics index (PCI) to quantitatively evaluate static seating comfort of motorcycle seats. This index expresses such characteristics as the shape and size of the posterior as well as the weight of a person. The static seating comfort of motorcycle seats was analyzed using the PCI and seated body pressure distribution data.

As a result, it was found that the contact surface area and level of surface area change in the body pressure distribution data obtained while the test subject was seated are effective indices for static seating comfort common to all test subjects.

1. Introduction

Static seating comfort, dynamic riding comfort, and fatigue caused by riding for long periods of time are important properties for motorcycle seats. Of these, static seating comfort is a particularly important factor when consumers select a motorcycle.

Motorcycle seats must be able to conform to the rider's movement (various riding positions) so that the driver can control the motorcycle. This type of seat performance is generally in conflict with static seating comfort performance. Therefore, motorcycle seats are characterized by having multiple shapes that provide a balance between the static seating comfort and the various riding positions required by the motorcycle model concept.

In conventional seat development, design parameters (e.g., hardness, shape, and materials) have been directly manipulated based on subjective feeling evaluations. However, this made it difficult to predict the effect of changes when multiple design parameters were changed simultaneously. On the other hand, body pressure distribution is influenced by personal differences of the test subject and seating area caused by, for example, differences in posterior size and bone structure. Therefore, conventional evaluations were performed with specified subjects. However, actual development requires a predictive equation which enables evaluation without specified subjects.

Therefore, the authors focused on body pressure distribution as a quantitative property intermediate between design parameters and the senses and developed the Posterior Characteristics Index (PCI). The PCI was then used for quantitative performance prediction of static seating comfort. The prediction flow is shown in Fig.1.

This paper describes an overview of the PCI and the results of quantitative prediction of static seating comfort.

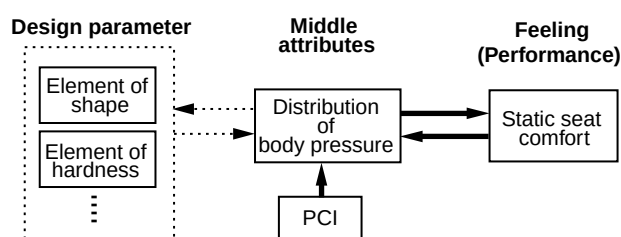


Fig.1 Prediction flow of static seat comfort

2. Static Seating Comfort Factor Analysis

2.1. Body Pressure Distribution of Motorcycle Seats

The shape of a motorcycle seat is designed taking into account not just sitting but also various riding positions, holding properties, and ease of foot-to-ground contact while stopping. The body pressure distributions of specified test subjects on scooters and on-road type and off-road type motorcycles are shown in Fig.2. These examples demonstrate the diversity of distributions obtained due to differences in seat shape.

2.2. Body Pressure Distribution of the Human Posterior

To learn the characteristics of the human posterior, body pressure distribution was measured with test subjects seated on a standard experimental seat (500 × 500 mm, 30 mm thick polyurethane foam) designed to contact the entire posterior area. The body pressure distributions of males and females by weight are shown in Fig.3. We can see from these results that there are differences in the characteristics of human posteriors, especially in terms of load, contact surface area, and level of contact surface pressure changes.

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The body pressure distributions of test subjects of different weights seated on the same motorcycle seat are shown in Fig.4. They indicate the same tendency as those of the subjects seated on the standard experimental seat though the absolute values are different.

Thus, when static seating comfort is evaluated using body pressure distribution data, it is necessary to minimize the effects of test subject posterior characteristics on evaluation results.

2.3. Hypothetical Equation for the Evaluation of Static Seating Comfort

First, we hypothesized that a seat with a large contact surface area and low levels of contact surface pressure changes throughout the body pressure distribution range has better static seating comfort. Next, we examined factors influencing static seating comfort, such as contact surface area, variations in contact surface pressure and the PCI of test riders, on different seats. Multiple regression analysis

was then performed ⁽¹⁾.

$$\text{Static seating comfort} = a_0 + a_1 \times (\text{contact surface area}) + a_2 \times (\text{contact surface pressure variations}) + a_3 \times (\text{PCI})$$

where,

Contact surface area is the total surface area of the portion of the subject's posterior in contact with the seat

Contact surface pressure variations are the variations in contact surface pressure of the contacting portion

PCI: Characteristics of the test subject's posterior

a_0 : Constant term

$a_1 \sim a_3$: Partial regression coefficients

3. Determining PCI

Prior to experimentation, the PCI for each test subject is established. Fifteen males and fifteen females were selected as test subjects based on the typical body weight distribution for Japanese. Fig.5 shows the body pressure weight

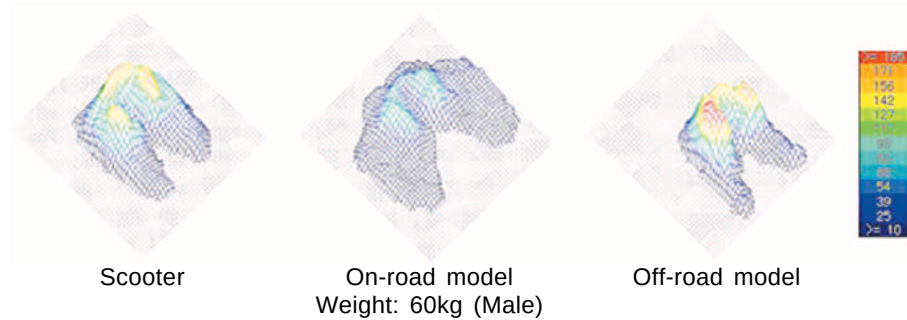


Fig.2 Body pressure distribution according to seat difference

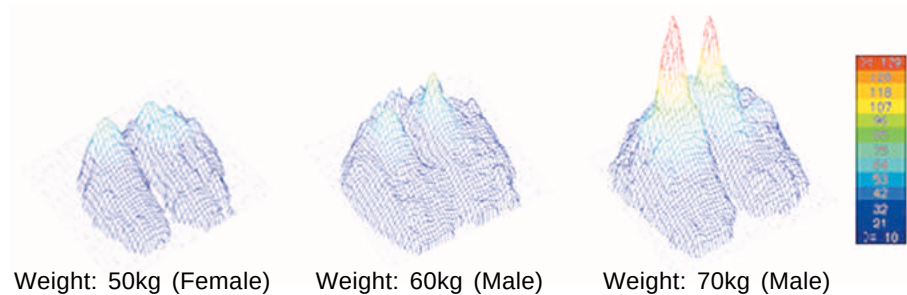


Fig.3 Body pressure distribution according to weight difference for test subjects seated on the standard experimental seat

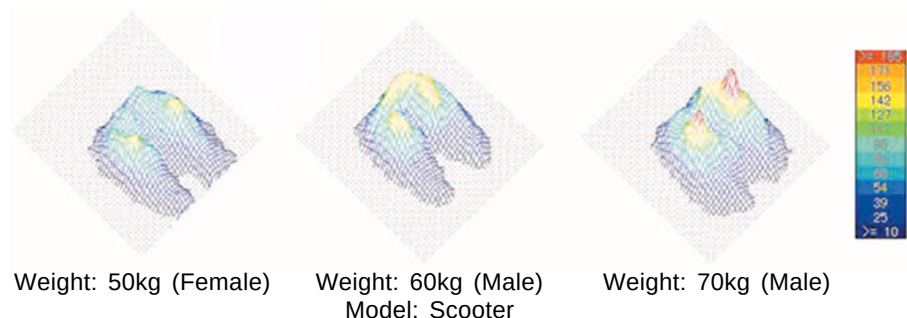


Fig.4 Body pressure distribution according to rider weight difference

distribution of each subject when seated on the standard experimental seat. We can see that there are considerable differences in contact surface area and contact surface pressure variations in these body pressure distributions. Based on this result, we judged that the overall characteristics of the posterior could be represented by the contact surface

area, the contact surface pressure variations and the test subject's weight, and principal component analysis was conducted. The results are shown in Table 1.

From the results of principal component analysis, we judged that the primary principal component expresses the total value for body weight, contact surface area, and contact

(F): Female (M): Male

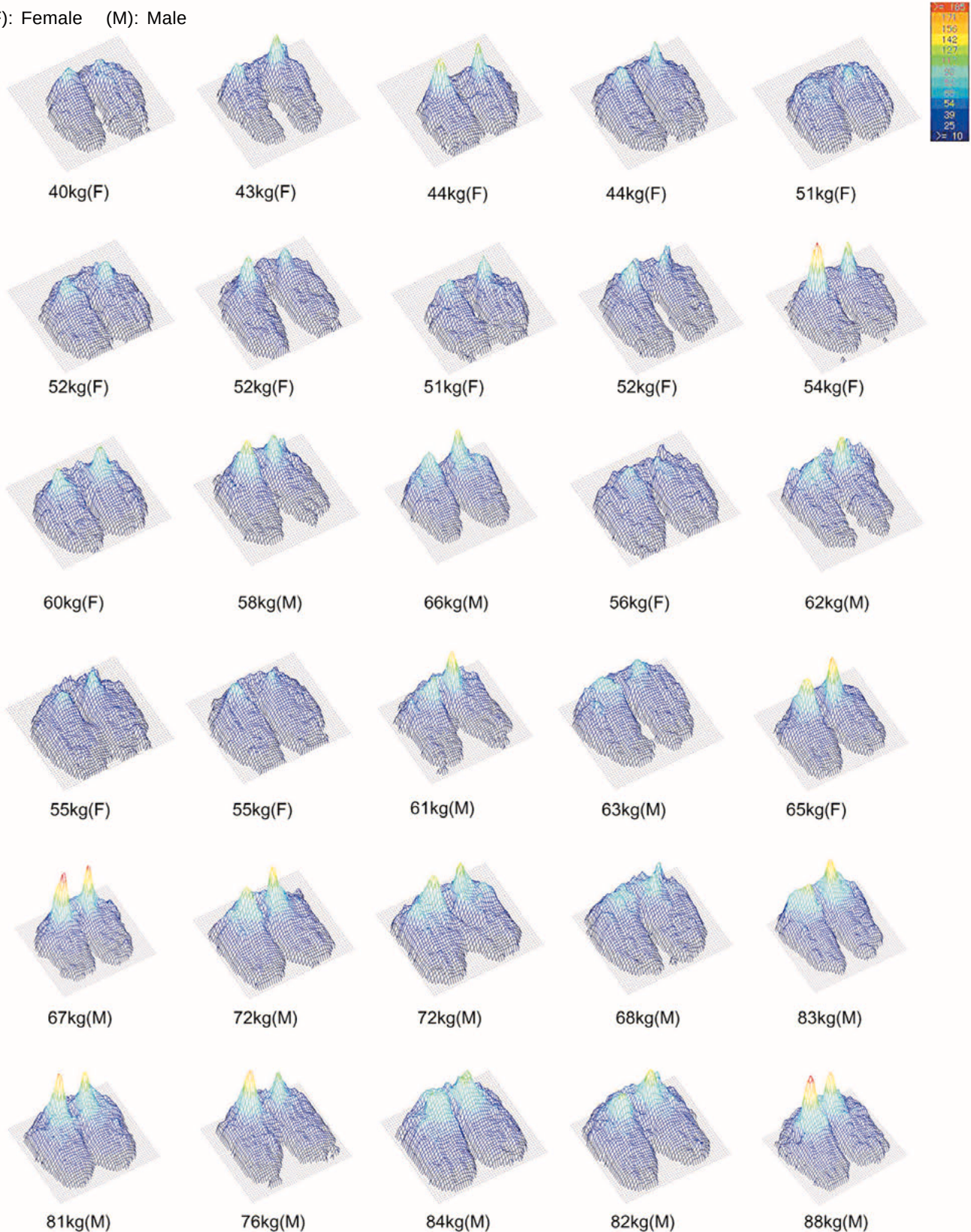


Fig.5 Body pressure distribution for various test subjects seated on the experimental standard seat

Table 1 Eigen value and eigen vector

Eigen value

No.	Eigen value	Contribution rate
Principal component 1	1.154	0.505
Principal component 2	1.129	0.376

Eigen vector

Items	Principal component 1	Principal component 2
Contact area	0.742	-0.005
Pressure changes	-0.123	0.917
Rider weight	0.686	-0.473

surface pressure variations and this component made the PCI. This PCI was then applied in a hypothetical equation. The principal component scores are shown in Fig.6.

4. Evaluation Test and Analysis

4.1. Selection of Subjects

A panel of five test subjects that reasonably represent the variations in the primary principle components, as shown in Fig.6, was selected from among the subjects used to establish the PCI.

4.2. Samples and Evaluation Test Method

Seven motorcycles and five subjects were used in the test. The subjects sat upright on each test motorcycle in a normal operating position. They took a normal riding posture as shown in Fig.7, and immediately thereafter evaluated

static seating comfort of the seat using the nine-point scale shown in Fig.8. In addition, body pressure distribution was measured under the same conditions as the static seating comfort evaluation. Fig.9 shows the system used to measure body pressure distribution.

4.3. Evaluation Test Results

Table 2 shows the results of the subjective feeling evaluation of static seating comfort. The range of standard deviation for the evaluation scores was 0.27 ~ 0.65.



Fig.7 Example of riding posture

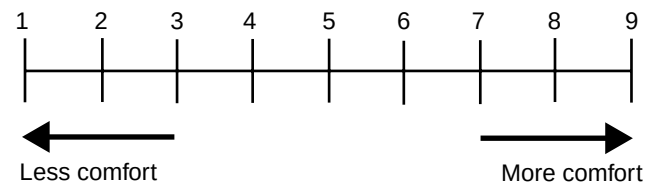


Fig.8 Evaluation scale for nine-point score

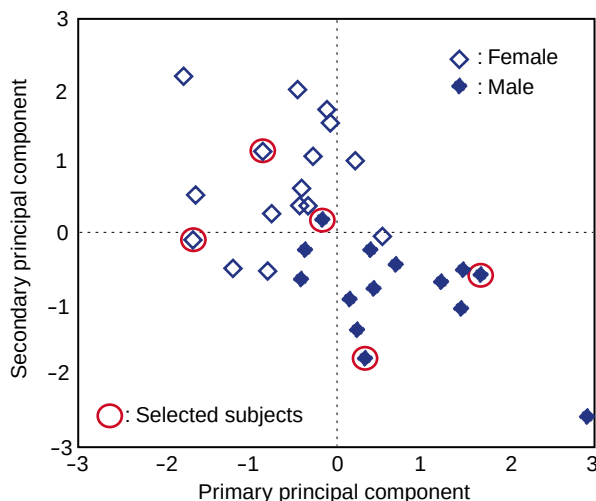


Fig.6 Graph of principal component scores and relation to each parameter

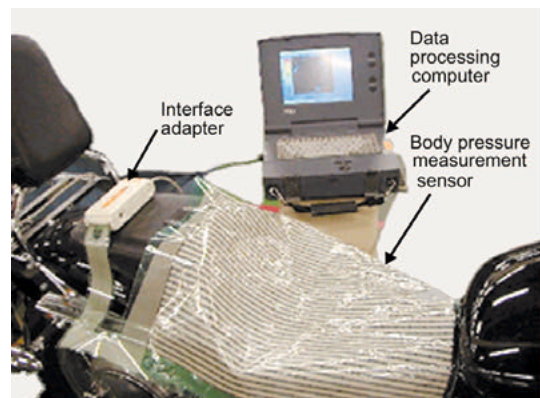


Fig.9 Body pressure measurement system

Table 2 Evaluation results

Sample motorcycles	Average of evaluation score	Standard deviation
Touring model A	8.2	0.55
Touring model B	7.5	0.27
American model A	7.9	0.41
American model B	6.3	0.54
European model	5.4	0.65
Scooter	3.4	0.54
Off road model	1.5	0.41

4.4. Results of Analysis

Multiple regression analysis was conducted for static seating comfort using the body pressure distribution data obtained during testing and the PCI for each subject. As a result, it was found that the contact surface area, contact surface pressure variations, and the PCI of each subject indicated by the hypothetical equation affected static seating comfort with a significance of 1%.

Thus, substituting body pressure distribution data into the hypothetical equation enabled the evaluation of static seating comfort. Fig.10 shows the relationship between prediction scores and evaluation scores.

Static seating comfort score = $1.733 + 0.008 \times (\text{contact surface area}) - 4.882 \times (\text{contact surface pressure variations}) - 0.594 \times \text{PCI}$

($R^2 = 0.91$, residual standard deviation = 0.74)

In the above equation, we can say that the contact surface area and contact surface pressure variations are valid indices for all subjects, and that PCI absorbs the individual posterior differences.

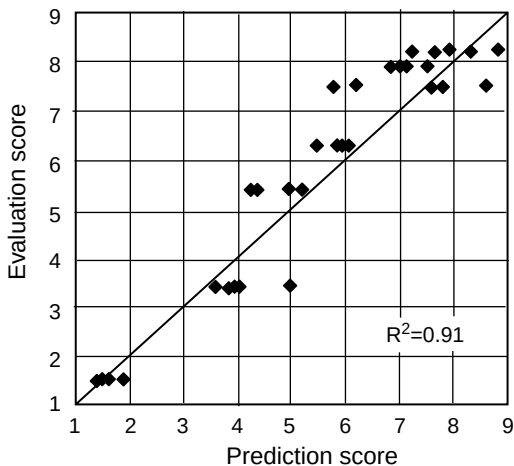


Fig.10 Correlation between prediction scores and evaluation scores

5. Validation with Other Types of Motorcycles

In order to validate the predictive equation, two on-road-touring motorcycles not used in the previous tests were evaluated in the same manner as those in the tests and body pressure distribution data were obtained. The body pressure distributions obtained through measurements on these two motorcycles are shown in Fig.11. Results of a comparison between the predicted score and the score obtained through evaluation show in Table 3 that the difference between the predicted score and subjective feeling evaluation scores are 0.2 and 0.1 in comparison to the 0.74 residual standard deviation derived from the predictive formula. This, thus, verifies the reasonableness of the predictive formula in motorcycle seat static seating comfort evaluations.

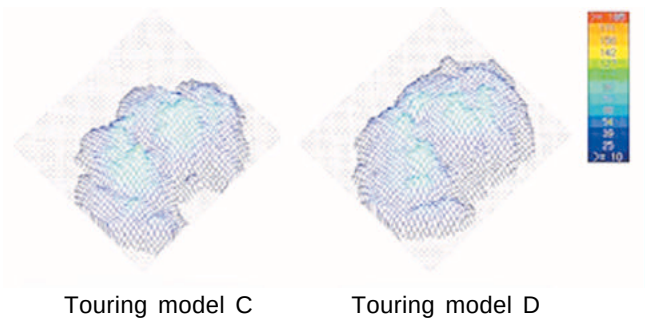


Fig.11 Results of body pressure distribution for different seats

Table 3 Comparison of evaluation scores and prediction scores

Sample motorcycles	Evaluation score	Prediction score	Residual error
Touring model C	6.0	6.2	0.2
Touring model D	7.0	7.1	0.1

6. Conclusion

1. An index of the characteristics of the human posterior was established by performing principal component analysis of body pressure distribution and body weight occurring when subjects are seated on a standard experimental seat.
2. It was found that the contact surface area and contact surface pressure variations of the body pressure distribution data obtained while the test subject was seated are effective indices for representing the static seating comfort of seats common to all subjects.

In the future, the authors will examine methods for predicting body pressure distribution from drawings.

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(Written in Japanese).

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Reducing Vibration of A 2-Cylinder Outboard Motor by the Pendulum Motion Mount Method

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ABSTRACT

First-order vibration, which an engine generates, has been isolated by applying the pendulum motion mount method to a 4-stroke 2-cylinder outboard motor. First, the vibration of the outboard motor was theoretically analyzed using a two-dimensional kinetic model. Through this analysis, the theory of isolating vibration using the pendulum motion mount method was established by finding the conditions which isolate vibration effectively. Next, this method was applied to the design of an outboard motor. The mount location was optimized in consideration of both the function and the vibration, and the mount structure was designed in consideration of endurance. The body of the outboard motor was designed by optimizing the vibratory characteristics in order to avoid the resonance. The experimental verification using the newly developed 4-stroke 2-cylinder small outboard motor revealed that the effectiveness of vibration reduction was equal to or better than that of an outboard motor equipped with a first-order balancer.

1. Introduction

Outboard motors are mounted as power sources on various types of small boats, and span a wide range of sizes from 2 PS class to over 200 PS. Engines also vary according to the boat size and application from single cylinder types to large scale V-6 type engines. Of these, 2-cylinder engines are mainly used in the 5 PS to 20 PS class, and are mounted on small fishing boats or pleasure boats, etc. Most outboard motors use 2-stroke engines, but the change to 4-stroke engines which have better emissions characteristics and fuel economy is spreading rapidly. 4-stroke 2-cylinder engines normally use a 360° crank which has one explosion per crank revolution.

In this case the exciting force (forces which induce vibration) generated by the engine is dominated by the first-order inertia force and driving torque reaction. Technologies used to reduce vibration include setting a soft mount rubber spring constant to improve the vibration isolating effects, or attaching a reciprocating balancer and rotating balancer to reduce the exciting force itself. The former method generally reduces the support stiffness of outboard motors, so simultaneously ensuring steerability and stability and isolating vibration becomes an issue. The latter method makes it possible to simultaneously isolate vibration and ensure steerability and stability, but adding balancers creates the new issues of increased weight, size, cost and mechanical noise.

In order to solve these problems of 4-stroke 2-cylinder outboard motors, vibration reduction technology for outboard motors was constructed using the pendulum motion mount method based on the theory previously reported in “Isolating Vibration through the Use of the Traveling Center of Engine Oscillation”⁽¹⁾. This paper presents the theory of reducing vibration by the pendulum motion mount method, mount and outboard motor body structure design methods, testing methods and results using the newly developed 4-stroke 2-cylinder small outboard motor BF9.9 as an example.

2. Vibration Isolating Methods and Aim

The outboard motor structure is shown in Fig. 1. The engine is integrally linked to the extension and the gear case, which is the drive portion of the system, to comprise the outboard motor. This outboard motor body is supported by upper and lower mounts which are attached to a swivel shaft (also called the mount frame). Fig. 2 shows the vibration transmission routes of outboard motors. The exciting force of 4-stroke 2-cylinder engines is dominated by the first-order inertia force and driving torque reaction as shown in this figure, and these forces become the vibration sources for boats equipped with outboard motors. The vibration of the excited outboard motor body is transmitted to the swivel shaft via the upper and lower mounts. A handle is integrated into this swivel shaft, and is supported by the swivel case in a manner that allows rotation. Vibration is transmitted to

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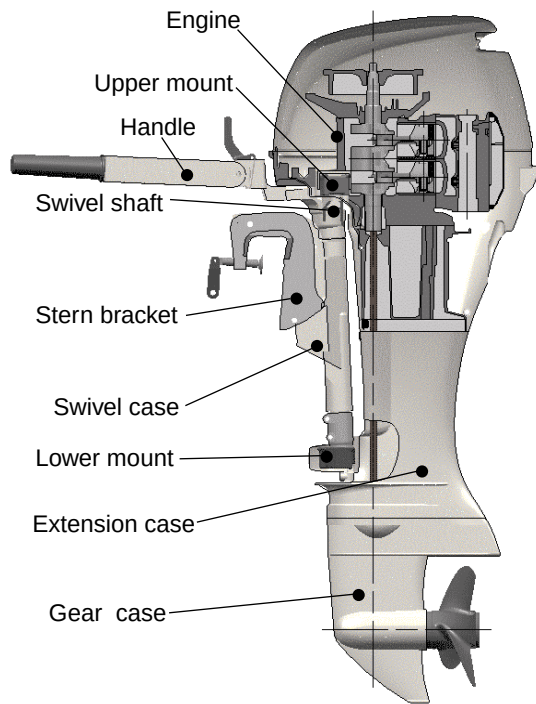


Fig.1 Cutaway view of outboard motor

the hull from the stern bracket to which this swivel case is fixed.

The aim of the pendulum motion mount method is to simultaneously achieve the same effective vibration and high support stiffness as when balancers are added. The basic concept is to generate lateral engine vibration which is perpendicular to the direction in which the thrust and self-weight act. Steerability, stability and reliability are assured by supporting the motor with high stiffness in the directions of the thrust, self-weight and steering effort, and sufficient vibration isolating effects are obtained by supporting the motor with low stiffness in the direction of the vibration. This concept is shown in Fig. 3.

Generating vibration in the lateral direction may adversely affect the hull and handle vibration, so suitable mount placement locations and characteristics must be set. In particular, these engines use a bar handle (hereafter "handle") which is directly supported by the operator's hand, so careful consideration must be given to vibration. In addition, the natural frequency of the outboard motor body in the lateral direction is generally near the normal engine speed, so countermeasures must be taken to eliminate the effects of resonance.

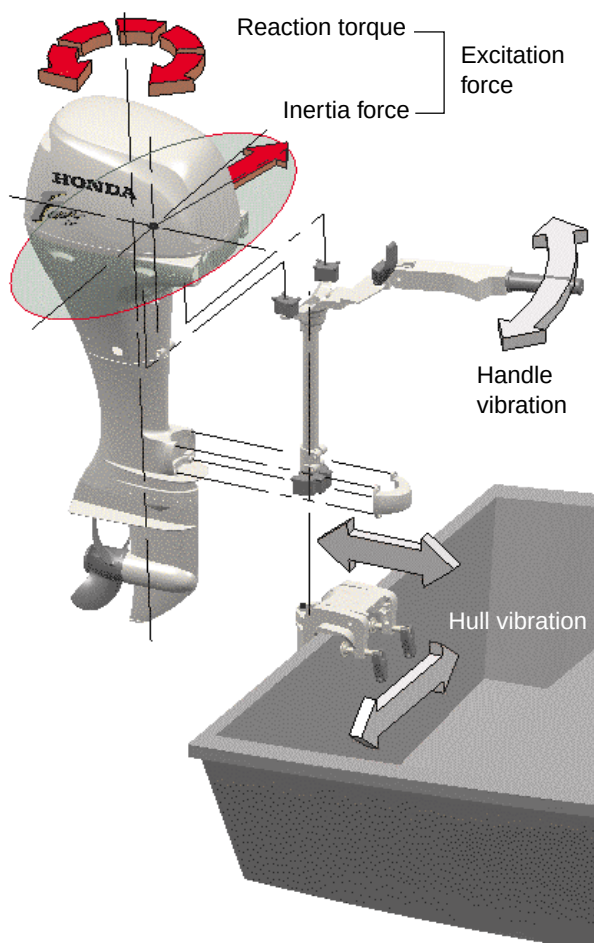


Fig.2 Vibration transmission to hull and handle

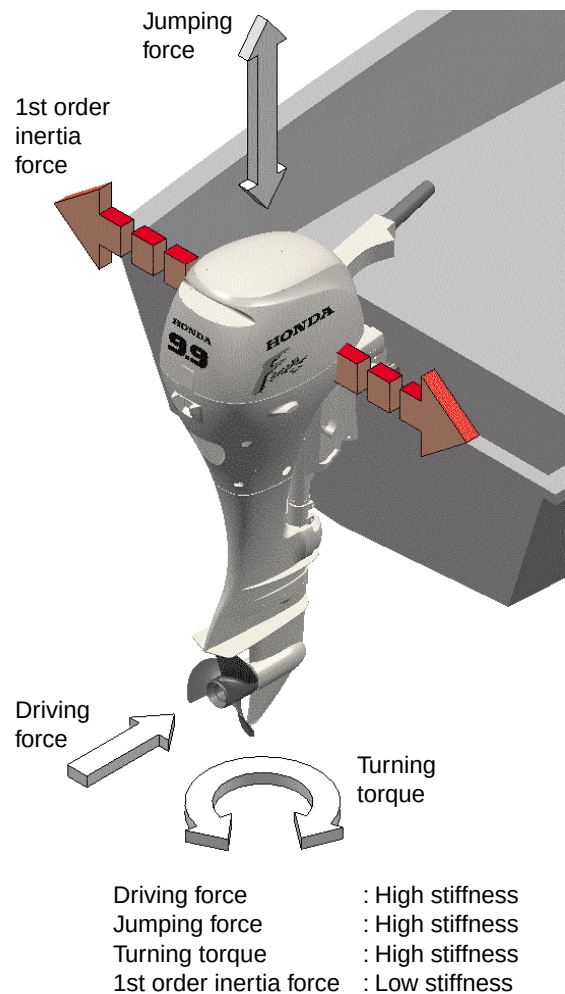


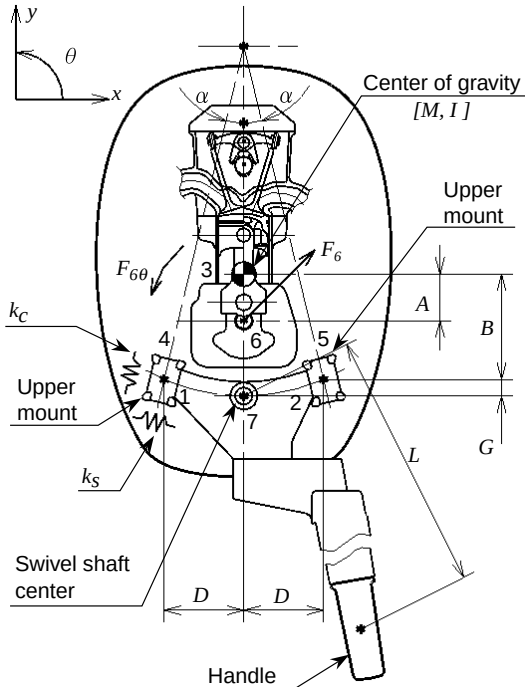
Fig.3 Concept of reducing vibration

3. Theory of Vibration Reduction by the Pendulum Motion Mount Method

The pendulum motion mount method has been reported to provide extremely high vibration isolating effects by generating the first-order inertia force of 4-stroke 2-cylinder engines in a single direction perpendicular to the cylinder axis, and supporting the engine by a mount having a small angle of inclination which satisfies the uncoupled condition.⁽²⁾ In this case, the first-order inertia force and driving torque reaction are dominant and also are approximately the same phase, therefore the engine vibration has a center of rotation (hereafter “center of vibration”) and the engine moves like a pendulum. Vibration is isolated utilizing this motion, so this method is named the pendulum motion mount method. This center of vibration moves according to the load and engine speed.

Fig. 4 shows an analysis model of an outboard motor as viewed from above, and the symbols used in this analysis are as follows.

- U_{ij} : Displacement of each nodal point
(i: Nodal point number, j: Direction)
- F_{ij} : Force acting on each nodal point
(i: Nodal point number, j: Direction)
- M : Engine mass
- I : Engine moment of inertia
- k_s : Spring constant in the shear direction
- k_c : Spring constant in the compressive direction
- α : Mount angle of inclination
- ω : Angular velocity



- Node number
- 1, 2 : Mount point at swivel shaft
- 4, 5 : Mount point at engine
- 3 : Center of engine gravity
- 6 : Rotating center of crankshaft
- 7 : Rotating center of swivel shaft

Fig.4 Two dimensional vibration analysis model

When the pendulum motion mount method is applied to outboard motors, the first-order inertia force produced by the engine is generated in the x direction in Fig. 4, so the outboard motor body should be supported from the swivel shaft by an engine mount having a small angle of inclination which satisfies the uncoupled condition. The swivel shaft is generally combined with the handle and attached to the stern bracket in a manner which allows rotation. The swivel case has a bolt for adjusting this friction, and increasing the friction constrains the rotating direction vibration of the swivel shaft. The following analysis was performed under these conditions, and handle vibration was evaluated by the constraining torque.

3.1. Exciting Force on the Hull

This section investigates the effects of the inertia force and driving torque reaction produced by the engine on the outboard motor and hull vibration. Structurally, the outboard motor has a small moment of inertia around the vertical axis compared to the moment of inertia around the other two axes, and exciting force is not generated in the vertical direction. Therefore, the outboard motor body was assumed to be a 2-dimensional rigid body, and its vibratory properties were investigated theoretically. The equation of motion without damping for the vibration system shown in Fig. 4 is as follows.

$$\begin{bmatrix} M & & \\ & M & \\ & & I \end{bmatrix} \begin{bmatrix} \ddot{U}_{3x} \\ \ddot{U}_{3y} \\ \ddot{U}_{3\theta} \end{bmatrix} + \begin{bmatrix} 2k_x & 0 & 2Bk_x - 2Dk_{xy} \\ & 2k_y & 0 \\ SYMM. & & 2B^2k_x - 4BDk_{xy} + 2D^2k_y \end{bmatrix} \begin{bmatrix} U_{3x} \\ U_{3y} \\ U_{3\theta} \end{bmatrix} = \begin{bmatrix} F_{6x} \\ F_{6y} \\ AF_{6x} + F_{6\theta} \end{bmatrix} \quad (1)$$

Where,

$$\begin{bmatrix} k_x \\ k_y \\ k_{xy} \end{bmatrix} = \begin{bmatrix} \cos^2 \alpha & \sin^2 \alpha \\ \sin^2 \alpha & \cos^2 \alpha \\ -\sin \alpha \cos \alpha & \sin \alpha \cos \alpha \end{bmatrix} \begin{bmatrix} k_s \\ k_c \end{bmatrix} \quad (2)$$

The uncoupled condition is shown in equation (3).

$$Bk_x = Dk_{xy} \quad (3)$$

At this time, the equation of motion (1) becomes as follows.

$$\begin{bmatrix} M & & \\ & M & \\ & & I \end{bmatrix} \begin{bmatrix} \ddot{U}_{3x} \\ \ddot{U}_{3y} \\ \ddot{U}_{3\theta} \end{bmatrix} + \begin{bmatrix} K_x & 0 & 0 \\ & K_y & 0 \\ SYMM. & & K_\theta \end{bmatrix} \begin{bmatrix} U_{3x} \\ U_{3y} \\ U_{3\theta} \end{bmatrix} = \begin{bmatrix} F_{6x} \\ F_{6y} \\ AF_{6x} + F_{6\theta} \end{bmatrix} \quad (4)$$

Where,

$$\begin{bmatrix} K_x \\ K_y \\ K_\theta \end{bmatrix} = \begin{bmatrix} 2k_x \\ 2k_y \\ -2B^2k_x + 2D^2k_y \end{bmatrix} \quad (5)$$

The frequency response in equation (5) is obtained as follows.

$$\begin{Bmatrix} U_{3x} \\ U_{3y} \\ U_{3\theta} \end{Bmatrix} = \begin{Bmatrix} \frac{F_{6x}}{-M\omega^2 + K_x} \\ \frac{F_{6y}}{-M\omega^2 + K_y} \\ \frac{AF_{6x} + F_{6\theta}}{-I\omega^2 + K_\theta} \end{Bmatrix} \quad (6)$$

On the other hand, the fulcrum reaction force in the swivel shaft constrained condition is expressed by the following equation.

$$\begin{Bmatrix} F_{7x} \\ F_{7y} \\ F_{7\theta} \end{Bmatrix} = \begin{bmatrix} -K_x & 0 & 0 \\ 0 & -K_y & 0 \\ (B+C)K_x & 0 & -K_\theta \end{bmatrix} \begin{Bmatrix} U_{3x} \\ U_{3y} \\ U_{3\theta} \end{Bmatrix} \quad (7)$$

In this study, the first-order inertia force produced by the engine is generated only in the x direction, so F_{7y} in equation (7) is 0. This shows that the exciting force F_{7x} on the hull becomes smaller as K_x becomes smaller. Therefore, extremely high vibration reduction effects with respect to hull vibration can be obtained by using a mount with a small angle of inclination.

3.2. Moment Exciting the Handle

The moment $F_{7\theta}$ which excites the handle is shown in equation (8).

$$F_{7\theta} = \left(\frac{B+C}{-(M/K_x)\omega^2 + 1} - \frac{A}{-(I/K_\theta)\omega^2 + 1} \right) F_{6x} - \frac{1}{-(I/K_\theta)\omega^2 + 1} F_{6\theta} \quad (8)$$

Furthermore, the properties of the moment $F_{7\theta}$ which excites the handle were investigated in detail by dividing equation (8) into the moment at low and high engine speeds as follows.

3.2.1. Moment Exciting the Handle at Low Engine Speeds

$AF_{6x} \ll F_{6\theta}$ at low engine speeds, so F_{6x} in equation (8) is assumed to be 0.

$$F_{7\theta} = -\frac{F_{6\theta}}{-(I/K_\theta)\omega^2 + 1} \quad (9)$$

In other words, this shows that as the rotating stiffness K_θ approaches 0, the exciting force on the handle also approaches 0. The following relationship is obtained from the uncoupled condition equation (3) and the condition where the rotating stiffness $K_\theta = 0$

$$k_x k_y = k_{xy}^2 \quad (10)$$

Equation (11) is obtained by substituting equation (2) into equation (10).

$$k_s k_c = 0 \quad (11)$$

The spring constant k_c in the compressive direction is generally a larger value than the shearing spring constant k_s , indicating that mount rubber with a small shearing spring constant is necessary. Setting $k_s = 0$ and substituting equation (2) into the uncoupled condition equation (3) results in the following relational equation.

$$(B \sin \alpha - D \cos \alpha) \sin \alpha = 0 \quad (12)$$

The above equation indicates the following two conditions.

(Condition I) $\tan \alpha = D/B$

(Condition II) $\alpha = 0, D = 0$

Condition I is the conventional inclined mount. Condition II is the mount with a small angle of inclination, and corresponds to the ultimate form when $k_c \gg k_s$. When using an engine mount with a small angle of inclination, the mount span D should be set as small as possible within the range that does not impair steerability and stability. Furthermore, in 4-stroke 2-cylinder engines $AF_{6x} = F_{6\theta}$ is true near the idle range, which means $U_{3\theta} = 0$. Therefore, it is predicted that handle vibration at low engine speeds can be vastly reduced by using a pendulum motion mount.

3.2.2. Moment Exciting the Handle at High Engine Speeds

$AF_{6x} \gg F_{6\theta}$ at high engine speeds, so $F_{6\theta}$ in equation (8) is assumed to be 0.

$$F_{7\theta} = \left(\frac{B+C}{-(M/K_x)\omega^2 + 1} - \frac{A}{-(I/K_\theta)\omega^2 + 1} \right) F_{6x} \quad (13)$$

Equation (13) can be transformed as follows.

$$F_{7\theta} = \left(\frac{B+C}{-(M/K_x) + 1/\omega^2} - \frac{A}{-(I/K_\theta) + 1/\omega^2} \right) \frac{F_{6x}}{\omega^2} \quad (14)$$

Assuming that $F_{7\theta} = 0$ in equation (14) is the condition required to cancel $F_{7\theta}$ at high engine speeds, the following equation is obtained.

$$\frac{K_x}{M} = \frac{A}{B+C} \frac{K_\theta}{I} \quad (15)$$

In other words, if the natural frequency in the translational direction can be matched with the frequency obtained by multiplying the natural frequency in the rotational direction by $\sqrt{A/(B+C)}$, the handle vibration at high rotation speeds can also be canceled.

3.3. Summary of Theoretical Analysis Results

Theoretical analysis showed that the pendulum motion mount method can reduce both handle vibration and hull vibration at all engine speeds. These results are summarized below.

- (1) The pendulum motion mount method can effectively reduce vibration caused by inertia force transmitted to the hull.
- (2) Vibration caused by the driving torque reaction at low engine speeds can be isolated equally by applying the conventional inclined mount or the pendulum motion mount method. In addition, with the pendulum motion mount method, the exciting force is reduced by having the driving torque reaction and inertia force cancel each other out near the idle range, thus vastly reducing vibration.

- (3) Handle vibration and the moment of excitation to the hull caused by the inertia force at high engine speeds can be canceled by applying the pendulum motion mount method.

4. Applied Design for Outboard Motors

4.1. Structural Design of the Engine Mount

As stated above, the BF9.9 design aimed to maximize vibration isolating performance while ensuring support stiffness with respect to self-weight, thrust and steering effort. Support stiffness with respect to thrust generally tends to be proportional to the product of the mount spring constant and the distance between the upper and lower mounts. On the other hand, with the pendulum motion mount method, the vibration isolating performance tends to increase as the lateral spring constant decreases. Therefore, the structure was provided with a long distance between the upper and lower mounts by lowering the lower mount within the range where problems with water resistance or spray do not occur. As a result, the partial load acting on each mount decreased. This makes it possible to set low mount spring constants while still maintaining the same support stiffness.

The lower mount shown in Fig. 5 was designed as a one-piece mount in consideration of water resistance and splashing conditions. The upper mount had a somewhat large mount span in consideration of steerability and stability, and used an angle of inclination α such that the mount rubber faced roughly toward the position of the center of percussion I/AM.

The mount rubber was selected to give the upper and lower mounts nonlinear spring characteristics to further reduce vibration at low speeds. Fig. 6 shows the upper mount structure. Here, rough-textured rubber is encased in a baked housing using a ribbed mandrel. Fig. 7 shows the nonlinear characteristics obtained by this structure.

4.2. Structural Design of the Outboard Motor

The theory of vibration isolating support described in Chapter 3 assumes that the outboard motor body, that is to say the sprung portion, is a rigid body. An outboard motor body has long vertical structures, so the natural frequency of the outboard motor body easily approaches the normal engine speed. If the engine speed approaches the resonance point and the outboard motor body oscillates as an elastic body instead of a rigid body, the above-mentioned pendulum motion mount method theory is not applicable. Furthermore, this vibration is transmitted via the mounts to the handle and the hull. This vibration causes a deterioration of the vibration feeling. Therefore, the authors focused on the fact that natural frequencies and natural vibration modes greatly affect outboard motor body vibration, and designed the structure to prevent the outboard motor body from resonating.

4.2.1. Natural Vibration Mode of the Outboard Motor Body

Methods for minimizing the effects of resonance were investigated. A general multi-degree of freedom equation of motion without damping is shown below.

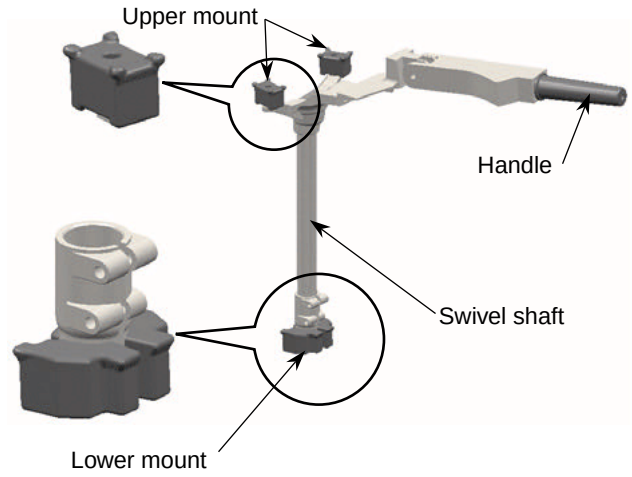


Fig.5 Structure around mount

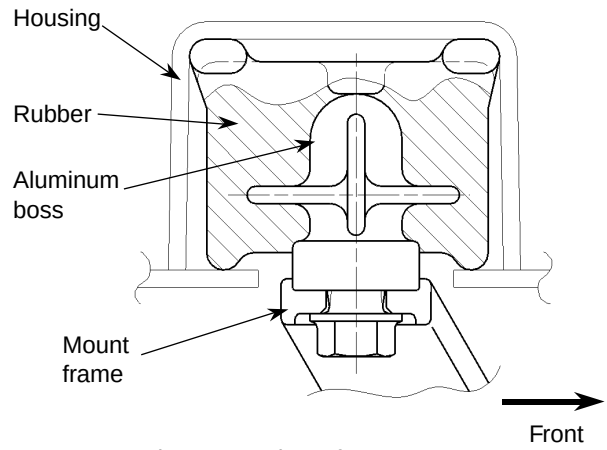


Fig.6 Section of upper mount

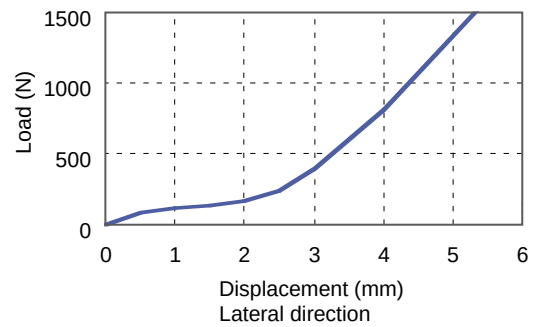


Fig.7 Spring rate of upper mount

$$[M]\{\ddot{U}\} + [K]\{U\} = \{F\} \quad (16)$$

Here, $[M]$ is the mass matrix, $[K]$ is the stiffness matrix, $\{U\}$ is the displacement vector, and $\{F\}$ is the exciting force vector.

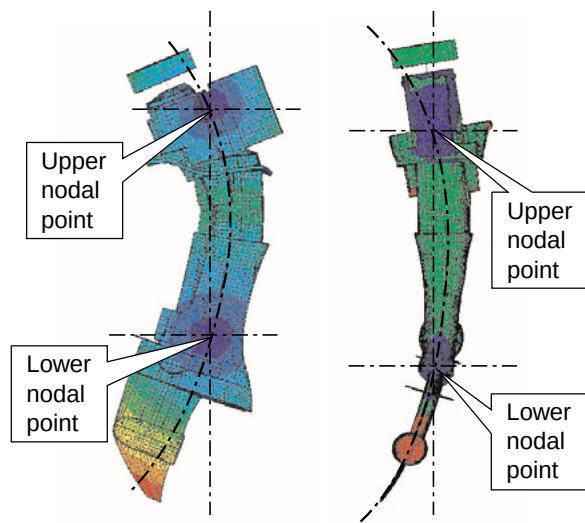
The above equation was converted to an equation of motion with modal coordinates to focus on the resonance of first-order lateral bending. The following is a generally known equation.⁽³⁾

$$[m_r]\{\ddot{\xi}_r\} + [k_r]\{\xi_r\} = [\phi]^{-1}\{F\} \quad (17)$$

Here, $[m_r]$ is the modal mass matrix, $[k_r]$ is the modal stiffness matrix, $\{\xi_r\}$ is the modal displacement vector, and $\{\phi\}$ is the natural vibration mode vector. The right side of equation (17) shows that the amplitude of vibration for that natural vibration mode is 0 when the product of the transposition of a certain natural vibration mode vector and the exciting force vector is 0. This is the principle for canceling resonance.

Fig. 8 shows the BF9.9 sprung bending first-order natural vibration mode calculated by the finite element method. The colors in the figure indicate the amplitude contour of these modes. The centers of the blue circles indicate the node positions for that mode. An outboard motor body is a continuous body, so there are an infinite number of natural vibration modes. However, the natural vibration mode shown in Fig. 8 with the lowest frequency has the greatest influence on vibration feeling. In case of the pendulum motion mount method, the exciting force is generated only in the lateral direction, so the authors focused on the lateral bending natural vibration mode shown in Fig. 8(b). This natural vibration mode has nodal points in the engine and gear case portions. The exciting force acts in the same phase at the center of the two cylinders, so according to the above mentioned resonance canceling principle, resonance should not occur if these nodal points are located approximately in the center of the two cylinders. This enables ideal elastic support with the outboard motor body moving approximately as a rigid body.

While developing the BF9.9, efforts were made to locate these nodal points close to the center of the two cylinders. For the final prototype specifications, eigenvalue analysis was performed using the finite element method for the S, L and UL specifications each possessing different extension case lengths. Table 1 shows the distance between the upper nodal point of the lateral bending mode and the center of the two cylinders. All three specifications placed the upper nodal point for lateral bending slightly above the center point of



(a) Longitudinal bending mode (b) Lateral bending mode

Fig.8 Shape of fundamental modes of the eigenvector

Table 1 Distance between upper nodal point and center of two cylinders

Type	Distance (m)
S (Short)	5.3×10^{-2}
L (Long)	4.5×10^{-2}
UL (Ultra long)	3.3×10^{-2}

the two cylinders which is the exciting force center, but prospects for achieving relatively small values were obtained.

4.2.2. Natural Frequency of the Outboard Motor Body

An additional method for eliminating the effects of outboard motor body resonance was to raise the natural frequency to a level that is much higher than the normal engine speed. This can be achieved by increasing the stiffness and reducing the mass of the outboard motor body.

In conventional outboard motors, the mount is often located near the center of gravity, but with the pendulum motion mount method the mount can be positioned much farther forward. This made it possible to use a structure whereby the secondary-moment of the extension case lower mount was improved a significant 2.5 times over the previous BF9.9 model. This structure is shown in Fig. 9. In addition, eliminating the balancers reduced the weight by approximately 2 kg which also contributed to increasing the natural frequency.

As a result, the lateral bending first-order natural frequency of the outboard motor body was increased to 218 Hz, which is over 1.5 times the 139 Hz of the conventional model.

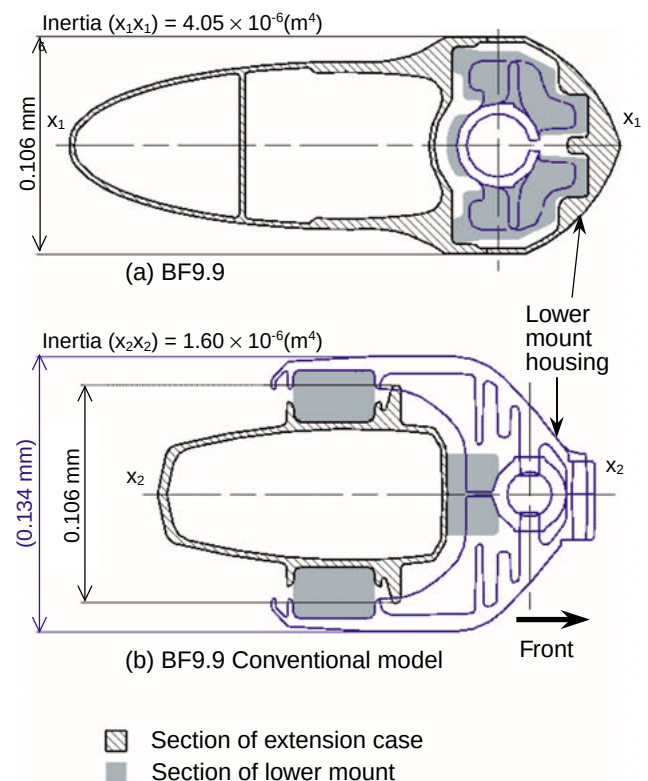


Fig.9 Comparison of section around lower mount

4.2.3. Evaluation of Outboard Motor Body Vibration

Frequency response analysis by MSC/NASTRAN was performed to investigate the degree to which natural vibration mode influences outboard motor body vibration. Outboard motor body vibration is a combination of rigid body vibration and natural vibration mode vibration as shown in Fig. 10.

The lateral acceleration value was divided into the vibration components of the rigid body vibration and natural vibration mode, and the contribution of each component at the normal engine speed was calculated. Table 2 shows the lateral acceleration at the mount positions for the S, L and UL specifications. At the upper mount, the vibration component of the natural vibration mode was kept to 5% or less of the total. This confirmed that the effects of the lateral bending first-order natural vibration mode had fallen to a level which could be largely ignored, and that the outboard motor body vibration had approached that of a rigid body.

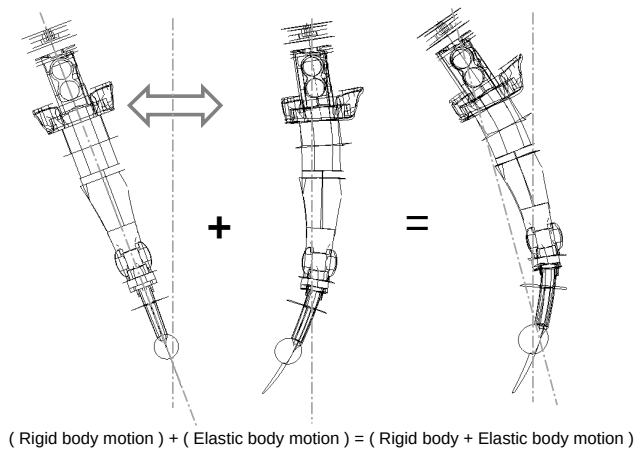


Fig.10 Lateral vibration of outboard motor

Table 2 Influence of elastic body motion lateral acceleration at mount points of S, L, and UL types (m/sec²)

Type	Mount	Rigid body motion	Elastic body motion	Total
S	Upper	120	-3	117
	Lower	64	-8	56
L	Upper	121	-4	117
	Lower	51	-10	42
UL	Upper	121	-4	118
	Lower	53	-14	39

5. Testing Methods and Results

Fig. 11 shows a photograph of the newly developed 4-stroke 2-cylinder outboard motor BF9.9 to which the pendulum motion mount method is applied. Vibration was compared using the conventional 4-stroke 2-cylinder outboard motor BF9.9. The conventional BF9.9 cancels the first-order inertia force generated by the engine and converts

it into a first-order inertia couple by adding a dual axis balancer to the top of the crankcase. Vibration was also measured without this dual axis balancer. Table 3 shows the basic specifications for the new and conventional BF9.9 models.

Vibration was measured using a 16ft. FRP Japanese boat. The conditions were gradual acceleration while cruising in a straight line on calm water; and, acceleration pickups were attached to the handle and seat for measurements. Fig. 12 shows the results for the first-order component obtained by performing vibration level analysis relative to engine speeds on the measurement data.

Handle vibration was almost the same as the outboard motor with the balancer at medium to low speeds, but was reduced by 50% at high speeds. In addition, significant vibration reducing effects were obtained at all operating speeds compared to the boat without the balancer. Hull vibration was also about the same as for the boat with the balancer, but significant vibration reducing effects were obtained compared to the boat without the balancer. Thus, the pendulum motion mount method was verified to significantly reduce both hull and handle vibration.



Fig.11 BF9.9

Table 3 Specification

Model	BF9.9 New model	BF9.9 Conventional model
Bore × Stroke (mm)	58 × 42	58 × 53
Displacement (cm ³)	222	280
Maximum engine output (kW/r/min)	7.4/6000	7.4/5500
Maximum Torque (N · m/r/min)	14.9/4000	16.6/3000
Reciprocating mass (kg)	0.386	0.349
Sprung mass (kg)	36	39

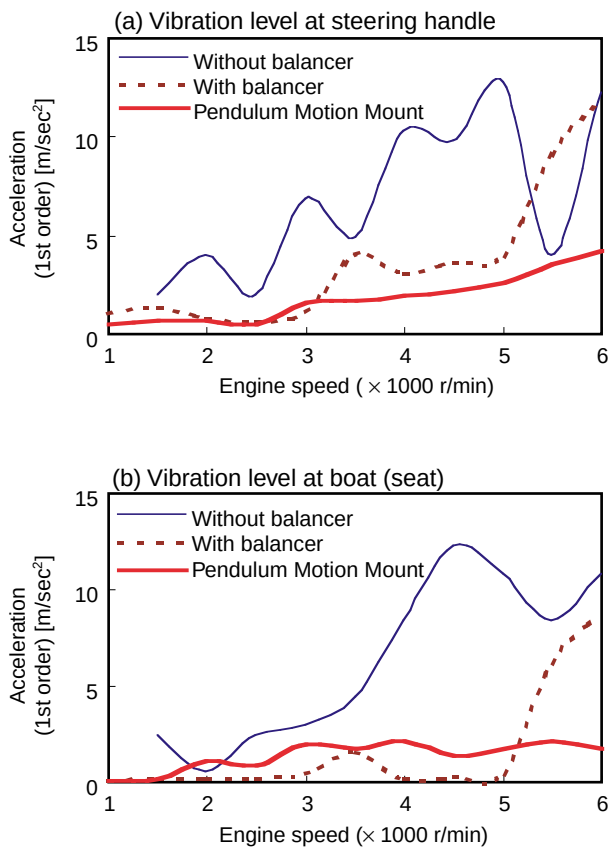


Fig.12 Effect of vibration reduction by Pendulum Motion Mount

6. Conclusions

Vibration reduction using the pendulum motion mount method was applied to a 4-stroke 2-cylinder outboard motor. The theory of isolating vibration was clarified, an actual outboard motor was manufactured, and the following results were obtained.

- (1) Hull and handle vibration when inclined mounts are applied to the outboard motor was clarified through 2-dimensional theoretical analysis.
- (2) Using a pendulum motion mount was theoretically shown to effectively isolate 4-stroke 2-cylinder outboard motor vibration at all engine speeds.
- (3) The conditions for canceling handle vibration caused by inertia force were clarified.
- (4) Outboard motor body and mount structures capable of realizing the pendulum motion mount method were presented.
- (5) An outboard motor based on this technology was newly developed, and vibration reduction effects exceeding those for a conventional outboard motor with a balancer were achieved.

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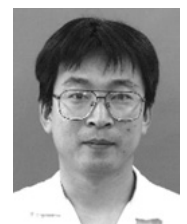
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Development of a Vibration Prediction Technique for Automobile Exhaust Systems

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ABSTRACT

A simulation technique that uses a finite element model for predicting vibration in the exhaust system was developed. The shapes of the flexible tube portions of the flexible joints, or vibration absorption joints, were faithfully modeled and the characteristics of the other parts were assigned numerical values. An accurate model of the ball joints was obtained by assigning numerical values to the dynamic properties of gaskets. The vibration force of the engine, which is the input for this system, was obtained using the sum of the inertial forces of the component parts and the rotational reaction force produced by cylinder combustion pressure. This made it possible to quantitatively predict the forces transmitted to the body from the exhaust hangers and to qualitatively understand exhaust system behavior. This in turn makes it possible to study the optimal locations for the exhaust hangers and the vibration absorption joints in the pre-development stages.

1. Introduction

One of the causes of noise and vibration in the cabin of a vehicle is the engine vibration that is transmitted to the body through the exhaust hangers. This often leads to booming noise, floor vibration, steering vibration and other noise and vibration phenomena that occur at idle.

Conventionally, prototype parts have been repeatedly fabricated during vehicle development to investigate and countermeasure the causes of the excessive noise and vibration (NV) described above in exhaust systems.

However, building and testing countermeasure parts takes considerable time and thereby forestalls the actual resolution of this issue. Thus, the application of simulation as represented by Computer Aided Engineering (CAE) has become a matter of importance.

Therefore, the authors have performed multiple verifications in order to grasp the characteristics and phenomena of each part with the aim of achieving highly accurate simulation technology results that can then be applied to actual development.

This paper reports the results of this verification process. That is, it reports that simulation modeling technology, which reliably simulates reality, has been established.

2. Overview of Calculation Technology

2.1. Modeling the Critical Structures of the Exhaust System

The basic exhaust system body Finite Element Model (FEM) used in this example is shown in Fig. 1.

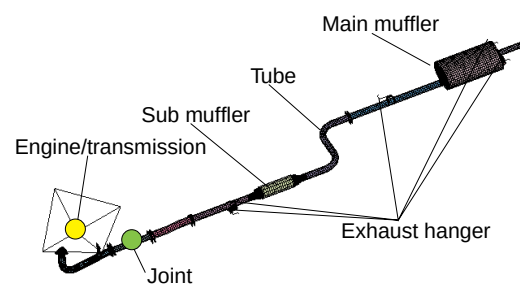


Fig.1 FE model of an exhaust system

All parts, whose shapes could be modeled, such as the exhaust system pipes and the mufflers, were modeled using shell elements.

The modeling know-how required to obtain the characteristics of the vibration absorption joints was reflected.

As the frequency range used in this research project is low for the engine and transmission, they were hypothesized to be within the rigid body range and the characteristics of the rigid bodies were expressed as a mass point.

2.2. Modeling of the Composite Joint Structure

The vibration absorption joints (flexible joints and ball joints) are the portion of the exhaust system that reduces overall exhaust system input and influences the vibration mode.

These joints are comprised of multiple parts. In this method, the mechanism of vibration absorption for each

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component part was analyzed to gain an understanding of the dynamic characteristics of the part. The obtained characteristics were then combined to create a model.

The vibration analysis was performed by verifying the directional characteristics of each joint's displacement.

(1) Flexible Joints

The component parts that comprise the flexible joints are shown in Fig. 2.

Test methods and restraining conditions as shown in Fig. 3 were used to analyze the characteristics of the flexible joints. The directions of displacement analyzed were both the expansion and contraction direction as well as the flex direction.

The frequency response function (FRF), which is the acceleration response level for each unit of load on each of the component parts, is shown in Fig. 4 for the expansion and contraction direction.

It is clear from the FRF peak in Fig. 4 that the flexible tube has dominant characteristics and that the other component parts are merely adding spring, attenuation and related characteristics.

The model of the dominant flexible tubing component was constructed by faithfully reproducing its actual shape. Modeling the shapes of the other component parts is difficult and thus the characteristics of these parts were expressed using numerical values.

A model was then built based on these results. Fig. 5 shows a comparison of the FRF for the actual measurements and for the calculations.

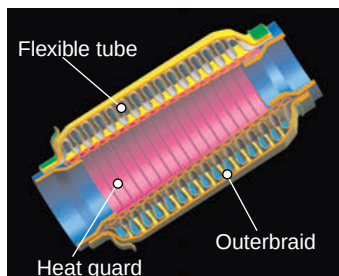


Fig.2 Flexible Joint

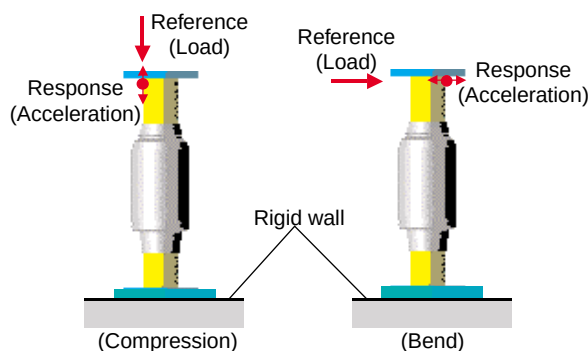


Fig.3 Test part for analyzing dynamic characteristics of the flexible joint phenomenon

(2) Ball Joints

Fig. 6 shows the parts that comprise the ball joints.

There are fewer parts comprising the ball joints than the flexible joints and, based on the behavior of the joints, the rotational (dynamic) spring is dominant. Thus, testing was performed on the dynamic spring characteristics focusing on the ball rotational center.

Fig. 7 shows the test method and restriction conditions and Fig. 8 shows the results of the tests.

As is clear from Fig. 8, the dynamic stiffness of the ball gasket and the dynamic stiffness for the entire ball joint are nearly equivalent.

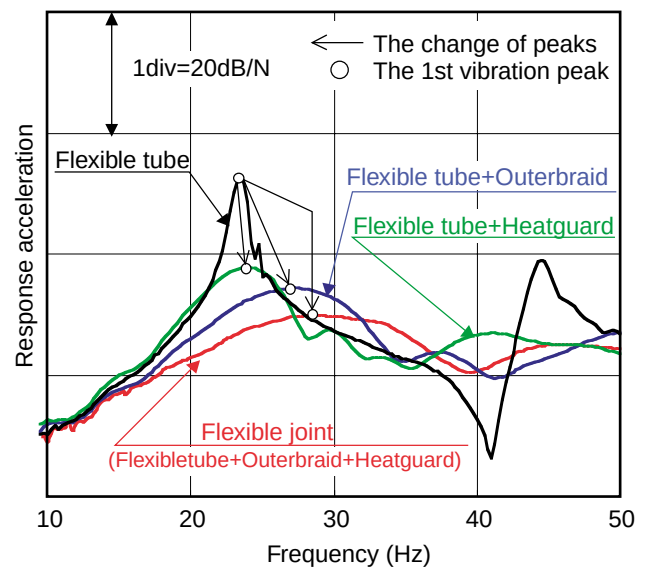


Fig.4 Measured FRF of flexible joints

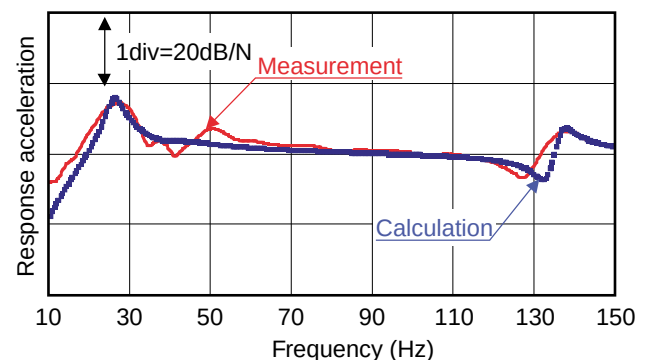


Fig.5 Comparison of FRF of the flexible joint

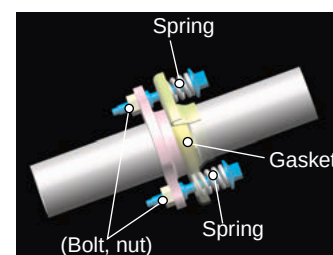


Fig.6 Ball Joint

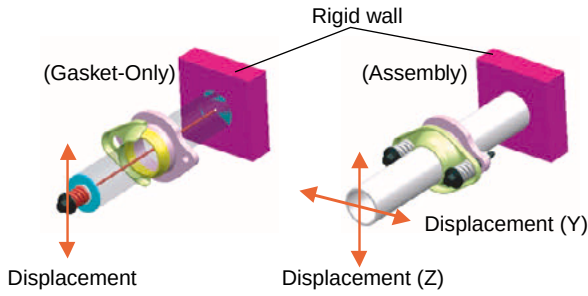


Fig.7 Test part for analyzing the mechanisms of the ball joint phenomenon

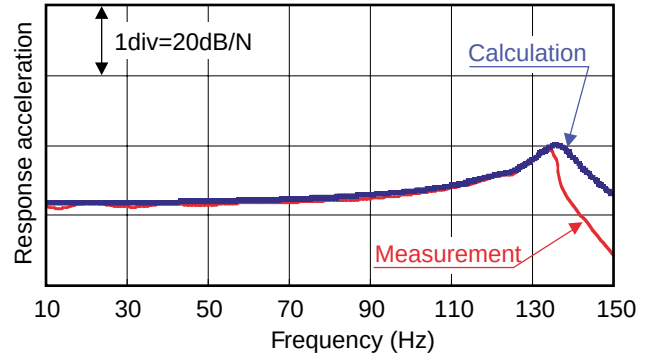


Fig.10 Comparison of FRF of the ball joint

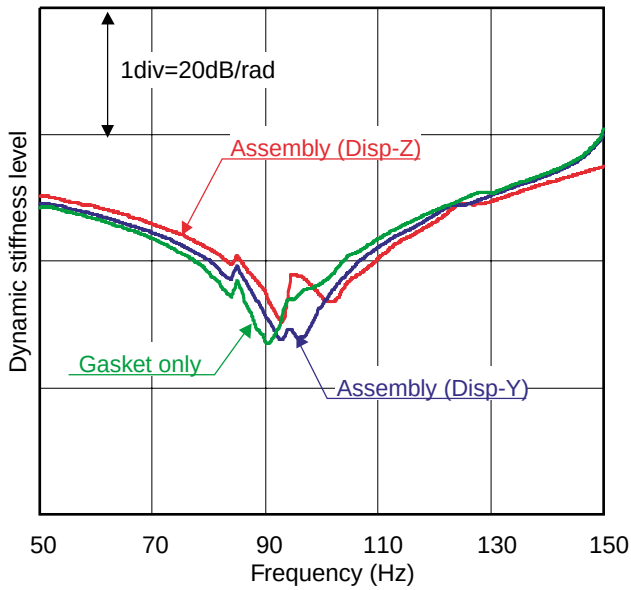


Fig.8 Measured dynamic stiffness of ball joints

Next, the test results of the FRF of the actual measurements and the calculations are discussed.

To assure consistent results, actual measurements were performed using the set-up shown in Fig. 9.

A highly accurate model was created through precise input of the spring characteristics of a ball gasket. Fig. 10 shows the results of the actual measurements and the calculated model.

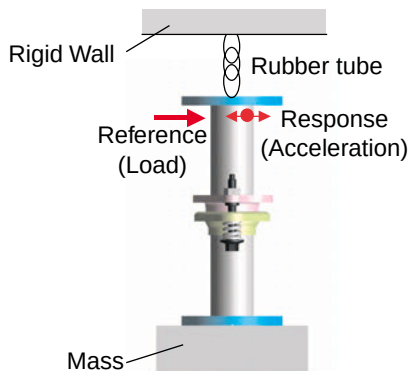


Fig.9 Test part for verifying correlation

As described above, the characteristics of each component part comprising the vibration absorption joints were understood through actual measurements. The characteristics of the parts were substituted into a calculation model. Further, a highly accurate model was obtained through the combination of these characteristics.

2.3 Examination and Identification Methods for Engine Modeling Application

The engine and transmission are critical environmental conditions for the exhaust system. We treated these areas as rigid bodies to identify the rigid characteristics from the FRF of the actual measurements.

The following describes the concept used to identify the rigid body concepts.

In a mass model of a spring with a one-degree of freedom system, with the frequency range above the resonant frequency, a rigid body movement frequency exists where the inertia level is constant. This level is determined by the mass value.⁽⁴⁾

This one-degree of freedom system is an expanded method in which there are a total of 6-degrees of freedom - 3 in the translational direction and 3 in the rotational direction.

If the six-degrees of freedom mass matrix characteristic is $[M]$ when the input force $[F]$ is applied, then the movement formula is:

$$[F] = [M][\ddot{x}]$$

In the equation above, $[F]$ is the input level and $\ddot{x}$ is the vibration acceleration response level; if these two are known, they lead to $[M]$, the answer to the linear equation system. Furthermore, if the coordinates of the input and response are known, it is then possible to attain $[M_c]$, the center of gravity by replacing the coordinates around the reference points, and calculating $[M]$ for these reference points.

2.4 Verifying the Accuracy of Calculations

Bench measurements were taken to verify correlation with the FEM.

Eigenvalues, natural frequency correlation coefficients, and FRF comparisons were used in the correlation verification. At least 120 FRFs were measured for the exhaust system in Fig. 1.

Also, a target of at least 0.8 was set for the native mode correlation coefficients (MAC).

Fig. 11 shows the calculation model accuracy verification flow.

Using the generated model, the engine and the exhaust system installed in a vehicle were reproduced to verify the FRF characteristics.

Vibrations were input from a shaker at the rear of the exhaust system, as shown in Fig. 12, for the measurement. Then, the acceleration levels at representative hanger points A and B were measured. Two vibration absorption joints were used in the measurements: a flexible joint and a ball joint.

The results of FRF verification on the exhaust system fitted with a flexible joint are shown in Fig. 13.

In the same manner, the results of FRF verifications on an exhaust system fitted with a ball joint are shown in Fig. 14.

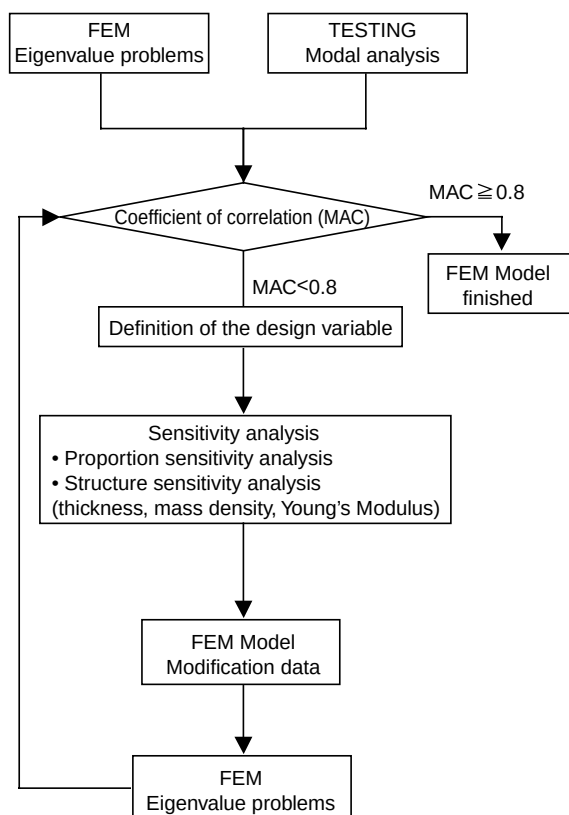


Fig.11 Flow chart for verifying correlation of accuracy

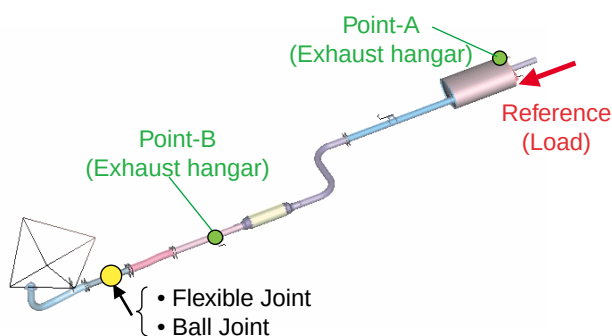


Fig.12 FE model of an exhaust system

Looking at Figs. 13 and 14, it is apparent that resonance peak level and frequency are both accurate.

Thus, based on the above, the behavior of the entire exhaust system including the engine could be quantitatively predicted.

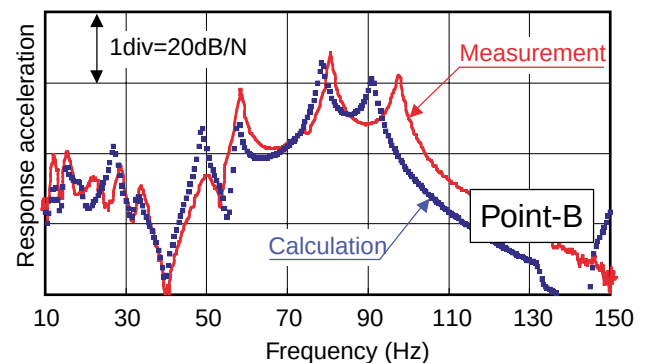
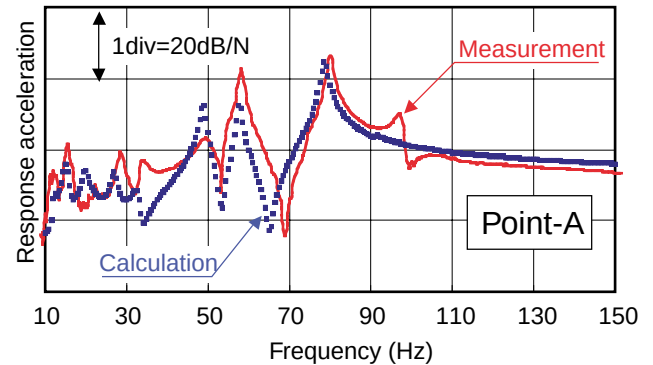


Fig.13 FRF comparison of point A and B in the flexible joint

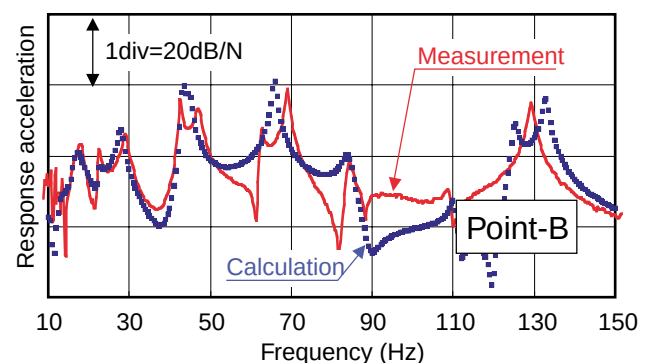
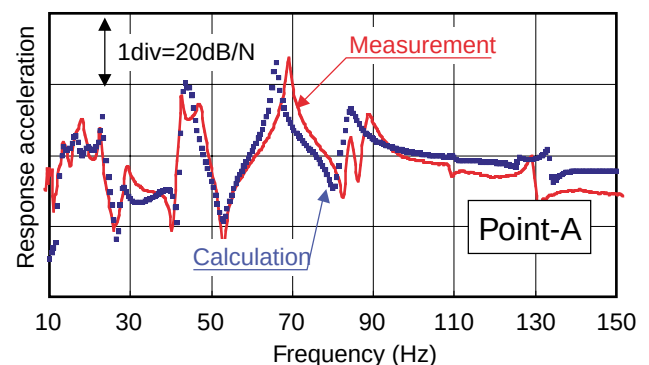


Fig.14 FRF comparison of point A and B in the ball joint

3. Engine Vibratory Force Calculation Method

Engine vibratory force was calculated by applying a model wherein all the parts, including the piston, connecting rod and cranks, were considered rigid bodies. ^{(2), (3)}

That is, the overall engine vibratory force was derived by calculating the total of the inertial force of both the reciprocating motion of the piston and the small end of the connecting rod and the rotary movement of the crank and the big end of the connecting rod, and the rotational reaction force of the crankshaft caused by the cylinder combustion pressure as determined for each cylinder, and finally by compensating for the phase differences between the cylinders.

The cylinder combustion pressure used was the average of the pressure data taken during test multiple cycles.

The following is a brief description of the equation used to calculate the inertial force and the rotational reaction force.

If the piston displacement from the piston top dead center is (S) in Fig. 15, then

$$S = r(1 - \cos \theta) + \frac{1}{4}r\rho(1 - \cos 2\theta)$$

where,

$$\rho = \frac{r}{l}$$

Acceleration would then be

$$\frac{d^2S}{dt^2} = r\omega^2(\cos \theta + \rho \cos 2\theta)$$

Therefore, if the mass of the reciprocating portion comprised of the piston and the small end of the connecting rod is M_1 , then the vertical inertial force of the reciprocating portion is derived using the following equation.

$$F_{1z} = -M_1r\omega^2(\cos \theta + \rho \cos 2\theta)$$

Next, if the mass of the rotating portion comprised of the crank and the large end of the connecting rod is M_2 , then the vertical inertial force and the lateral inertial force of the rotating portion would be found using the following equations.

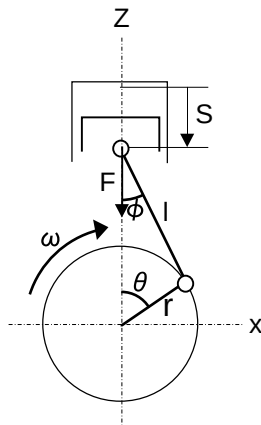


Fig.15

$$F_{2z} = M_2r\omega^2 \sin \theta$$

$$F_{2x} = M_2r\omega^2 \cos \theta$$

Crank rotational reaction force is derived using the following equation, when the sum of the cylinder combustion pressure and the vertical inertial force for the reciprocating portion as determined above is defined as F.

$$F_c = F \cdot \frac{\sin \theta}{\sqrt{\frac{1}{\rho^2} - \sin^2 \theta}} \cdot (r \cos \theta + l \sqrt{1 - \rho^2 \sin^2 \theta})$$

Fig. 16 shows the results of the actual measurements and calculations of the vibration acceleration of the engine mounts caused by the firing of the engine.

We compared the response at points A and B in the exhaust model shown in Fig.12 when the engine vibratory

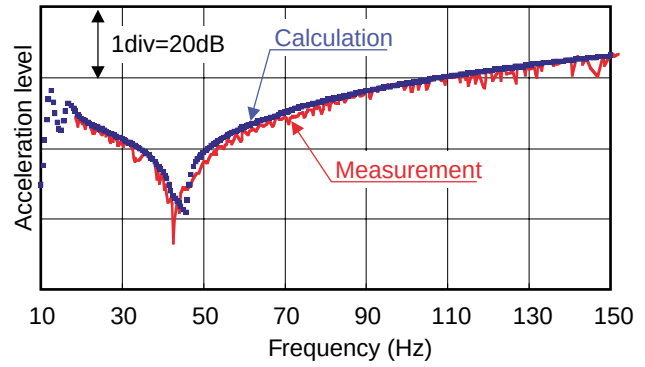


Fig.16 Response comparison of the engine mount during firing

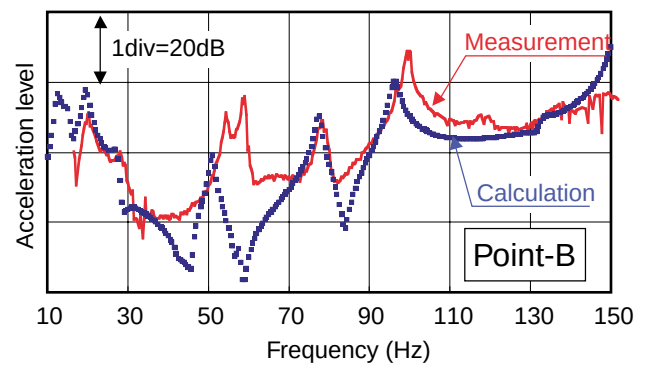
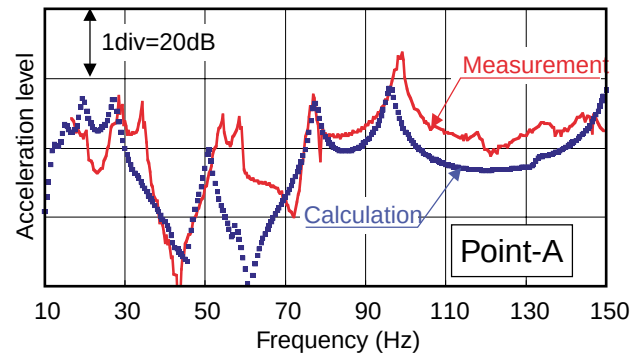


Fig.17 Response comparison of point A and B with a flexible joint during firing

force was applied with the response measured during firing.

The results of the exhaust system acceleration fitted with a flexible joint are shown in Fig. 17.

In the same manner, the results of the exhaust system acceleration with a ball joint are shown in Fig. 18.

From the results of Figs. 17 and 18, we see that the trend of the waveforms, the resonance peak levels, and the frequencies are accurate.

Therefore, vibration transmitted from the exhaust system hanger to the body can be obtained as absolute values, and judgment is rendered possible.

4. Application to Exhaust System Vibration Reduction Methods

Next, we will discuss an example wherein this method is applied to an FE model of the exhaust system, as shown in Fig. 19.

Here, we investigated the optimum positioning of the exhaust system hangers - the points at which the exhaust system vibration is transmitted to the body.

In this method, positions that could easily be nodes were determined based on multiple exhaust system mode shapes; the hanger positions were then shifted to those positions. Fig. 20 shows the results of calculations of the exhaust system acceleration at the hangers of the main muffler with input from the rear of the exhaust system. Here, it is possible to infer a reduction of the response levels at the exhaust system hanger points, seen in Fig. 20, and therefore we were able to verify the effectiveness of this predictive technology.

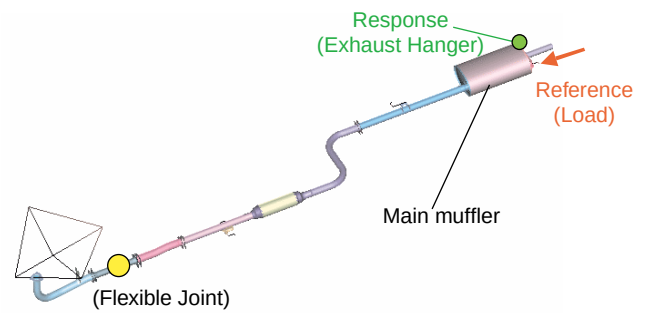


Fig.19 Structural model of an exhaust system

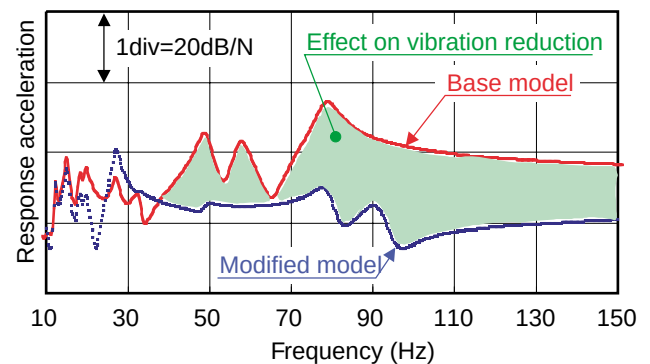


Fig.20 Response comparison at the hanger points

5. Conclusions

A highly reliable exhaust model that conforms to the real world was constructed using the following methods:

- (1) A faithful model of the flexible tube shape was created, as the flexible tube is the dominant characteristic of a flexible joint. Also, numerical values were assigned to the dynamic characteristics of the other component parts.
- (2) As the ball joint characteristics are basically determined by the ball gasket characteristics, the dynamic characteristics are assigned numerical values.
- (3) An engine area model was constructed by treating the engine and the transmission as rigid bodies and then obtaining the center of gravity mass matrix characteristics using the FRF and the measurement positions from the tests.
- (4) Engine vibratory force was derived by considering the phase differences between the individual cylinders based on the inertial force, caused by the reciprocating and rotary motions of the component parts for each cylinder, and the rotational reaction force, caused by the actual cylinder combustion pressure.

As a result, analysis of the phenomena of the entire exhaust system, including the engine, could be performed due to the highly accurate FE model of the exhaust system and the modeling of the engine area; the area that sets the limit conditions for the system. Further, by applying rigid body engine vibratory force, we were able to predict the

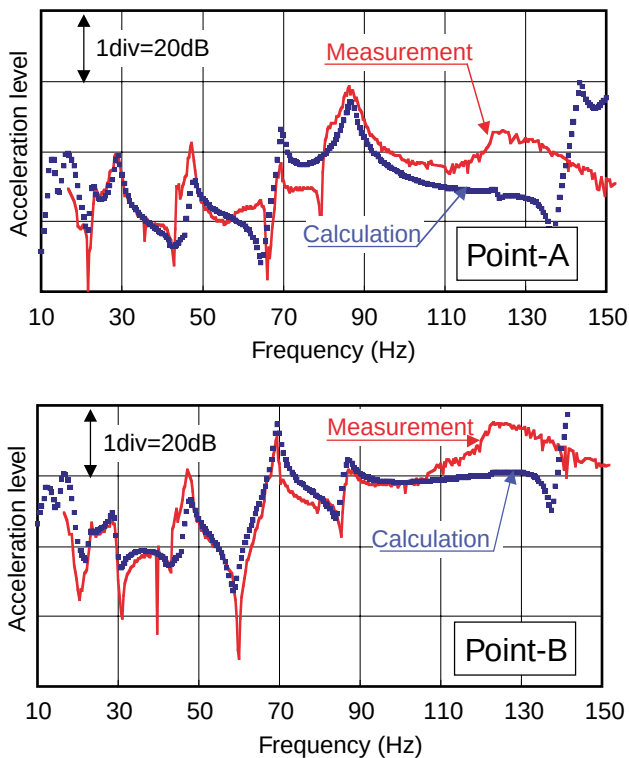


Fig.18 Response comparison of point A and B with a ball joint during firing

exhaust vibration that affects the noise and vibration in an evaluation vehicle with enough accuracy for real world application.

6. Prospects

Prediction engineering tools and engine vibratory force derivation tools that enable coherent analyses from eigenvalue analysis to sensitivity and structural analysis were provided to development engineers for effective use in the calculation and design stages of development that come before prototype builds.

The technologies were applied to actual vehicles so that exhaust system vibration could be reduced, thus contributing to improvements in automobile noise and vibration performance.

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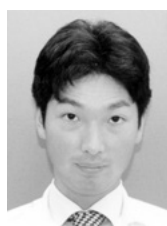
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Development of Simulation Technology for the Dynamic Behavior of Crankshaft Systems in Motorcycle Engines

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Tatsuya IWAMOTO*

Shingo UEDA*

ABSTRACT

In order to improve durability and marketability of an engine and reduce the development period, crankshaft system simulation technology was developed.

A crankshaft and a crankcase are modeled using the Finite Element Method (FEM). The models and a calculation model of the hydrodynamic reaction force of bearings are linked and the crankshaft behavior is calculated using numerical integration within the time domain. This simulation technology enabled accurate prediction of crankshaft behavior and stress through the high operating speeds that characterize motorcycle engines.

1. Introduction

Today's market demands that high-performance motorcycle engines with reduced weight and more compact dimensions be developed and that they be developed in a significantly shortened period of time. Since the key parts of engines, such as the crankshafts, connecting rods and pistons, have a major impact on the durability, output characteristics and noise and vibration of these engines, it is absolutely essential for the specifications of the parts to reach a level of maturity during the initial stage of development ⁽¹⁾.

The crankshaft held by the crankcase bearing receives the explosion force and inertia force exerted from the piston through the connecting rod in each cycle.

This load is generally at 10^4 N and higher, and it acts cyclically with each cycle. Resonance is induced, and an excessively high bending load and torsional moment are exerted on the crankshaft with this load especially at high engine speeds. In order to create lightweight and compact engines that will function under these severe conditions, the behavior of the crankshaft, including the pistons, connecting rods, and ACG rotors, is identified and each part must be designed for those conditions.

Based on this background, the authors have developed a simulation technique for calculating very precisely the stress that occurs in the crankshaft under firing operation conditions.

2. Stress Occurring at High Engine Speeds Operation

Fig. 1 shows the structure of the V type, 2-cylinder, 4-stroke engine used in this research. As shown in the figure, this engine has two connecting rods for each pin journal, and the cylinders have a 90-degree phase angle.

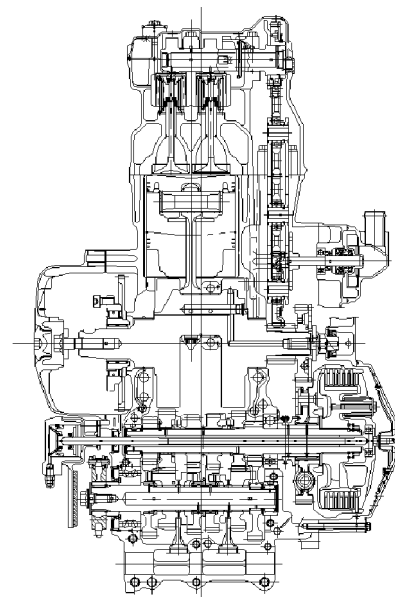


Fig.1 Engine structure

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The stresses that occur in the pin journal fillet area of the crankshaft under the firing operation conditions was measured using strain gauges.

Fig. 2 shows the locations of the measuring points.

Fig. 3 shows the stress amplitude extracted from the stress values obtained for each engine speed.

The following can be said about these results.

- (1) Compared with low engine speeds, the stress amplitude increases significantly at high engine speeds.
- (2) At specific engine speeds, the stress amplitude indicates a maximum value.

The following two factors are likely the causes of the stress amplitude occurring in these areas:

- (1) The inertia force of the reciprocating part acts upon the crankshaft during each cycle
- (2) The load produced by the pressure inside the cylinders acts upon the crankshaft through the piston and connecting rod during each cycle.

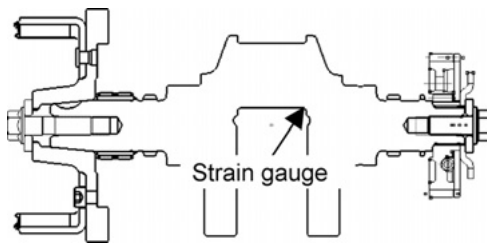


Fig.2 Stress measurement points

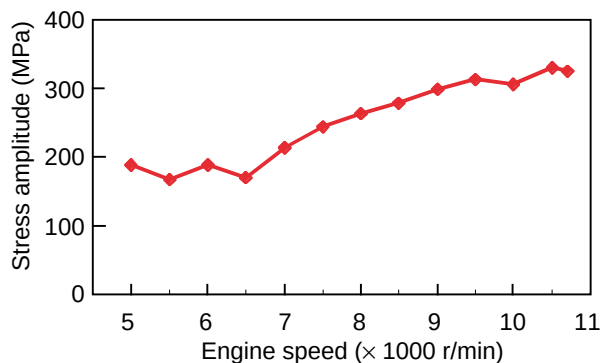


Fig.3 Stress amplitude

3. Static Stress Analysis

In order to analyze the causes for the increase in the stress amplitude at high engine speeds and for the peak stress amplitude exhibited at specific engine speeds, the stress calculation under the static conditions was performed using the finite element method.

Fig. 4 shows the finite element model and the boundary conditions used for the static calculations.

5,000 rpm was selected as the low engine speed and

10,800 rpm was selected as the high engine speed.

Fig. 5 shows a comparison between the static calculation results and the experiment results.

At low speeds, these two sets of results are virtually equal. However, at high speeds, the experiment results were 52% higher. It is considered that the stress amplitude increased at high engine speeds as a result of the dynamic behavior, which occurred in the crankshaft at these engine speeds.

This indicates that adequate precision cannot be obtained using this kind of static calculation method to design the strength of the crankshafts used at high engine speeds.

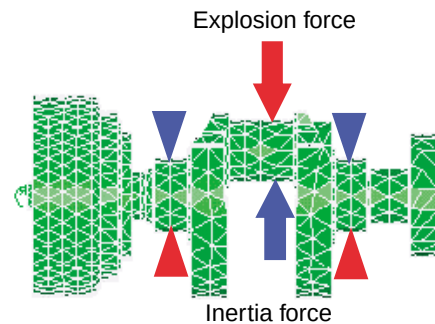


Fig.4 Static FEM model and boundary conditions

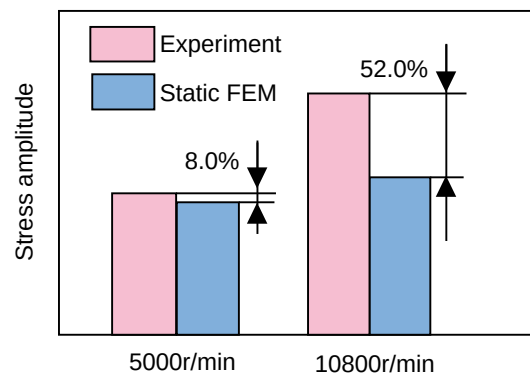


Fig.5 Comparison between experiment measurement and static FEM

4. Dynamic Calculation Methods

The numerical integration method in the time domain, which is widely employed, was used for simulating dynamic behavior. Fig. 6 shows the steps for this calculation method.

The calculation method consists of the following four parts.

- (1) Generating part of finite element data
- (2) Generating part of behavior
- (3) Generating part of oil film pressure
- (4) Generating part of stress

4.1. Generating part of Finite Element Data

The finite element data was generated by the automatic element generating function of a software program available on the market using a 3D solid model. Since the data obtained by this process has a substantial degree of freedom, the modal reduction method was used to eliminate the degree of freedom which was not required for the purposes of the analysis, and the degree of freedom of the finite element data overall was reduced.

4.2. Calculation of Behavior

Fig. 7 shows the simulation model of the behavior calculation. This model consists of the structural elements and the load elements that make the connections between the structural elements.

The following components were modeled.

- Crankshaft Finite element model
- Crankcase Finite element model
- Piston Rigid body model
- Piston-pin Rigid body model
- Connecting rod Rigid body model

The load elements were defined into three categories as follows.

1. In the parts components, a non-linear load is defined for relative displacement, and so on. By this way, the contact of the connecting parts is expressed.
2. The load created by the pressure inside the cylinders is exerted on the pistons and the head part of the cylinder head. The pressure inside the cylinders obtained by actual measurements is used as a numerical formula for the function of the crank rotation displacement angle.
3. Torque is applied in the reverse direction of rotation to the power output gear at the top of the crankshaft. The engine's operation speed for the calculation is determined by adjusting the magnitude of this torque.

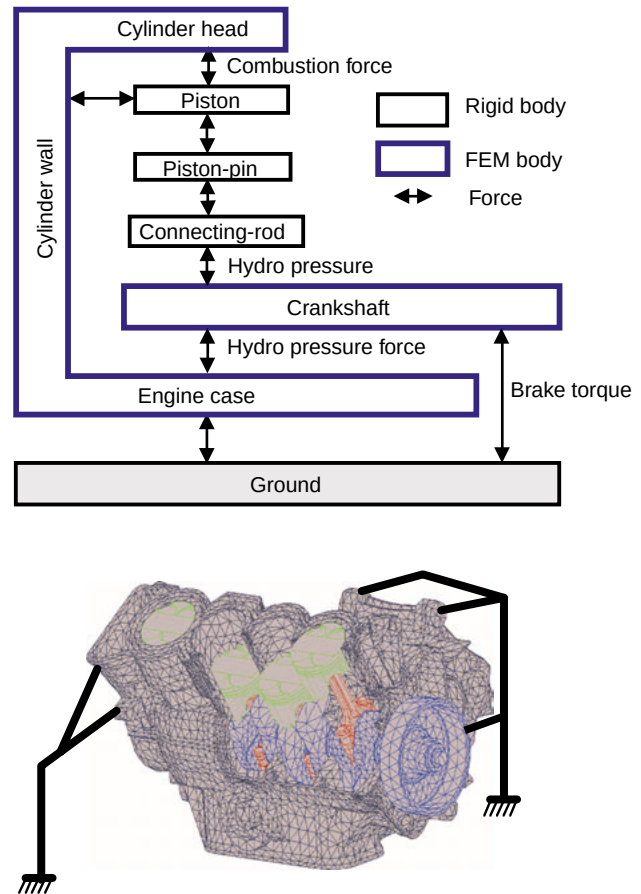


Fig.7 Simulation model

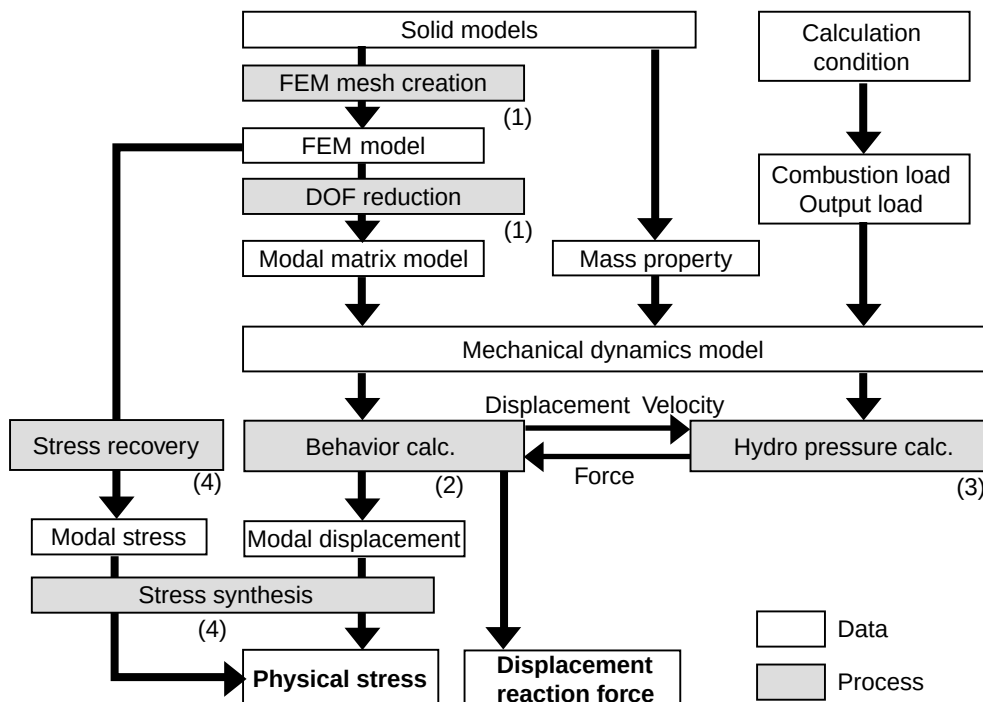


Fig.6 Analysis flow

4.3. Calculating Part of Oil Film Pressure

Based on the fluid lubrication theory, the load calculated from the pressure distribution resulting from the wedge force action and the squeeze force action of the oil was applied between the crank journals and the crankcase bearings and between the crank pin and the connecting rod large-end bearing. It was necessary to create a particularly exact model because it influences directly to the deformation behavior of the crankshaft and the behavior of the journal's axial center.

4.4. Calculating Part of Stress

The stress at each point in time was calculated from the displacement for each mode at each point in time, which was obtained from the behavior calculations, and from the crankshaft stress of each unit mode. At time "t", the displacement of the "i-nth" mode is represented by "ait", the stress of the "i-nth" mode is represented by " σ_i ", and the stress " σ_t " at time "t" is defined by the following formula:

$$\sigma_t = \sum ait \cdot \sigma_i$$

Since no static substitutions are involved during processing, this technique has two advantages: a high accuracy for the solution can be obtained, and the calculation time and data volume can be reduced when the stress at each of many different points in time is calculated.

5. Comparison of Calculation and Experiment Results

The stresses in the crank pin fillet shown in Fig.2 were compared using the aforementioned type 2-cylinder engine to verify the calculating results.

5.1. Engine Operation Conditions

The engine was operated from 5,000 rpm to 11,000 rpm by the full load driving firing.

5.2. Stress during One Cycle at Low Engine Speeds

Fig. 8 shows the results obtained by comparing the stresses of the crank pin fillet area at an engine speed of 5,000 rpm.

The stresses of the crank pin fillet in the above operation condition begin to generate significantly immediately after the crank angle of 0 and 450 degrees.

From here, it was found that the stresses of the crank pin fillet were dominated by the pressure inside the cylinder.

5.3. Stress during One Cycle at High Engine Speeds

Fig. 9 shows the results obtained by comparing the stresses of the crank pin fillet area at an engine speed of 10,800 rpm.

Compared with low engine speeds, it is clear that vibration is generated in the stress waveforms. Additionally, it can be seen that the absolute value of the stress increases significantly by the vibration causes compared with low engine speeds.

5.4. Comparison of Correlation

Based on the stress waveforms of the crank pin fillet

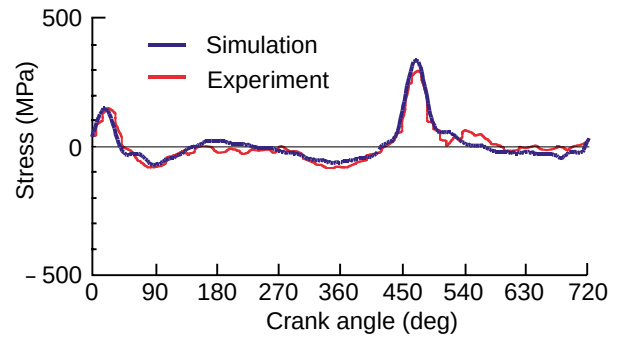


Fig.8 Comparison of stress at 5000r/min

during one cycle as shown in Figs. 8 and 9, it can be stated that the calculation and experiment results correspond very closely. Fig. 10 shows the stress amplitude of the crank pin fillet versus the engine speed for both the calculation and experiment results.

In each engine speed range, it can be stated that the stress of the calculation results of the crank pin fillet area stress practically coincide with the experiment results.

From the above mentioned data, the developed simulation technique verified that the numerical model for the behavior of the crankshaft system was highly accurate.⁽³⁾

Moreover, the effectivity of this simulation technique was verified even in high engine speeds; that was the feature of the motorcycle engine which had been difficult to simulate up to now.

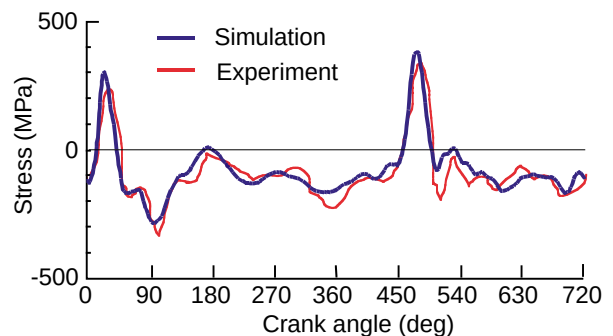


Fig.9 Comparison of stress at 10800r/min

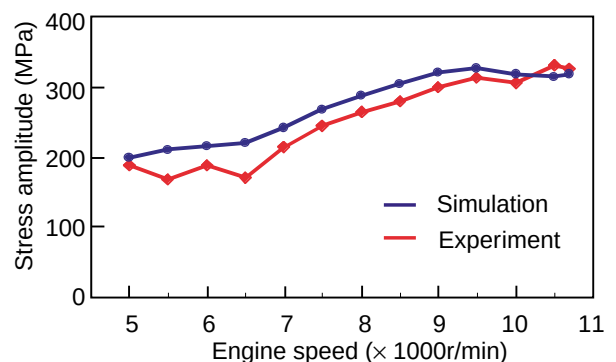


Fig.10 Comparison of stress amplitude

6. Conclusions

- (1) A numerical analysis technique was constructed using the numerical integration method in the time domain that linked the oil film pressure calculation of the bearings with the models of the crankshaft and crankcase using the finite element method.
- (2) Applying the technique to motorcycle engines, it was found that the simulation technique can be used sufficiently in the design stage because it faithfully simulates the stress occurring in crankshafts even at high engine speeds.

Acknowledgments

The authors wish to express their sincere gratitude to all those at Trans Cosmos Inc. and Information Services International- Denstu, Ltd. (iSiD) who cooperated with their research.

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Establishment of Valve Displacement Measurement by Laser Equipment

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ABSTRACT

In order to precisely analyze valve behavior of an automobile engine valve train, we applied a laser displacement meter. We were able to demonstrate that laser measurements are more precise and that lasers produce less noise than a conventional electro optical displacement transducer. However, when the laser method is applied, such problems as irregular reflection of noise on walls around the measured object, and digital conversion noise that is inherent to laser measurements, were encountered. We resolved these problems and improved the laser measurement by changing the shape of the measured surface and the surrounding walls to decrease the noise, and through zero point shift of the input voltage.

As a result of reducing the noise, it became possible to precisely analyze valve behavior, and we isolated the rocker arm re-collision with the valve due to the delay caused by the rocker arm inertia. Additionally, because the laser is more compact than the conventional optical instrument, it is possible to simultaneously measure multiple cylinders.

1. Introduction

Stabilizing valve train behavior is essential for improving engine performance, especially as the required maximum engine speeds are increasing. Valve trains have also become higher performance, as typified by the DOHC coaxial roller bearing VTEC rocker arm shown in Fig. 1.⁽¹⁾

Accordingly, various valve train behaviors are exhibited at high engine speeds, and technical issues must be overcome through detailed analyses.

To achieve this, development periods must be shortened using measurement technology, which provides the highest

accuracy possible, and performance prediction.

Particularly, in terms of valve lift, destruction of the valve train may occur if the bounce phenomenon during valve closing shown in Fig. 2 is 100 to 300 μm or more. Thus it is this phenomenon that determines the maximum engine speed. When analyzing bounce of a few hundred μm , the required measurement accuracy is 3% assuming a full scale of a few hundred μm . That is to say, the noise level must be 10 μm or less. However, achieving a high accuracy measurement of 10 μm within a valve lift displacement of 10 mm or more is extremely difficult, and thus far there are no reported cases of success^{(2)~(6)}

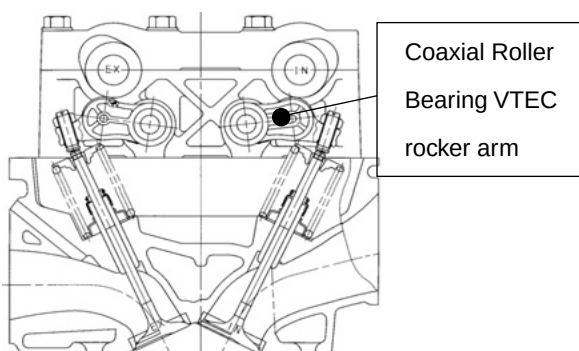


Fig.1 Cylinder head and valve train

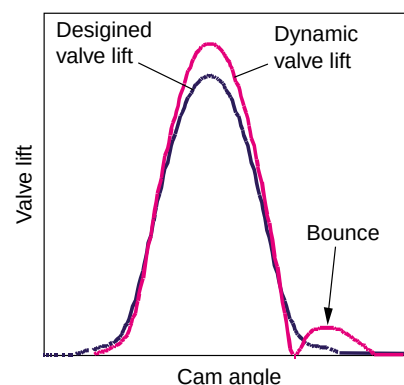


Fig.2 Valve bounce phenomenon at high speed

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The authors, like other researchers, performed measurements using a conventional electro optical displacement transducer, but there was too much noise which made it difficult to analyze phenomena and verify simulations for predicting behavior.

Therefore, the authors focused on laser triangulation method displacement meters as a means of overcoming the issues of electro optical displacement transducers. However, this creates new issues of irregular reflection noise and digital conversion noise, so various measures were taken to resolve these new issues. As a result, the advantages of laser displacement meters allowed accurate measurement of valve displacement, and also fully satisfied the conditions for analysis of phenomena and verification through simulations. The details are reported below.

2. Overview

Semiconductor laser triangulation method displacement meters (hereafter “laser method”) offer the following advantages over electro optical displacement transducers (hereafter “optical method”).

- (1) The required noise level for analyzing bounce is 10 μm or less. The measured noise level of the optical method is 50 μm , and the laser method level is 10 μm or less, which thus satisfies the required accuracy.
- (2) The required sensor head width for simultaneous measurement of multiple cylinders, which is used to analyze and verify the effects of the simultaneous behavior of multiple cylinders, is 30 mm or less. However, although simultaneous measurement of multiple valves is not possible with the optical method due to the camera size, it is possible with the laser method, thus satisfying the requirements.
- (3) The camera set up time is also 30 to 40 minutes per test for the optical method, so the number of test times is also limited. In contrast, the laser method requires only mounting, so set up can be completed in 5 minutes.

However, applying the laser method to the measurement of engine valve train behavior creates the following two issues that had to be overcome before the system could be applied for precise measurement.

2.1. Irregular Reflection Noise due to the Triangulation Method

Measurement is performed from below the valve, that is, from inside the cylinder, so reflection noise is generated by the laser light scattering in the closed area.

- (1) Reflection noise due to the wall shape between the sensor head and the measured object
- (2) Reflection noise due to concave shape of the measured object
- (3) Reflection noise from other walls

2.2. Reducing Zero Cross Noise through Displacement Meter Digital Processing

Zero cross noise is the noise produced by digital-analog conversion when the $\pm$ sign, the maximum digit, changes.

These issues were solved by the methods described

hereafter, enabling accurate analysis of valve behavior. Fig. 3 shows the device fabricated to directly compare the two methods.

Fig. 4 shows a comparison of valve lift measurements for the two methods. Compared to the waveform measured with the optical method, the bounce measured with the laser method clearly has lower noise.

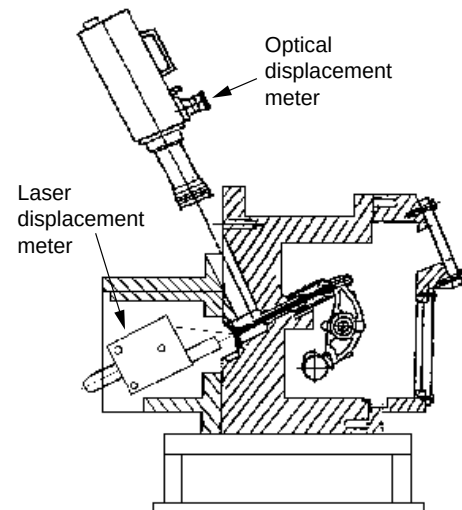


Fig.3 Comparison of optical and laser meter

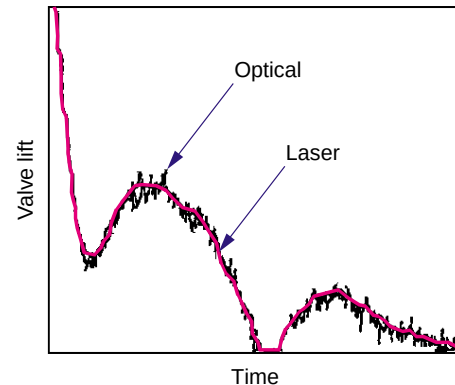


Fig.4 Simultaneous measurement results of optical and laser meter

3. Solution for Displacement Meter Issues

3.1. Reduction of Irregular Reflection Noise

3.1.1. Basic Principle of Laser Triangulation Method

Fig. 5 shows the laser measurement principle.

First, the light from the semiconductor laser strikes the measured object. Part of this scattered light is collected by an imaging lens and it is then formed into an image at the focal position. When the measured object moves in the axial direction of the laser light, the image-forming position changes, and this change in position is sensed by an optical detector.

Therefore, when the sensor head is installed in a closed

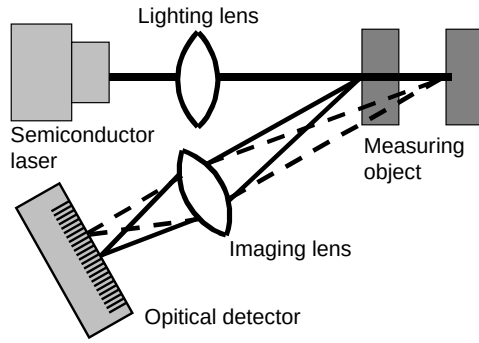


Fig.5 Locational detection of triangular method

area such as inside the combustion chamber, the secondary reflected light produced by the surrounding walls enters the imaging lens in addition to the reflected light from the measured object. This secondary reflected light also strikes the optical detector, creating irregular reflection noise.

Fig. 6 shows data measured by a laser displacement meter without countermeasures. If a narrow channel exists between the measured object and the laser sensor head, noise equivalent to displacement of $400\text{ }\mu\text{m}$ is produced.

Therefore, the measures described in the following section were taken to reduce irregular reflection noise.

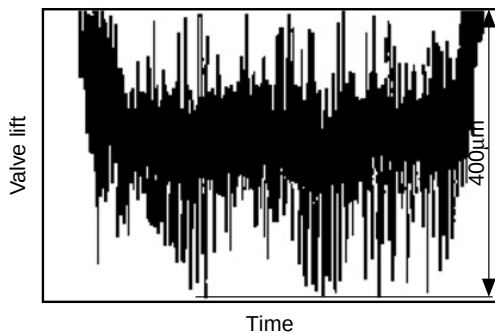


Fig.6 Measured data by laser method (Initial data)

3.1.2. Reduction of Irregular Reflection from around the Measured Object

Scattered light from the valve being measured reflects off the narrow channel walls, passes through the imaging lens, and enters the optical detector. As a result, light is irradiated to a different optical detector position than the light from the measured object, creating noise. Therefore, countermeasures were taken by changing the surrounding wall shape from that shown in Fig. 7 to the concave shape shown in Fig. 8.

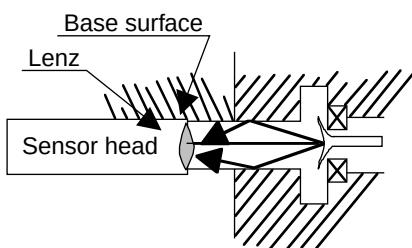


Fig.7 Wall reflection in narrow channel

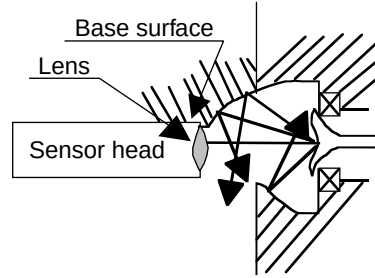


Fig.8 Wall reflection in wide channel

As a result, the noise was reduced by approximately half, to $230\text{ }\mu\text{m}$ as shown in Fig. 9, but this still does not meet the target accuracy of $10\text{ }\mu\text{m}$ or less. Furthermore, it is approximately five times greater than the noise level obtained through simultaneous measurement using the optical method (Fig. 10), and thus is not at a practical level.

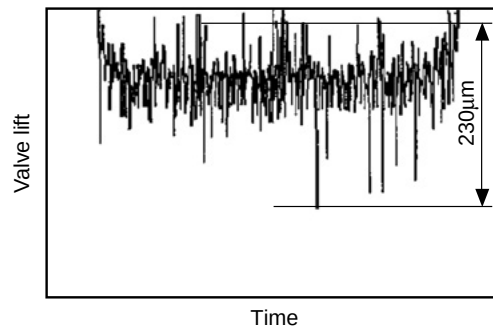


Fig.9 Data measured by laser method (modified wall shape)

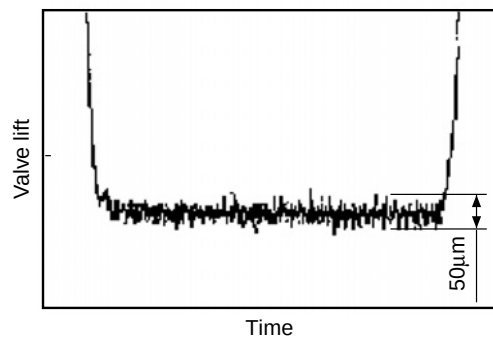


Fig.10 Data measured by optical displacement meter

3.1.3. Eliminating the Concave Shape of the Measured Object

When the measured surface has a concave area, a round spot reflection of the laser is observed. This is thought to increase noise produced by reflection, even in the circular portion around the bottom surface. Therefore, the concave shape was eliminated as shown in Fig. 11. This reduced the noise to $25\text{ }\mu\text{m}$ as shown in Fig. 12.

However, even when the countermeasures through section 3.1.3 were implemented, the noise was still not

reduced to 10 μm or less. The cause for this is thought to be the entry of irregularly reflected light from other surrounding walls. Therefore, countermeasures were taken, as shown in Fig. 13, whereby a conical tube with a rough, non-reflective inner surface was used to prevent the entry of external irregularly reflected light from outside of the cone.

As a result, the noise level was reduced to 15 μm as shown in Fig. 14. However, noise spikes sometimes still occurred. This issue was resolved by the method described in the following section, 3.2.

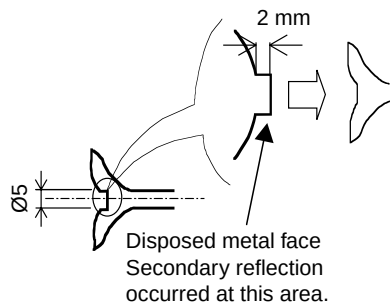


Fig.11 Abolishment of concave shape

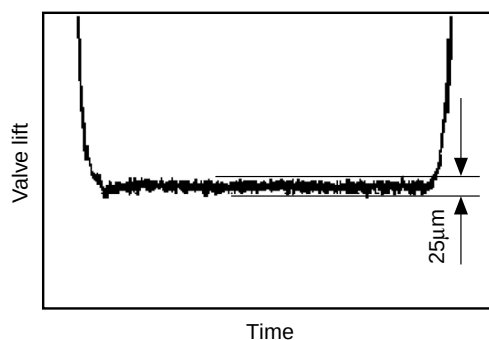


Fig.12 Data measured by optical method

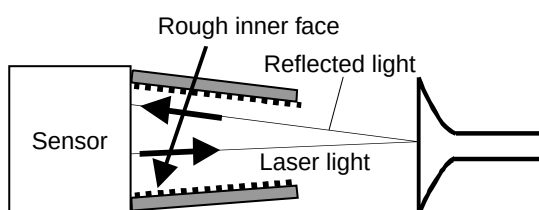


Fig.13 Non-reflective pipe

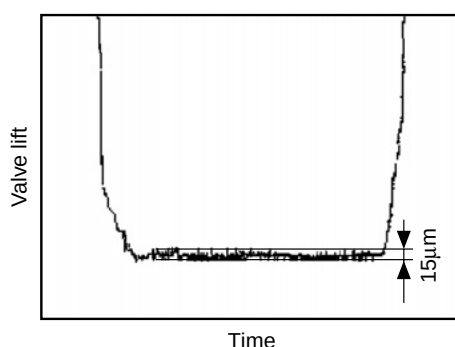


Fig.14 Data measured by laser method (Non-reflective pipe)

3.2. Reduction of Zero Cross Noise

When performing triangulation method using digital circuits, the analog output from the detector amplifier is converted from analog to digital, translated by a calibration map, and then converted from digital to analog again and output as shown in Fig. 15 below. Therefore, the noise produced by the detector amplifier is not amplified, and the analog output noise is minimized.

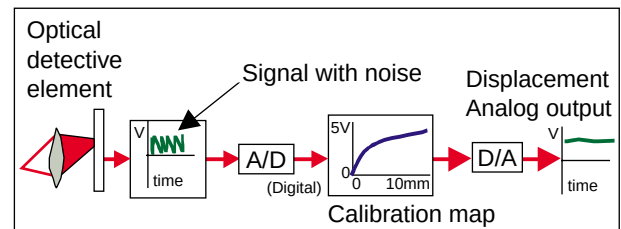


Fig.15 Block diagram of semiconductor laser amplifier

However, digital measurement is subject to the phenomenon known as zero cross noise. When displacement changes from a positive to a negative or vice versa, the sign changes accordingly. In digital representation, the sign ($\pm$) digit is the maximum digit, and switching this digit causes all bits to change from 0 to 1 as shown in Fig. 16. This is converted from digital to analog, so a large instantaneous voltage change occurs, producing noise spikes called "gritty noise." Therefore, when continuously measuring displacement around zero, noise spikes constantly appear on the waveform. As a countermeasure, when measurement accuracy is needed near zero, noise is reduced by shifting the zero point and then reshifting this same amount during a calibration processing as shown in Fig. 17.

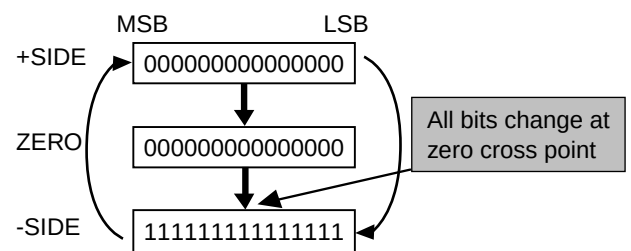


Fig.16 Figure of "Zero cross noise"

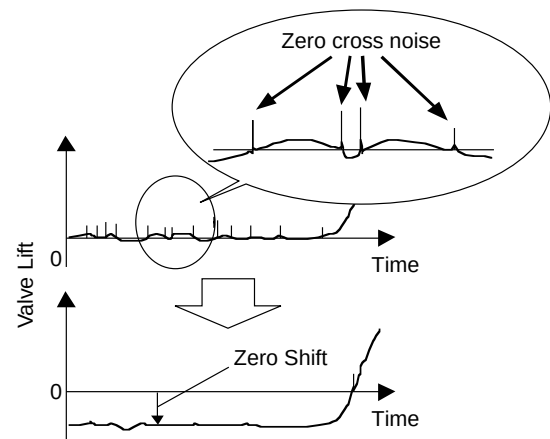


Fig.17 Improved pulse shape of "Zero cross noise"

Implementing the above four countermeasures reduced the noise to 10 μm or less as shown in Fig. 18.

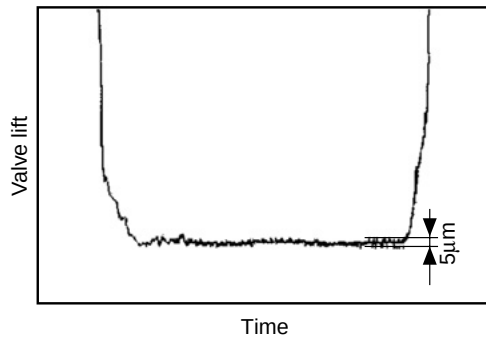


Fig.18 After reduced zero cross noise (final)

3.3. Summary of Noise Reduction Effects

As shown in Fig. 19, the above improvements enabled a reduction in noise, compared to the optical method, sufficient for detecting valve bounce.

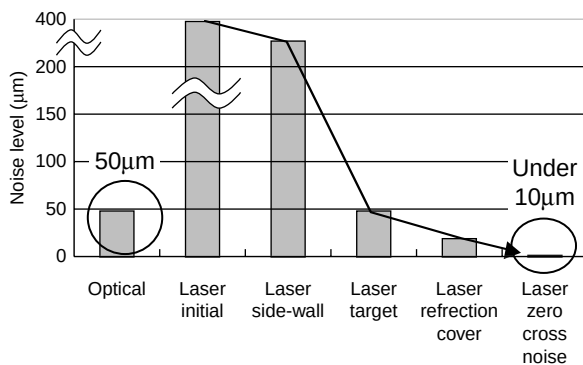


Fig.19 Transition of noise reduction

4. Effects of this Measuring Method

4.1. Improved Analysis Capabilities at High Engine Speeds

Fig. 20 shows the results of bounce during valve closing as measured by the optical method. Here, the details of why a large bounce occurs under these measurement conditions are unclear.

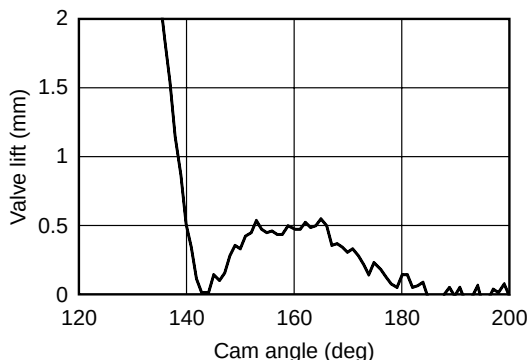


Fig.20 Valve bounce measured by optical method

However, when the laser method developed through this research is used, a small jump can be confirmed as shown in Fig. 21.

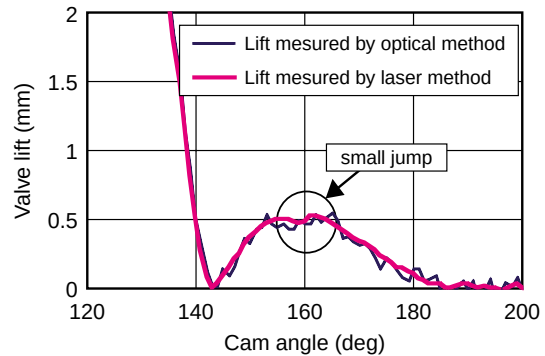


Fig.21 Valve bounce measured by optical & laser method

From this motion, it was thought that collision with some other part occurs at this time. Therefore, simulation calculations were carried out, whereby it was confirmed that similar jumps were reproduced as shown in Fig. 22.

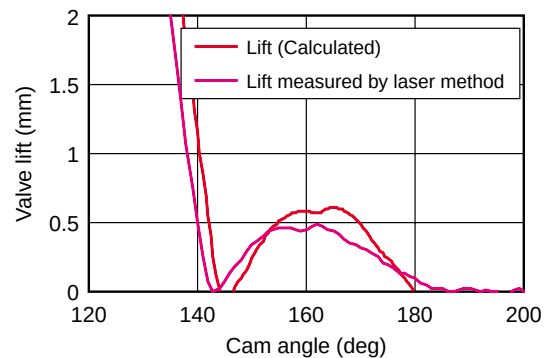


Fig.22 Bounce measured by laser & simulated result

Furthermore, the behavior of adjoining parts was also analyzed using simulation models. These results showed that re-collision with the valve due to the delay of the rocker arm inertia as shown in Fig. 23 caused the valve to bounce.

In this manner, at high engine speeds, the valve makes slight collisions during the period of valve opening to closing. Therefore, when analyzing the detailed causes of behavior,

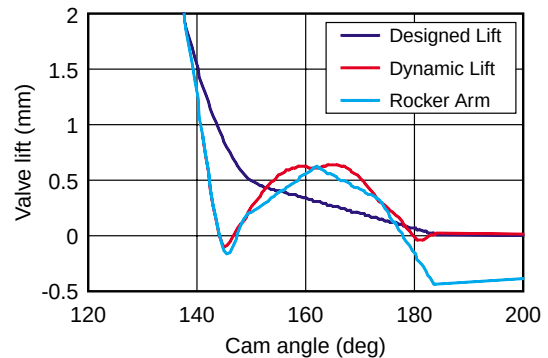


Fig.23 Valve simulation analysis

this laser measurement method is an essential technology for understanding changes in behavior of 10 μm or less.

4.2. Simultaneous Measurement of Multiple Cylinders

Two laser displacement meters can be mounted to each cylinder, so eight valves can be measured simultaneously as shown in Fig. 24. This enables comparison of behavior differences between cylinders, and makes it possible to confirm the effects of cam torsional and flexural rigidity between cylinders. It also enables analysis of the effect of rocker arm shaft flexural rigidity when constructing simulation models for analyzing the behavior of multiple cylinders simultaneously. In addition, fluctuations such as that shown in Fig. 25 can be simultaneously measured, so that this method can also be utilized to analyze the causes of abnormal phenomena resulting from the compounded behavior of multiple cylinders, rendering it a highly effective method.

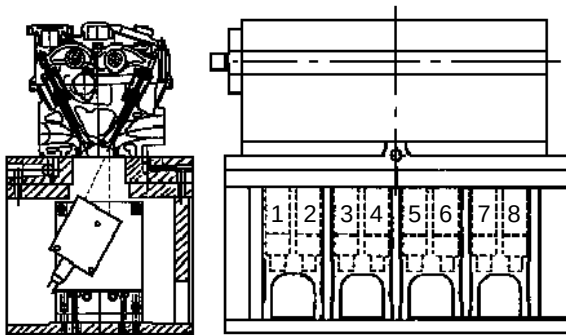


Fig.24 Position of laser sensor heads

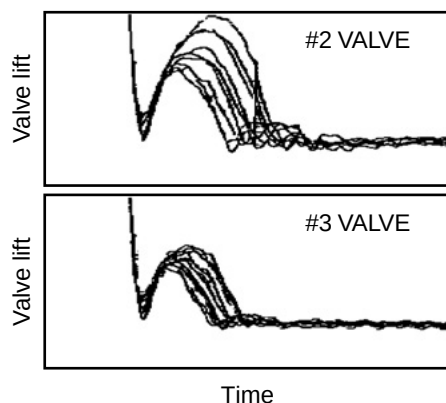


Fig.25 Multi-cylinder simultaneous data

5. Conclusions

The following issues arise when measuring valve train behavior with a laser triangulation method displacement meter.

- (1) Irregular reflection noise produced by walls
- (2) Zero cross noise during analog-digital conversion

Solving these issues made it possible to assure the required measurement accuracy of 10 μm or less.

This allowed detailed valve train collisions occurring at high revolutions to be extracted, and enabled highly efficient analysis of compound issues and other phenomena through simultaneous measurement of multiple cylinders.

Efforts will be made in the future to apply this technology to the measurement of other parts as well.

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Development of a Strength Prediction Technique for Automotive Exhaust Systems

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Ayumu ISHIZUKA*

ABSTRACT

A technique has been devised for predicting the strength of an automotive exhaust system in relation to the rotational force input from the torque reaction force of the engine.

Conventional static analysis conducted using the Finite Element Method (FEM) has been replaced by techniques for dynamic analysis that treat the force input accompanying large transient displacement as a forced displacement, and techniques that take into account the nonlinear nature of mounting rubbers, flexible joints, and so on. As a result, it has become possible to predict the strength of exhaust systems with greater accuracy, and to apply these techniques as development support tools.

1. Introduction

Although the development period for new model vehicles is decreasing, there are major demands for improved quality, decreased weight, lower cost, etc. Utilizing CAE is an effective means for meeting these needs in the development process, and the Finite Element Method (FEM) is generally used as a method for analyzing the vibration and strength of automotive components.

The exhaust system is subject to vibration from the road surface, vibration from the engine, and engine rotation while the engine is running; in the front wheel drive vehicle, the source of the greatest stress input is this engine rotation while the engine is running. Further, because the input displacement from this stress is large, the response of the exhaust system exhibits nonlinear characteristics. The input mode also has transient characteristics. These points are extremely important in predicting the strength of the exhaust system.

This paper describes how we applied NASTRAN transient response analysis and expressed component characteristics in terms of two opposing forces equivalent to action and reaction, developed a means for predicting transient stresses on an automotive exhaust system by applying nonlinear characteristics as a displacement/load map, and how we were able to apply these technologies as strength development tools.

2. Modeling a Nonlinear Exhaust System

The means for calculating the nonlinear strength of an exhaust system used in this paper is based on a linear vibration model, with the addition of select items that are essential for strength analysis. Table 1 describes the differences between our model and the base model.

Since it is necessary to determine the engine rotational

Table 1 Differences between vibration prediction FE model and nonlinear stress prediction FE model

		Vibration FE model	Stress FE model
Estimation value		Eigen value MAC value	Displacement response Stress
FE model basic element		Shell	Shell+solid
Displacement area of input		Small	Large
Restriction condition	Mounting rubber	Elastic numerical value of spring	Displacement-Load nonlinear MAP
	Flexible joint	Elastic numerical value + of spring Shape model	Displacement-Load + nonlinear MAP Shape model

input in the time domain, we elected to use transient response analysis. For the input conditions, we used engine rotational input equivalent to what would be generated while driving for the forced displacement, derived using the Lagrange multiplier method.

In the model, two opposing forces are represented in an action/reaction relationship. Because components that have a displacement-limiting structure, such as flexible joints and mounting rubbers as shown in Fig. 1, exhibit nonlinear characteristics in response to a large displacement input, the spring characteristics of such components are represented by two relative forces in an action/reaction relationship based on a linear numeric spring (vibration model). In addition, the model uses a pseudo-nonlinear spring with the nonlinear characteristics given as a displacement/load map ("nonlinear map," hereafter).

One feature of this method is that the pseudoforce method is used to model components that have a nonlinear spring. This makes it possible to represent the nonlinearity of the exhaust system materials within the scope of linear analysis without a major increase in analysis resources.

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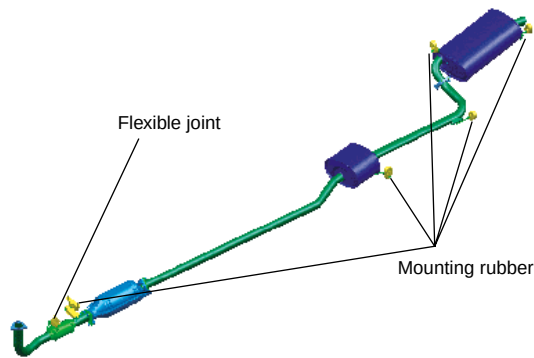


Fig.1 Nonlinear components of an exhaust system

3. Nonlinear Phenomena in an Exhaust System

3.1. Nonlinear Components in an Exhaust System

Two components in an exhaust system that exhibit nonlinear characteristics are shown in Fig. 2: the mounting rubbers and the flexible joint.

Fig. 3 shows the nonlinear map of a mounting rubber, and Fig. 4 shows the nonlinear map of a flexible joint. The flexible joint in particular exhibits strong nonlinearity.

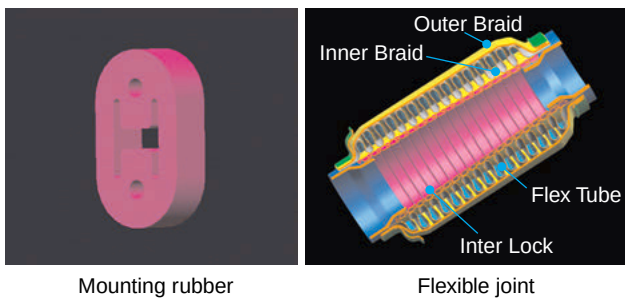


Fig.2 Nonlinear elements

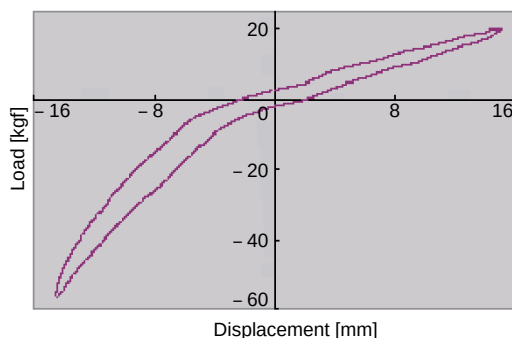


Fig.3 Hysteresis curve (Mounting rubber)

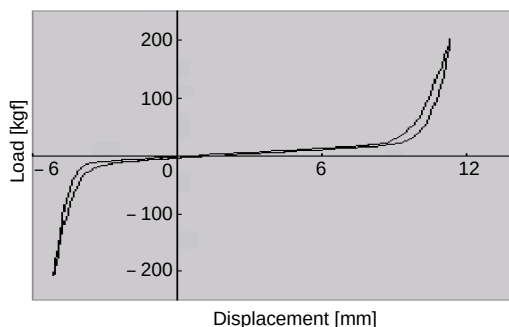


Fig.4 Hysteresis curve (Flexible joint)

3.2. Analysis of Nonlinear Phenomena

In order to understand the effect that the nonlinear characteristics of these components have on the behavior of the exhaust system, we conducted a vibration test in which we applied rotational input equivalent to that generated in an actual vehicle to the exhaust system through a hydraulic shaker, as shown in Fig. 5.

Fig. 6 shows the change in displacement behavior in the vertical direction of the end of the exhaust mounting points when the mounting rubber displacement stopper has been removed. Fig. 7 shows the horizontal displacement of the rearmost joint end when the outer braid and interlock components (the components that demonstrate nonlinear characteristics) are removed from the flexible joint, leaving only the tube.

These figures show that in large displacement areas, the nonlinear characteristics of components have a major effect on the displacement behavior of the exhaust system as a whole. Accordingly, the modeling of components that have nonlinear characteristics, such as mounting rubbers and flexible joints is an extremely important element in predicting the strength of an exhaust system.

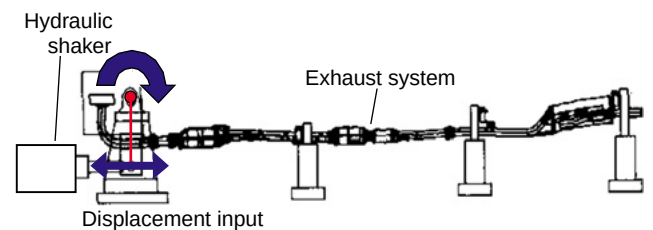


Fig.5 Rig test

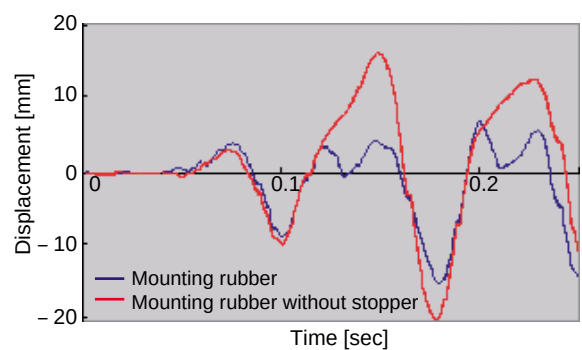


Fig.6 Influence of mounting rubber stopper

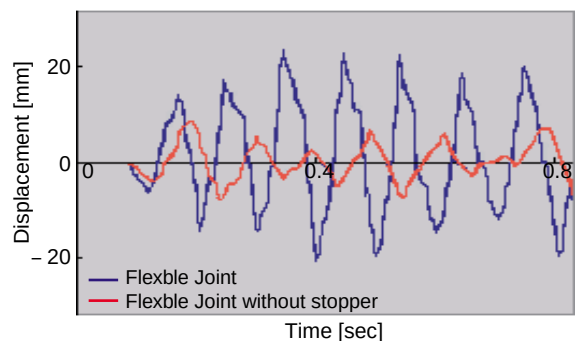


Fig.7 Influence of flexible joint stopper

4. Modeling Techniques

4.1. Overview of the Linear Model and Its Precision

Fig. 8 shows the Finite Element (FE) model that we used for the basic exhaust system (vibration model).

We used shell elements to model all exhaust system components that permitted shape modeling, such as the pipes and mufflers, and used solid elements only in the vicinity of welds.

Fig. 9 shows a comparison of the model with the frequency response level (up to 100Hz) generated by shaker input obtained in experimental results. The model matches the experimental results closely.

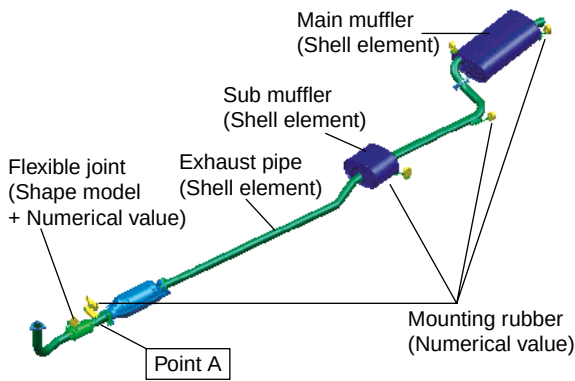


Fig.8 Exhaust vibration prediction model in linear area

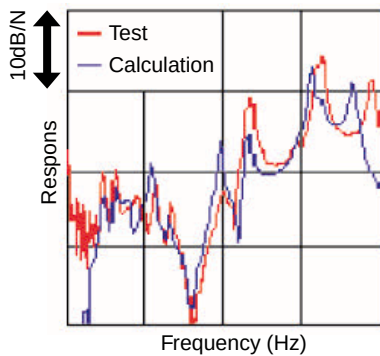


Fig.9 Results of the frequency response test (Point A)

4.2. Mounting Rubber Modeling

4.2.1. Overview of the Model

In order to understand the characteristics in the strength range, the mounting rubber modeling technique involved testing the spring characteristics in a wide displacement

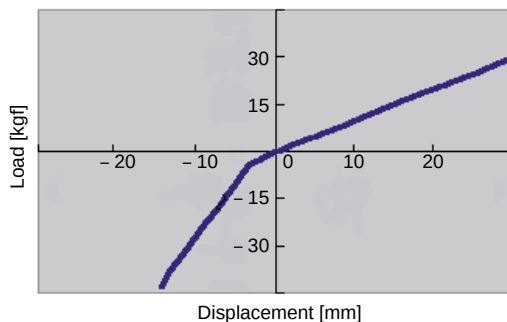


Fig.10 Nonlinear map of mounting rubber

amplitude area. We then created a spring characteristics map (Fig. 10) by performing linear approximation for both the linear area and the nonlinear area of the displacement-load curve that was obtained from testing (Fig. 3). Attenuation characteristics were also identified from the curve in Fig. 3.

4.2.2. Model Verification Results

Fig. 11 shows the results calculated for the standalone characteristics produced by the above models. The calculated results closely match the standalone test results.

Dynamic verification was performed by setting one mounting point at the end of the system and creating a straight pipe exhaust model (Fig. 12) supported by mounting rubbers, and then comparing the calculated results with the results produced by a rig test. Fig. 13 shows the results of a comparison of vertical displacement in the mounts as a time series. The calculated results correspond closely with the results of the rig test.

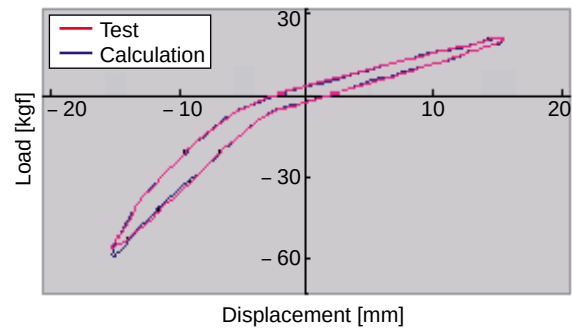


Fig.11 Hysteresis curve (Mounting rubber)

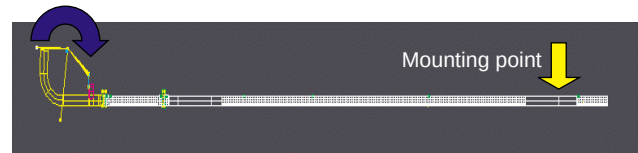


Fig.12 Simple exhaust FE model

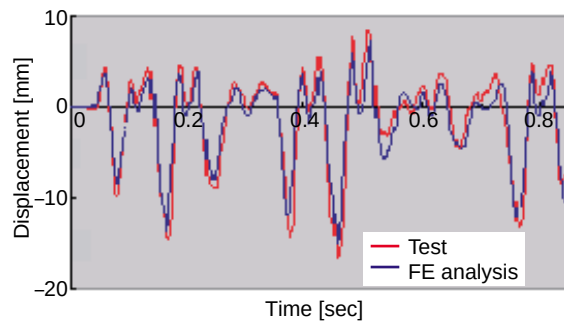


Fig.13 Response in the vertical direction at the mounting point

4.3. Flexible Joint Modeling

4.3.1. Overview of the Model

In modeling the flexible joint, we used a shape model for the flexible tube as shown in Fig. 14, and we used a nonlinear map similar to that used for the mounting rubbers for other elements.

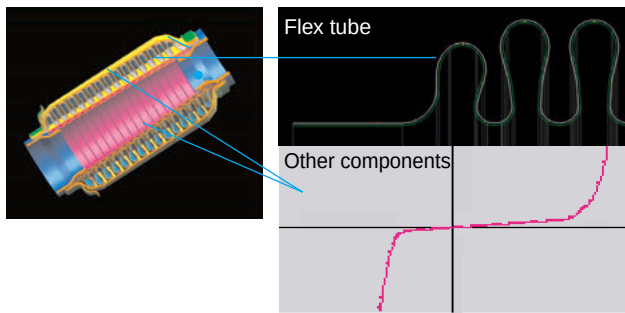


Fig.14 Flexible joint model

4.3.2. Flexible Joint Relative Displacement Calculation Technique

In order to use a nonlinear map to identify counterforces that are generated in the flexible joint and mounting rubbers, it is necessary to derive the amount of relative displacement of each component corresponding to the nonlinear map.

In the case of a mounting rubber, because the body side is regarded as a fixed point, as shown in Fig. 15, the displacement of the movable point of the exhaust side becomes the relative displacement of the mounting rubber.

On the other hand, in the case of a flexible joint, because both ends are movable points, it is necessary to derive the

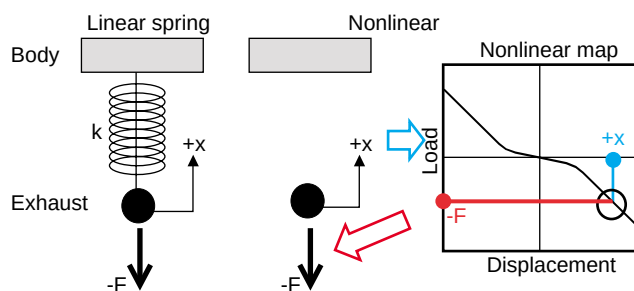


Fig.15 Load reference method for mounting rubber

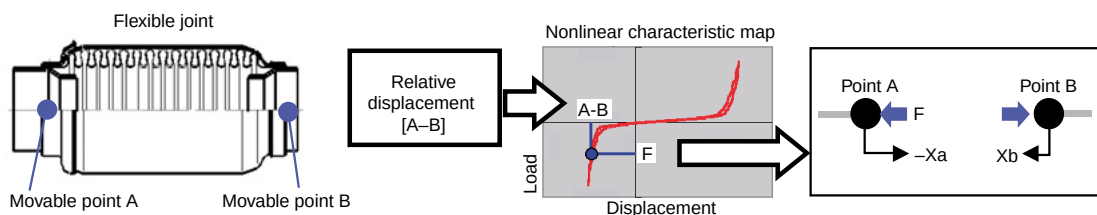


Fig.16 Relative displacement in the flexible joint

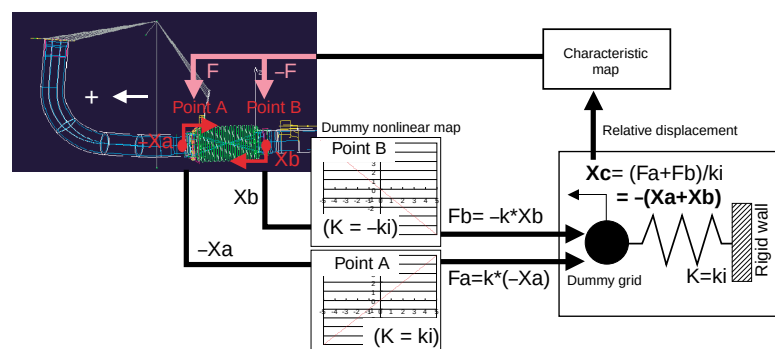


Fig.17 Dummy grid method

relative displacement from the difference in movement between the two points. However, it is difficult in NASTRAN to extract the relative displacement directly and to mathematically calculate the difference in movement from the amount of displacement of each point. Therefore, we proposed a method (the “dummy grid technique”) in which we would establish suspended structural nodes in the same analysis space as the exhaust system analysis model in order to calculate the relative displacement of both ends of the flexible joint, and then extract the relative displacement through the suspended structure linked to the ground by linear springs.

Fig. 17 shows an overview of the dummy grid technique. First, we created a dummy grid with a single degree of freedom that was independent from the exhaust system analysis model, and then linked that grid to the ground via a linear spring. Furthermore, the forces corresponding to the displacement of the movable points in front of and behind the flexible joint were applied to this system simultaneously. As a result, the displacement of the dummy grid becomes proportional to the relative displacement of the movable points in front of and behind the flexible joint, and generating forces in accordance with the dummy grid displacement become equivalent to generating forces in accordance with the relative displacement of the movable points in front of and behind the flexible joint.

4.3.4. Flexible Joint Model Verification Results

For the purpose of dynamic verification, we created a simple model (Fig. 18) that includes a flexible joint and then compared the results produced by a rig test and by calculation. Fig. 19 shows the results of a time series comparison of the relative displacement (in the vertical and horizontal directions) of the front and rear ends of the flexible joint. The results produced by calculation compare well to the results of the rig test, which means that it is possible to predict complicated flexible joint behavior through this modeling technique.

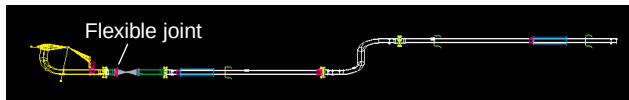


Fig.18 Simple exhaust FE model with flexible joint

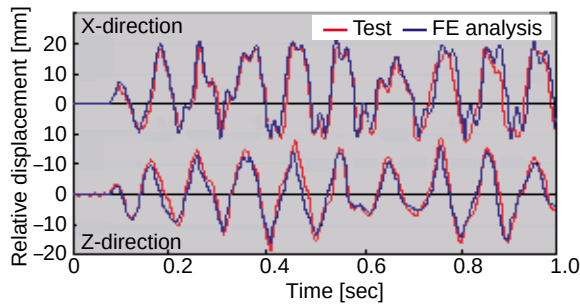


Fig.19 Relative displacement of flexible joint

5. Comparison of Results with Actual Tests

We applied all of the modeling techniques for each of the elements described above to an overall model of the exhaust system, then used the rig test apparatus shown in Fig. 20 to apply rotational input equivalent to that which would be generated in a real car, and compared the rig test results with the calculated results.

Fig. 21 shows a point-by-point comparison of the stress values predicted by the calculations and the stress values actually measured in the rig test. The trends in stress distribution obtained by the two methods match, with a high correlation coefficient of 0.96 obtained.

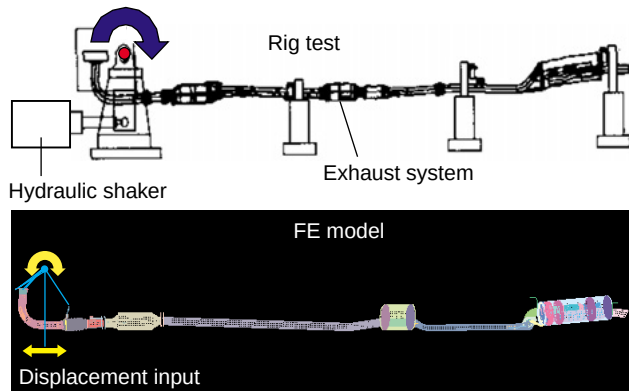


Fig.20 Rig test and FE model

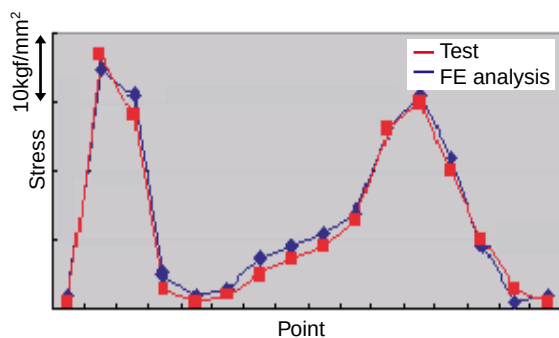


Fig.21 Exhaust system stress distribution

Figs. 22 and 23 show examples of the calculated results. The calculated results make it possible to understand the stress behavior corresponding to the displacement behavior of the flexible tube and the exhaust system as a whole, an understanding that is difficult to obtain experimentally.

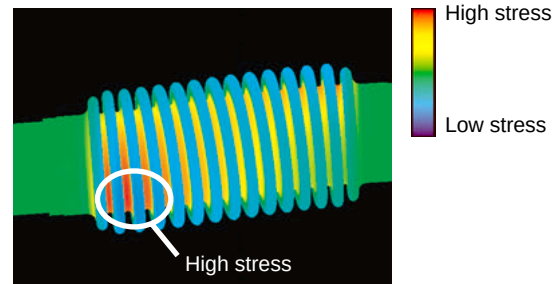


Fig.22 Flexible tube behavior

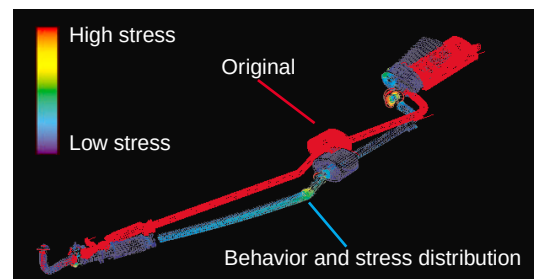


Fig.23 Exhaust pipe behavior

6. Case Study of Application

CAE is an effective tool for predicting stress, making small layout changes, conducting parameter studies, etc., during the drawing stage.

We predicted the differences in the stresses generated in the bellows with the flexible joint positioned in two different arrangements, Type A and Type B, as shown in Fig. 24.

The calculated results are shown in Figs. 25, 26, and 27. Figs. 25 and 26 show the behavior when the maximum stress is generated in the flexible joint, and Fig. 27 plots the stress values in that situation. These results show that Type B, with the flexible joint positioned in the front, is superior to Type A, which exhibited high stress due to the shearing mode. Fig. 28 shows that similar trends were obtained through rig tests, verifying the validity of the calculations.

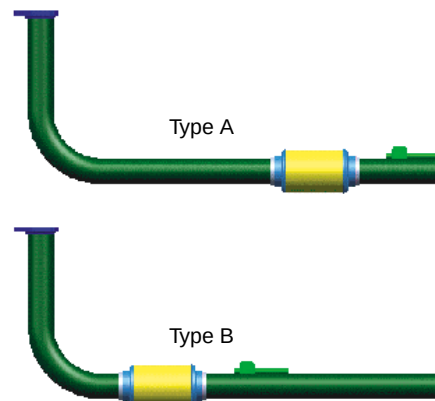


Fig.24 Comparative exhaust model

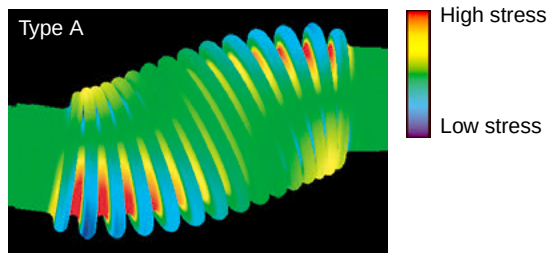


Fig.25 Flexible tube (typeA) stress distribution and behavior

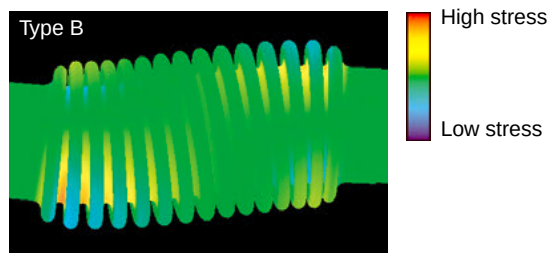


Fig.26 Flexible tube (typeB) stress distribution and behavior

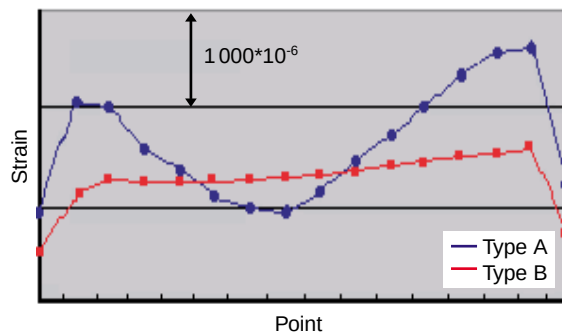


Fig.27 FE Analysis results

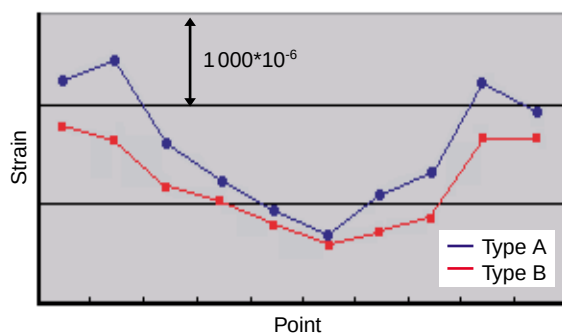


Fig.28 Rig test results

In this way, we were able, while in the drawing stage, to predict and analyze the behavior of dynamic stress in an exhaust system that includes a flexible joint, making efficient development possible.

7. Conclusions

- (1) By utilizing nonlinear maps and the proposed “dummy grid technique”, we were able to model nonlinear components, such as flexible joints and mounting rubbers, making it possible to predict stresses in a manner that takes the nonlinearity of the exhaust system

into account.

- (2) Detailed analysis of the locations where stress is concentrated was made possible by using shell elements or solid elements to represent all components that permitted modeling by shape, such as the flexible tube, pipes, and muffler.
- (3) By making nonlinear analysis possible for the FE model, we were also able to adapt certain parts of the model for a vibration analysis model.

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authors



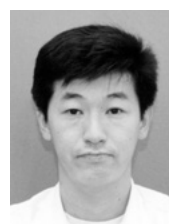
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Development of a Long Life Continuously Variable Transmission Fluid

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Junpei YOSHIDA* Yasuo HIRAO*

ABSTRACT

With the purposes of making the maintenance interval of continuously variable transmission (CVT) fluid as long as that of automatic transmission (AT) vehicles and of making its use common to AT vehicles, an ester friction modifier and high FW amide were used to develop a fluid that is optimally adjusted to friction characteristics by means of a metallic detergent. This fluid has at least 4 times the longevity of current products with respect to friction characteristics.

The initial vehicle quality such as the shift quality of AT vehicles, which is affected by the friction characteristics of a wet clutch transmission, and acceleration smoothness from stop of CVT vehicles are maintained by this new fluid, which has made it possible to extend the maintenance intervals of CVT vehicles. Also, the extended maintenance schedule has occasioned improved convenience for the user and contributed to reduction in the amount of liquid waste disposal because of the reduced frequency of fluid change. Therefore, it is expected that this new fluid will contribute to the prevention of pollution of the global environment.

1. Introduction

Automobile development has long been focused on the satisfaction of demands for both performance and environmental protection, together with the goals of more economical use of fuel and resources, as well as higher quality. Against this background, the automatic transmission fluid (ATF) used in automatic transmission vehicles has been the subject of continuous research in the attempt to improve friction characteristics, low-temperature viscosity characteristics, and oxidation stability.⁽¹⁾⁻⁽⁶⁾

Fig. 1 illustrates the progress of research into the improvement of automatic transmission fluid.

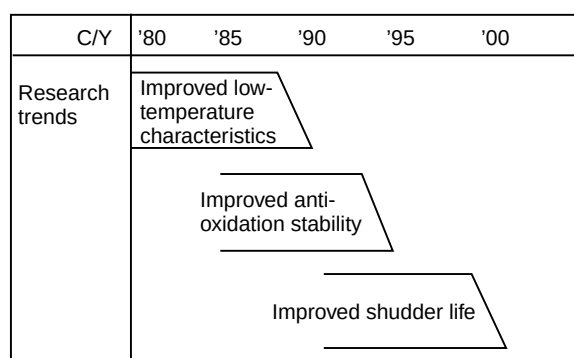


Fig.1 Research trends for ATF

On the other hand, global concerns for the protection of the environment have led to demands for the reduction in the amount of oils and fluids discarded, and the desire to respond better to increasing user needs have led to calls for longer service lifetimes for the various oils and fluids used in automobiles.

In response to such social concerns and the need to enhance the commercial competitiveness of continuously variable transmission (CVT) vehicles, we have engaged in the development of a new continuously variable transmission fluid (CVTF) for such CVT vehicles. Our aim has been to improve vehicle performance characteristics, including the shift quality of conventional automatic transmission vehicles, and the acceleration smoothness from start of vehicles with continuously variable transmissions.

In the development project discussed here, we analyzed the mechanism of deterioration whereby friction characteristics deteriorated in the start clutch of CVT vehicles as a result of the use of automatic transmission fluid. To counter such deterioration, we worked to develop a friction modifier that led to a means of extending the service life of the fluid. Further, we studied how that fluid might be adopted not only in CVT vehicles, but also to those with conventional automatic transmissions. By adjusting the friction characteristics that would allow application to conventional automatic transmissions, we achieved an extension of the maintenance interval of both continuously variable transmission and automatic transmission vehicles.

* Tochigi R&D Center

2. Development Objectives

With the ultimate aim of extending the service life of automatic transmission fluid, we set the following development objectives, focusing on improving the durability of the friction characteristics (particularly μ -V values) which have an important influence on product appeal:

- (1) To improve the durability of μ -V characteristics by a factor of 4 times or more compared to current automatic transmission fluids.
- (2) To provide the fluid friction characteristics usable in conventional automatic transmission vehicles.
- (3) To maintain low-temperature viscosity characteristics, anti-oxidation stability, and abrasion resistance performance at or above levels currently available in automatic transmission fluids.

3. Maintenance Intervals for Continuously Variable Transmission and Automatic Transmission Vehicles

Fig. 2 displays a comparison of current maintenance intervals for continuously variable transmission and automatic transmission (AT) vehicles commercially available in Japan.

As indicated by this chart, the maintenance interval for CVT vehicles is set lower than that for conventional AT vehicles.

One reason for the shorter maintenance interval for CVT vehicles is the fact that while automatic transmissions achieve a power-transmission efficiency of around 95%, the efficiency of continuously variable transmissions is only around 85%, meaning that a range exists within which efficiency is lower than with conventional automatic transmissions under conditions of a high stable speed. This lower efficiency produces higher fluid temperatures, thus adversely affecting the working life of the fluid.

As a result, use over long periods of time results in a deterioration of the friction characteristics of the automatic transmission fluid, leading to the potential for a lowering of the product value of CVT vehicles, and thus necessitating the setting of a routine maintenance schedule with shorter intervals.

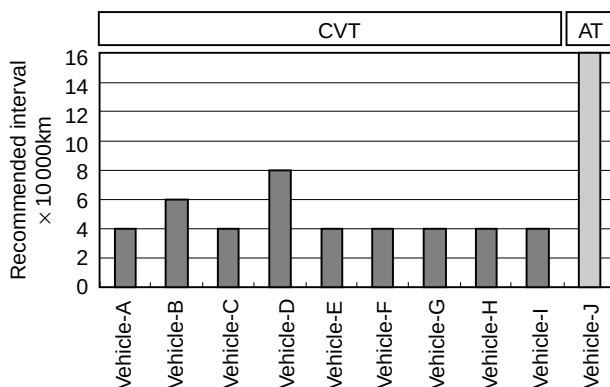


Fig.2 Maintenance interval recommended in Japan

4. Analysis of Deterioration of Friction Characteristics

In order to study the factors leading to the deterioration of friction characteristics over long periods of use, a vehicle endurance test was performed followed by an analysis of the vehicle's oil, in an effort to study ways of extending life.

4.1. Analysis of the Deterioration of Friction Characteristics

Fig. 3 shows the results of analysis of the friction characteristics of transmission fluid after a vehicle endurance test. As demonstrated on the chart of friction characteristics, the μ -V characteristics, which are measured in the low sliding velocity range, change over the course of the vehicle endurance test. The results shown here are the actual measurements of μ -V characteristics resulting from the combination of the automatic transmission fluid and wet clutch material recovered from the vehicle following the endurance test. The friction characteristics prior to the test in general demonstrate an inclination (upslope), but following the test, the μ -V characteristics of the aged material products indicate a down sloping or declination.

Specifically, when comparing the ratios of the two friction coefficients at the sliding velocities 2 rpm and 50 rpm (μ ratio: μ_2 / μ_{50}), the μ ratio of the new product (μ_2 / μ_{50}) is 0.92 and the μ ratio of the deteriorated product (μ_2 / μ_{50}) is 1.1, indicating that as the μ -V characteristics deteriorate, the μ ratio grows larger (greater than 1.0).

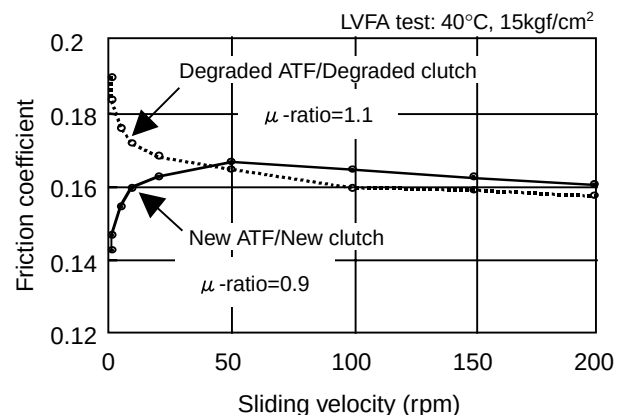
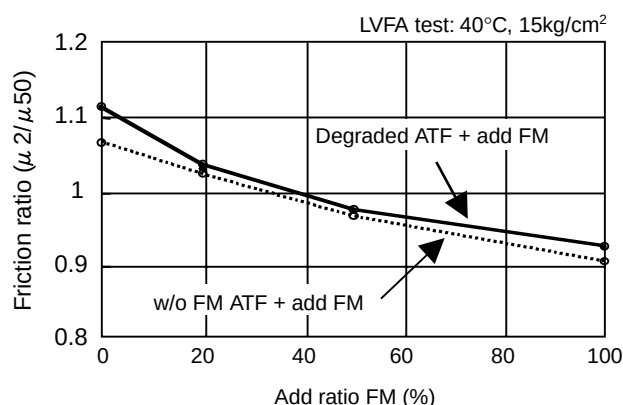


Fig.3 Friction characteristics of degraded ATF

4.2. Analysis of Deterioration Factors

Next, in order to make an analysis of factors involved in deterioration, we investigated the μ -V characteristics when 20%, 40%, 60% and 100% of the amount of friction modifier added to automatic transmission fluid was added to non-additive fluid (fluid to which no friction modifier had earlier been added), and the used fluid. The results are shown in Fig. 4. As friction modifier was added to the non-additive fluid and to the used oil, both came to produce the same kind of μ ratio as the new automatic transmission fluid with its additives.

From these facts, it became clear that the deterioration

Fig.4 Relationship of μ -ratio and FM

of friction characteristics was caused by reduced performance of the friction modifier added to the fluid.

5. Extending the Life of Friction Characteristics

Based on the understanding gained in the previously described factor analysis, we next studied how to extend the service life of the friction modifiers.

5.1. Oxidation and Friction Modifiers

First, we performed a study to find friction modifiers whose products would still demonstrate high friction characteristics even when oxidized.

Conventional carboxylic acid additives perform well at controlling friction characteristics, but they are also known for the friction performance they bring about as the result of oxidation reactions. In contrast, as indicated in Fig. 5, the Indiana Staring Oxidation Test (ISOT) demonstrates that ester additives maintain friction characteristics for more than twice as long as carboxylic acid additives.

In order to clarify the factors behind these results, we studied the heat-degradation products of oil treated with an ester friction modifier and detected carboxylic acid with the same structure as that possessed by carboxylic acid friction modifiers. From this fact, we deduced that the thermal cracking substance of ester additives may also function as friction modifiers.

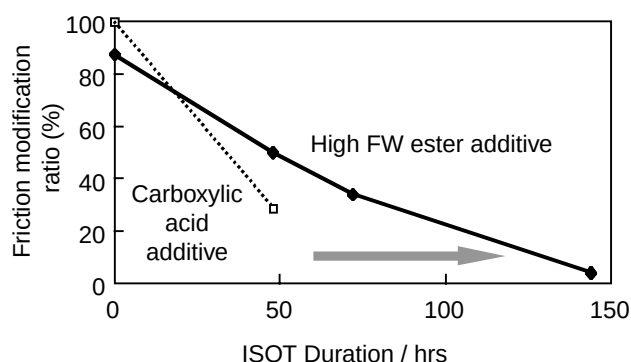


Fig.5 Durability of friction modification

5.2. Slow-Oxidizing Additives

Amide radicals are compounds produced as the result of the acid-base reaction between amines and carboxylic acid. While demonstrating lower polarity and adsorption than the group's amines and carboxylic acid, they have low reactivity since they are produced through a neutralizing reaction. As a result, they have less ability to modify friction characteristics, but are able to maintain those friction characteristics for an extended period of time.

We thereupon measured these radicals' life-extending properties by means of a part heating deterioration test apparatus equipped with a local heater as shown in Fig. 6.

The results of the test are shown in Fig. 7.

From our test we learned that of all the additives considered, the high-polymer amide compound possessed the greatest ability to extend longevity, extending the life some two times longer than conventional carboxylic acid additives.

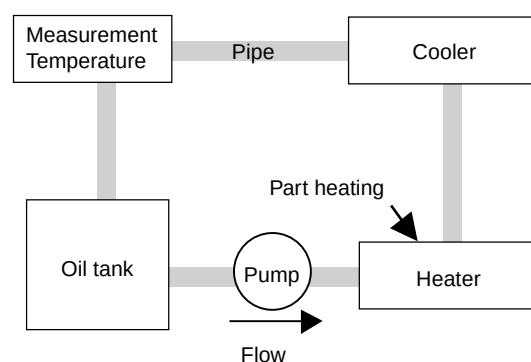


Fig.6 Part heating deterioration test device

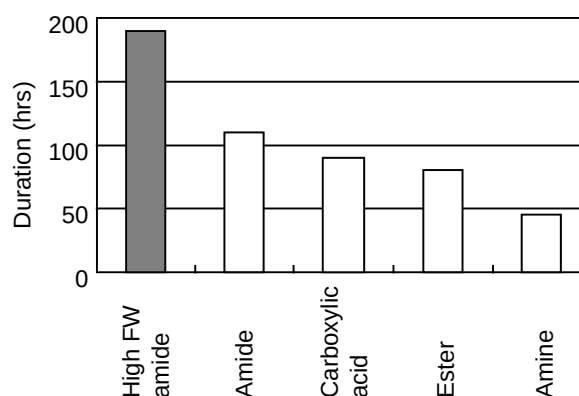


Fig.7 Durability of friction modification

6. Optimization of Friction Characteristics

The foregoing studies oriented us to the proper direction necessary for achieving our goal of anti-judder life, with the result that we now turned our attention to studying the optimum formulation of a friction modifier which would satisfy our friction characteristics, including application to conventional automatic transmission requirements.

6.1. Modification of Friction Characteristics

While attempting to achieve our goal of extended life by addition of a friction modifier, we discovered that the static friction coefficient μ_s , an index of transmission torque capacity, was lower than that produced with conventional automatic transmission fluid.

As a result, we reconsidered the composition of additives for automatic transmission fluid. We focused on the fact that in general, lowering the amount of friction modifier raises the μ_s , but since that method would not permit us to reach our goal of anti-judder life, we considered how to formulate an additive which would raise the μ_s while minimizing any fluctuations in friction modifier. An example of our formulation investigation procedure is shown in Table 1.

The results of our compounds and their friction characteristics are shown in Fig. 8.

The horizontal axis indicates the μ ratio (μ_0/μ_d), a general index of product appeal when used with a wet clutch, while the vertical axis indicates the wet clutch's transmission torque capacity, μ_s .

Compared to conventional automatic transmission fluid, prototype B has a lower friction ratio (μ_0/μ_d), thus indicating higher product appeal, but the μ_s value is also lower, thus giving it lower transmission torque capacity.

As a result of considering the various prototype fluids as listed in Table 1, we found that adjusting the ratio of friction modifiers and metallic detergents to produce prototype D resulted in a formulation which was capable of fulfilling our desired requirements for both μ ratio and μ_s .

Table 1 Optimal additive combination and volume

Evaluation fluid	Prototype B	Prototype C	Prototype D	Prototype E
FM-A	○	○	Increase	Decrease
FM-B	○	○	Decrease	Decrease
Detergent-A	○	Increase	○	Decrease
Detergent-B	○	Decrease	○	Decrease

○ : basic

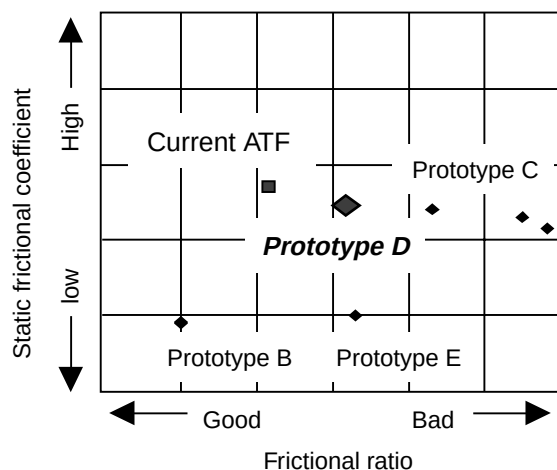


Fig.8 Friction characteristics of prototype ATF

7. Specifications for the Developed CVTF

Table 2 shows the final specifications which we arrived at based on the above studies. These specifications are for a continuously variable transmission fluid which possesses viscosity characteristics and shear stability performance equaling that of current automatic transmission fluid, while also achieving our goals of anti-judder life and friction characteristics.

8. Effects Extending the Life of Friction Characteristics

8.1. Results of Bench Evaluation

The newly developed continuously variable transmission fluid was used in an anti-judder life evaluation test using the start clutch from a continuously variable transmission automobile. The bench test conditions are shown in Table 3. The test involved use of an actual engine and continuously variable transmission, adopting engine firing to measure mode operation, and checking the continuously variable transmission fluid's friction characteristics (μ -V characteristics) after each elapsed operation cycle. The results of this test are shown in Fig. 9. Compared to conventional automatic transmission fluid, the newly developed continuously variable transmission fluid demonstrated an effect of extended life represented by an approximate four-fold increase in μ -V characteristics.

Table 2 General characteristics of new CVTF

		Current ATF	New CVTF
Kinematic Viscosity cst	40°C	29.51	29.55
	100°C	7.285	7.208
Low Temp. Viscosity cp	- 40°C	6 800	8 550
TAN	mgKOH/g	0.51	0.73
TBN	mgKOH/g	3.72	4.12
Aniline point	°C	95.2	100.7
Metal contents ppm	Ca	800	400
	Mg	—	230
	P	200	170

Table 3 Bench endurance test condition

Item		Condition
Transmission		CVT mission
Engine		1.6L SOHC
Test mode		High speed endurance mode
1 Cycle	Distance	60 km
	Av. speed	150 km/hr
	Max. speed	180 km/hr

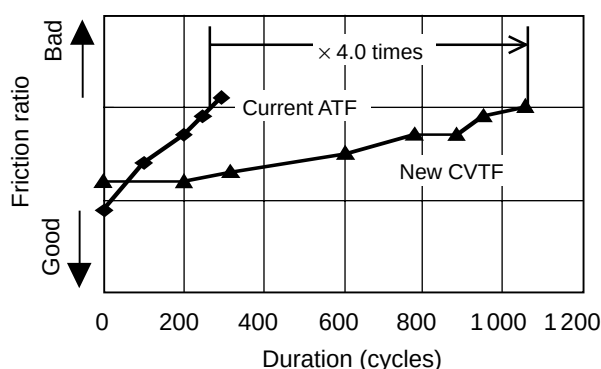


Fig.9 Bench endurance test result

8.2. Results of Road Testing

Next, an actual vehicle was used to road test the endurance effects of the fluid under conditions of high-speed driving. Utilizing a 1996 model automobile equipped with continuously variable transmission, an evaluation was performed involving high-speed driving conditions on an expressway, focusing on a measurement of μ ratio values of the used CVT fluid. As shown in Fig. 10, the results of the test indicate that the newly developed continuously variable transmission fluid achieves roughly a three-fold anti-judder life compared to current conventional automatic transmission fluids.

In short, the newly developed continuously variable transmission fluid possesses the hoped-for effects of extended friction characteristics and applicability to conventional automatic transmission vehicles, while also allowing continuously variable transmission vehicles to be given maintenance schedules equivalent to those for conventional automatic transmissions.

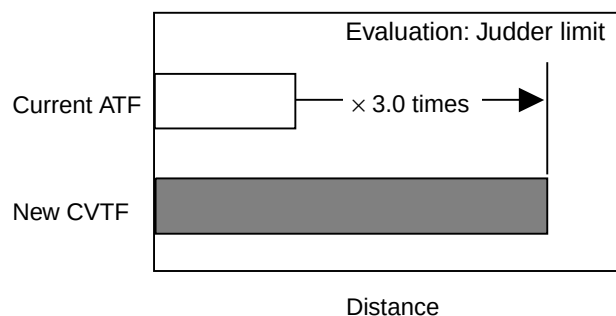


Fig.10 Vehicle endurance test result

9. Conclusions

The continuously variable transmission fluid development project discussed here achieved the following goals:

- (1) clarified that the cause of deterioration of automatic transmission fluid friction characteristics is the degraded performance of the friction modifier due to oxidation;
- (2) extension of the life of friction characteristics, to a

maximum 4 times can be achieved by a compound additive formulated from ester friction modifiers with excellent heat-resistance, and high-polymer amides;

- (3) by adjusting the formula of friction modifiers and metallic detergents, a continuously variable transmission fluid was produced which possesses friction characteristics equivalent to those of conventional automatic transmission fluid, and which can be applied to current automatic transmission vehicles.

Plans are being made to apply the newly developed continuously variable transmission fluid to new automobile models which will be marketed this fall. Adoption of this long-life continuously variable transmission fluid will not only improve vehicles' product appeal, but also extend their maintenance intervals, thus enhancing convenience for owners, and reducing the amount of waste fluid, to thereby contribute to lower loads being placed on the global environment.

Acknowledgments

In conclusion, we would like to express our deep appreciation to Nippon Mitsubishi Oil Corporation and Idemitsu Kosan Co., Ltd for their cooperation in the preparation of prototype formulas.

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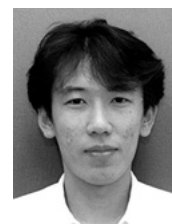
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Development of a Long Life Engine Coolant

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ABSTRACT

We have developed a long life engine coolant that has performance equivalent to current coolants with respect to impact on cooling system parts and anti-corrosion performance for extended metal life through the use of organic carboxylate, inorganic oxychloride inhibitors, and other compounds added in appropriate amounts.

Vehicles utilizing this coolant realized maintenance intervals more than two times longer than with current coolant. In addition, this maintenance reduction reduces the amount of coolant waste generated with regular replacement by approximately 70% or more, and makes it possible to reduce disposal costs and incineration volumes.

1. Introduction

Global concerns to reduce environmental waste load have led to increasing calls for recycling and reuse. In addition, the crucial issue of the cost for disposal of waste fluids remains. Further, demands are also being heard for reducing the maintenance costs associated with automobile ownership. Thus, there are strong needs for long life consumables. As one response to those needs, we set our sights on extending the longevity of various oils used in automobiles.

As shown in Fig. 1 as a result of revisions in control laws and systems of regular vehicle inspection, non-amine types of corrosion inhibitors were developed and adopted for use in automobile coolants. In the 1990s, maintenance intervals were set at 2-3 years after the initial application, but recently manufacturers have more commonly begun recommending intervals of once every five years.

In the face of such universal social needs, as well as

demands for extended coolant maintenance intervals, the current project aimed at the development of a long life engine coolant.

In this article, we introduce the development of this new coolant through a discussion of inhibitor longevity performance and anti-corrosion mechanisms, together with an evaluation of parts compatibility.

2. Development Objectives

Numerous demands are placed on the coolant used in internal-combustion engines,⁽¹⁾ as summarized below:

- (1) to prevent freezing of the coolant system passages under low-temperature conditions, or when used in cold-winter areas;
- (2) to maintain the temperature differential between the coolant and ambient temperature by the effect of raising the boiling point of the coolant;
- (3) to help prevent metal corrosion of the cooling system;
- (4) to be compatible with the various parts used in the cooling system.

Of these demands, (1) and (2) are innate characteristics of the combination of the coolant's basic ingredients ethylene glycol (EG) and water, and have no impact on the longevity of the fluid. On the other hand, (3) and (4) are decisive factors in coolant longevity, with the result that we set the following development goals with regard to these two points:

- to produce a coolant that would provide corrosion prevention for at least twice as long as current coolants, at current coolant ratios;
- to suppress any adverse influences on cooling system parts to current levels.

	'85	'90	'95	'00
Law and technical history	● US ★ Toxicity examination of Inhibitor in EPA ('88-) ★ Acceptance of bitterness additive ('93-) ☆ Maintenance interval 2 years → 3-5 years	● JPN ★ Amine export regulation ('88-) (Amine LLC → Non-amine LLC) ☆ Maintenance interval 2 years → 3-5 years	● EU ★ Correspondence to non-amine ('87-)	
Maintenance interval tendency	2 years/30000-60000km	3 years/60000-100000km	5 years/100000-240000km	
Honda response	☆ Non-amine regulation conformance (Conventional LLC) ☆ Maintenance interval increase			

Fig.1 Coolant technical and maintenance history

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3. Extending Coolant Longevity

The basis for coolant maintenance interval settings is the deterioration of anti-corrosion performance. Thus, based on current coolants, we studied the anti-corrosion mechanism of current coolants, together with means of extending its longevity.

3.1. Analysis of Recovered Coolants

In order to understand trends in the deterioration of currently available coolants, we performed coolant sampling of currently operating vehicles. The sample totaled more than 300 vehicles from all over the world.

(1) Inhibitor changes

Figs. 2 and 3 portray examples of changes over time to the inhibitor within actual automobiles. Inside an automobile, inhibitors are exposed to heat, metals, and polymers, subjecting them to deterioration and adsorption, resulting in substantial volumetric decline from their initial levels. From our analyses of these phenomena, we learned that inhibitors can be divided into inorganic acids which are consumed quickly, and organic acids which undergo almost no change in level over long periods of time. Based on these facts, we discovered that the use of organic acid inhibitors is effective in extending the longevity of coolant anti-corrosion properties.

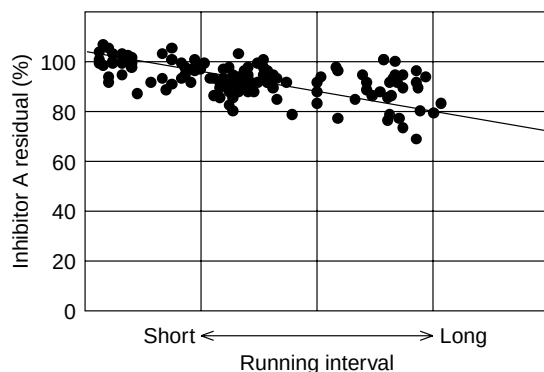


Fig.2 Market deterioration of conventional Honda Genuine coolant inhibitor A

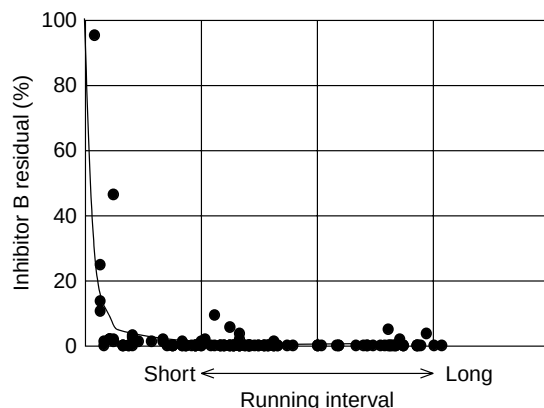


Fig.3 Market deterioration of conventional Honda Genuine coolant inhibitor B

(2) Changes in by-products of degradation

When the ethylene glycol comprising the basic coolant ingredient degrades, it changes into its by-products, formic acid and glycolic acid,⁽²⁾ but these cause a lowering of the solution's pH, leading to corrosion. The causal factor in this degradation is high-temperature oxidation caused by heat, dissolved oxygen and contact with metal.

We then examined time-based changes in the production of degradation by-products in currently used coolants. The results of that study are shown in Fig. 4. These results show that by-products of degradation increase in direct proportion to elapsed engine operating time. In sum, we learned that to prevent degradation, it was necessary to formulate a coolant with pH buffering properties capable of suppressing oxidation.

(3) Changes in Metal Ions

Next, we studied changes in metal ions released over time. Fig. 5 depicts the volume of aluminum (Al) ion generated, while Fig. 6 depicts the same for copper (Cu) ions generated. The results showed only slight generation of aluminum ions, while the volume of copper ions produced tended to rise with long elapsed operating times. As a result, we learned that one issue in extending the longevity of current coolants is the necessity of extending the coolant's ability to resist copper corrosion.

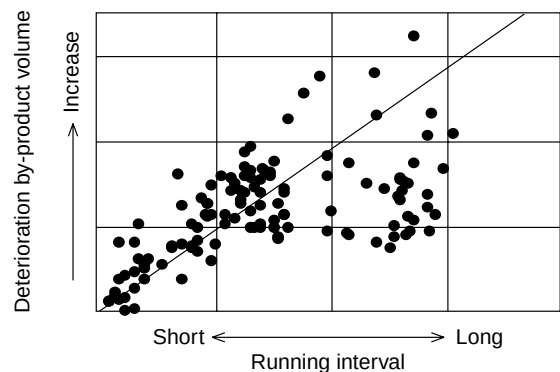


Fig.4 Market behavior of by-product in engines using conventional Honda Genuine coolant

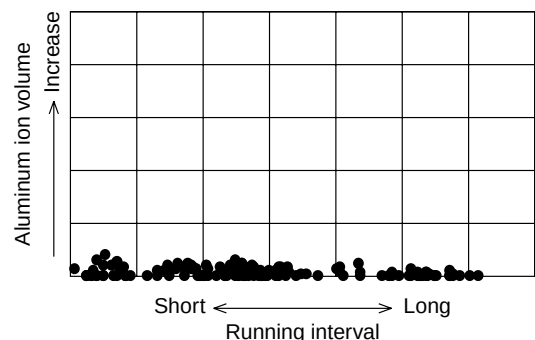


Fig.5 Aluminum ion occurrence in engines using conventional coolant

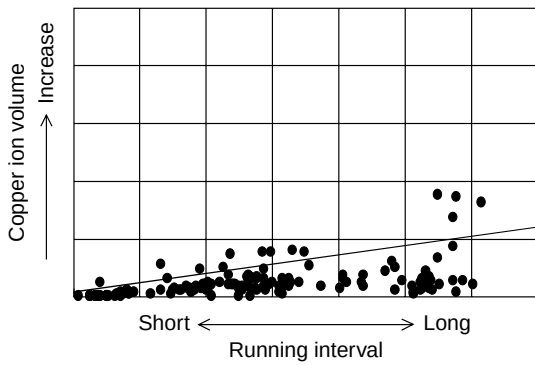


Fig.6 Brass ion occurrence in engines using conventional coolant

4. Consideration and Evaluation of a New Formula Coolant

When formulating the ingredients for a new coolant, we focused on the three points noted above, namely, use of long life inhibitors, improvement of pH buffering performance, and extending the longevity of the coolant's copper corrosion resistance.

Specifically, we selected a coolant based mainly on an organic acid inhibitor, also possessing heat-resistance, corrosion resistance, and compatibility with coolant system

	Metal appearance after being dipped in coolant					
	Aluminum	Cast Iron	Steel	Brass	Solder	Copper
Organic acid prototype coolant						
Weight loss toughness	OK	OK	OK	OK	OK	OK
Current coolant						
Weight loss toughness	OK	OK	OK	OK	OK	OK

Fig.7 Metal corrosion test using the organic acid prototype and the current coolant

	Radiator lower tank after operation
Organic acid prototype coolant	
Current coolant	

Fig.8 Parts recovered from each engine test using the organic acid prototype and the current coolant

materials. In our selection, we spent the most time confirmation testing all metal parts throughout the coolant circulation system for anti-corrosion efficiency under circulating conditions, anti-cavitation corrosion characteristics, and changes in the physical properties of polymers.

4.1. Evaluation of Metal Corrosion Resistance

Selecting the optimum organic acid as our long life inhibitor, we formulated a prototype organic-acid type coolant and then evaluated the prototype's corrosion-prevention properties. The results of our metal corrosion testing are shown in Fig. 7, while results from parts collected after bench testing are shown in Fig. 8 compared with the existing coolant.

According to these tests, no unnatural phenomena could be detected using currently available conventional coolants. On the other hand, laboratory tests of metal corrosion using the prototype coolant resulted in some blackening of the aluminum sample, but the prototype still appeared good, producing no change in mass. When parts from actual vehicles were recovered and examined, however, whitish sediment was found. An analysis of the sediment showed it to be aluminum salts resulting from corrosion produced within the engine environment.

As a countermeasure, we changed the inhibitor. To measure initial corrosion resistance, we, for example, set up a circulation corrosion test which could simulate the conditions of an actual engine, and thus formulated a new coolant which could maintain the anti-corrosion function for aluminum parts.

To evaluate the new coolant's anti-corrosion performance, we performed anti-corrosion tests for aluminum heat-transfer surfaces⁽³⁾ as shown in Fig. 9; the results of the Circulation Corrosion Test are shown in Fig. 10. As indicated in Fig. 9, we confirmed no remarkable discoloration or aluminum ion elution. Fig. 10 shows that under the conditions of coolant circulation, the new coolant possesses good aluminum anti-corrosion properties. On the other hand, the aforementioned prototype organic acid coolant did not possess sufficient aluminum corrosion prevention properties, and significant blackening and coolant flow erosion could be detected on the aluminum heat-transfer surfaces.

Coolant	New coolant	Organic acid prototype coolant
Appearance		
Weight loss by corrosion	No problem	No problem
Aluminum anti blacking	Good	Bad

Fig.9 Anti-corrosion performance on aluminum heat-transfer surfaces

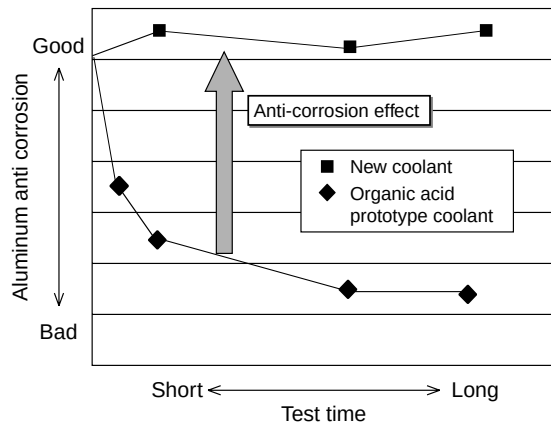


Fig.10 Circulation corrosion test results

4.2. Compatibility with Polymer Materials

Since it is difficult to suppress polymer oxidation degradation through the deliberate use of additives, it is necessary to evaluate the formulated coolant's resistance to polymer degradation and to confirm that it presents no adverse impact on polymer service life. To test this property of the coolant, we used rubber and nylon resin parts as our polymer materials for testing. One example of each of these is shown in Figs. 11 and 12.

In the case of rubber parts, we confirmed that the newly formulated coolant's compatibility with rubber is equivalent to current conventional coolants in terms of changes in hardness, volume, elongation, and tensile strength. Further, nylon radiator parts likewise demonstrated no change in their fulfillment of minimum creep resistance standards, confirming that they are not adversely influenced in terms of their ability to withstand heat and pressure.

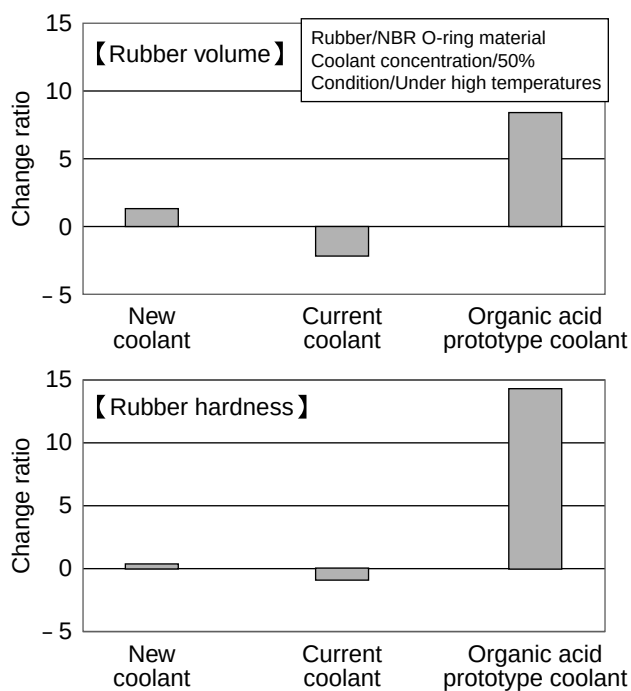


Fig.11 Coolant performance with rubber

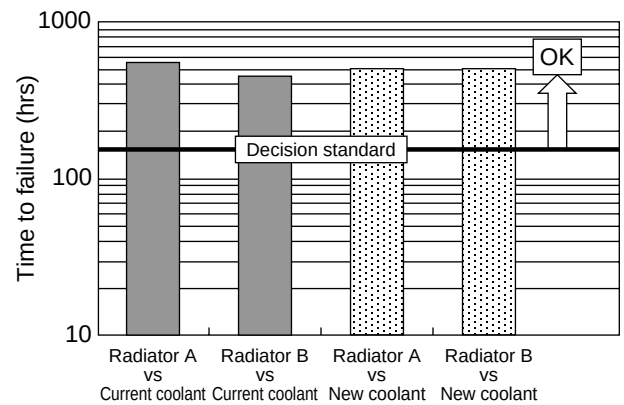


Fig.12 Creep performance of plastics radiator tanks

4.3. Compatibility with Cooling System Parts

Before adopting a new coolant, it is necessary to confirm that the coolant is fully compatible with all the parts composing the cooling system.

In particular, the water pump must be able to circulate the coolant while remaining sealed, and of all the parts of the cooling system, the mechanical seal at the pump's core is considered the most sensitive to new coolant formulas.⁽⁴⁾ Fig. 13 depicts the mechanism of leakage from the sliding surfaces of the pump. Fig. 14 shows the results of an evaluation of the sealing characteristics of the rotating

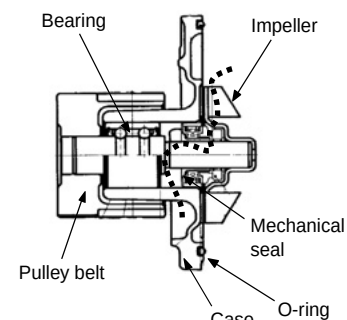


Fig.13 Water pump coolant leak mechanism

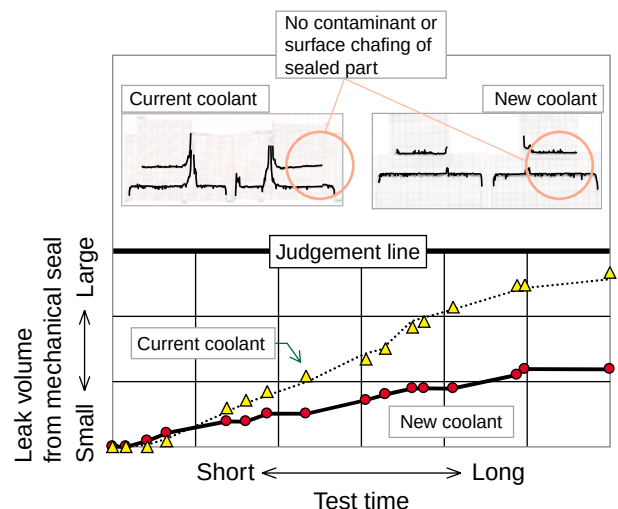


Fig.14 Compatibility of the new coolant with the water pump mechanical seal

mechanical seal when considered in isolation. From this, it can be seen that the new coolant formula demonstrates performance equivalent to conventional coolants in fulfilling the minimum criteria for coolant leakage with no damage to seal performance. It also demonstrated no problem of contaminant buildup or chafing on the mating ring and seal ring sliding surfaces, allowing us to determine that it fulfilled all requirements for compatibility.

4.4. Road Test Results

Next, a vehicle engine was used to test for overall engine compatibility, the results of which are shown in Figs. 15-17. Fig. 15 displays photographs of the engine water jacket. The photographs show no corrosion on the aluminum heat-transfer surfaces, indicating good anti-corrosion properties. Fig. 16 is a cross-section cutaway of one part of the water outlet made to expose the surface layer. The new coolant demonstrates no noteworthy corrosion pitting, and based on the analysis indicated in Fig. 17, it can be confirmed that the surface aluminum oxide film has been maintained,

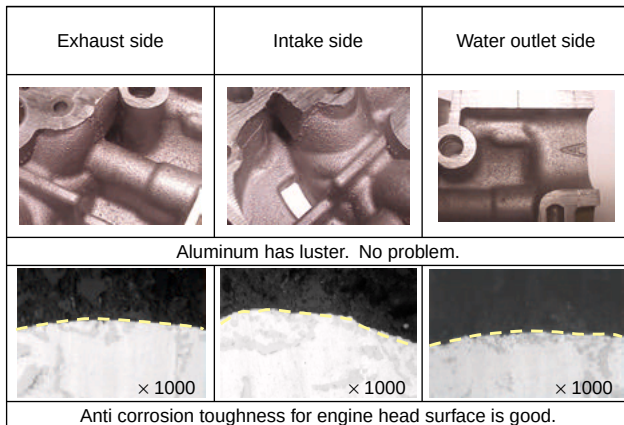


Fig.15 Heated surface condition of the engine head after bench testing using the new coolant

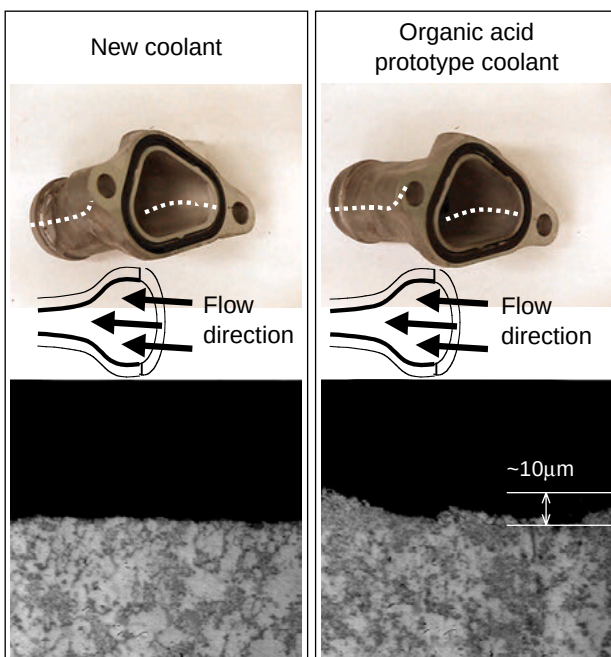


Fig.16 Surface section of water outlet area

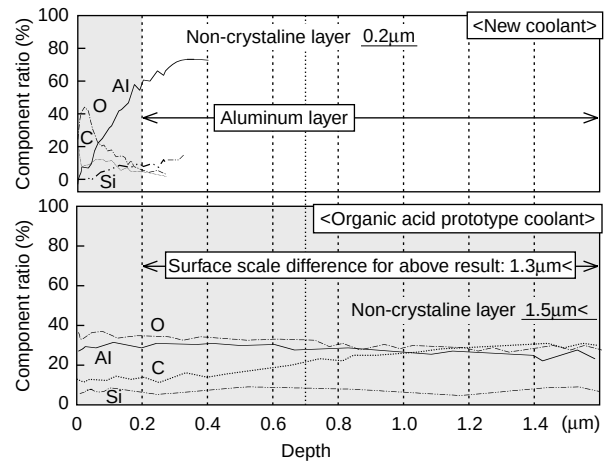


Fig.17 Element composition of the layers on the water outlet surface

indicating excellent compatibility with engine parts. On the other hand, the prototype organic acid coolant provided as a control demonstrates advancing corrosion of the surface aluminum, a sign of coolant flow erosion.

4.5. Results of Endurance Evaluations and Longevity

While confirming the initial performance characteristics of the new coolant fluid, we also performed corrosion testing to consider the load on an actual vehicle to evaluate potential coolant degradation. Table 1 shows anti-corrosion results when metal is soaked in coolants aged to a condition equivalent to 10 years old. According to these results, the new coolant formula caused no corrosion as demonstrated by gain or loss of metal volumes, for all the metals present in the cooling system, including aluminum, cast iron, steel, solder, bronze, and copper, thus demonstrating excellent retention of its anti-corrosion performance.

Fig. 18 shows the results of a simulation of coolant degradation in an actual vehicle. Compared to currently available conventional coolants, the new coolant delivers more than twice the anti-corrosion service life. In addition, this fact should make it possible to extend the regular maintenance interval for coolant.

Table 1 Corrosion toughness for equivalent to a coolant used for roughly 10 years

	Alumi-num	Cast Iron	Steel	Brass	Solder	Copper
General organic acid type coolant	×	×	○	○	×	○
Current coolant	○	×	○	○	×	○
New coolant	○	○	○	○	○	○

○: Good anti-corrosion

×: Poor anti-corrosion

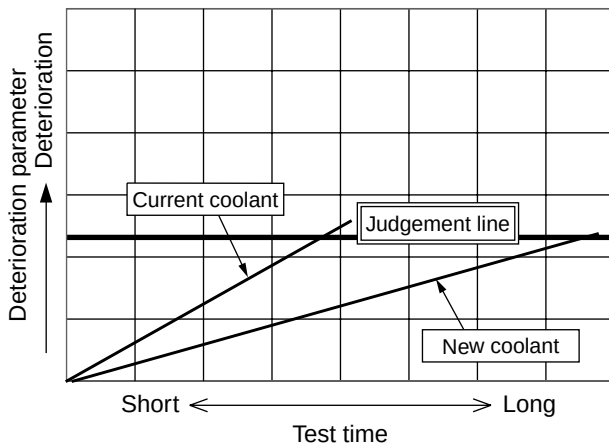


Fig.18 Long life performance of the new coolant compared with the current coolant

5. Effect of Reducing Discard Waste and Improved Quality

Finally, we calculated the effect that extended service life would have on reducing discarded coolant fluids throughout the world. These results are portrayed in Fig. 19. According to our calculations, based on 14-year automobile ownership with the same coolant fluid, the amount discarded would be reduced by 70% over current levels. As a result, by reducing the amount of such discard waste, switching from current conventional coolants to the newly developed coolant can be expected to contribute to a substantial lessening of environmental waste load and an economizing of resources.

Further, the extension of coolant service life means that it may be possible for users to merely make occasional additions to the engine coolant, avoiding the draining and changing of coolants throughout the entire life of the vehicle. This could promote higher performance through the avoidance of inferior aftermarket coolants and poor-quality water.

The newly developed coolant is planned to be installed on all new models from the 2001 model year, and its distribution will be expanded throughout the world.

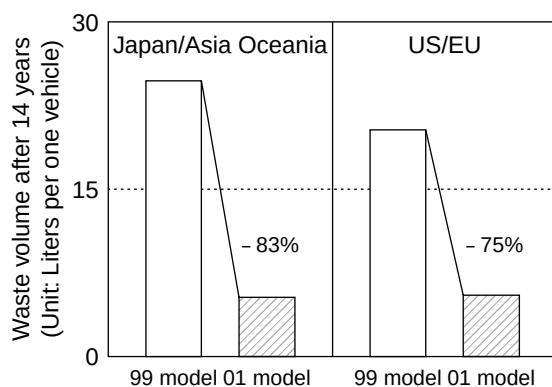


Fig.19 Coolant waste reduction volume through use of the new coolant

6. Conclusion

The current study can be summarized in the following points:

- (1) by adding appropriate amounts of organic carboxylate and inorganic oxychloride inhibitors, we were able to develop a new coolant with twice the metal-corrosion prevention performance of currently available conventional coolants;
- (2) we confirmed that the new coolant has compatibility to polymers and other parts of the cooling system equal to currently available coolants;
- (3) by using the new coolant, a reduction of more than 70% of the waste fluids represented by coolants could be achieved throughout the world (based on the assumption of 14-year automobile ownership).

7. Acknowledgments

We would like to take this opportunity to express our deep appreciation to coolant manufacturer C.C.I., Inc., and to the various manufacturers of cooling system parts for their cooperation in our formulation and compatibility testing.

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Development of a Lead-Free Fuel Tank

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ABSTRACT

A hot dip Sn-Zn alloy (8% Zn) coated steel sheet having superior corrosion resistance, formability and processability was developed as a lead-free fuel tank. As soldering material, an Sn-Ag alloy (3.5% Ag) having low temperature soldering characteristics and superior galvanic corrosion characteristics with the coating was developed.

An issue in the production engineering area was that consecutive spot welding quantity declined to 1/2 the number of current welds. The cause was degradation of the electrode caused by an alloy reaction between the metallic electrode and the coated metal. The technical improvement measures to handle this were the surface contact resistance was controlled, the weld current was reduced, and the shape of the electrode tip was optimized.

This enabled attainment of processability equivalent to current levels using existing equipment, and development of a fuel tank having superior corrosive durability, formability and welding characteristics, and overall performance.

1. Background

Laws and regulations that call for reducing the use of substances such as lead that place a burden on the environment are becoming more common. Examples of which include the European waste regulations (ELV) and the Japanese Ministry of International Trade and Industry laws related to waste treatment and clean up. Control standards for products containing lead are scheduled to be tightened. An example is the announcement in May 1997 of Japanese government targets as a part of the end-of-life automobile recycling initiative.

The Voluntary Recycling Action Plan calls for a reduction in the amount of lead (chemical symbol Pb) used per automobile to less than 1/3 of the 1996 level by the end

of 2003. Fig. 1 shows the major parts that use lead and the amounts of lead used per automobile (source: Japan Automobile Manufacturers Association).

The amount of lead used in a fuel tank is typically 200 g, which is equivalent to roughly 11% of the total amount used per automobile. The use of lead-free fuel tanks will therefore be indispensable for reducing the amount of lead used to 1/3 of the 1996 level.

2. Present State and Important Issues

As shown in Fig. 2, the lead used in fuel tanks is in the form of steel sheet coating components, solder metal and coating components for pipes and other parts.

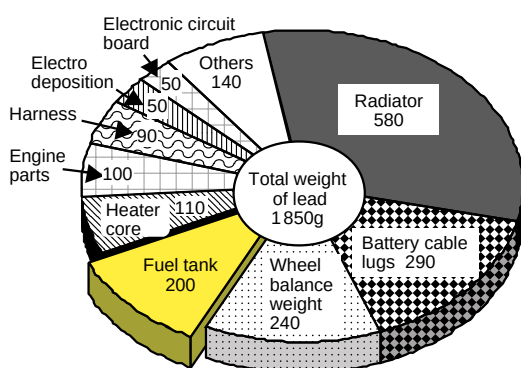


Fig.1 Automobile parts containing lead and amount they contain

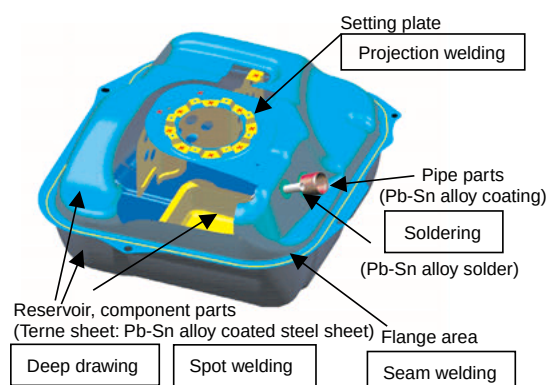


Fig.2 Areas where lead alloys are used, the alloy type is in parenthesis and the production technology is in the box

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*** Nippon Steel Corporation

Fuel tanks are required to have high corrosion resistance on both their inner and outer surfaces, and as shown in Fig. 2, deep drawing formability, as well as, for example, suitability for various types of welding and joining techniques. The terne sheet currently used as material for fuel tanks inherently possesses outstanding suitability with respect to all of these requirements.

(1) Internal Corrosion Resistance

Soured gasoline, which contains numerous olefin components and so forth, degrades due to the formation of organic acids resulting from oxidation. Moreover, the moisture that condenses in the tank results in a high acidic concentration. ⁽¹⁾⁽²⁾ Terne sheet is chemically stable against these organic acids.

(2) External Corrosion Resistance

A high level of corrosion resistance is required in environments where the vehicle is splashed by water containing rock salt and other salts that are sprinkled on roads to melt snow. The chemical stability and outstanding formability of terne sheet ensures little damage to the coating and a minimal decrease in its corrosion resistance.

(3) Formability

The physical properties of fuel tank materials are required to provide high levels of elongation, flexibility (deformation resistance) and solid lubricity. Terne sheet demonstrates outstanding characteristics with its high levels of elongation, rollability and flexibility for press forming and other plastic processing.

(4) Weldability

Stable weldability is required for seam, spot, projection and other types of resistance welding, and it is necessary for suppressing any deterioration of the welding electrodes. The lead contained in terne sheet has the characteristic of resisting the formation of an alloy with the copper (chemical symbol Cu) of the electrode, thereby demonstrating satisfactory weldability.

(5) Solderability

Fuel tank materials are also required to have such characteristics as high mutual reactivity with the solder metal and low coating surface energy that inhibits the formation of an oxide film and results in good wettability. In addition, the materials are also required to inhibit bimetallic corrosion with the solder. The solder used with terne sheet is the same type of Pb-Sn material as that of the coating, resulting in satisfactory solderability and resistance to bimetallic corrosion.

The objective of this present development was to obtain levels of corrosion resistance, weldability, formability and other workability equal to or better than the current levels, as well as a level of processability equal to or better than current with existing production equipment.

With respect to corrosion resistance in particular, there is growing need for a relative increase in the corrosion life of the fuel tank to accompany the general increase in service life of automobiles in recent years due to the improved corrosion resistance of automobile bodies. Not only is this need growing in North America and other regions having a severe corrosion environment, but a similar need has recently become conspicuous in Japan due to the significant increase in the amount of road salt sprinkled on expressways and

major trunk roads in snowbound regions since the use of studded tires was prohibited.

In addition, there is increasing attention being focused on the inside of the fuel tank due to the problem of soured gasoline accompanying the expanding automobile demand in developing countries, and the use of this gasoline resulting in the need for a higher level of corrosion resistance.

3. Design of Metallic Coatings

3.1. Selection of Coating Metal

Two important elements of a highly corrosion-resistant coating are as follows:

- high chemical stability against soured gasoline on the inner surface of the fuel tank and against salt damage to the outer surface
- an anodic corrosion potential of the coating relative to iron (chemical symbol Fe); namely, demonstration of sacrificial corrosion prevention by the coating

The relationship between the corrosion potential relative to iron and the eluted volume in soured gasoline (containing 10% water) for the typical metals used for coating is shown in Fig. 3. Moreover, the relationship between the corrosion potential and eluted volume in sea water (as an estimate of the external corrosion resistance) is shown in Fig. 4.

Aluminum (chemical symbol Al) demonstrates the best performance among metals, this being followed by tin

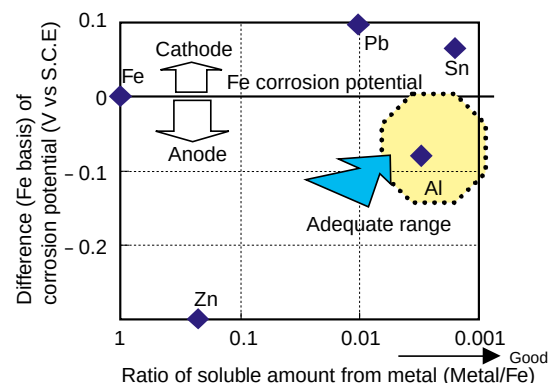


Fig.3 Comparison of the corrosion potential and dissolution in soured gasoline solution of various metals

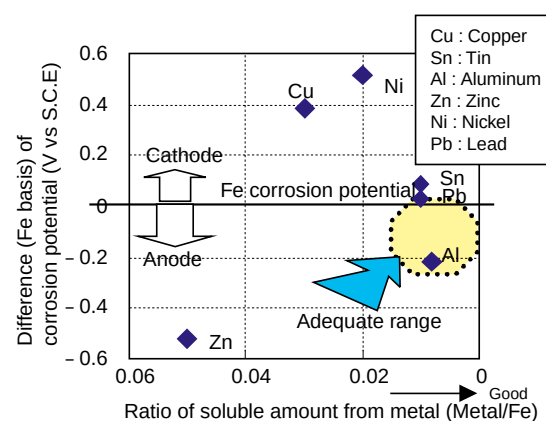


Fig.4 Comparison of the corrosion potential and dissolution in sea water of various metals

(chemical symbol Sn). However, aluminum was discounted because soldering is difficult due to the formation of a hard oxide film on the surface. Since the potential of tin is cathodic, it is necessary to make the potential more anodic in order to improve the corrosion resistance. Tin inherently has excellent soldering properties, and is likely to be highly suitable for fuel tank use due to its formability based on its outstanding flexibility and elongation.

Thus, in this study, we attempted to optimize the potential by using an anodic metal alloy based on tin.

Although examples of anodic metals that can be used for coating include magnesium (Mg), aluminum and zinc (Zn), it is difficult to obtain satisfactory coating processability with magnesium, and a high content is required to adjust the corrosion potential in the case of aluminum. Since problems with solderability also exist, zinc was selected and its anodization examined. In addition, hot dip coating was selected for the coating method due to its high processability and the potential use of existing coating equipment.

3.2. Alloy Design ⁽³⁾⁽⁴⁾

Fig. 5 shows the change in corrosion potential and metal eluted volume in soured gasoline versus the zinc content of the zinc-tin alloy.

The potential moves toward the anodic side as the amount of zinc is increased, becoming anodic at 4%. Since the potential stabilizes when the amount of zinc exceeds 6%, a zinc content of 7% or more was considered to provide the optimum potential.

On the other hand, although the eluted volume was low and remained stable until a zinc content of 10% was reached, it tended to increase rapidly with a zinc content of around 11%. This phenomenon can be interpreted from the two-element equilibrium diagram for Sn-Zn shown in Fig. 6 and from the eutectic states shown in Fig. 7.

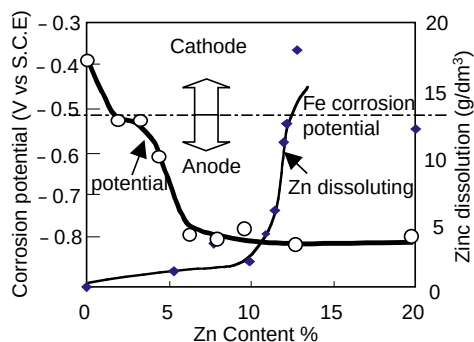


Fig.5 Relationship between Zn content and corrosion potential and Zn dissolution

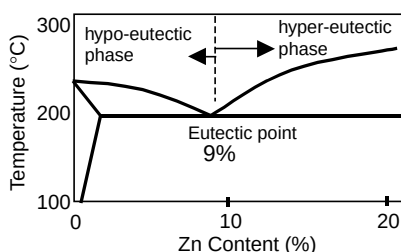


Fig.6 Equilibrium diagram of Sn-Zn alloy

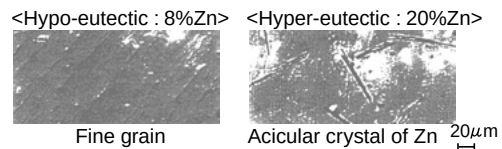


Fig.7 Appearance of Sn-Zn alloy crystal

The eutectic point for alloys of tin and zinc is at a zinc content of 9%. Coarse zinc crystals are formed in the hyper-eutectic range in excess of this zinc content, and it is assumed that the excess zinc dissolves to result in an increase in the eluted volume.

Thus, it is necessary for the zinc content to be less than the eutectic point of 9% in order to inhibit the formation of coarse zinc crystals.

Summary:

The optimum range for the zinc content is thus 7-9% based on the corrosion potential and the elution properties (to inhibit the formation of coarse zinc crystals). Coating was performed in molten Sn-Zn alloy containing a median value of 8% zinc, this being subsequently referred to as the Sn-Zn coating.

The Sn-Zn coating was enhanced by chemical conversion coating to further improve the corrosion resistance and adhesion of the coating.⁽⁵⁾

Fig. 8 shows a cross-section of this newly developed coated steel sheet.

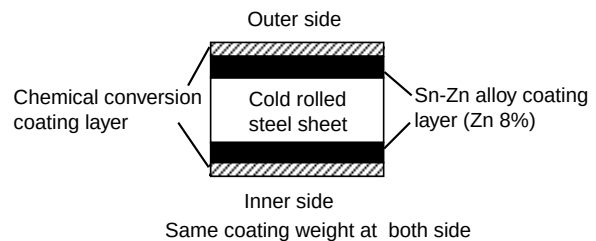


Fig.8 Schematic illustration of the Sn-Zn coated steel sheets for fuel tank applications

4. Major Technical Elements and Test Results

The following section describes the results of examining the major fuel tank technical elements of corrosion resistance, spot weldability, seam weldability and solderability.

4.1. Corrosion Resistance

4.1.1. External Corrosion Resistance

Photographs taken after a corrosion test on a test piece incorporating the variation in sheet thickness equivalent to that of a fuel tank formed by press working a coated steel sheet are shown in Fig. 9.

The Sn-Zn coating provided particularly good corrosion resistance at the drawn portions. This is the result of the good elongation of the coating and its having suitable sacrificial corrosion prevention properties. In the case of the current material, rust started to form where the coating was damaged due to the absence of any sacrificial corrosion prevention effect.

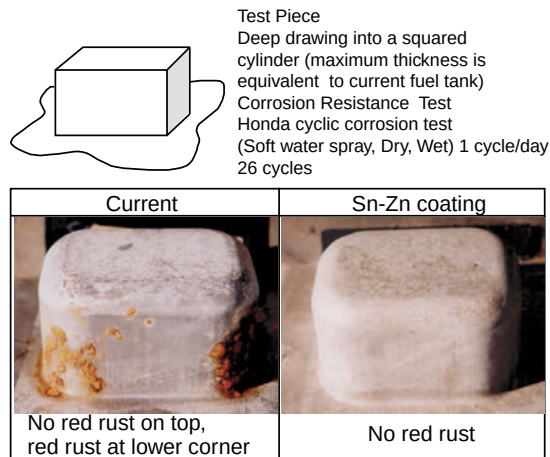


Fig.9 Appearance after corrosion test

4.1.2. Internal Corrosion Resistance

The results for corrosion resistance against soured gasoline are shown in Fig. 10 for a test piece worked in the same manner as that described in section 4.1.1 above.

In comparison with the current material, the Sn-Zn coating provided superior corrosion resistance.

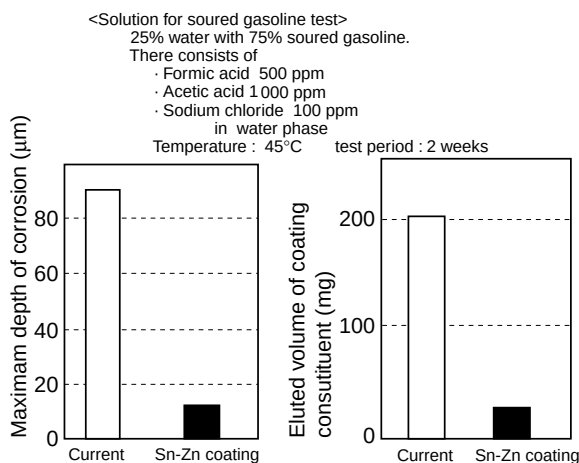


Fig.10 Result of corrosion resistance by soured gasoline test

4.2. Spot Welding

Fig. 11 shows the results of an electrode tip life test on the spot weldability of the current material and the Sn-Zn coated material, while Fig. 12 shows the states of the weld at the start and completion of an electrode tip life test of the Sn-Zn coated material.

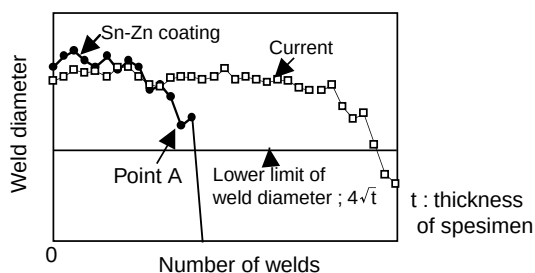


Fig.11 Changes of weld diameter during electrode tip life test

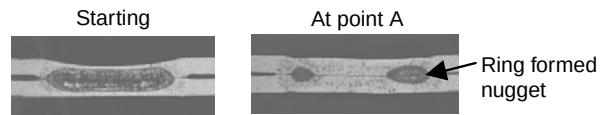


Fig.12 Cross-sectional view of nugget (Sn-Zn)

The Sn-Zn coated material demonstrated roughly half the electrode tip life of the current material, and at the end of the electrode tip life test (point A), the nugget exhibited an unstable ring shape.

An analysis of the cause was made, and the change in the shape of the electrode tip faces occurring in the test was impressed on a piece of pressure-sensitive paper. The results are shown in Fig. 13, while the appearance of the electrode after the test is shown in Fig. 14.

Number of welds	1	100	point A
Before modifying			
After modifying			

Fig.13 Change of electrode tip face shape during electrode tip life test

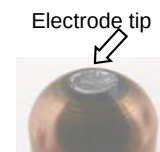


Fig.14 Appearance of electrode after welding

The electrode tip was worn down during the electrode tip life test, and the area of the contact surface can be seen to have become larger.

The results of an EPMA analysis of the electrode cross-section after the electrode tip life test are shown in Fig. 15.

An indentation had formed in the center of the electrode tip, and Cu-Sn or Cu-Sn-Fe alloy was deposited progressively deeper toward the center. The top of this deposit consisted of a coated metal or heterogeneous alloy layer, and is believed to have been easily peeled. It is thought that the electrode deteriorated in this manner, inhibiting the formation stability of the nugget and reducing the electrode tip life.

In contrast, the current material deposited a thin, uniform layer of alloy on the electrode and there was only slight deterioration as shown in Fig. 16. This is because a lead and copper alloy is difficult to form.

Summary: Causes of Decreased Electrode Tip Life

The cause of the deterioration in electrode tip life of the Sn-Zn coated material relative to the current material is that, during the course of the electrode tip life test, an alloy formation reaction occurred between the electrode and coating, causing a deterioration in the form of an indentation at the center of the electrode which expanded the tip and

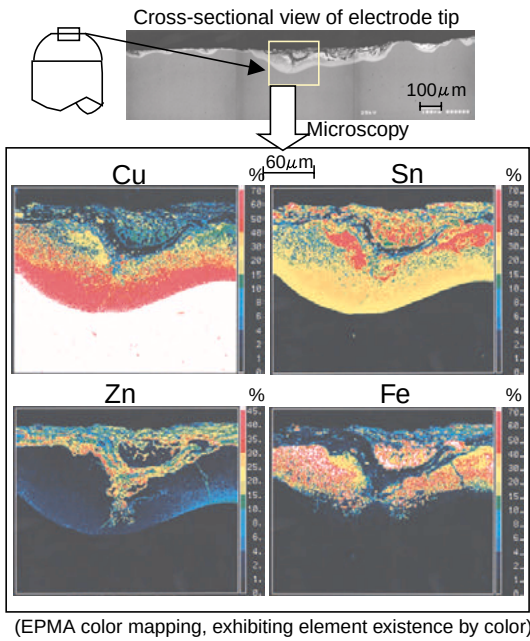


Fig.15 Metal distribution at cross-section of deposition electrode tip



Fig.16 Cross-sectional view of electrode tip used for welding the current material

resulted in an increase in the contact surface area of the electrode tip. This decreased the current density, which is thought to have caused insufficient nugget formation. A schematic diagram of this is shown in Fig. 17.

Thus, the corrective action required to improve the electrode tip life involves the three factors listed below.

- (1) Reducing the amount of wear (rate of progression) caused by alloy formation on the electrode by using a lower current, etc.
- (2) Reducing the coating weight to reduce the absolute amount of alloy formed.
- (3) Inhibiting the increase in contact surface area of the tip caused by wear through optimization of the electrode tip shape.

The results of our studies conducted based on the above viewpoint are described in section 4.2.1.

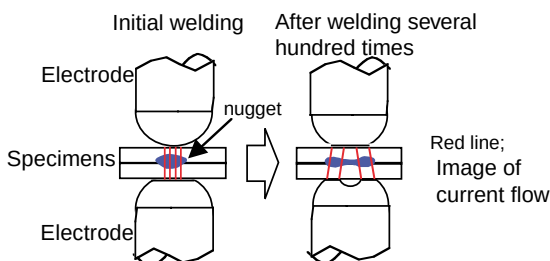


Fig.17 Model of degenerated electrode

4.2.1. Results of a Study to Improve the Spot Weldability

(1) Use of a Lower Current

One way of allowing a lower current to be used while still producing an acceptable nugget is to increase the value of resistance (R) shown in equation 1. This can be achieved by increasing the contact resistance between the steel sheets.

$$\text{Heat quantity equation: } Q (\text{cal}) = 0.24 \cdot I^2 \cdot R \cdot t \quad (1)$$

where I is the current (A), R is the resistance (Ω) and t is the time (s).

Two factors that govern the contact resistance are the coating weight (thickness) of the chemical conversion coating and the condition of that coated surface.

More specifically, increasing the coating weight and decreasing the surface roughness are presumed to contribute to increasing the resistance value. (Inhibiting the alloy formation reaction between the electrode and coating is also expected to contribute to this effect.)

An image of the condition of the surface is shown in Fig. 18, and the change in contact resistance when the amount of the chemical conversion coating and surface roughness were changed is shown in Fig. 19. In addition, the relationship with the contact resistance is shown in Fig. 20 by expressing the two variables of coating weight and surface roughness as the shield ratio (coating weight/surface roughness).

Although the contact resistance increases as the coating weight increases, if the surface roughness increases, this contribution is considerably decreased. Thus, it is effective to reduce the surface roughness. In addition, it was found that the contact resistance could be controlled by adjusting the shield ratio.

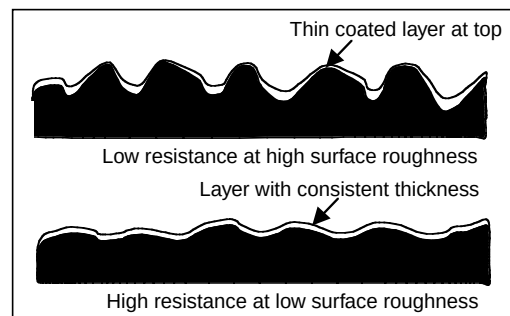


Fig.18 Image of cross-sectional view of coating

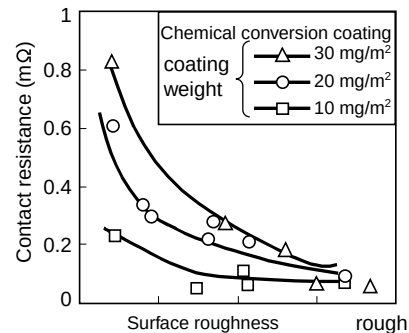


Fig.19 Relationship between surface roughness and coating weight and contact resistance

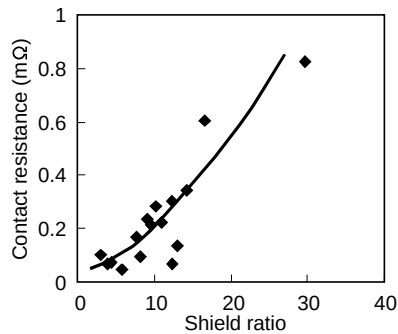


Fig.20 Relationship between shield ratio (coating weight/surface roughness) and contact resistance

The effect of contact resistance on the lower limit of the welding current over the weldable range was confirmed on the basis of the foregoing results. Fig. 21 shows the relationship between the contact resistance and the lower limit of the welding current, while the effect on improving electrode tip life is illustrated in Fig. 22.

Summary: Use of Lower Current

The contact surface condition can thus be optimized by controlling the surface roughness, while the electrode tip life can be improved by reducing the welding current.

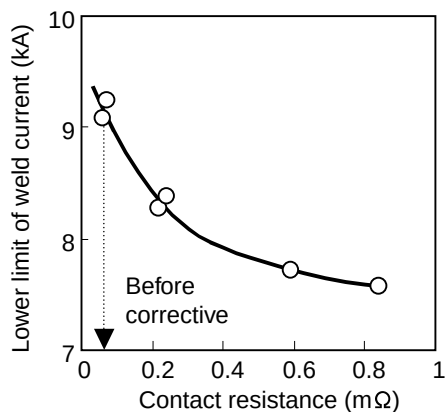


Fig.21 Relationship between contact resistance and lower limit weld current

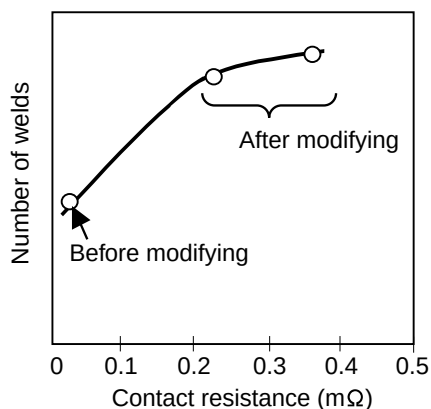


Fig.22 Relationship between contact resistance and electrode tip life

(2) Coating Weight

It is necessary to consider the effect on corrosion resistance when reducing the coating weight. The relationship between electrode tip life and corrosion resistance (the area over which red rust had formed after 72 hours of the salt spray test) according to the coating weight is shown in Fig. 23.

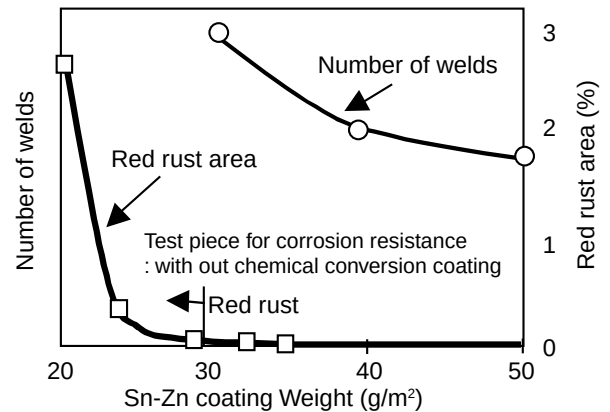


Fig.23 Relationship between Sn-Zn coating weight and electrode tip life and corrosion resistance

The thinner the coating, the better the electrode tip life. However, the lower limit was set at 30 g/m² to achieve a suitable balance with the corrosion resistance.

(3) Optimization of the Electrode Shape

Reducing the diameter of the electrode tip was most effective for suppressing the increase in contact surface area. Fig. 24 shows the relationships between the electrode tip diameter and the electrode tip life and lower limit of the welding current, while Fig. 25 shows on the change in contact surface area of the electrode tip.

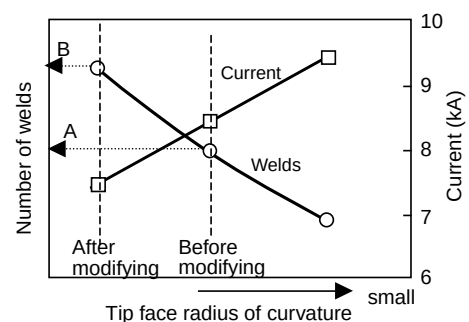


Fig.24 Relationship between shape of electrode and electrode tip life and lower limit of welding current

Number of weld	1	100	A	B
Before modifying				
After modifying				

Fig.25 Change of electrode face shape during electrode tip life test

Together with inhibiting the increase in contact surface area over the span of the electrode tip life, reducing the diameter of the electrode tip also improved the electrode tip life due to the synergistic effects resulting from a decrease in the initial contact surface area (see the first weld in Fig. 25), the lower welding current, and the reduced electrode deterioration. However, since reducing the diameter of the electrode tip also results in such a problem as small nugget diameters, the tip diameter was ultimately set to maintain the optimum balance.

Summary: Improvement of Spot Weldability

Spot weldability could therefore be significantly improved by optimizing the surface contact resistance, coating weight and shape of the electrode, to attain a performance equivalent to the current level.

4.3. Seam Welding

Although there was expected to be a decrease in weldability due to alloy formation, since copper was used for the electrode in the same manner as that for spot welding, weldability equal to the current level was obtained. Fig. 26 shows a photograph of the test, while Fig. 27 shows the state of the nugget during the electrode tip life test.

The quality of the nugget was favorable even after 2000 m of welding, indicating long electrode tip life.

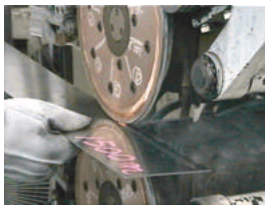


Fig.26 Seam welding test

Distance of consecutive welding		Cross direction	Longitudinal direction
Sn-Zn coating	100 m		
	500 m		
	2000 m		
Current Over 1000 m (mass production)			

Fig.27 Cross-sectional view of nugget during electrode tip life test

Fig. 28 shows the appearance of the electrode after 10 m of welding, while Fig. 29 shows the state of metal distribution at the electrode tip as determined by an EPMA analysis during electrode tip life tests.

An alloy consisting mainly of tin was formed as blisters on the surface (arrow “a” in Fig. 29) and an alloy consisting mainly of iron was formed near the electrode interface. Both

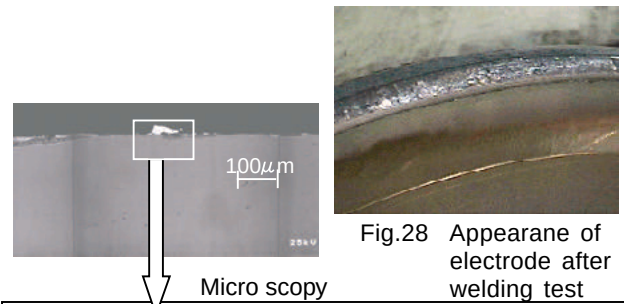


Fig.28 Appearance of electrode after welding test

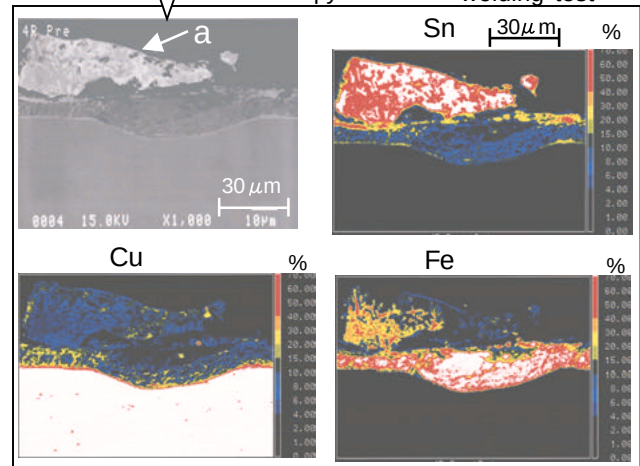


Fig.29 Metal distribution in cross-sectional view of electrode tip (EPMA color mapping, exhibiting element existence by color)

of these are components of the coated steel sheet material, and the phenomenon of alloy formation with copper, as occurred on the spot welding electrode, was only observed to an extremely small degree. In addition, intermittent peeling and separation of adhered substances from the electrode were apparent in the electrode tip life test. The appearance and components of the substances that peeled off are shown in Fig. 30.

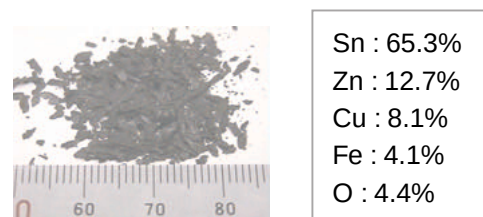


Fig.30 Appearance of peeled sample and its composition

The substances that peeled off consisted primarily of coating components and a small amount of copper (electrode material), indicating that the coating repeatedly accumulated and then separated from the electrode.

On the basis of these findings, it is believed that electrode deterioration was inhibited as a result of self-dressing by which the coating components peeled off the electrode before an alloy could be formed between the electrode and coating, thereby resulting in satisfactory electrode tip life.

The mechanism by which coating components peeled off the electrode is illustrated in Fig. 32.

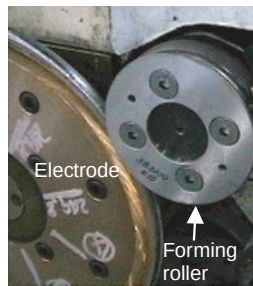


Fig.31 Layout of forming roller

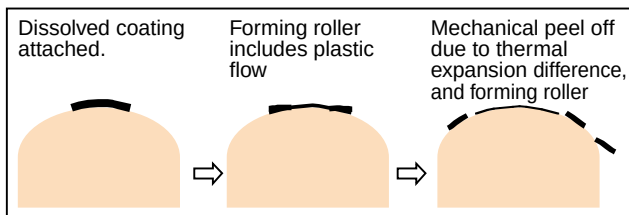


Fig.32 Behaviour of by-products on electrode tip

- (1) Effects appear in which the forming roller, a component part of the welding machine (the position is shown in Fig. 31), physically forms the tip of the electrode while ejecting the adhered coating.
- (2) The adhered coating is induced to plastic flow by the forming roller due to its low melting point and low deformation resistance.

4.4. Soldering

4.4.1. Selection of the Solder Material

When selecting a solder, in addition to its spreadability, wettability and physical properties, the solder should satisfy the following requirements for adequate corrosion resistance:

- (1) A low degree of potential difference corrosion caused by bimetallic contact between the solder material and coating (steel sheet and soldered tube)
- (2) A low soldering temperature, namely a low melting point for the solder material, and a resulting resistance to deterioration of the Sn-Zn coating of the steel sheet and tube coating caused by heat (melting and oxide formation).

The corrosion potential series of typical metals used for soldering and brazing along with their melting points are shown in Table 1.

An Sn-Zn alloy also consisting primarily of the Sn was the most favorable solder material based on its low melting point and position in the corrosion potential series. This

Table 1 Typical solder, metal and corrosion potential series and melting point

Corrosion potential series in sea water		Melting point (°C)
Zn-Al	Anode ↑ ↓ Cathode	390
Sn-Zn		200
Pb-Sn (current)		191-215
Sn-Ag (Ag3.5%)		221
Cu-Zn		850-900
Cu	Cathode	1 100
Ag-Cu		650-750

was followed by Sn-Ag (containing 3.5% Ag), the potential candidates being limited to these two types.

In addition, an Sn-Zn coating of the same type as that used for the steel sheets (electroplated, with a Zn content of 30-45%) was used for the tube coating.

The wettability performance was next evaluated by a soldering simulation test with the high frequency induction heating method as shown in Fig. 33. A photograph of the resulting cross-sections is shown in Fig. 34.

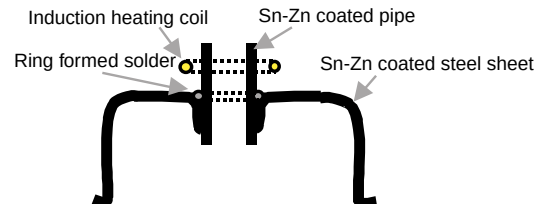


Fig.33 Solderability evaluation model (Induction heating method)

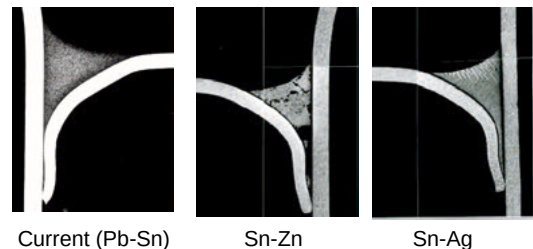


Fig.34 Cross-sectional microstructure of soldering

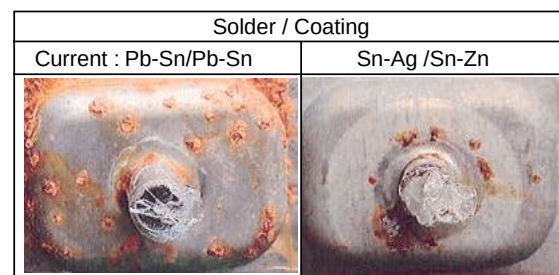
Although Sn-Ag exhibited favorable wettability comparable to the current level, adequate wettability could not be obtained with Sn-Zn, even though it was heated to 150°C above its melting point.

This is presumed to have been due to oxidation of the Zn component, resulting in entrapment of oxide and increased viscosity, which in turn increased the surface tension and decreased the wettability.

A corrosion resistance test was next performed after soldering, a photograph of the results obtained after the test being shown in Fig. 35.

The combination of Sn-Ag solder and Sn-Zn coating was favorable in that it reduced corrosion around the solder to a greater extent than the current soldering method.

Tests were also conducted on other parameters including



Corrosion test : CCT 15 cycles

Fig.35 Appearance after corrosion test on soldered area

the adhesion strength and creep characteristics. As a result, the Sn-Ag solder was confirmed to be equal to or better than current solder materials.

On the basis of these results, it was decided to use an Sn-Ag alloy (containing 3.5% Ag) as the solder material.

5. Evaluation of Suitability as a Fuel Tank Material

A prototype fuel tank was produced using a mass-production line and equipment, and the quality and performance of this prototype tank were evaluated in order to identify its suitability for use as a fuel tank. The results of the technical evaluation consisting of formability, spot weldability and corrosion resistance are indicated.

5.1. Formability

The sheet thickness at individual points was measured and a comparison was made with current materials. Sheet thickness data are shown in Fig. 36 as standard deviations.

Since the sheet thickness distribution of the Sn-Zn coating is equivalent to that of current materials, it was determined to provide satisfactory formability.

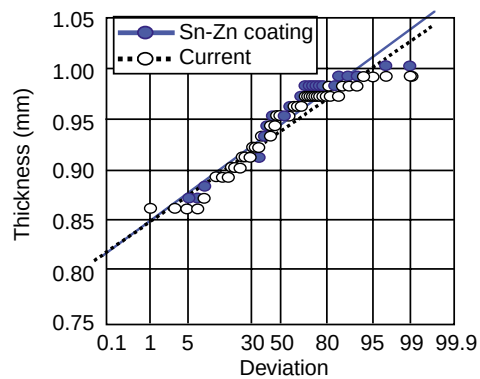


Fig.36 Sheet thickness distribution on formed work

5.2. Weldability

Both the welding quality and welding properties were favorable. Fig. 37 shows the change in the number of welds and the mean value of the weld diameter at spot-welded positions in the electrode tip life test. A photograph of the cross-section of the weld nugget is shown in Fig. 38.

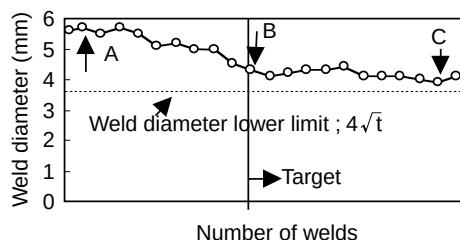


Fig.37 Changes of weld diameter during electrode tip life test



Fig.38 Cross sectional view of nugget

Electrode tip life was determined to be more than twice the target value with the developed material and welding specifications, while the quality of the nugget and so forth was also favorable.

5.3. External Corrosion Resistance

Fig. 39 shows the appearance of a painted fuel tank after the corrosion resistance test.

The Sn-Zn coating demonstrated good coatability, and provided outstanding corrosion resistance in comparison with that of current materials. More than twice the number of test cycles as that of current materials could be performed before red rust was formed in the deep-drawn portions.

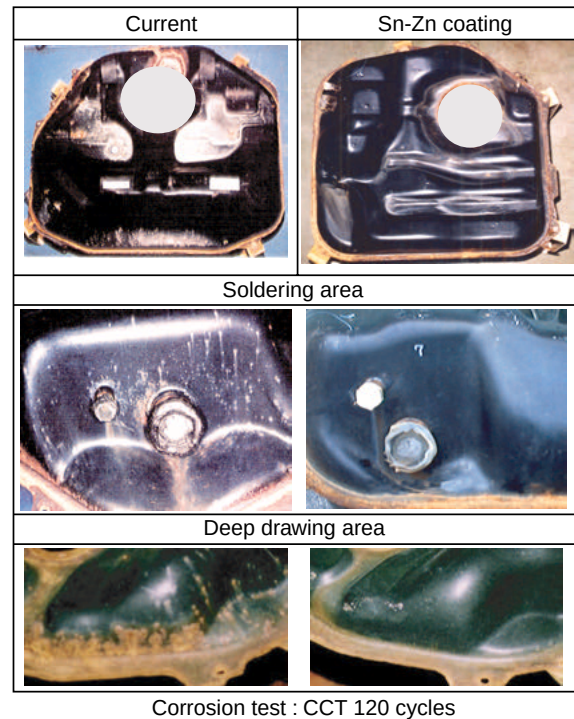


Fig.39 Appearance of fuel tank after corrosion test

5.4. Internal Corrosion Resistance

A circulation test was conducted on an Sn-Zn coated fuel tank with a soured gasoline test solution containing organic acids and salt water. The appearance of the fuel tank after the test is shown in Fig. 40, while the volume of metal eluted into the solution, the total weight of precipitated products and the maximum corrosion depth at each site are shown in Fig. 41.

In comparison with the current material, the Sn-Zn coating demonstrated lower levels of corrosion weight and

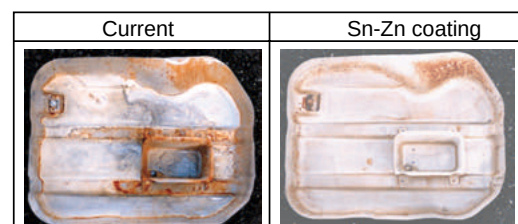


Fig.40 Appearance of fuel tank after soured gasoline test

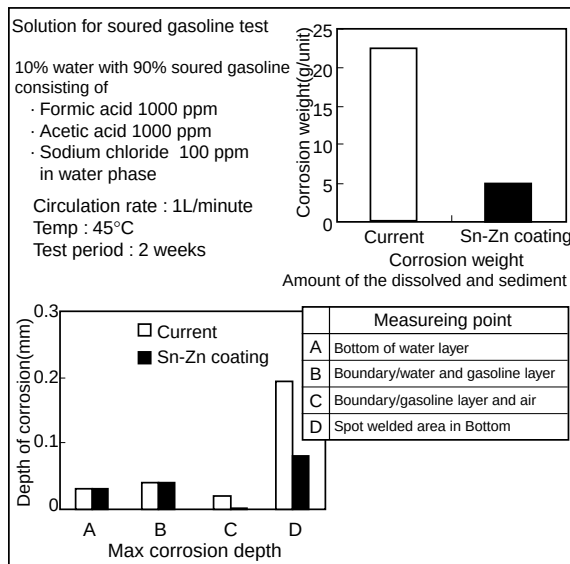


Fig.41 Result of soured gasoline test

corrosion depth, and thus was confirmed to have outstanding corrosion resistance.

6. Conclusions

This paper has presented conclusions of material and production technology development for a lead-free fuel tank, along with the results of evaluating its suitability. A summary of the findings obtained through this research is provided below.

- (1) A hot-dip Sn-Zn alloy coating was developed for coated steel sheet containing 8% Zn which demonstrated superior corrosion resistance, workability, processability and so forth. In addition, an Sn-Ag alloy containing 3.5% Ag was developed for use as solder material on the basis of the melting point, corrosion potential difference with the coating and so on.
- (2) With respect to the issue of spot weldability caused by the formation of an alloy between the copper electrode and the Sn-Zn coating, processability equal to or better than the current level could be obtained by controlling the surface contact resistance. This was done by improving the surface roughness of the coated steel sheet and by optimizing the shape of the electrode tip.
- (3) An Sn-Zn coating was confirmed to be equal to or better than current coatings in various suitability tests for use as a fuel tank, including tests of formability, welding quality, corrosion resistance and processability. The coating was determined to exhibit particularly excellent corrosion resistance.

This lead-free fuel tank is scheduled to be sequentially applied in automobiles beginning in 2001 models, with the goal that all fuel tanks be lead-free by the end of 2003.

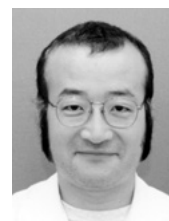
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Development of a Novel Bumper Adhesion Promoter

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ABSTRACT

A new type of adhesion promoter for bumper painting was developed to eliminate conventional primer spray application. The importance of this new material allowed for reduced cost and improvement in process efficiency. The new adhesion promoter (HX) is a waterborne emulsion incorporating chlorinated polyolefin, a key enabler of adhesion of base coat to thermoplastic olefin. As a consequence of the material design, the adhesion promoter can be applied as a final rinse, resulting in the primer spray application to become unnecessary. Furthermore, the primer spray booth became available for additional topcoat application, increasing production volumes. Overall, this material has proven to meet rigorous testing, and has been successfully implemented in mass production.

1. Introduction

In the past five years, there has been a large move towards using thermoplastic olefins (TPO) substrates by OEMs for bumper fascias. The drive towards lower cost, reduced weight, and continuing environmental consciousness has been paralleled by coatings materials development.

In general TPO, having low surface energy, is a difficult substrate to paint while meeting severe exterior specifications. Earlier systems used trichloroethane vapor (TCE) to clean and swell the substrate in order to accept paint materials. Due to environmental regulations and the hazardous nature of TCE, solvent washing has virtually been eliminated from Japan, North America, and Europe. Since the elimination of the solvent cleaning, various methods have been used to overcome the nonpolar substrate characteristics. Some examples of such methods include, an aqueous degreasing followed by an adhesion promoter, flame treatment, and plasma treatment. However, the overall output of the manufacturer, and the environmental restrictions usually dictate the type of TPO surface activation process.

In North America, the conventional approach to painting TPO bumpers has been aqueous degreasing and then applying a solvent borne primer; followed by base coat, and clear coat. The cure can vary depending on the substrate materials. However, the typical cure is over 100 degrees Celsius. In general, OEM bumper paint process has not changed drastically for several years.

Primers have been the source of attack due to the typical

high volatile organic compound (VOC) content, and the lack of intrinsic value to the customer. Consequently, the energy to abate VOC and to apply the paint is a burden that no OEM desires. Currently, waterborne primers are available, and have been used successfully. These waterborne systems still require an application step that is energy consuming. OEMs, paint, and substrate suppliers have researched various methods to change primer application.

Although there has been a desire to eliminate primer, the majority of the research has been focused on making base coats directly adhere to TPO. Unfortunately, this approach has two main concerns. First, each base coat color development would potentially be affected to enable direct adhesion to the substrate; meaning less freedom in color selection. This becomes a very difficult task when considering the number of colors and the frequent color and model changes. Second, TPO is naturally nonconductive. This would reduce base coat application transfer efficiency, the most costly item. Most attempts to eliminate primer spray application have been ineffective or unreasonable.

2. Function of Primer

Primer has three functions: to promote adhesion, to assist in appearance, and if necessary, to promote conductivity for high volume situations.

In general, TPO has a low surface energy, preventing paint from adhering properly. Both additives in the substrate and primer allow for chemical and mechanical bonds to occur. The primer contains chlorinated polyolefins that have

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both functions as a long chain polymer to entangle itself with the TPO, and chemically reactive sites for chemical bonds.

Appearance of the bumper is generally noted after the application of the topcoat. The appearance can be defined in two different ways: color and smoothness. Although the topcoat has a major role in the final appearance, smoothness and color below the base coat may influence the outcome in mass production.

Since TPO is naturally nonconductive, primer can add function as a conductive base. Therefore, the base coat and clear coat transfer efficiency increases by means of electrostatic application; optimizing the materials cost. However, this may be a function of the operating parameters. Therefore the spray methods and the operator efficiency plays a critical role. In general conductivity and transfer efficiency is dependent on the various manufacturing demands specific to each facility.

3. Development Approach

Adhesion promoters for TPO have three key resins: Chlorinated polyolefin (CPO), Polyurethane (PUR), and Acrylics. Most importantly, CPO penetrates into the TPO to achieve the necessary adhesion. The other two key resins, Polyurethane and Acrylics have film building and reinforcing function. Figure 1 illustrates the resin system concept. Solvents carry these resins. For this adhesion promoter, HX, mostly water is used and a small amount of other solvents. In addition, some additives are necessary for smooth mass production, such as a defoamer. The HX formulation breakdown is clarified in Figure 2.

The main function of HX is to promote adhesion of base coat to substrate. By testing a variety of resin combinations, a window was developed for the resin ratios. This window is quantified in Figure 3. In addition to the resin balance, the total resin content was formulated for optimal physical

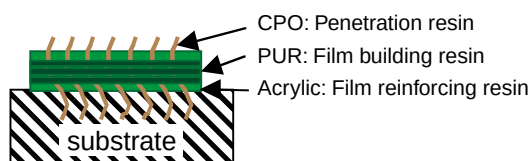


Fig.1 Schematic of resin function

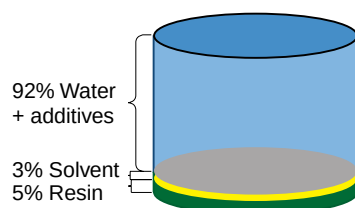


Fig.2 HX formulation

properties and appearance; demonstrated in Table 4. Tests determined that the Acrylic resin should be around 10% of the resin content, and the remaining should be the CPO and PUR for best gasahol resistance, and good adhesion. Additional testing showed that higher resin content resulted in better adhesion. However, the higher solids level changed the wetting properties of the materials, leading to sagging appearance failures, shown in appearance column of Table 5.

Mass production trials further tuned the materials to meet optimal appearance. Evaluation of the surface after HX application determined the roughness should be less than 1.0 μ m. The trials and microscopic investigations, in Table 5, resulted in the final HX resin concentration to be 5 weight percent.

At the 5% HX solids concentration, the physical properties were acceptable on current regular modulus TPO, "A", and on high modulus TPO, "B", with a functional group additive. This functional group is an additive to assist in paint adhesion. Accordingly, the high modulus TPO without the functional group additive, substrate "C", showed some

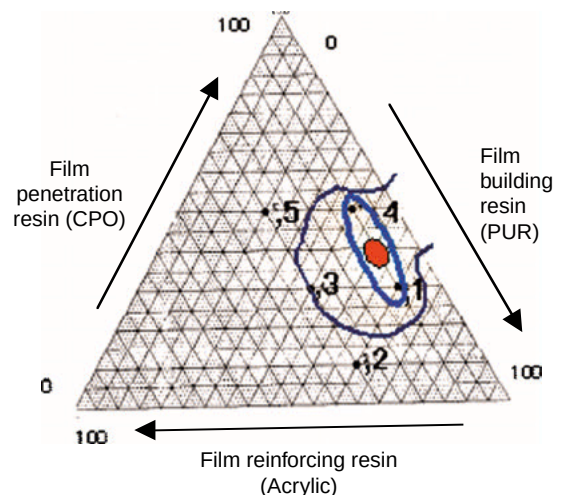


Fig.3 Resin ratio investigation

Table 4 Confirmation of various resin ratios

Formulation	Resin	Resin wt%	Initial adhesion on 90°	Initial adhesion 45°	Hot water 80°Cx3hr	Gasahol (E10)
1	CPO Acrylic PUR	30 10 60	100/100	25/25	100/100	120min
2	CPO Acrylic PUR	10 30 60	100/100	25/25	100/100	21min
3	CPO Acrylic PUR	30 30 40	100/100	25/25	100/100	55min
4	CPO Acrylic PUR	50 10 40	100/100	25/25	100/100	120min
5	CPO Acrylic PUR	50 30 20	83/100	25/25	98/100	45min

adhesion weakness. Table 6 shows the comparative results and substrate “C” with failures in high pressure washing and gasohol tests. Therefore, for future model developments and high modulus tool design, the high modulus TPO with the functional group additive is recommended. This functional group additive is currently used, and is adaptable to high modulus TPO without changing the physical properties drastically.

In addition to physical properties, accelerated weathering results were positive. For this research, typically incorporated UV absorbers, and hindered amine light stabilizers were removed from the topcoat formulations, to instigate degradation of the paint system. Without the UV package, the HX layer had no protection against weathering. However, as Figure 7 shows, there was no significant degradation of appearance, or adhesion. The concluding

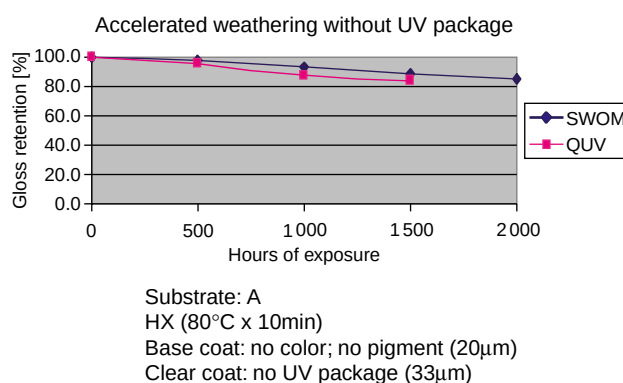


Fig.7 Weathering

Table 5 HX Solids versus appearance

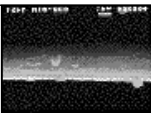
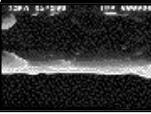
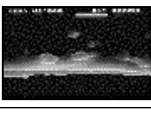
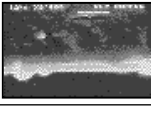
%NV HX	Microscope photo	Film thickness	Appearance		Adhesion		Overall
			dry HX only	HX + topcoat	100/100 90°cut	25/25 45°cut	
2%	cannot measure	—	OK	OK	OK	OK	OK; meets expectations
4%		0.20µm	OK	OK	OK	OK	OK; meets expectations
6%		0.40µm	Acceptable	OK	OK	Very Good	OK; meets expectations
8%		0.75µm	Not good	Not good	OK	Very Good	Not good; not acceptable
10%		1.25µm	Not good	Not good	—	—	—

Table 6 Physical properties confirmation

Test materials			Adhesion		High pressure wash (n=2)	Water soak		Humidity		Gasahol (min)
Substrate	Primer	Topcoat	90°	45°		Appearance	Adhesion 90°	Appearance	Adhesion 90°	
Specification			100/100	>20/25	0mm	>8VS	100/100	>8VS	100/100	>30min
Current substrate	A	5% HX Beige metallic	100/100	25/25	0mm	8VS	100/100	8VS	100/100	60min+
High modulus with functional group	B	5% HX Beige metallic	100/100	25/25	0mm	8VS	100/100	8VS	100/100	60min+
High modulus no functional group	C	5% HX Beige metallic	100/100	13/25	12.5mm	8VS	100/100	8VS	100/100	8min

formula was set to meet physical properties as well as show a smooth appearance, and good workability. Although UV absorbers, and hindered amine light stabilizers were removed for testing, they are critical additives for the overall protection from weathering of color and durability of the paint. This accelerated testing showed that the HX material would not adversely be affected by UV penetration in a worse case scenario.

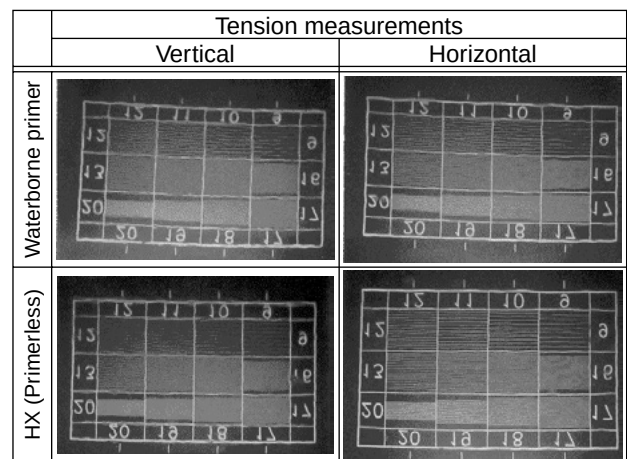
4. Test Results

Both bumpers and test plaques were evaluated according to the Bumper Coating Specifications, on current bumper substrate (A) and high modulus with functional group (B) met current levels of chemical, physical, and weathering properties.

HX was developed to meet rigorous exterior physical properties. However, appearance, could only be measured accurately on line under mass production conditions. Appearance requirements were set to be equivalent to the original primer system. Base coat hiding power and color were not affected by the lack of color by HX. All colors on the current pallet were tested by filmbuild wedge method, where a higher amount of paint is applied to one end compared to the other to show a gradient in film thickness. Figure 8 illustrates the certain film thickness of the paint will hide the substrate or black and white indicator below. The definition each hiding point is dependent on the color of the base coat, and the color of the substrate below. The “process hiding”, “black and white hiding”, and the “line film thickness” were defined for a variety of colors. In most cases the current base coat application was sufficient to hide the substrate color. Therefore, matching the color of body

to bumper and base coat material usage was not a concern.

Another appearance related issue was the leveling ability and smoothness of the bumper surface after topcoat. Under mass production environment, the HX bumpers showed an equal or improved appearance according to tension measurements. In Figure 9, the tensionmeter photographs determine the DOI by the degree of clarity of the lines photographed. Experts judged visual appearance, and all agreed the appearance meets or exceeds current mass production bumpers.



Judgement Criteria

Clarity indicated DOI in these photos. Non continuous lines indicate the DOI rating

Fig.9 Appearance-tensionmeter measurements (D.O.I.)

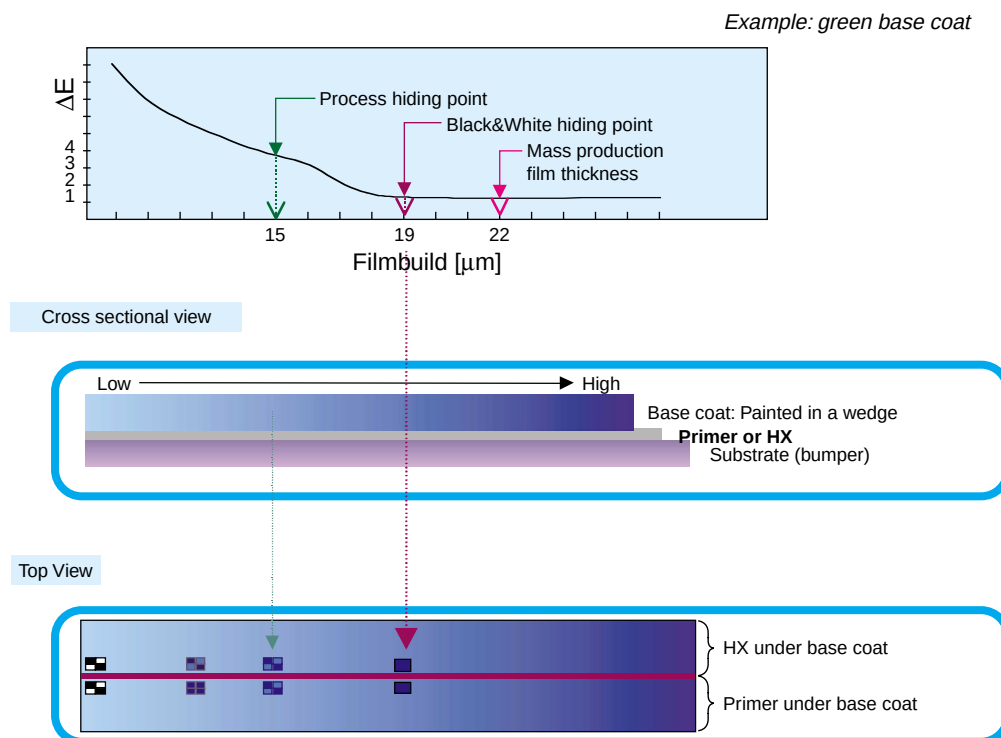


Fig.8 Appearance-hiding

5. Application Process

The material properties and the design of the formulation allow HX to be applied similar to a final rinse. These attributes include the viscosity, and solids content. In conventional North American mass production processes bumpers are cleaned by aqueous power wash. As a final stage of pretreatment before topcoat, a surface conditioner is applied to aid in the uniform removal of water from the substrate. It is in this stage that HX can be applied instead of surface conditioner. The HX solids concentration is such that the material wets evenly. Also, the solution in HX is mostly water; greater than 90%. Therefore, the liquid properties also behave similarly.

Another benefit to the HX material is the ability to be recycled. Material not deposited on the surface of the bumper is recollected and reapplied. In addition, the degradation of the material is monitored and controlled. This effectively keeps waste and cost low.

After the HX is poured throughout the part, it is cured at 80 °C for 10 minutes. The heat allows the material to dry and function as an adhesion promoter for the base coat. Furthermore, this cure fits into current hot air blow after pretreatment to dry off the bumper prior to paint. No additional equipment is necessary to cure the HX.

Conceptually, as Figure 10 shows this process is simple. HX adhesion promoter being applied within pretreatment allows for current primer areas to be converted into more valuable topcoat areas. The cost of rerouting the conveyors and added application equipment is far less than a new line build.

6. Conclusion

HX adhesion promoter was shown to have excellent physical properties. The primary development was around the resin content and concentration. The developed material meets current requirements and performance of mass production bumpers for physical properties. Controlling the resins enabled appearance to be optimized to meet targets.

Currently, the mass production process has achieved the goals set forth. A dramatic increase in production capacity, and decreased unit costs, has proven this materials development a success. In addition unit emissions of VOC were reduced due to the elimination of the primer spray. In conclusion, Honda strives to be the industry leader of efficient and environmentally conscious manufacturing and paint processes.

7. Acknowledgements

The HX system would not have been possible without the contributions by Honda of America Manufacturing Technical Planning, Honda de Mexico, and Nippon Bee Research of America.

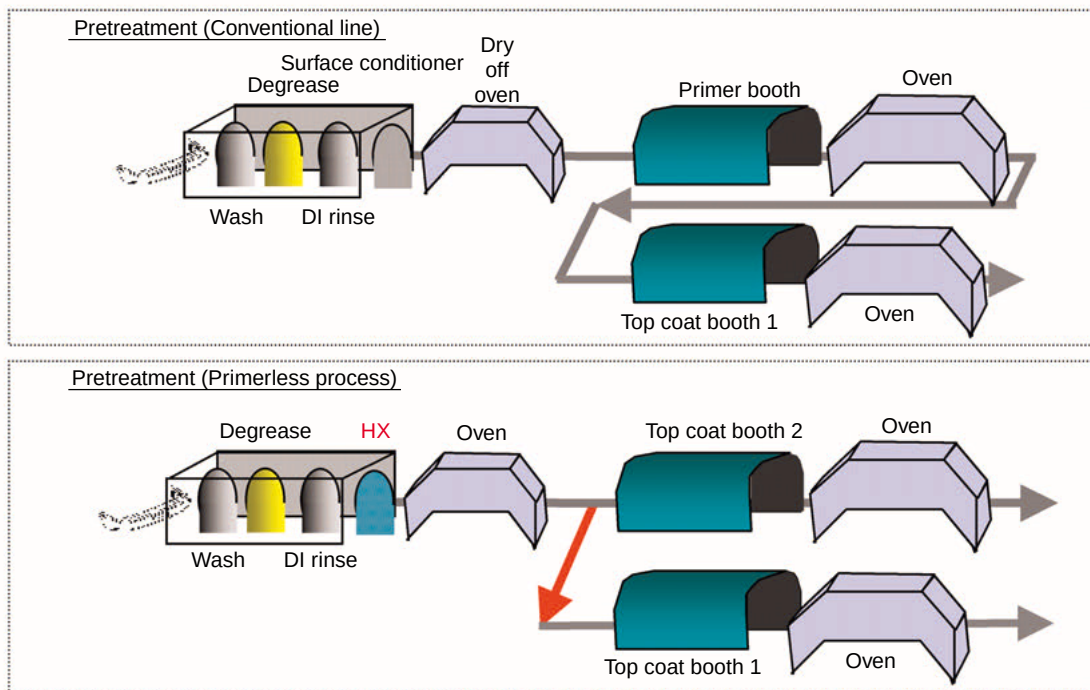
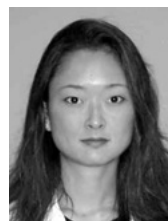
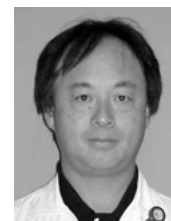


Fig.10 Process Comparison

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Development of Weight Reduction Technology for Door Trim using Foamed PP

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ABSTRACT

To reduce vehicle weight, a foaming PP resin was developed as a substrate for the injection molding of door trim. This material was formed by mixing a PP resin that acts as a base with a master batch of foaming agent. The PP resin used as the base is low viscosity to reduce the amount of the injected resin. Non-filled PP resin is also used to lower the specific gravity. Impact-resistance is improved by adding an elastomer. The foaming agent was selected after considering such factors as the uniformity of foam cells, effects on pollution, and the associated work environment. The ratio of foaming agent was selected for a balance between product stiffness and low-temperature impact-resistance. The result is a weight reduction of approximately 25% compared to conventional door-trim substrate material.

1. Introduction

Weight reduction of automobiles is important for improving fuel consumption and addressing environmental issues. Weight reduction of resin components consisting primarily of polypropylene (PP) plays a significant role in this.

The base materials used for automobile interiors is normally PP, and the specific gravity frequently exceeds 1.0 due to the addition of fillers to improve the stiffness, heat resistance and dimensional stability of the base material. Studies have been conducted on reducing vehicle weight by using filled PP to improve the stiffness by increasing the degree of crystallization. Such increased stiffness makes it possible to reduce weight by reducing the thickness of the base material. However, since stiffness is typically proportional to the cube of the sheet thickness, there is the concern of overall stiffness of the base material decreasing as the sheet becomes thinner. When a solution is provided such as incorporating ribs on the back of the product to compensate for this, it becomes difficult to meet the objectives of weight reduction.

By introduction of this new foaming technique, a technology has been developed to obtain greater weight reduction than was possible in the past, while ensuring equivalent levels of stiffness and low-temperature impact resistance (face impact resistance) to those of conventional door trim base materials (filled PP). This paper provides a description of the technology.

2. Development Aim and Concept

2.1. Development Aim

In the case of conventional weight reduction by high crystallization, the amount of filler was increased and the thickness was reduced. As a result, the specific gravity was also increased and the weight reduction effect was 10-15% at the most. The goal of this newly developed foaming technology was to achieve a weight reduction of 20-30%, equivalent to twice the current level of weight reduction, by reducing the thickness due to the effect of high crystallization.

2.2. Weight Reduction Concept

The amount of resin injected must be reduced in order to generate a weight reduction effect. In order to accomplish this, it was decided to pour in resin while using an initial sheet thickness of 2.0 mm instead of the normal 2.5 mm. Although the stiffness of the base material would be decreased as a result of reducing this sheet thickness, it was considered that this loss of stiffness could be compensated for by effectively increasing the sheet thickness through foaming.

Foaming injection molding is performed by opening the mold several millimeters to increase the sheet thickness after the material has been injected. The injection molding process is shown in Fig. 1. Immediately after injecting resin into a mold providing a typical sheet thickness of 2.0 mm, the mold is opened to 4.0 mm to perform foaming to twice the volume

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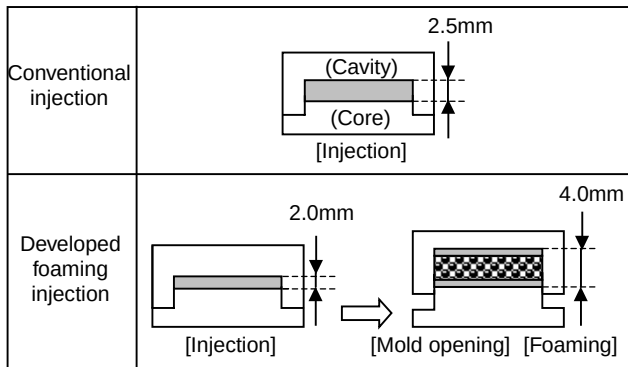


Fig.1 Molding process

of the cavity. Increasing the sheet thickness also increases stiffness. The amount of resin injected could be reduced by 20% as a result of this reduction in the initial sheet thickness.

In order to achieve further weight reduction, a non-filled PP resin was selected. In contrast to the 0.99 specific gravity of the filled PP, the specific gravity of the non-filled PP resin was 0.9. Thus, a weight reduction of approximately 10% could be achieved by applying the material with this lower specific gravity. Thus, theoretically, a total weight reduction of approximately 30% should be possible by applying the measures just described.

2.3. Concept for Improving Stiffness

Fig. 2 shows the relationship between the weight (per square meter) and maximum flex load of non-filled and filled PP resin. The maximum flex load and the slope of the flex modulus incline of three-point bending of a 50-mm test piece were compared to evaluate the product stiffness. The goal was a maximum flex load of 110 N/50 mm for the filled PP having a thickness of 2.5 mm that is used as a conventional door trim base material.

Thickness (mm)	2.0	2.5	3.0
Filled	△	▲	◆
Non-Filled	○	●	

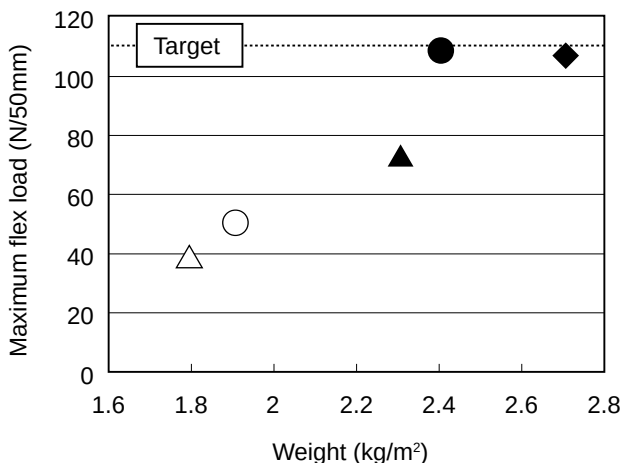


Fig.2 Relationship between weight and maximum flex load

Since the sheet thickness needs to be 3 mm in order to provide a stiffness equal to that of a solid non-filled PP product, the weight ends up increasing. The goal was to obtain a maximum flex load of 110 N/50 mm with an amount of resin equivalent to a sheet thickness of 2.0 mm by using foamed PP resin.

2.4. Concept for Impact Resistance

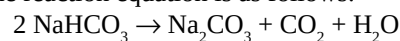
Low-temperature impact resistance (face impact resistance) is one of the requirements of a door trim base material. A base material is considered to be satisfactory if it does not split or crack when subjected to an impact test at -30°C. However, since this test is conducted at the finished product stage, testing was conducted on a conventional door trim base material by using the Dupont impact tester (face impact test) with the finished product as the test piece. The results of the Dupont impact test for filled PP having a thickness of 2.5 mm were 5J at room temperature and 1J at -30°C. These figures were accordingly set as the targets for impact resistance.

3. Material Specifications

3.1. Selection of Material Specifications

3.1.1. Foaming Agent

A foaming agent with a high expansion ratio, uniformly foaming cells and which does not generate noxious gases is ideal. Current foaming agents are classified as either endothermic or exothermic. A typical example of the former agent is sodium bicarbonate which generates sodium carbonate and carbon dioxide gas in an endothermic reaction. The reaction equation is as follows:⁽¹⁾



The expansion ratio of the endothermic type is just under 2 times, and although the foaming strength is slightly inadequate, the foam cells are comparatively uniform.

A typical example of the latter exothermic type is azodicarbon amide (ADCA). Since ammonia gas is produced in an exothermic reaction, there are concerns about such issues as mold corrosion and a poor working environment. In addition, though this type of agent achieves an expansion ratio of 2 times or more, its foam cells tend not to be uniform. The endothermic type was selected as a result of a comprehensive assessment of the available foaming agents.

The PP resin used as the base material and the foaming agent are separately prepared in the form of master batches so that they can be easily handled at the injection molding plant. This is because mixing the two in advance causes a thermal cracking reaction resulting in foaming.

The sheet thickness of the foaming injection molded product was approximately 4 mm, which was obtained by setting the expansion ratio at 2 times, this being the foaming limit of the endothermic type, and the initial sheet thickness was 2 mm to achieve the desired weight reduction. The resulting specific gravity was approximately 0.6.

3.1.2. Specifications of the PP Resin for Injection Molding

The mold clearance is minimized to reduce the amount of resin injected. Thus, if the fluidity of the resin material is

too low, it starts to solidify before it fills the mold resulting in the occurrence of a short shot. It was therefore necessary to increase the fluidity.

Necessary fluidity was determined by using a prototype door mold. Fluidity of resins conventionally used in injection molding exhibit a melt flow rate of approximately 20 g/10 min. These resins easily resulted in a short shot, and thus it was difficult to obtain a satisfactory product. A satisfactory product could be obtained with a resin having a fluidity of 70 g/10 min, so this was used as the basic specification.

There were concerns about insufficient stiffness due to the use of a non-filled PP resin and degraded impact resistance due to foaming. A PP resin was selected which achieved a balance between these two factors. The results are shown in Fig. 3. The relationship between the stiffness and impact resistance was confirmed by varying the amounts of foaming agent (3% and 5%) and the amount of elastomer (0%, 10% and 20%). An amount of foaming agent of 3% or more was needed to satisfy the stiffness criterion of a maximum flex load of 110 N/50 mm. In addition, since the impact resistance was affected by the use of a highly fluid PP resin for the base material, an elastomer was added. The addition of the elastomer at 20% made it possible to meet the target Dupont impact strength of 5J. On the basis of the foregoing results, the specifications of the PP resin for

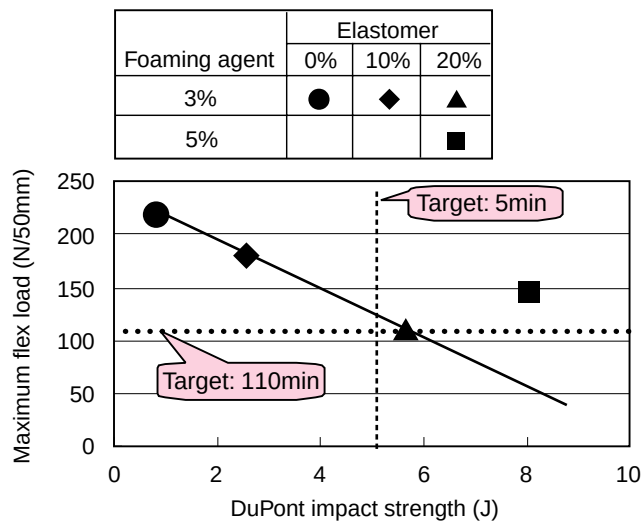


Fig.3 Stiffness and impact absorption of the foamed product

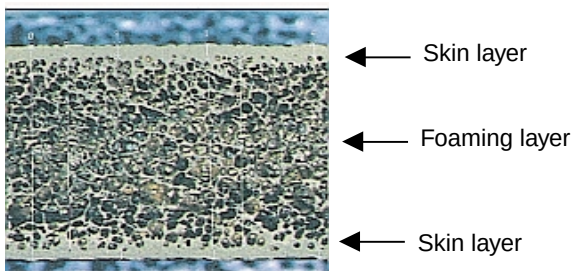


Fig.4 Cross section

Table 1 Typical properties

Properties		Typical value
Melt flow rate 230°C × 21.2N		73g/10min
Specific gravity		0.61
DuPont impact strength	23°C	9J
	- 30°C	1.5J
Tensile yield strength		10.9MPa
Tensile elongation at break		16%
Maximum flex load		113N
Flex modulus incline		92N/cm
Flammability		19mm/min
Fogging value		2.30%

injection molding were set at a foaming agent content of 3% or more and an elastomer content of 20% or more.

A cross-sectional photograph of the foamed product obtained with these material specifications is shown in Fig. 4. Skin layers were formed on both sides of the foaming layer, which was comprised of uniformly foamed cells. The typical properties of this foamed product are shown in Table 1.

3.2. Production Engineering

3.2.1. Molding Conditions

Foaming injection molding is characterized by the foaming step during which the mold is opened immediately after injection. The process is shown in Fig. 5. The process follows the normal injection compression molding procedure through to the final molding step. Following completion of PP resin injection, the mold on the movable side is opened several millimeters. This step is referred to as initial mold opening. In this step, the resin is foamed and then enters a cooling step, after which the mold is fully opened and the product is removed.

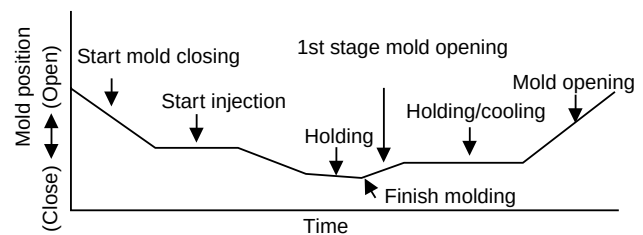


Fig.5 Molding cycle

Since the conditions of the initial mold opening have a considerable effect on the molded product, the molding conditions were checked. The expansion ratio, sheet thickness, skin layer thickness and other parameters vary according to the position of the mold at the start of opening, the initial mold opening velocity, the mold holding time, etc. For example, if the amount of mold opening at the start of injection is 20-30 mm and the mold holding time is increased, the thickness of the skin layers increases and the

foaming layer is not adequately foamed as a result of the resin temperature failing to rise sufficiently. This results in decreased thickness of the product sheet. In addition, if the final molding time after injection is decreased, the thickness of the skin layers decreases. Since this results in the foaming layer undergoing adequate foaming, the product sheet thickness increases.

3.2.2. Effect of the Initial Mold Opening

The conditions of the initial mold opening on the foaming status were investigated, and the effects of the initial mold opening velocity on the skin layer thickness and foam cell diameter are shown in Figs. 6 and 7. Tests were conducted at velocities of 0.5 mm/sec and above. It was determined that as velocity was increased, the skin layers became thinner and the foam cell diameter increased. As the mold opening velocity increased, the surface layer of the resin was unable to keep up with the movement of the mold, and a void was formed between the mold and the resin surface layer. This

void caused the pressure to decrease which in turn facilitated foaming. This resulted in the foam cells becoming larger. However, if the mold opening velocity was too high, the foam cells began to rupture, preventing the formation of a closed cell foam. In contrast, if the mold opening velocity was too slow, the mold and resin surface layers remained in constant contact, making the skin layers start to solidify and become too thick.

Next, the effects of the mold holding time are shown in Figs. 8, 9 and 10. If the holding time exceeded a certain predetermined time, the skin layers became thick and the expansion ratio decreased. As a result, foaming was inhibited and the product sheet thickness decreased. If the skin layers became excessively thick, the resin temperature in the foam layer fell and foaming was unable to adequately proceed. Therefore, a mold holding time of 1-2 seconds was determined to be most favorable.

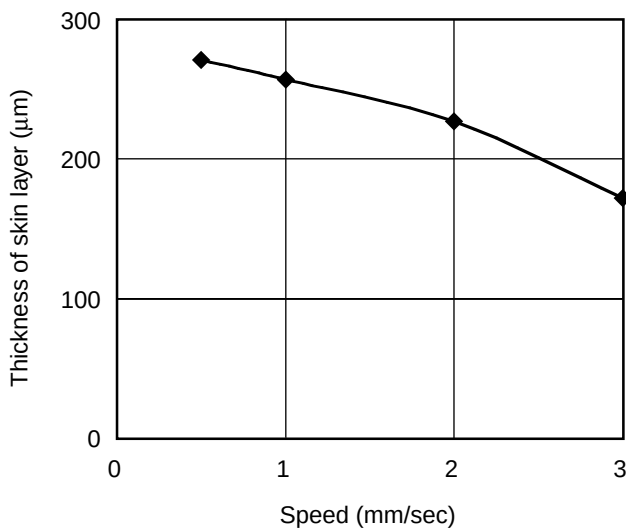


Fig.6 Initial mold opening speed and thickness of skin layer

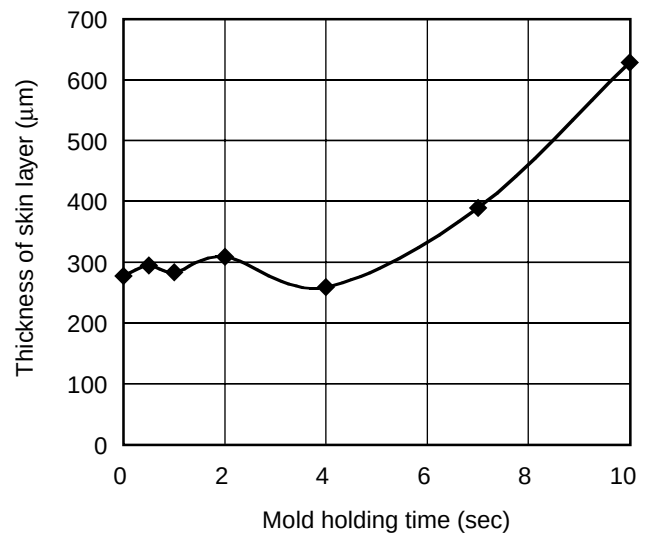


Fig.8 Mold holding time and thickness of skin layer

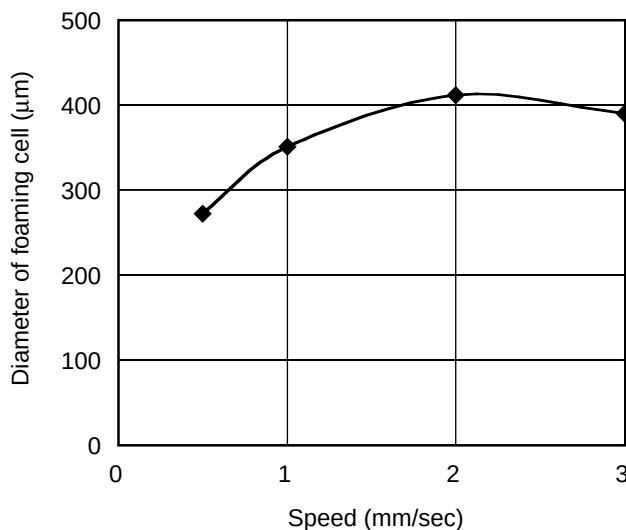


Fig.7 Initial mold opening speed and foam cell size

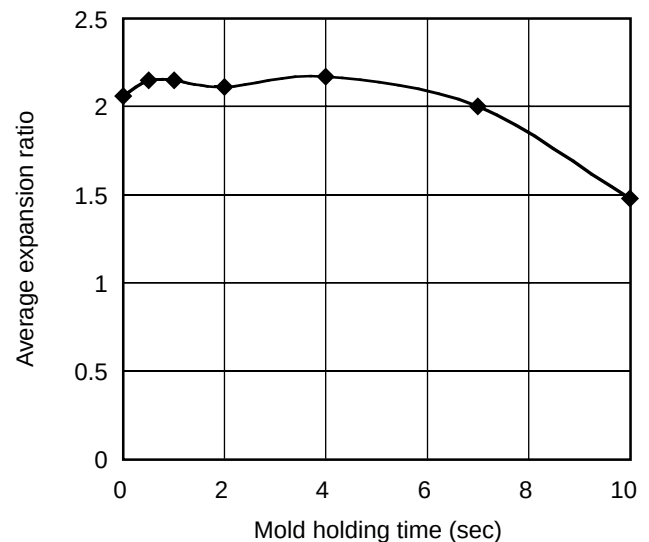


Fig.9 Mold holding time and foaming expansion

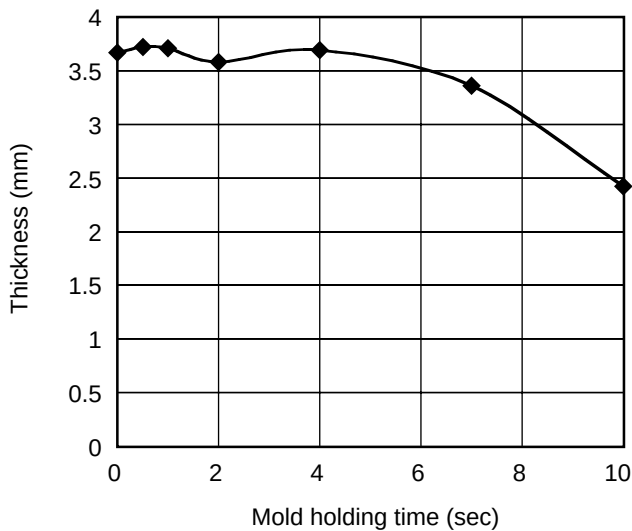


Fig.10 Mold holding time and product thickness

3.2.3. Fabric Application Conditions

Door trim employing this newly developed foamed PP resin is applied at the same time as the fabric. Therefore, the effect of fabric application on the foaming status was investigated. In the case of fabric application with ordinary injection compression molding, if the fabric is damaged by the resin heat and pressure, those fibers that are originally upright are pushed down, resulting in a dull appearance. In the case of foaming injection molding, however, it was found that damage to the fabric was less severe due to the occurrence of the initial mold opening performed immediately after the injection process, thereby proving the advantages of the process.

In addition, since the fabric is placed between the resin and mold, no well-defined skin layer is formed on the movable mold side. However, the resin impregnates the fabric to serve as a substitute skin layer, so the stiffness is maintained. Fig. 11 shows a cross-sectional photograph of a fabric-applied product. The solid layer of resin that has impregnated the fabric can be seen.

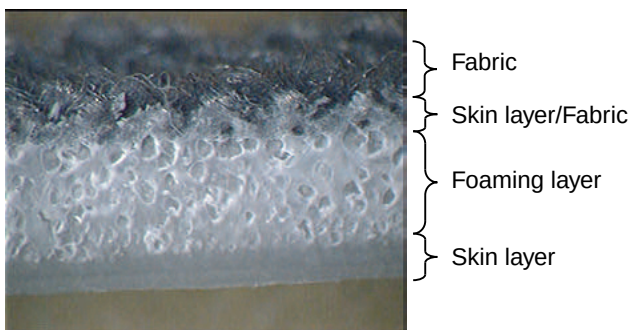


Fig.11 Cross section of product

4. Component Performance

4.1. Stiffness

Various stiffness tests were conducted on door trim that had been injection molded with foamed PP. There were no particular concerns and the product performance was satisfactory.

The amount of deflection at various locations was determined in a static load test. The results of measuring with a 30 mm diameter pressing tool supporting a weight of 10 kg indicated that standards were achieved at all locations measured. A load was also cyclically applied in the vertical direction to the armrest and the breaking load was measured. The required standard was satisfied in all cases. Since foaming does not take place on vertical surfaces with foaming injection molding, stiffness is comparable to that of the solid product.

4.2. Impact Resistance

A 1 kg steel ball was dropped from a height of 30 cm on to door trim installed on a door panel in a low-temperature environment (-30°C), which is considered to represent the extreme conditions to which the automobile will be subjected. A subsequent evaluation revealed no splitting or cracking of the door trim, confirming that the impact resistance performance was equal to that of conventional products.

In addition, no cracking or splitting of the base material was apparent in side collision tests, so the requirements were satisfied.

4.3. Confirmation of the Deformation Behavior due to Environmental Temperature Conditions

The effect of environmental temperature changes on the foamed base material was also investigated. The test consisted of 5 cycles of hot (4 hours at 80°C), cold (4 hours at -30°C), humid (4 hours at 50°C and 95% RH) and cold (4 hours at -30°C). Although a maximum gap of 1 mm occurred when the base material was returned to room temperature after completing the 5 cycles, this was determined to not present any concerns in terms of product appearance, and there was no deformation or any other apparent issues. In addition, since no secondary foaming or any other phenomena was apparent following injection molding, the foamed base material was determined to satisfy the required level of performance for use as a door trim base material.

5. Conclusions

We developed a foamed PP material for application as door trim base material as a part of the program for reducing vehicle weight. Material development included an examination of the foaming agent blending ratio and improvement of the impact resistance of the PP resin used as the base material for injection molding. Studies were also conducted on production engineering to establish a method for foaming injection molding. The following results were obtained during the course of this development:

- (1) An endothermic foaming agent prepared in the form of master batches allowed uniform foam cells to be

produced with an expansion ratio of 2.

- (2) Thin sheets of PP resin for base material injection molding could be molded by increasing the fluidity of the mix.
- (3) The addition of an elastomer in a later step at a rate of 10-20% restored the impact resistance that had been decreased by the fluidity requirement.
- (4) Products foamed to roughly twice their volume had stiffness equal to that of filled PP.
- (5) The conditions for the initial mold opening were determined to have an important effect on the expansion ratio, skin layer thickness and sheet thickness.

Vehicles in which door trim base materials produced from this newly developed foamed PP resin showed a weight reduction of 400 g per vehicle. This represents a weight savings of approximately 25% as compared with filled PP.

6. Acknowledgments

The author expresses his deep appreciation to those persons involved in conducting this development study at Japan Polychem Co., Ltd. and Nissen Chemical Co., Ltd. for their immense cooperation in examining modifications to the resin materials, producing prototype parts and numerous other activities.

7. References

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■ author ■



Kenji HIKITA

Technical Papers Submitted to Out-of-Company (2000-January to -June)

(English Papers only)

Date	Conference	Title	Author	Organization
Jan.15, 2000	JSAE/JSAE Review/Vol.21, No.1	The Research of Zero Level Emission Vehicle Using Gasoline Engine: Efficient Method for Using the Hybrid Catalyst and EHC	Noriyuki Kishi Shinnichi Kikuchi Hideo Kaiho Tadayoshi Hayashi	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		An Automotive Lateral Controller Based on Generalized Predictive Control Theory	Shinnichirou Horiuchi Toshiyuki Tamatsukuri Sinnya Nohtomi	Nihon University Nihon University Tochigi R&D Center
Feb.18, 2000	Technical University of Berlin/Invited talk for the “Neural computation day”/(Berlin, Germany)	Architecture of Intelligent Systems	Edgar Koerner	Honda R&D Europe
Mar.6-9, 2000	SAE/2000 International Congress & Exposition/(Detroit, USA)	A Quick Warm-Up System During Engine Startup Period Using Adaptive Control of Intake Air and Ignition Timing	Masaki Ueno Yuji Yasui Shusuke Akazaki Yoshihisa Iwaki	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		The Secondary O ₂ Feedback Using Predication and Identification Type Sliding Mode Control	Yuji Yasui Shusuke Akazaki Masaki Ueno Yoshihisa Iwaki	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		L4-Engine Development for a Super Ultra Low Emissions Vehicle	Hiroshi Kitagawa Toshihiro Mibe Kazunori Okamatsu Yuji Yasui	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		A New Concept for Occupant Deceleration Control in a Crash	Yasuki Motozawa Takahiro Kamei	Tochigi R&D Center Tochigi R&D Center
		Development of the High-power, Low-emission Engine for the “Honda S2000”	Yutaka Otake Hiroshi Kawaguchi Hideo Ueshima	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Development of Valve Seat Material for Gas Fueled Engine	Katsuaki Sato Teruaki Midorikawa Teruo Takahashi Hiroshi Oshige	Tochigi R&D Center Tochigi R&D Center Nippon Piston Ring Co., Ltd. Nippon Piston Ring Co., Ltd.
		A New Heat Resistant Magnesium Alloy for Oil Pan	Seiichi Koike Kazuhiro Washizu Tsuyoshi Baba	Tochigi R&D Center Tochigi R&D Center Wako Research Center
		The Development of a High Efficient Lean NOx Catalyst for a Lean Burn Engine	Hiroshi Ohno Toshikatsu Takanohashi Nobuaki Takaoka Kenji Sugiura	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center

Date	Conference	Title	Author	Organization
Mar.6-9, 2000	SAE/2000 International Congress & Exposition/(Detroit, USA)	A Vibration Transfer Reduction Technique, Making Use of the Directivity of the Force Transmitted from Road Surface to Tire	Masaki Minakawa Jiro Ninomiya Jun Nakahara	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Cyclic Variations of Initial Flame Kernel Growth in a Honda VTEC-E Lean-Burn Spark-Ignition Engine	Pavlos G.Aleiferis Alex M.K.P.Taylor James H.Whitelaw Kiyoshi Ishii Yasuhiro Urata	Imperial College Of Science, Tech. & Med. Imperial College Of Science, Tech. & Med. Imperial College Of Science, Tech. & Med. Tochigi R&D Center Tochigi R&D Center
		Shifting Mechanisms and Variation of Frictional Coefficients for CVT Using Metal Pushing V-belts	Osamu Fujimura Shigeru Kanehara Kazuya Okubo Toru Fujii	Doshisha University Tochigi R&D Center Doshisha University Doshisha University
		Measurement of Ambient Roadway and Vehicle Exhaust Emissions - An Assessment of Instrument Capability and Initial On-road Test Results with an Advanced Low Emission Vehicle	Timothy J. Truex John F. Collins Jeff J. Jetter Benjamin Knight Tadayoshi Hayashi Noriyuki Kishi Norio Suzuki	University of California HONDA R&D AMERICAS Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Development of an On-Board Analyzer for Use on Advanced Low Emission Vehicles	Jeff J. Jetter Shinji Maeshiro Seiji Hatcho Robert E. Klebba	Honda R&D Americas Honda R&D Americas Tochigi R&D Center Nicolet Instrument Corp
		Trends in Engine Valve Development for Automobiles and Motorcycles	Hiroyuki Oketani Masao Ishida Toshiharu Noda Shigeki Ueta Mikio Kiriyaama	Asaka R&D Center Asaka R&D Center Daido Steel Co., Ltd. Daido Steel Co., Ltd. Nittan Valve Co., Ltd.
		Relationship between Occupment Constraint System and Injury Criteria	Hideoki Yajima Atsunori Nozaki Qjang Yu Masaki Shiratori	Wako Research Center Yokohama National University Yokohama National University Yokohama National University Yokohama National University
Mar.7-9, 2000	DARATECH/Product Process Management CAD/ CAM, CAE 2000/(Boston, USA)	Hope for 3D CAD in the 21th Century	Hiroshi Ueda	Tochigi R&D Center
Mar.28, 2000	Haus Der Technik/ Pkw-Lenksysteme im Jahr 2000/ (Essen, Germany)	EPS (Electric Power Steering) System for Honda Accord	Yasuo Shimizu Katuji Watanabe Mitsuo Sugino	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center

Date	Conference	Title	Author	Organization
Mar.29, 2000	Elsevier/Applied Mathematics and Computation/No.111	Hypothetical Reasoning and Brainware	Kouji Tsujino Edgar Koerner Hiroshi Kondo	Wako Research Center Honda R&D Europe Wako Research Center
Apr.15, 2000	JSAE/JSAE Review/Vol.21, No.2	Thermal Behavior Analysis of Nickel Metal Hybrid Batteries for Electric Vehicles	Noboru Sato Kazuhiko Yagi	Tochigi R&D Center Tochigi R&D Center
		Reducing the Effects of Driving Fatigue with the Adoption of a Lane Following Assistance System	Koji Tanida	Tochigi R&D Center
		Development of Oxygen Sensor for Compressed Natural Gas Engine	Hiroshi Kubota Hiroyuki Fujita Teppei Okawa Akio Mizutani	Tochigi R&D Center Tochigi R&D Center NGK Spark Plug Co., Ltd. NGK Spark Plug Co., Ltd.
		Development of a New Six-speed Manual Transmission	Kyoji Kitajima	Tochigi R&D Center
Apr.20, 2000	Ministry of Education, National Science Council/3rd International Conference on TCD/(Taipei, Taiwan)	R&D Management System in Product Development	Seiichi Koike	Tochigi R&D Center
Apr.26-28, 2000	ACDM2000/Fourth inf. Conf. On Adaptive Computing in Design and Manufacture/(Plymouth, UK)	Optimization of a Stator Blade Used in a Transonic Compressor Cascade with Evolution Strategies	Markus Olhofer Toshiyuki Arima Toyotaka Sonoda Bernhard Sendhoff	Honda R&D Europe Wako Research Center Wako Research Center Honda R&D Europe
May.5, 2000	Austrian Society of Automotive/Vienna Engine Symposium/(Vienna, Austria)	NOx Elimination Performance of a Mixed Metal Oxide Type De-NOx Catalyst for Lean Exhaust Gas	Shigeru Ozuhata Hisao Yamashita Akio Mukao Hidehiro Iizuka Osamu Kuroda Koujiro Okude Hiroshi Ohno Toshikatsu Takanohashi Nobuaki Takaoka	Hitachi, Ltd. Hitachi, Ltd. Hitachi, Ltd. Hitachi, Ltd. Hitachi, Ltd. Hitachi, Ltd. Hitachi, Ltd. Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
May.11-12, 2000	IEEE/IEEE Symposium/(San Antonio, USA)	Optimization of Neural Networks for Problem Classes	Michael Husken Jens Gayko Bernhard Sendhoff	Ruhr-Universitat Bochum Ruhr-Universitat Bochum Honda R&D Europe
May.15-17, 2000	SAE/2000 SAE Automotive Dynamics & Stability Congress/(Troy, USA)	A Vibration Transfer Reduction Technique Making Use of the Directivity of the Force Transmitted From Road Surface to Tire	Masaaki Minakawa Jirou Ninomiya Atsushi Nakahara	Honda R&D Europe Tochigi R&D Center Tochigi R&D Center

Date	Conference	Title	Author	Organization
May.21-23, 2000	International Magnesium Association/IMA2000 (Vancouver, Canada)	New Creep-Resistant Magnesium Alloy for Automobile Engin Components	Seiichi Koike Kazuhiro Washizu Kouhei Kubota Youichi Nozaka	Tochigi R&D Center Tochigi R&D Center Mitsui Mining & Smelting Co., Ltd. Mitsui Mining & Smelting Co., Ltd.
May.24-26, 2000	JSAE Annual Congress (Yokohama)	Development Introduction of Aerodynamic Drag Reduction in Low Fuel Economy Car	Michiharu Imayasu	Tochigi R&D Center
		Objective Description of the Optimum Interior Sound Character of Sports Cars	Franz.K.Brandl Yasuo Ishii Tsuyoshi Yamashita Siegmund Thomann Werner Biermayer	AVL List Tochigi R&D Center Tochigi R&D Center AVL List AVL List
Jun.12-15, 2000	FISITA/2000 FISITA World Automotive Congress/(Seoul, Korea)	Analysis of the Influence on Fuel Economy by Transmission Type and the Estimation of Fuel Economy	Kazuhisa Yamamoto Takashi Aoki	Tochigi R&D Center Tochigi R&D Center
		Evaluation of Thermal Fatigue Life on the Exhaust Manifold by Analysing Restraint Ratio	Kazuo Ishii Makoto Nakada Shinnichi Takahashi Masaru Enomoto Yasunori Konishi	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		A New Concept for Occupant Deceleration Control in a Crash	Yasuki Motozawa Takahiro Kamei	Tochigi R&D Center Tochigi R&D Center
		The Three-dimensional Pulsation Flow Simulation and its Application for Engine Intake and Exhaust Systems	Toru Takabayashi Yasuaki Hotozuka Hiroto Tsushima	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		The Computer Simulation Process for Pedestrian Protection Structure	Suguru Yoshida Satoru Takeishi Kouji Enomoto Toru Kikuchi	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Analysis on Steering Gain and Vehicle Handling Performance with Variable Gear- ratio Steering System(VGS)	Masato Abe Kouji Shibahata Yasuo Shimizu Shoichi Sano	Kanagawa Institute of Technology Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center

Date	Conference	Title	Author	Organization
Jun.12-15, 2000	FISITA/2000 FISITA World Automotive Congress/(Seoul, Korea)	Research of the Ultimate Cleanness of Internal Combustion Engine and the Application for Mass Production Vehicles	Noriyuki Kishi Shinichi Kikuchi Norio Suzuki Tadayosi Hayashi	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Transition of the Development of Aluminum Frame and its Future	Eitaro Koya Syuichi Kawamoto Toshihisa Nagatshii Yuji Kadono Koichi Chiba	Asaka R&D Center Asaka R&D Center Asaka R&D Center Asaka R&D Center Asaka R&D Center
Jun.19-21, 2000	SAE/Government/ Industry Meeting/ (Washington D.C., USA)	An Integrated Motor Assist Hybrid System Development of the 'Insight', a Personal Hybrid Coupe	Kenji Nakano Shinobu Ochiai Benjamin Knight Shigemasa Kajiwara Kaoru Aoki Hiromitsu Sato	Tochigi R&D Center Tochigi R&D Center Honda R&D Americas Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
	SAE/"Spring" Fuels & Lubricants Meeting & Exposition /(Paris, France)	Development of Integrated Motor Assist Hybrid System Development of the 'Insight', a Personal Hybrid Coupe	Shigetaka Kuroda Kaoru Aoki Benjamin Knight Shigemasa Kajiwara Mitsuhiro Sato Yoshio Yamamoto	Tochigi R&D Center Tochigi R&D Center Honda R&D Americas Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Potential of a Natural Gas Vehicle as EEV	Toshiyuki Suga Tadashi Muraishi Thomas Brachmann Fumio Yatabe	Tochigi R&D Center Tochigi R&D Center Honda R&D Europe Honda R&D Europe
		Basic Understanding of Activated Radical Combustion and Its Two-Stroke Engine Application and Benefits	Youichi Ishibashi	Asaka R&D Center

Editor's Notes

Vol.12 No.2

We have finished our editorial work on the final issue of the Honda R&D Technical Review in the 20th century, the second year for the Review to be published biannually. It goes without saying that in the twelve years since our inaugural issue, we have seen an increase in the number of articles we publish as well as progress in the quality of the articles. The Technical Review has made great strides as is evident from the recognition it receives both inside and outside Honda.

Looking back to the days when we began publishing the Review, we recall how we had to fumble through the unfamiliar editorial work, and how the review committee struggled to revise the wording in order to improve the level of the articles published. This is all just a memory now, including how frantically we had to work to bring out one issue a year.

Recently, though, the Technical Review has developed to the point of meeting the twice-yearly deadlines on schedule, thanks to the untiring work of our review committee and the efforts of all the others involved. The growing trend of publishing information for people outside the company has also aided us in the publication of the Review. A growing part of the Review's contents now touch on environmental, energy, and safety-related topics from a global perspective, so that the Technical Review can fulfil its role as a record of Honda's research and development activities addressing the needs of society.

The Technical Review is intended to serve not only as a communications tool that conveys Honda's new technologies to the world at large, but also to function more than ever as an arena for exchange among specialists inside and outside Honda, and as a forum to refine technologies that will contribute to the human society as a whole. Let's continue joining forces in working toward this goal in the 21st century.

(Masayasu Nishikawa: Wako Research Center)

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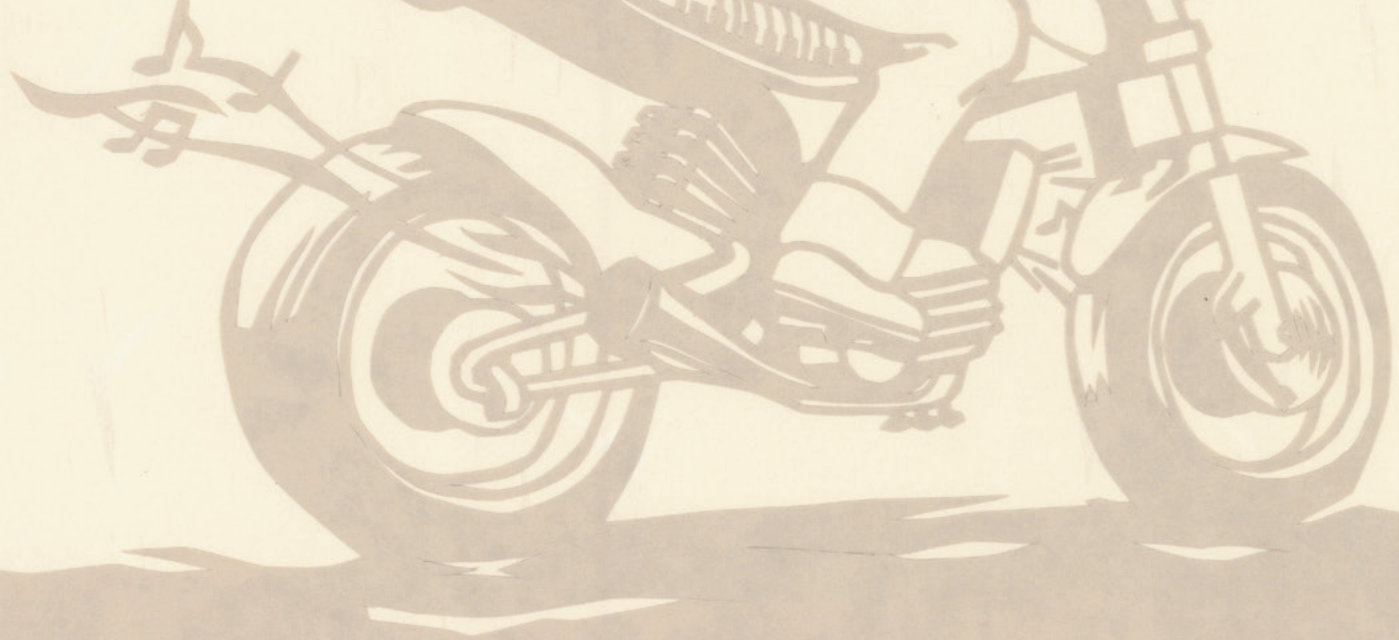
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