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Development of the Humanoid Robot ASIMO

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ABSTRACT

Honda has been conducting research on biped walking humanoid robots as a challenge toward a new field of mobility. This paper discusses ASIMO, the Advanced Step in Innovative MObility developed as a result of our efforts in this field. Our target in this development was a robot that could co-exist with humans and that could actually be useful to humans. This target was achieved through the development of a minimally sized robot (height: 1200 mm, weight: 43 kg) that could conform to a real living environment. Also, by further fostering past research we were able to advance the functionality of the robot. In particular, by adding “free walking technology,” which includes predictive control technology, to the existing walking stability control, we were able to achieve a biped robot that can move more smoothly and continuously.

1. Introduction

Since 1986, Honda has been engaged in research on a humanoid robot that would generate new value to human society through coexistence and useful cooperation with human beings. Honda’s aim has been to develop a robot that would be able to function in people’s living spaces, as part of its program to create innovative mobility. In the course of that research, we obtained gait data from an analysis of human walking, and made use of designs for actuators and mechanisms for bipedal walking, control processes for bipedal walking, and other such technologies to create our P2 prototype cordless biped humanoid robot.⁽¹⁾ The first such robot in the world, the P2, was announced in 1996. In 1997, after studying the place of the humanoid robot in the human life environment and in society, we further completed the P3 prototype model, which is closer to human size.

At the end of October 2000, we brought out the humanoid robot ASIMO (Fig. 1). This is of the minimum size suited to the human life environment (1200 mm in height, 43 kg in weight), and our goal was both to further develop the above technologies and to make a robot that could actually perform useful functions in close association with people. ASIMO, which stands for Advanced Step in Innovative MObility, is the collective name for Honda’s humanoid robots. Here we will introduce ASIMO’s specifications and the walking technology that gives it the capability for smooth, continuous motion.

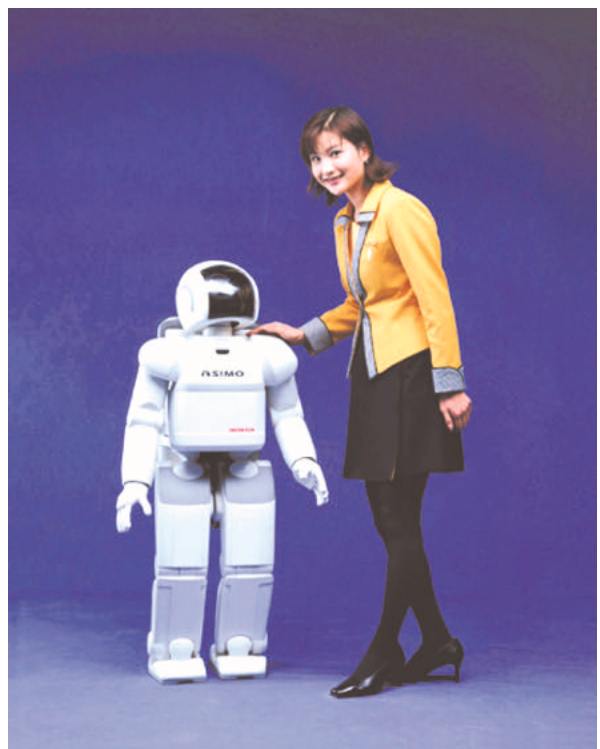


Fig. 1 Humanoid Robot ASIMO

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2. Development concept

This project has engaged in research and development of new technologies for the practical application of ASIMO, and particularly the resolution of technical issues in the following three areas:

- (1) Compactness and light weight
- (2) Advanced arm and hand functions
- (3) Enhanced intelligent walking

Size and weight are the most important factors in having a robot function in close association with people in their life environment. It will be necessary to seek greater compactness and lighter weight relative to the previous P3 in order for ASIMO to demonstrate its practicality in the human life environment.

It will also be necessary to expand the compact robot's range of arm motion in order to minimize limitations on its range of activities. In addition, we will have to develop simple mechanisms to make hands capable of grasping objects gently with uniform force.

In order for ASIMO to work in collaboration with humans, we must also develop walking functions and control that enable smoothness of movement in addition to stability in walking. This will be crucial in achieving practical application.

3. Main specifications

3.1 Size

The objective in fabricating the previous P3 prototype was to create a humanoid robot equivalent in size to an adult human. With ASIMO, however, we sought to determine what size a robot should be to actually be useful in the human life environment. The point was for ASIMO not simply to move and transport objects, but also to be of a size suited to its environment. Therefore, we studied the vertical height of arm extension and the squatting position in terms of enabling access to items such as doorknobs, light switches, electric outlets, and other such items in the life environment, as shown in Fig. 2. We also determined the elbow and shoulder locations for ASIMO from work posture at desks and workbenches that we ordinarily use.

ASIMO was given a shoulder height of 910 mm when it stands erect. That location served as the criterion for an

upper working space that extends to a height of 1290 mm with ASIMO's arms extended above, and a lower working space that allows ASIMO to work below while crouching. As shown in Fig. 3, the legs had to be 610 mm in length from the hip joint down in order to enable the robot to go up and down stairs. The results of this study led us to make ASIMO 1200 mm in height. The robot's width was set at 450 mm and its depth at 440 mm in consideration of its passage through narrow corridors and doorways. Table 1 compares these sizes with those of the P3. Assigning these dimensions also made the robot more approachable and allows easier communication with it when we are seated on chairs.

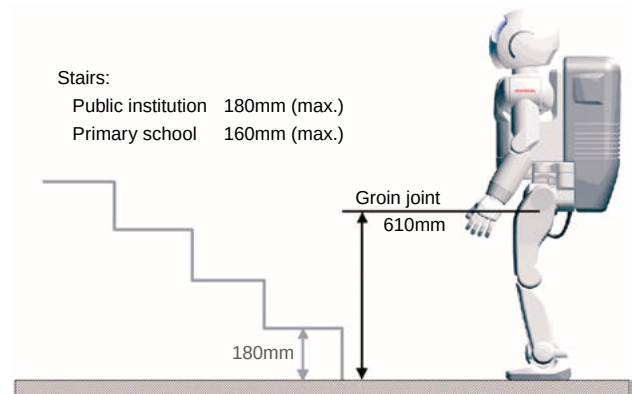


Fig. 3 Stairs

Table 1 Size of ASIMO/P3

	ASIMO	P3
Height (mm)	1200	1600
Width (mm)	450	600
Depth (mm)	440	555
Weight (kg)	43	130

3.2 Weight

As shown in Fig. 4, we reduced ASIMO's bulk and weight relative to the P3, which was 1600 mm in height and 130 kg in weight. Where a weight of 54kg would have followed proportionately from the reduction in bulk, we pressed to make ASIMO lighter by another 20% and achieved a weight of 43 kg. The frame utilizes magnesium alloy materials in an H-shaped cross-sectional structure made up of three-sided elements. The thin-wall casting method was used for the 1.5-mm thick basic structure. The electrical equipment, control units, and so on were also made compact and lightweight by means of optimum design. This robot weighs approximately twice as much as the 22–23 kg of a human being of the same height, but it can be transported by two adults.

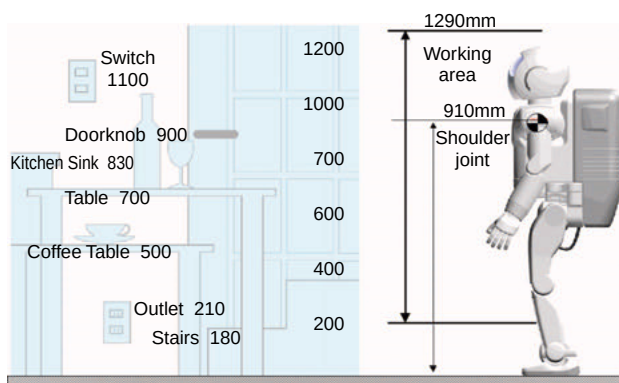


Fig. 2 Life environment and working space

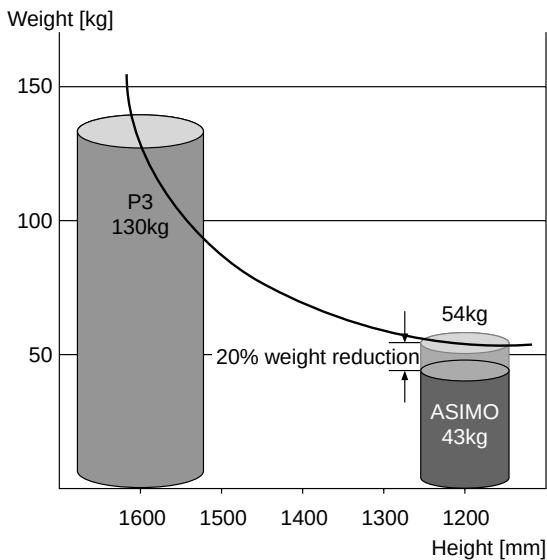


Fig. 4 Weight reduction

3.3 Joint structure

The structure of the joints was determined with reference to the human skeletal structure, while maintaining simplicity and basic movement functions. Fig. 5 shows the joint structures and degrees of freedom. The legs are similar to those of the previous P3 robot in that the ankle, knee, and hip joint structures have six degrees of freedom, and they are of course capable of walking as well as such actions as going up and down stairs and striding over obstacles. The hands, which can open and close their five fingers, have one degree of freedom. The arms are constructed to have five degrees of freedom in the shoulder, elbow, and wrist joints to allow free positioning of handheld implements. The head is constructed with two degrees of freedom to give it the ability to nod.

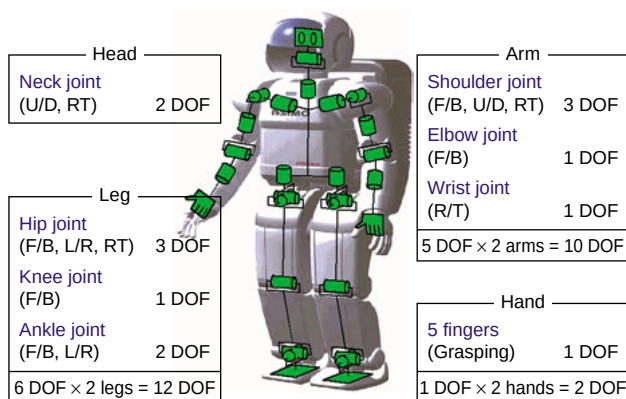


Fig. 5 Joint structure and DOF (Degree of Freedom)

3.4 Arm and hand functions

Study of the work posture of a 1200-mm high robot showed that movements at a high workbench resulted in a posture that brings the elbow joint immediately beside the shoulder joint, as shown in Fig. 6. With this kind of movement, the direction of axial rotation of the shoulder joint and of the elbow joint are in a straight line. This can cause points where their directions of movement overlap, thus restricting the degree of freedom. In order to avoid this limitation, the shoulder joint was built to tilt upward at an angle of 20°, which allows the arm a greater range of movement. As a result, ASIMO can work with its elbow raised as high as 105°, giving it the same working area as the 1600-mm tall P3.

The hand is configured so the five fingers are actuated by pulling a single wire. This makes both grasping and opening movements possible. When the robot starts to grasp an object, one finger will stop on that spot as soon as it comes in contact with the object and the remaining fingers then come in contact with it one after the other in an enfolding movement. In other words, the hand is capable of grasping objects gently with uniform force. The simplicity of the mechanism also improves its access in confined spaces.

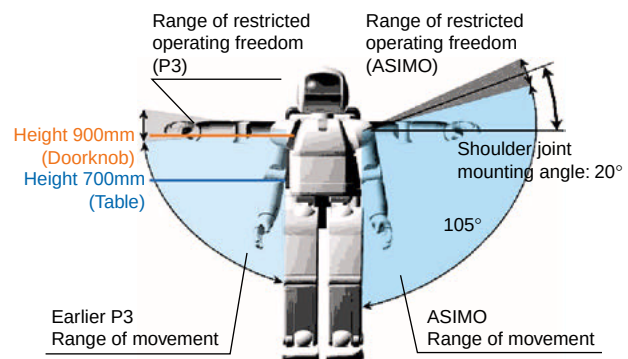


Fig. 6 Range of arm movement

3.5 Operating method

The previous P3 was operated entirely from a workstation. In addition to that method, ASIMO utilizes another, portable remote controller, as shown in Fig. 7, to achieve more intuitive, direct operation.

The use of the workstation as in the past allows startup and standard movements to be carried out automatically. The use of a portable controller makes it possible for the robot to easily perform flexible walking (forward, reverse, sideways, diagonal, turning). Stored movements (grasping with the hand, waving both hands, waving goodbye, responding with a bow, etc.) can be selected and executed by pressing a button on the portable controller.

3.6 Walking method

As will be discussed in detail in the next section, ASIMO has realized a flexible walking technology that incorporates

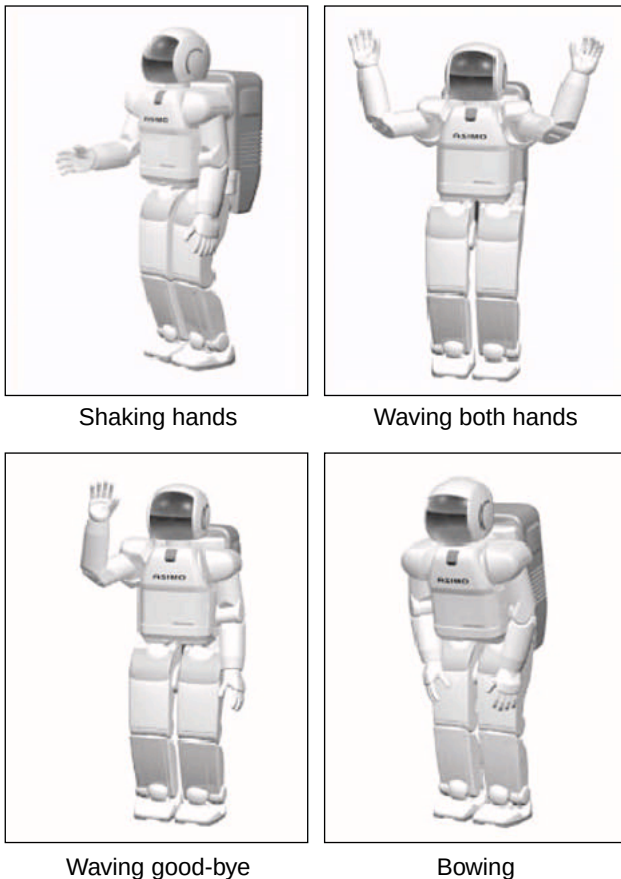
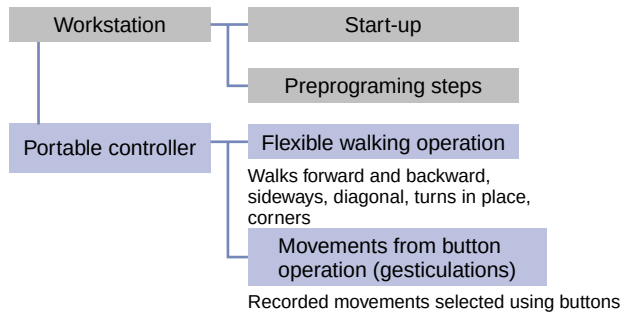


Fig. 7 Operation system

Table 2 Specifications for ASIMO

Height	1200mm
Weight	43kg
Walking speed	0 ~ 1.6km/h
Walking cycle	Cycle adjustable, Stride adjustable
Grasping force	0.5kg/hand (5-finger hand)
Actuator	Servomotor + Harmonic speed reducer + Drive unit
Control unit	Walk/Operating control unit, Wireless transmission unit
Sensors	Foot 6-Axis foot area sensor
	Torso Gyroscope & acceleration sensor
Power section	38.4V/10AH (Ni-NH)
Operating section	Workstation or portable controller

predictive movement control. This is in addition to the walk stabilizing control that was developed in previous research. The new technology has made it possible for ASIMO to change its walking pattern continuously and freely at any moment.

The main specifications of ASIMO described above are shown together in Table 2.

4. Walking technology

4.1 Walk stabilization

When humans are on the point of falling over while walking or standing erect, they (a) push down forcefully on the ground with one part of the sole of the foot, and if that is not enough to hold them up, they (b) change the way they are moving or take a step forward to try and recover their posture. These human movements can be analyzed according to the study of dynamics as follows:

The point where the line of action for the combined gravity and inertia forces intersects with the ground surface is termed the zero moment point (ZMP).⁽²⁾⁽³⁾ On the other hand, the point where the ground reaction force operates is known as the central point of the ground reaction force (C-ATGRF). Basically, robots generate their desired walking pattern from that dynamic model using their CPUs, and they walk by making their joints track through the resulting pattern. The ZMP of the desired walking pattern is called the desired ZMP.

While an actual robot is walking in a desirable manner, the line of action of the desired combination of gravity and inertia forces and the line of action of the actual ground reaction force coincide. However, when the robot steps on an uneven ground surface, for example, the lines of action diverge. Then, as shown in Fig. 8, the robot becomes unbalanced and a tipping moment is imposed. This tipping moment is roughly proportional to the discrepancy between the desired ZMP and the central point of the actual reaction force. In other words, the discrepancy between the desired ZMP and the central point of the actual ground reaction force is the greatest cause for loss of balance. The walk stabilization control used in the P2 and P3 instead restored

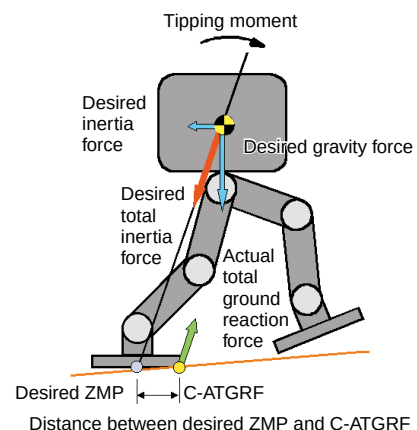


Fig. 8 Unbalance of the legged robot

the posture when the robot lost balance and was on the point of falling over by actively controlling that discrepancy. Ground reaction force control, desired ZMP control and foot landing position control, which are the key technologies in posture stabilizing control, will be discussed below.

4.1.1 Ground reaction force control

Not only does the ground reaction force control absorb unevenness in the ground, but it also acts by controlling the force with which the robot presses the sole of its foot down when it is on the point of falling over. Thus it can be considered a kind of active suspension. In concrete terms, the robot uses the 6-axis force sensor in each ankle to detect the central point of the actual reaction force. While doing this, it changes the positions and the attitudes of the front of the feet, rotating them around the central point of the desired ZMP, in order to control the correct location of the central point of the actual ground reaction force and thus keep a balanced posture. On the kind of unfamiliar upward slope shown in Fig. 9, for example, the central point of the actual ground reaction force shifts toward the front of the feet. However, the ground reaction force control function raises the fronts of the feet slightly to return the central point of the actual ground reaction force to the desired ZMP. When both feet are touching the ground on an unfamiliar upward slope as shown in Fig. 10, the ground reaction force in the forward foot increases and that in the rear foot decreases so

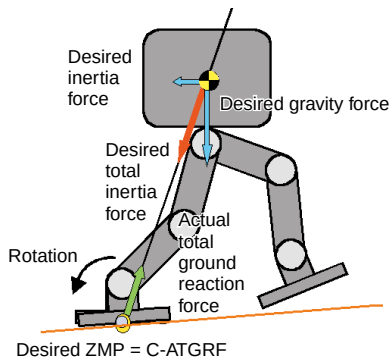


Fig. 9 Compliant motion during single support phase

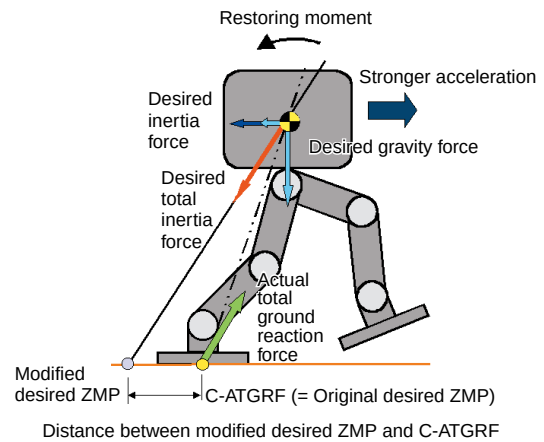


Fig. 11 Model ZMP control

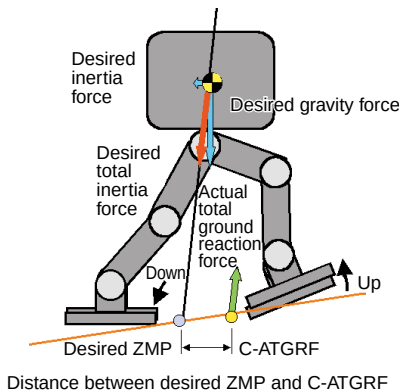


Fig. 10 Compliant motion during double support phase

that the central point of the actual reaction force ends up shifting forward. The control function then raises the front of the leading foot and lowers the front of the rear foot in order to rotate the fronts of the feet around the desired ZMP and in this way bring the discrepancy back to zero. In addition to making movements that absorb the influence of such unevenness in the ground, the control function also acts when the robot tips forward for some reason, by shifting the central point of the actual ground reaction force to be farther forward than the desired ZMP in order to generate a posture-restoring force. However, the central point of the actual ground reaction force cannot go beyond the area where the soles of the feet are touching the ground. Therefore there are limits to the posture-restoring force, and if the robot tilts very far, it will fall over.

4.1.2 Desired ZMP control

The desired ZMP control operates when the robot tilts significantly to prevent it from falling over. If the robot is in danger of falling forward, for example, the desired ZMP control accelerates the upper body trajectory of the desired walking pattern more strongly forward than the ideal trajectory, as shown in Fig. 11. As a result, the desired ZMP is shifted to the rear of the central point of the actual ground reaction force so that a force is applied to tilt the robot backward and the posture is restored. In other words, the control restores the robot's posture by deliberately unbalancing the desired walking pattern.

4.1.3 Foot landing position control

The operation of the desired ZMP control causes the desired position of the upper body to be accelerated more forcefully forward and so to be displaced. When this happens, and the next step is taken with the stride length that was originally ideal, the feet end up being left behind by the upper body. The foot landing position control then operates to correct the stride length in order to bring the upper body and the feet back into an ideal spatial configuration.

4.2 Enhanced intelligent walking

The previous method for generating the robot walking pattern involved designing several dozen kinds of basic walking patterns offline and having the robot memorize them as time-series data. The combination of these data (weighted averages) would then be used to generate flexible walking patterns. There are, however, several drawbacks arising from this principle, such as: (a) the robot must come to a temporary stop while making the transition from straight-ahead walking to a turn; (b) its turning walk is flat-footed; and (c) it is limited to several kinds of walking cycles. These drawbacks were resolved in ASIMO by the development of a new predictive movement control technology that seeks to improve the intelligence of walking.

Let us take the example of a continuous transition from straight-ahead walking to turning. As shown in Fig. 12, it is necessary to shift the center of gravity toward the inside of the turn in order to balance the centrifugal force. When ASIMO makes the transition from straight-ahead walking to a turn, predictive calculations are made starting before the turn to determine the optimum shift of the center of gravity at every instant while generating a walking pattern in real time.

In this way, the previous walk stabilizing control is augmented by incorporating additional predictive movement control to form a new flexible walking technology. Realization of this technology has made it possible to change the robot's walking pattern smoothly, freely, and flexibly at any moment without first coming to a stop.

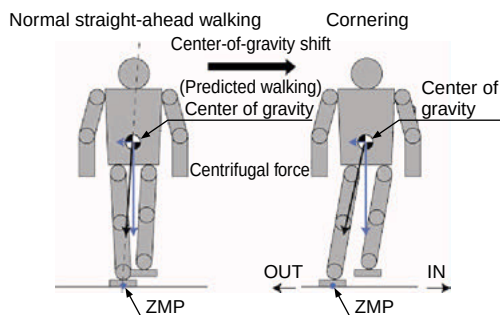


Fig. 12 Center-of-gravity shift when cornering

5. Conclusion

The development of ASIMO realized significant advances in compactness and lighter weight by means of our reassessment of dedicated electrical equipment designs and structures. The newly developed walking technology also gave it the capability for walking flexibly. For the future, we are planning to make further improvements in compactness and motor functions for a robot that will be useful in our life environments, and we will also promote development of advanced intelligence for a robot that can perform a variety of tasks.

We wish to express our gratitude to the many people here and elsewhere who cooperated with us in numerous and important ways on bringing this robot to completion.

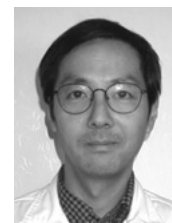
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Styling Design of the Humanoid Robot ASIMO

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ABSTRACT

The Honda humanoid robot ASIMO was developed as a challenge to create robots with a higher level of mobility than has existed before. For the first, the automotive styling division took part in this project from the initial planning stages. Furthermore, this humanoid robot is a product in a brand new category that does not have an established market. We proceeded in our development with our own unique proposals while at the same time we listened to the opinions of people, both inside and outside Honda, in order to determine our styling direction. Also, as ASIMO is a humanoid robot comprised of a multitude of moving elements, we made maximum use of computer technology from the initial stages of development. We were able to simulate, for example, walking motions with computer graphics to achieve an overall balance between styling and structure. By adding automotive styling experience to the robotic development cultivated to date, and by including new styling methods, we achieved a unique sense of value as well as familiarity and friendliness with the ASIMO styling.

1. Introduction

No commercial market for humanoid robots has yet been established. However, the development of many robots were announced at the end of the 20th century, prominent among them the AIBO⁽¹⁾ in the entertainment sector, re-igniting the dream of coexisting with robots. Honda has been moving forward with research on humanoid robots since 1986 as part of its program for innovative mobility (mobile equipment). The styling design division first joined this development effort in connection with the P2 (Fig. 1), which was announced in 1996. The Honda humanoid robot, including its styling, is gradually gaining the awareness of society at large, and the styling design is also becoming an extremely important element of the project in addition to the world-class technologies involved.

Honda's objective from the beginning has been to develop a humanoid robot (Fig. 1) that would coexist and collaborate with human beings and serve a useful function in society. The thinking was that in order for it to actually become useful, the robot should more easily blend into society, so it was necessary to give it a greater sense of friendly appeal with people. Development of ASIMO⁽²⁾ accordingly proceeded with the goals of further advanced innovativeness and approachability. ASIMO is the new

general name that Honda is applying to its humanoid robots, and at the same time we have also designed it as a logo (Fig. 2) that expresses the creation of new value, which is the key ASIMO concept. This represents one further step in laying the groundwork for Honda's challenging project of taking the humanoid robot out of the laboratory and positioning it as a new industrial product accepted by the world at large.

Industrial products developed during the 20th century, as typified by the automobile, evolved through the pursuit of such qualities as efficiency. In the 21st century, the robot will attain a much greater presence with human beings, and it will require new criteria for value. This motivated research for the appearance of humanoid robots created by Honda. This paper introduces the ASIMO styling design development effort and the techniques we employed, which included making the greatest possible use of computer technology, in the course of that effort.

2. Styling design concept

The goal for ASIMO was for it to be a gentle companion for people that has a real practical use. The basic conceptual approach to the styling design, therefore, was "to make a humanoid robot, which embodies the dream-like mobility, a gentler and closer presence."

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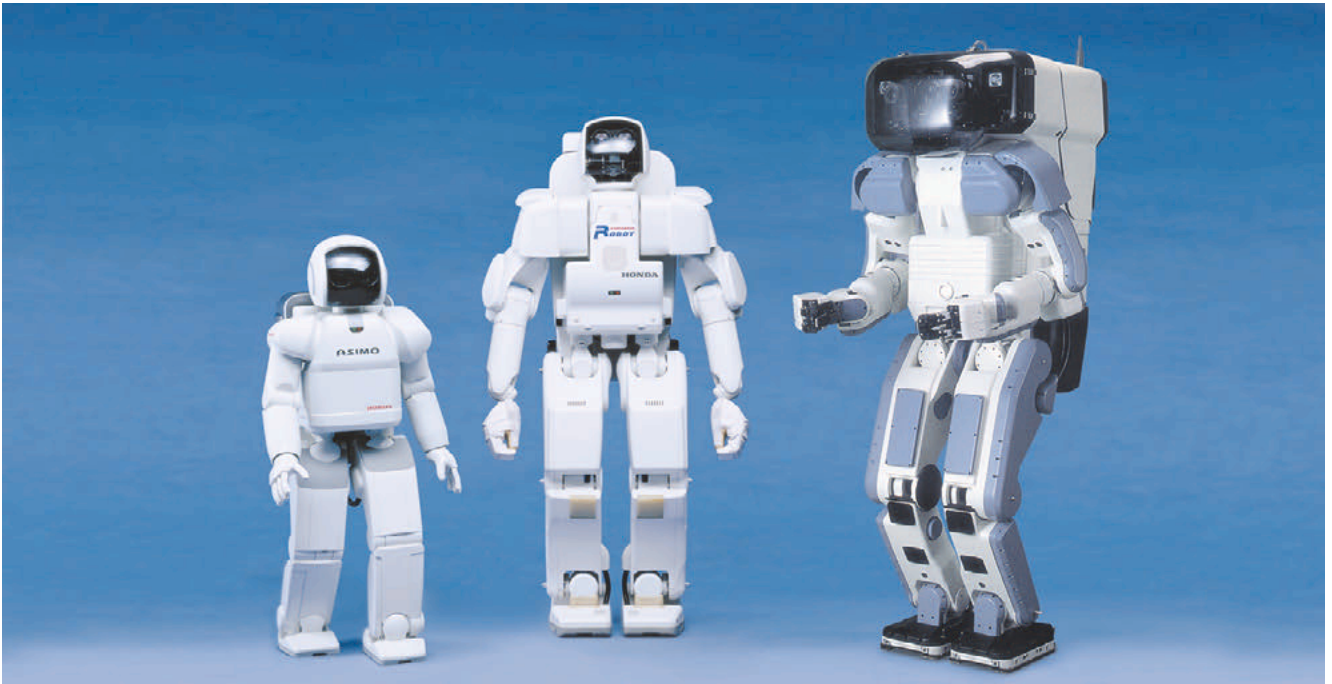


Fig. 1 The Honda Humanoid Robots Left: ASIMO (2000), Center: P3 (1997), Right: P2 (1996)



Fig. 2 ASIMO Logo styling design

One answer to achieving a mobility that is suited to coexisting with human beings in their life environment is to give the humanoid robot a form close to that of human beings. The robot should be able to go wherever people can go, and it should also embody possibilities that transcend time and space. Such a humanoid robot will, of course, be equipped with the same motor functions as humans, but the eventual goal is for it to surpass human capabilities.

Although the external styling design takes the human form as a motif, opinions differ on whether it is most appropriate for the robot to be given a realistic human appearance. It should go without saying that human beings should remain in the dominant role. The relationship between human beings and humanoid robots, which are machines, must remain comfortable, without being either too close or too distant.

With ASIMO, our basic approach was to avoid a rote anthropomorphism while creating a smooth and simple form that emphasizes a sense of integration and that avoids any superficial features. This approach would utilize the styling approach we had cultivated in our work on automobiles and make it part of the mobility design. We also eliminated any intimidating or fearsome aspects, and pursued a styling design for a cheerful, enjoyable approachability that would be acceptable to anybody.

3. Styling design development

3.1 Preliminary studies

Before beginning the ASIMO styling design, we examined its size and proportions from the perspective of friendly appeal, which is one of the ASIMO concepts. In light of that data, we also used computer graphics (CG) to create walking animations in order to develop preliminary design proposals.

3.1.1 Size

As shown in Fig. 3, we built mockups that more or less replicated Honda's previous humanoid robot, the P3, in reduced scale, and studied their fit and feel in actual living spaces. We also took accessibility to the environment and functionality into consideration, and settled on a height of 120 cm⁽²⁾ to give a sense of approachability within a human living space. In addition, the selection of this size eliminated any potential intimidation that might be caused by the robot when in motion.

3.1.2 Proportions

The proportions were studied in two ways. One method was to try changing the overall balance using CG. The other method was to try different sizes of heads on the mockups. The P3, which is 160 cm high overall, is close to the ideal proportion of eight head-lengths for overall height, but when

the mockup modeled on it was reduced in size, the proportions did not seem quite right. However, when only the size of the head was changed to create a body with a height of approximately six head-lengths, the result gave a sense of approachability. The proportions were ultimately those of an elementary school first or second-grader. The importance of proportions was confirmed in the course of studies using 3-dimensional models as mockups, in addition to this study of synthesized images made using CG, as shown in Fig. 4.

3.1.3 Preliminary styling proposals

The P2 and P3 models were a development of Honda's image of a humanoid robot, which was clean and simple and built on the previous astronaut image (Fig. 5). Based on this styling design, a walking animation (Fig. 6) was created by CG using P3 behavior data. This allowed visualization of the possibility, though only as a screen image, of a compact robot walking together with the P3.

3.2 Idea development

The styling concepts from the preliminary studies and the previous data on proportions were used as the basis for hand-drawn or CG image ideas developed by a number of stylists (Fig. 7). The proposals in CG form were not only realistic with static views, but they could also be presented in motion. These images therefore carried considerable persuasive force and had an impact even at the initial stage of development.

3.3 Transitional proposals for mockups

Styling proposal A, shown in Fig. 8, addressed the issue of friendly appeal, which was a major issue in the initial planning. By contrast, styling proposal B, shown in Fig. 9, went in a more neutral, simple direction. These proposals were put together, and both were taken to the next step of mockup model creation (Fig. 10). The aims of proposals A and B are as shown below.

Aims of styling proposal A:

- Presentation of a cheerful, enjoyable atmosphere that would be acceptable to anybody
- Embody the motif of a child that expresses cuteness, buoyancy, and a sense of future potential
- Create a facial expression to allow eye contact and enhance communicative qualities

Aims of styling proposal B:

- Instantly recognizable as being in the Honda humanoid robot lineage
- A sense of unlimited application possibilities embodied in a simple, neutral image
- Pursuit of the functional beauty of a machine

3.4 Styling design clinic

Obviously, as a completely new product, the humanoid robot does not yet have an established market. Therefore, image surveys were conducted concerning ways it would be used in the market, and styling design clinics were also conducted to hear views inside and outside the company.

Models of both styling proposals were built as 1:1 mockups, and CG walking animations were presented. The following results were obtained:

Reactions to styling proposal A:

- Very cute, approachable, and good communicative quality
- Distinctive mascot-like character means reactions are swayed by personal taste
- Possibility that its image and uses will be limited
- Image as the Honda humanoid robot is weak
- Seems to contradict the typical robot image

Styling proposal A will work best when the product use and purpose are narrowed down.

Reactions to styling proposal B:

- Cute despite having an industrial design (ID) look, and it has a strong sense of friendly appeal
- Acceptable to many people, and conveys good basic quality
- Gives a feeling of embodying implicit potential and universal use
- Anyone who looks at it will recognize it as Honda's humanoid robot
- Has a fully realized robot quality

Styling proposal B is acceptable to many people, and it quickly conveys the image of a Honda humanoid robot.

3.5 Final styling proposal

Opinions diverged regarding styling proposals A and B. Overall, however, a vigorous discussion was carried out that gave a renewed sense of the heightened attention directed at humanoid robots. With regard to styling proposal B, in particular, the opinion that "it is a cute humanoid robot despite its industrial design (ID) approach," made a strong impression. We confirmed that the goal of friendly appeal could be fully realized by size, proportions, and simple facial structure even without making an overt point of the robot's approachability.

In addition, the kinds of TV commercials and events shown in Fig. 12 provided opportunities to display the P3 to many more people. The image of the Honda humanoid robot was in the process of taking shape. Given these circumstances, among other factors, the styling was converging on styling proposal B and we reached the point of the final proposal. The course of events is shown in Fig. 13.

3.6 Development of variations

The styling design clinics were also used as an avenue for future studies regarding diverse needs. The final proposal was set as the basic model. This effectively allowed the possibility for a broad variety of choices that could be offered by changing the head design and the color coordination without altering the basic body. Thus, variations of the model were built by the methods shown in Fig. 14.

The styling design development of ASIMO took place as described above. The characteristic design features are shown in Fig. 15.

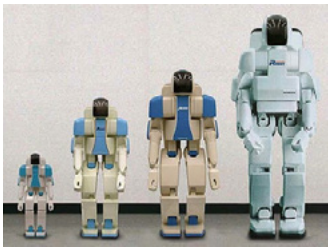


Fig. 3 Size study (Mockup)

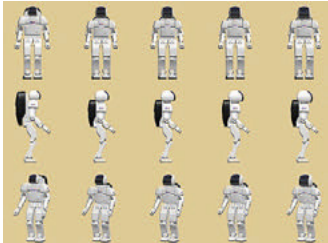


Fig. 4 Proportion study (CG)



Fig. 5 Styling design study (CG)

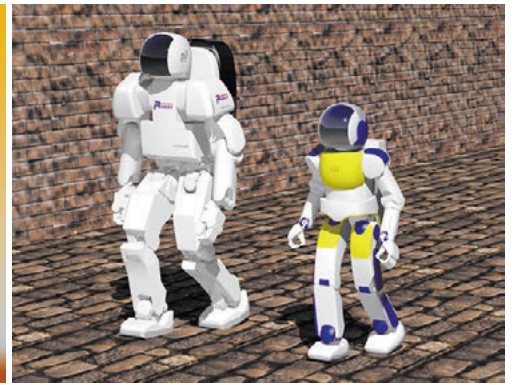


Fig. 6 Styling design study (CG animation)

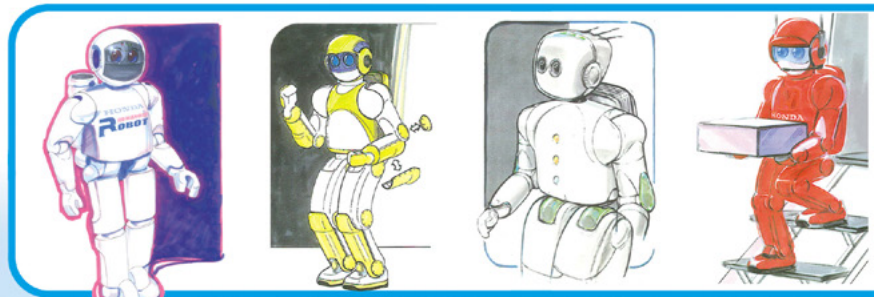


Fig. 7 Styling design ideas (Sketch and CG)

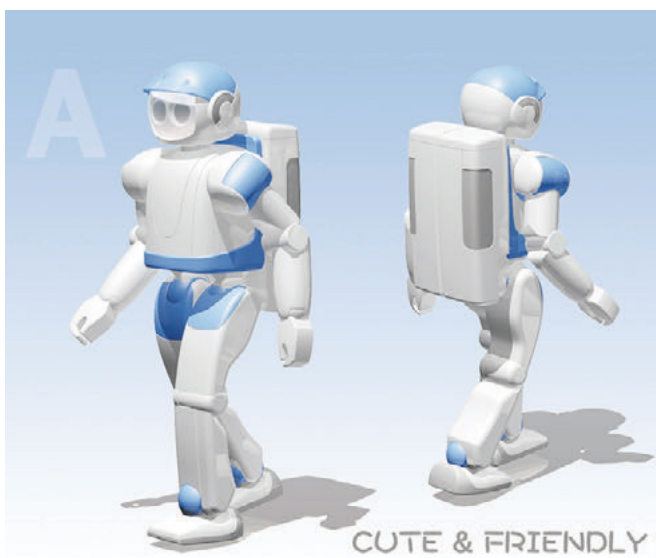
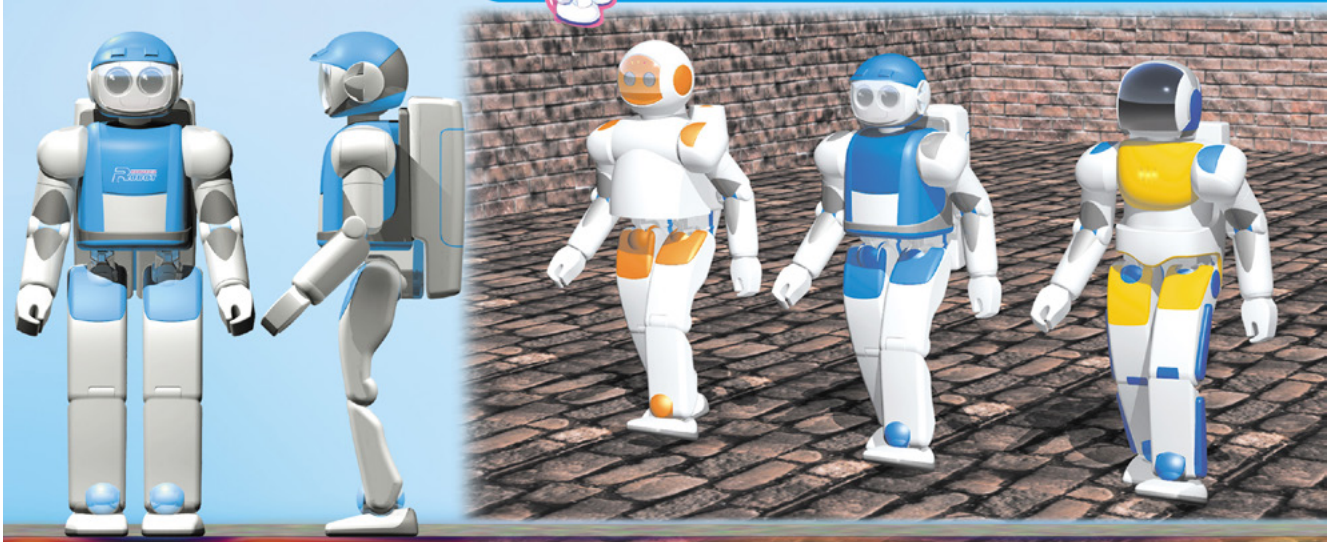


Fig. 8 Proposal A (CG)

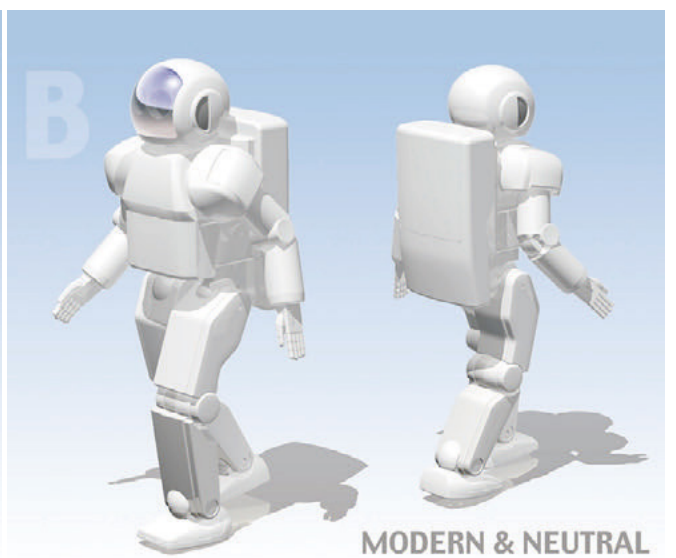


Fig. 9 Proposal B (CG)

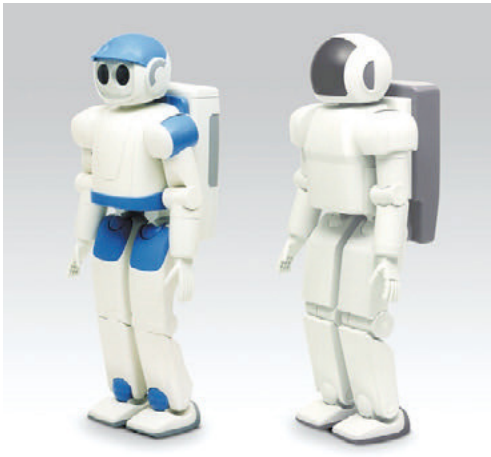


Fig. 10 Proposal A, Proposal B (Mockup)

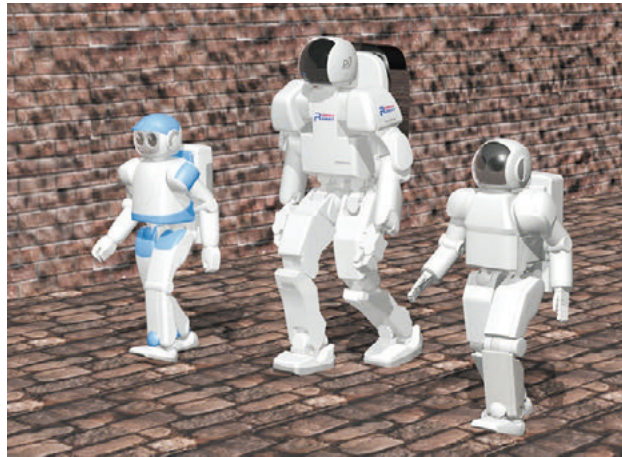


Fig. 11 Proposal A, Proposal B (CG animation)

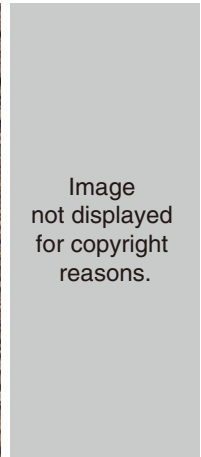


Fig. 12 P3 image

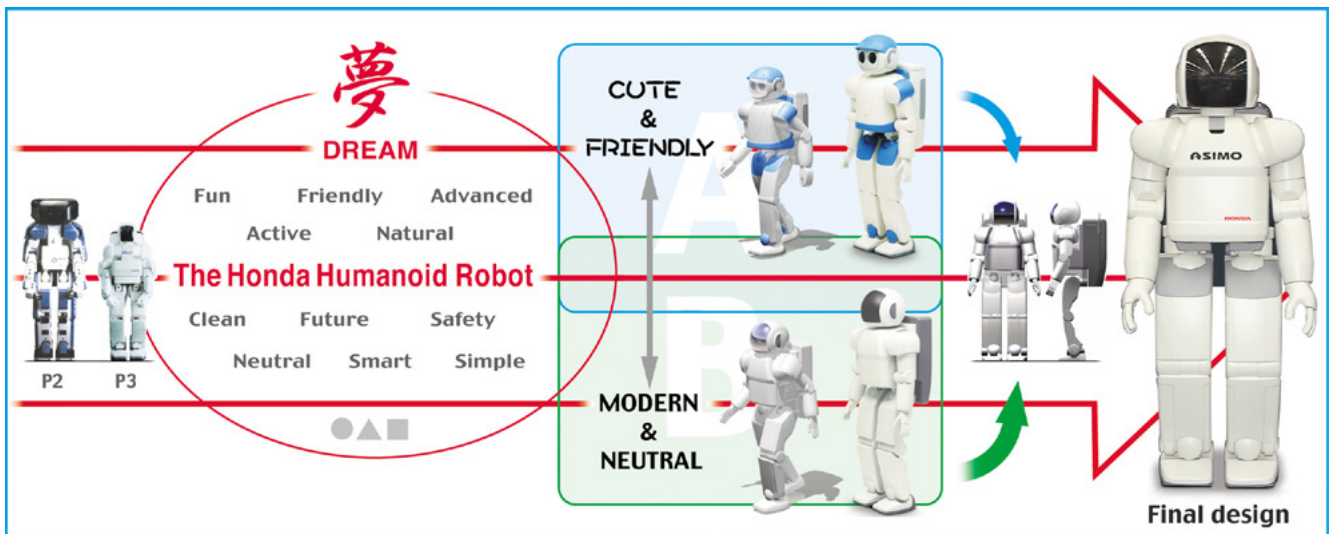


Fig. 13 Styling design process

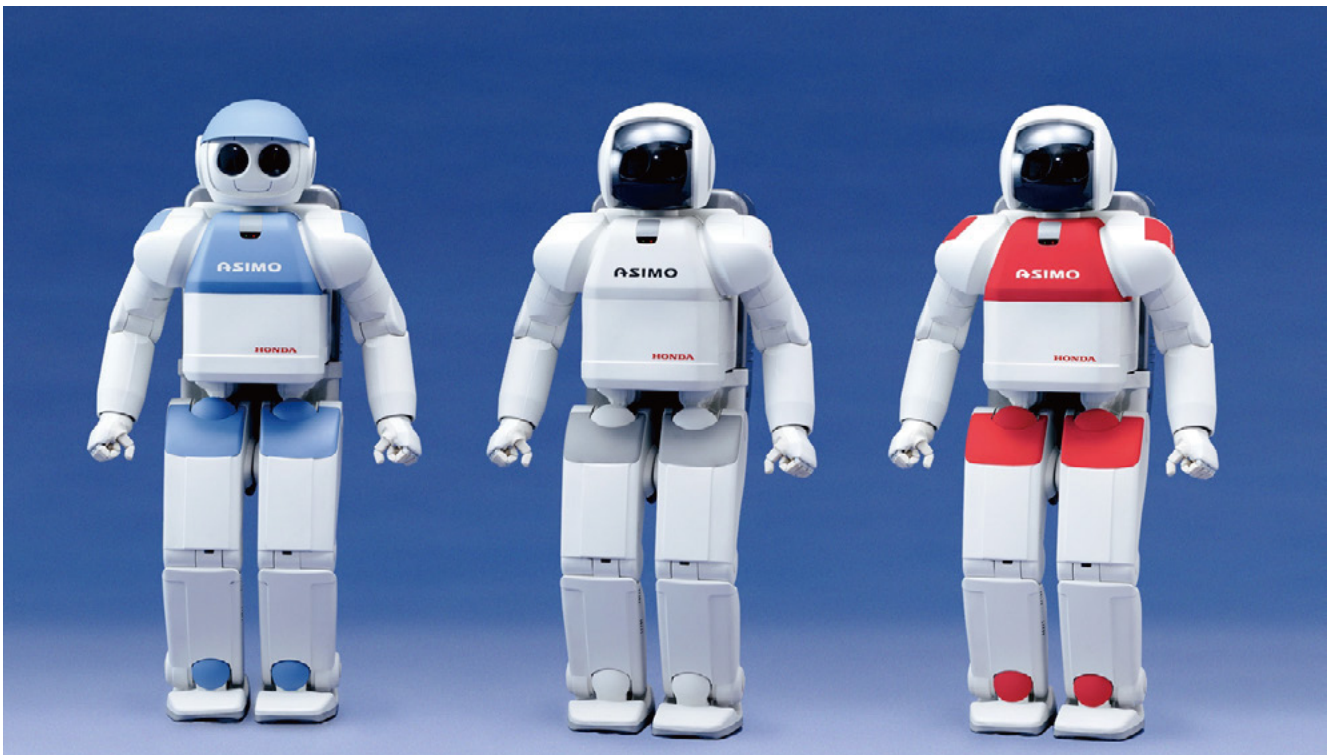


Fig. 14 Variations of the final styling design Left: Face and color variation model, Center: Basic model, Right: Color variation model

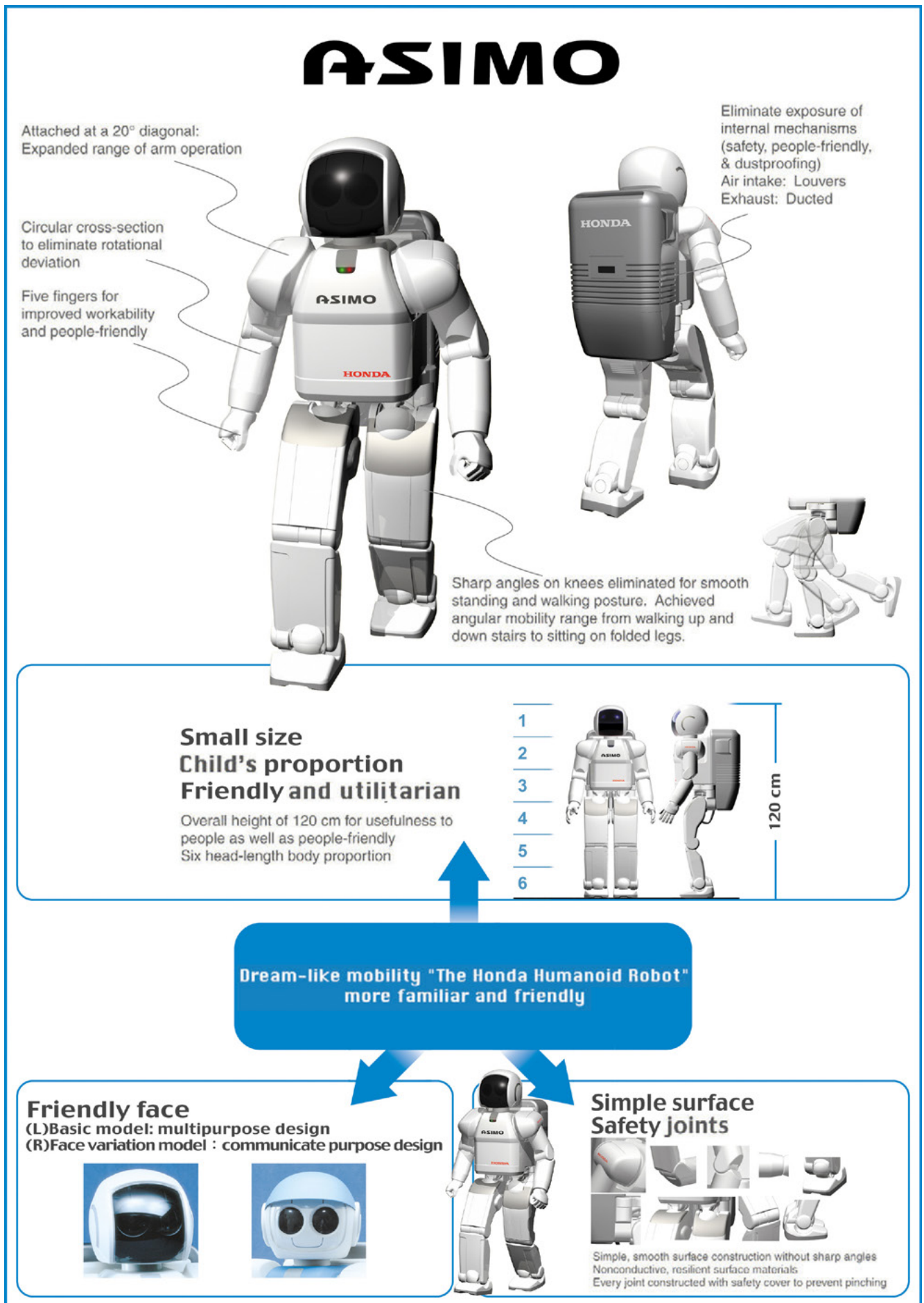


Fig. 15 Concept and final styling design

4. Styling design utilizing computer technology

The major difference in construction between a humanoid robot and an automobile, as shown in Fig. 16, is that the robot has more observable complex movable parts. An automobile receives its basic character through packaging design. The surrounding body is designed to be aerodynamic, and the overall automobile is expressed as a basically monolithic mass. Moreover, the constituent elements are not very different, and there is a limited number of openings used for access. The movement of doors and other parts is in two dimensions, and in general, parts with different shapes are only rarely built up from elements in ways that involve complex movements.

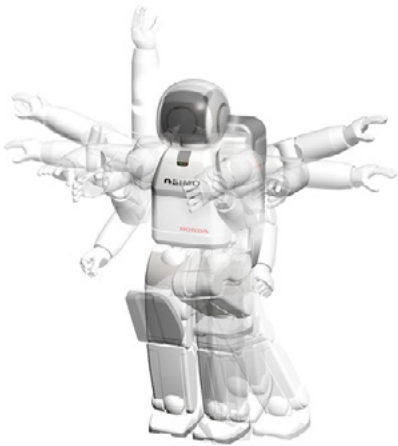


Fig. 16 Movable parts in the humanoid robot

In contrast, the humanoid robot has multiple movable parts. (The ASIMO body has 26 degrees of freedom.) The styling design had to establish the mechanisms and the overall form while envisioning the diverse states of these many movable parts. In this sense, too, it had already become too cumbersome to provide adequate simulation by means of hand-drawn sketches alone. For this project, the stylists themselves used computers directly to carry out the styling design, and doing this yielded significant results under the four headings below:

- (1) Increased speed and accuracy through coordination with engineering design
- (2) Styling design that takes dynamic states into consideration
- (3) Effective presentation
- (4) Smooth transition to 3-dimensional form

4.1 Increased speed and accuracy through coordination with engineering design

The styling designers received study data provided by the engineering designers. They then used Alias software, which has seen widespread use in automobile design, to develop their styling ideas in the form of 3D data while taking engineering design requirements into consideration.

The engineering designers, on the other hand, received 3D data from the stylists at an early stage and used CATIA to design structures and mechanisms. This made it possible

for them to incorporate styling design at the stage when they are engineering the skeletal structure, something that used to be more difficult to do.

As shown in Fig. 17, the styling designers and engineering designers proceeded with development through the shared language of data. This made the exchange of information smoother and served to raise development efficiency. As a result, there was an increase in speed and accuracy in the move toward project completion while compatibility was maintained between styling design and mechanisms.

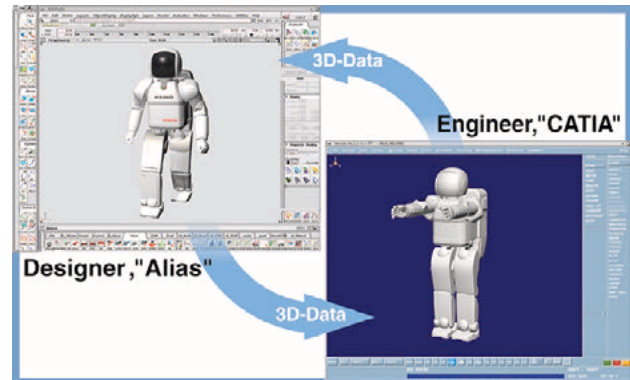


Fig. 17 Digital styling design

Fig. 18 contrasts wire-frame data, a CG image, and an actual robot. The styling design process is not simply about creating shapes and ideas. It is vital that the styling reflect a variety of requirements while creating a simulation of the final form. As shown clearly in this figure, we were able to represent a finished form realistically and very accurately from the initial stage by developing a data-centered styling design in coordination with the engineering designers.

4.2 Styling design that takes dynamic states into consideration

As mentioned earlier, the humanoid robot requires styling design that takes into consideration not only its static states, of course, but also its moving states. Up to the P3, styling designers and engineering designers had no technique for confirming moving states until the actual robot was completed and walking. Styling design of ASIMO proceeded while also making use of computers to envision the robot's moving states. This design work was carried out in ways that made the robot's posture when walking, in particular, more easily visible. The walking posture is not subject to improvement by styling design alone. However, as shown in Fig. 19, we worked out ways in the styling design to use shape processing on every part possible, in other words, to draw in the chin, to straighten the back, to throw out the chest, to minimize the bend in the knees, and so on. The engineering designer in charge of robot behavior also wrote a program to make the robot walk more energetically and with better posture. It is possible to verify that the ASIMO walking posture is improved relative to the P3.

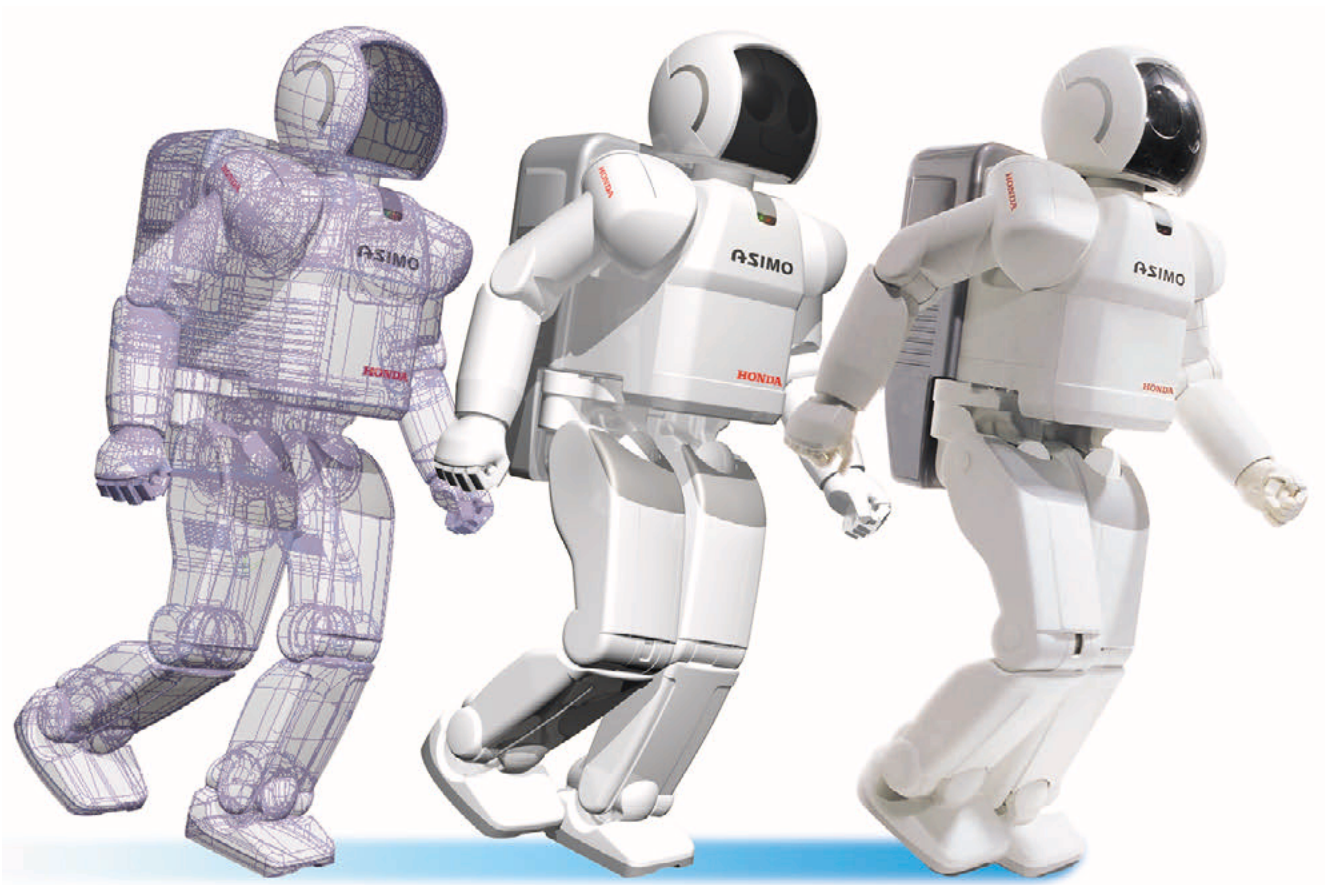


Fig. 18 Digital styling design process

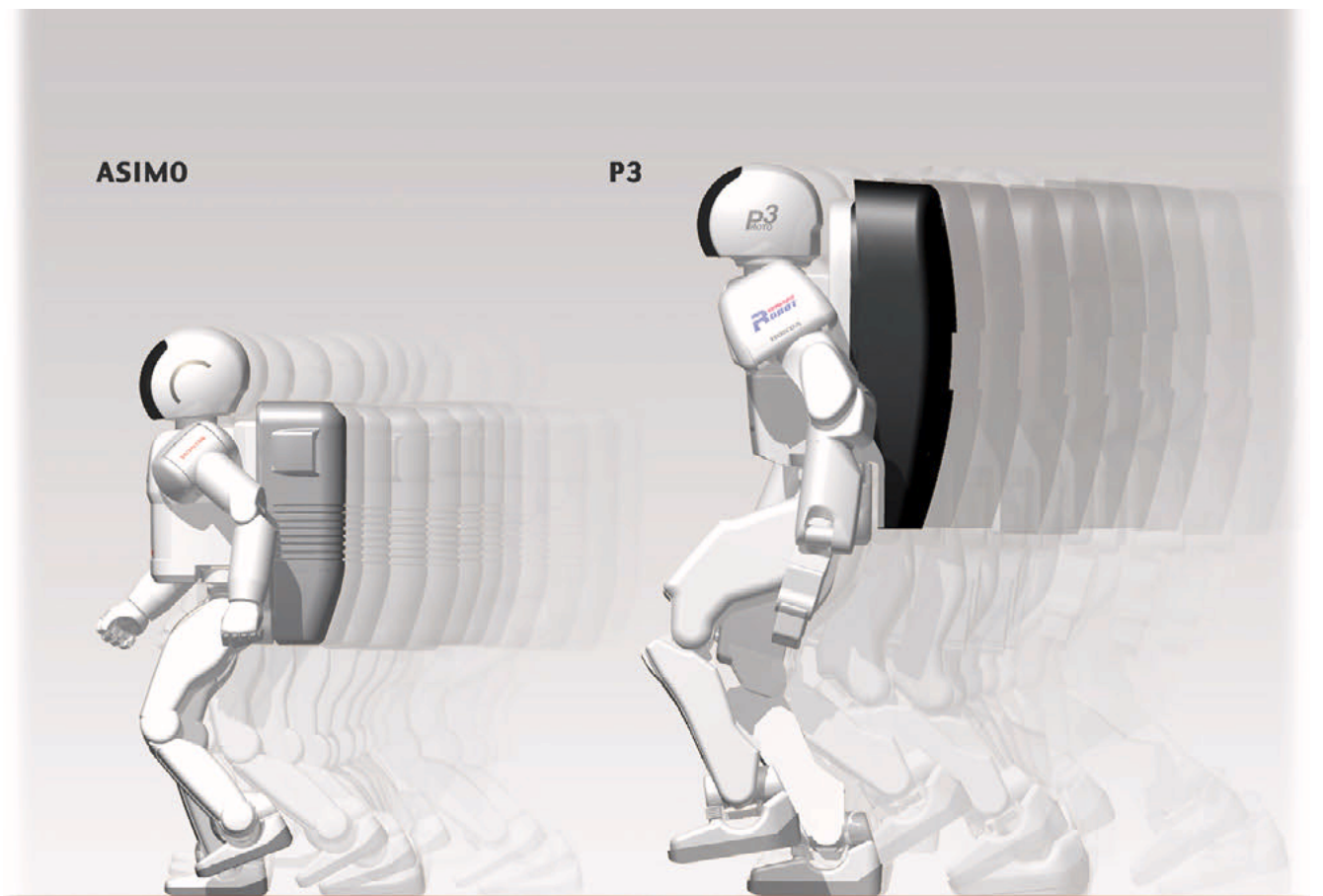


Fig. 19 Dynamic robot walking styling design

4.3 Effective presentation

A three-dimensional image projector used by automobile designers to check their data was used to project CG images of ASIMO. We thus verified images of walking and other behavior in nearly life size. As a result, it became apparent that the walk lacked sufficient energy, so that program was improved and in the actual robot we achieved an ASIMO that has a cute, energetic walk. Such applications of CG made it possible to expose problems at an early stage.

Fig. 20 shows ASIMO's distinctive bowing gesture. For this, too, we used the CG image on the left to generate the proposal, and it was reflected in the gesture performed by the actual robot, as shown on the right.

We also created images of different uses for ASIMO in many different settings (Fig. 21). This allowed us to give

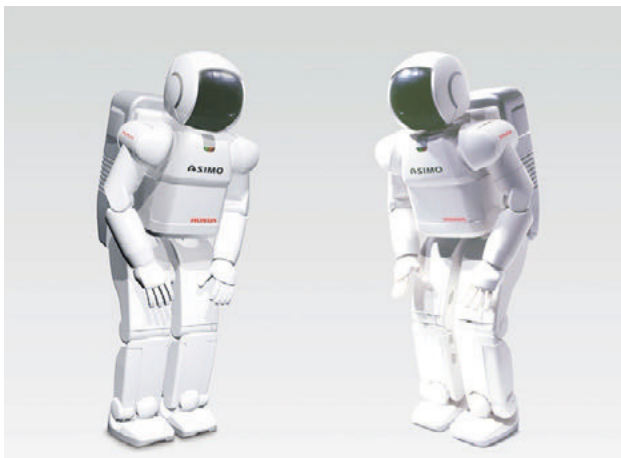


Fig. 20 Dynamic robot bowing styling design

more effective presentations. These same images ended up being used at the time of the public announcement with only some partial adjustments being made.

There was not enough time for taking photographs and creating printed press materials for use when the public announcement was made. However, use of CG screen images made it possible to handle Web pages, technical information, press materials, advertisement formats for use in sales promotion (Fig. 22), presentations at exhibition sites (Fig. 23), and other media presentations, in a short time frame.

4.4 Smooth transition to 3-dimensional form

As explained above, this project used CG for styling development methods that yielded many results. Ultimately, however, the final styling verification has to be done with full-scale models that provide a sense of size and presence so that the styling designs can be judged by their immediate impact. The last stage in creating a 3-dimensional form also involves final finishing work done by a seasoned hand. Those measurement data are reflected in the final data. At the present time, therefore, the creation of 3-dimensional models is an essential process.

The fabrication of clay models like those well-known in automobile design development took place up to the development of the P3. With the ASIMO development, however, verification of the shape was done on the CG screen and with mockup models, and the result was fed back to generate data for the following process. The humanoid robot design project required prompt and flexible responses to engineering design changes and the study of designs in motion. Therefore this method is more efficient.

Image not displayed for copyright reasons.

Fig. 21 Image movie



Fig. 22 Advertisement format

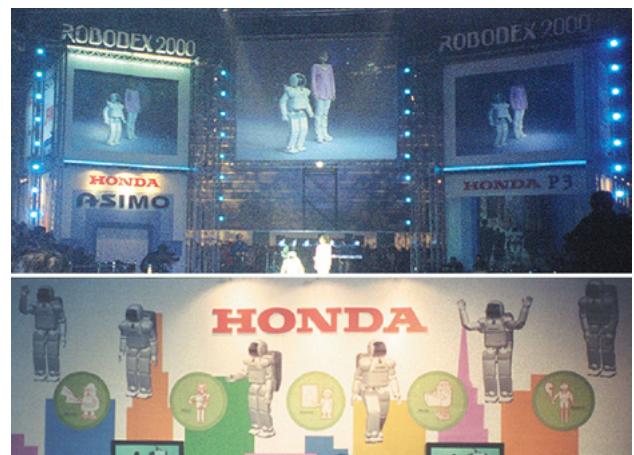


Fig. 23 Presentation for ROBODEX 2000

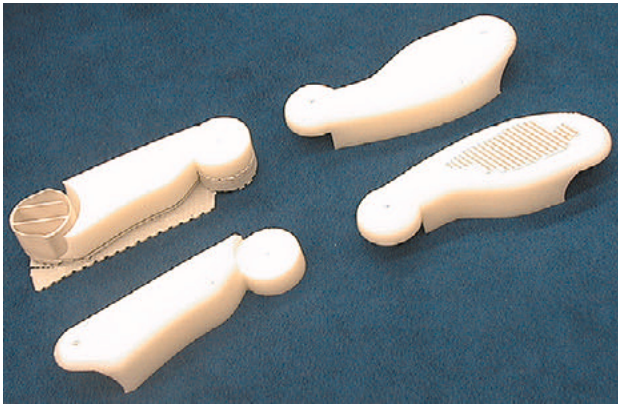


Fig. 24 Rapid prototyping (FDM)



Fig. 25 Rapid prototyping (SLA)

In the interest of making mockup models lighter and easier to fabricate, different fabrication methods were assigned to different parts. The main body was made by working resin with a milling machine. The head, arms, legs and other such hollow, enclosure-like sections, as shown in Fig. 24, were fabricated by the fused deposition method (FDM), which involves lamination of melted ABS. Stereolithography (Apparatus) (SLA), as shown in Fig. 25, involves directing a laser beam onto liquid resin that hardens when exposed to ultraviolet light in order to build up the part by lamination. These and other such methods were combined for greater efficiency and speed.

5. Summary

From the initial stages of the ASIMO styling design development project, we utilized the maximum extent of computer technology. The design data was generated by building on styling image proposals. The final verification was also carried out using 3-dimensional 1:1 mockups. In this way, the advantages of both digital and analog methods were used effectively together to further the project. As a result, a high-order of compatibility was maintained between styling design and mechanisms, and we were able to realize the initial aim of a more familiar, approachable design that has a novel sense of value.

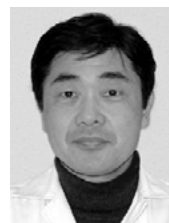
6. Conclusion

Let us share something that happened at a styling design clinic. Thinking nothing of it, during the clinic, we casually removed the head from the mockup so it could be studied, when, all of a sudden, one of the participants screamed in shock. This illustrates the anomalous character of the humanoid robot, which has the form of a human being even though it is a machine. Humanoid robots should be designed not just because such robots would be handy to have around, but because their presence strikes a chord in the human spirit. The humanoid robot is assigned parameters that cover a broadened deep range. Clearly, it is expected to become something more than the typical industrial product of the 20th century. Born at the end of the 19th century, the automobile underwent striking development as a vehicle that is useful to people. It may be that the humanoid robot, like the automobile of the past, will have both successes and failures, bringing both joy and sadness as it evolves. Nevertheless, we are committed to this development effort, and we realize as we proceed that the creator's responsibilities are by far greater when the product is such a familiar presence in people's lives.

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- (1) Sony, AIBO public relations materials (1999)
- (2) Hirose and Takenaka, "Development of the Humanoid Robot ASIMO," Honda R&D Technical Review, Vol. 13, No. 1, p. 1-6 (2001)

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ABSTRACT

In developing the new model Civic, high performance targets were set for comfort, efficiency, and style based on the overall concept of a “comfortable life,” as well as for collision safety, driving stability, and noise and vibration. The engine and chassis have a compact design to ensure a wide cabin space relative to the vehicle’s outer dimensions, and a flat floor was achieved with a centralized under-floor layout for the exhaust and fuel systems, thus achieving a “comfortable space.” In addition excellent fuel economy of 19.4 km/L in the 10-15 mode was achieved, and the Civic also obtained J-LEV certification. Further, the high performance targets were achieved in the areas of driving stability, noise and vibration performance, and collision safety performance.

1. Introduction

Since the first generation of the nameplate, Honda’s concept for the Civic has been “an advanced compact car for the people.”

Twenty-eight years after the first Civic was introduced, many changes in society have resulted in numerous changes in the role of the automobile. As we look toward the future, we must ask ourselves: What will constitute “advanced?” What will signify “performance for the people?”

While passenger vehicles in general have not changed much over the past 10 years, a new type of vehicle, the sport-utility vehicle (Light Truck), has gained much popularity. We decided that this trend signified a growing desire among the car-buying public for a more comfortable, useful automobile that was suited to their lifestyle.

We therefore constructed a concept that focused on what was essential in a lifestyle-oriented vehicle; which is different than a sport-utility vehicle that offers performance that goes beyond what is needed in everyday life. In addition, we continued to focus on collision safety performance.

2. Concept and development objectives

One conclusion reached by our development team was that consumer values were changing toward an emphasis not on the specific pluses and minuses of a given product,

but on whether the product improved one’s overall quality of life. This includes consideration of the utility of the product. Therefore, we established the concept of a “comfortable life” as one of our main aims. In terms of achieving this aim, we realized that a car should provide “comfort,” “efficiency,” and “stylishness” as indicated in Fig. 1.

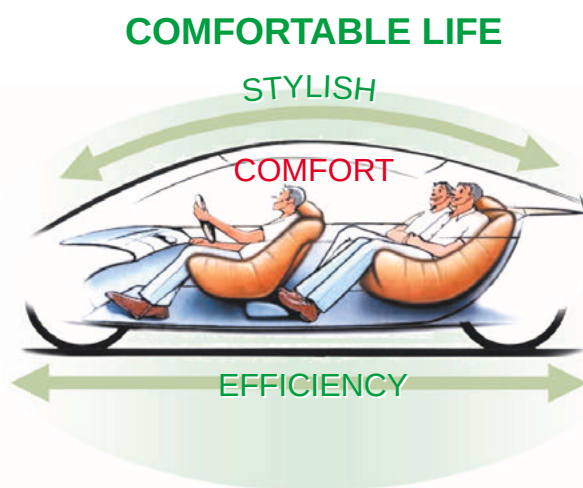


Fig. 1 Product concept

* Honda R&D Europe (UK) Ltd.

** Tochigi R&D Center

- **Comfort:** We decided that the car should be spacious enough for four adults to relax in comfort, a degree of roominess not previously available in earlier cars of this class. We also sought to achieve a full flat floor and a visually pleasing space.
- **Efficiency:** We wanted to make the external dimensions as compact as possible. In order to reconcile this goal with our aim for a spacious interior, we created a new layout for the engine compartment.

We also established a fuel economy target of 19.4 km/L, which exceeds the standard of 1.5L for this class.

- **Stylishness:** We sought to achieve a sporty styling that would identify the car as a passenger vehicle, and not as a minivan or a Light Truck.

In regard to safety performance, we sought to achieve a high level of safety performance through many objectives, including but not limited to the following:

- Further advances to our pedestrian injury reduction technology.
- Achieving a four-star EU-NCAP rating.
- Satisfying internal safety requirements for car to car frontal collisions with a two-ton passenger vehicle.

3. Package Layout

In configuring the package layout, we sought to achieve both a spaciousness that surpassed that of other passenger vehicles, as well as a comfortable driving position.

We developed an efficient package layout that would accommodate four adults comfortably even though the body of the vehicle is compact. By saving space in the engine compartment and reducing the amount of space occupied by the rear suspension, we were able to extend the spacing between the front and rear seats by 70 mm while reducing the exterior length by 165 mm (compared with the earlier Civic Ferio), as shown in Fig. 2.

In addition to enlarging the luggage capacity to a level that is equivalent to or better than that of competitor vehicles in Europe, our efforts to reduce the amount of space occupied by the rear suspension resulted in a widening of the distance between the wheel wells by 142 mm and a further reducing of the protrusions into the luggage space, which improved the utility of this area.

In order to enhance visibility and to make it easier to get in and out of the vehicle, we raised the seats by 50 mm compared with the earlier Civic Ferio. However, the seats

are still 40 to 120 mm lower than those in a Light Truck, allowing us to retain a passenger vehicle-like driving position.

We kept the floor at the same height as in earlier model Civics and eliminated the floor tunnel, allowing us to achieve the world's lowest flat floor. In order to take advantage of the flat floor, we incorporated the dashboard-mounted gearshift and adopted a foot-pedal parking brake. Furthermore, we raised the ceiling in the passenger compartment by 90 mm in order to make movement inside the compartment easier. As a result, it is possible to walk between the left and right seats and between the front and back seats. The flat floor and the higher ceiling also contribute to the increased leg and head room.

4. Engine

4.1 Development aims

In order to achieve some of the development concepts established for the new Civic, we re-examined the powertrain, starting with its basic framework. Basing our efforts on a compact design suited for the vehicle package, we decided to try to improve fuel economy by improving combustion and reducing friction in the engine and by improving the efficiency of the CVT. We also decided to try to reduce emissions by adopting a lean NOx catalyst and by enhancing the exhaust system.

Our specific engine development targets were as follows:

- To achieve the best fuel economy in the class of mass market vehicles in Japan.
- To achieve low emissions (JLEV) for all grades of 2001 model year Civics. (acquire J-LEV certification).
- To develop a compact design that is 10% smaller than the 2000 model Civic.
- To achieve high responsiveness within the range of normal operation.

4.2 Engine overview and major specifications

Fig. 3 shows the engine, and Table 1 lists the major specifications. We made many changes in the layout of the intake system to produce a more compact design, including changing the positions of the components, making associated enhancements in the cylinder block, and adopting a plastic intake manifold. Furthermore, we placed the air cleaner and the intake resonator chamber above the engine and

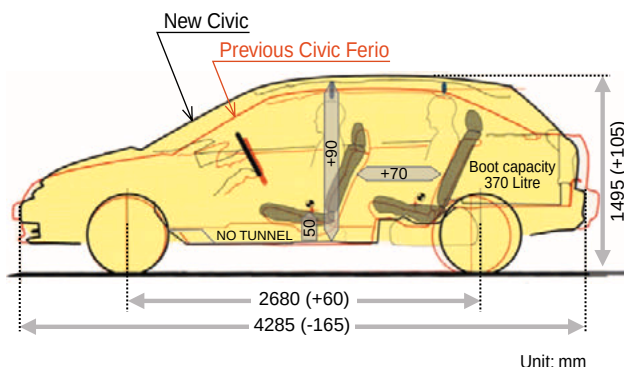


Fig. 2 Package layout



Fig. 3 View of 2001 model year lean-burn engine

Table 1 Engine specifications

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	

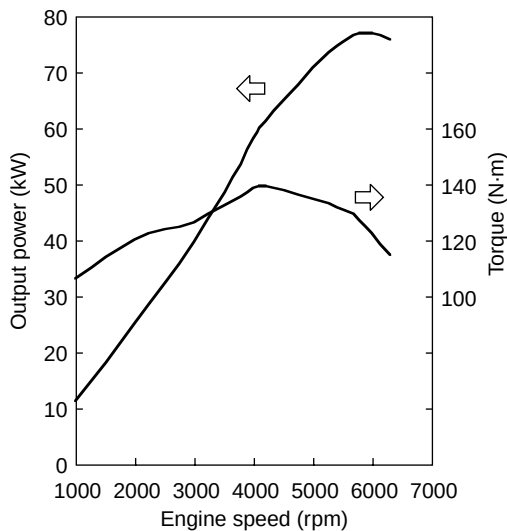


Fig. 4 Engine performance

transmission in order to make the powertrain, as a whole, more compact, while simultaneously allowing us to modularize the design.

Many of the components in the engine have been newly designed. For example, the design of the combustion chamber configuration was changed by modifying the shape of the piston crown and various parts were modified to reduce friction in each area and to reduce emissions.

Fig. 4 shows the engine performance curve. The output characteristics are set with an emphasis on low-speed torque in accordance with control of the gear ratio in the CVT. The result is a balance between high fuel economy and linear driveability within the range of normal operation.

4.3 Higher Fuel Economy Technology

In the course of developing our higher fuel economy technology, we set high objectives for ourselves in regards to the engine, the CVT, and body, as shown in Fig. 5, and we achieved a fuel economy of 19.4 km/L.

4.3.1 Positioning of mode fuel economy

Fig. 6 shows the relationship between vehicle weight and 10-15 mode fuel economy. In general, a vehicle

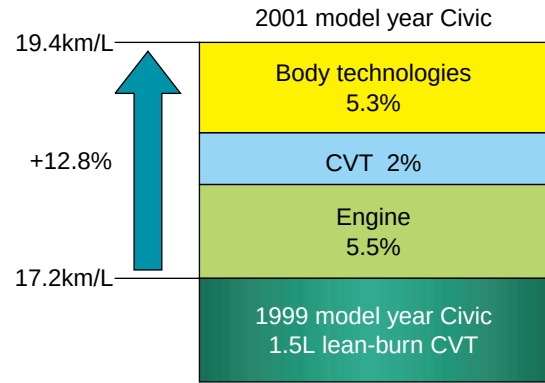


Fig. 5 Ratio of contribution to 10-15 mode fuel economy improvements

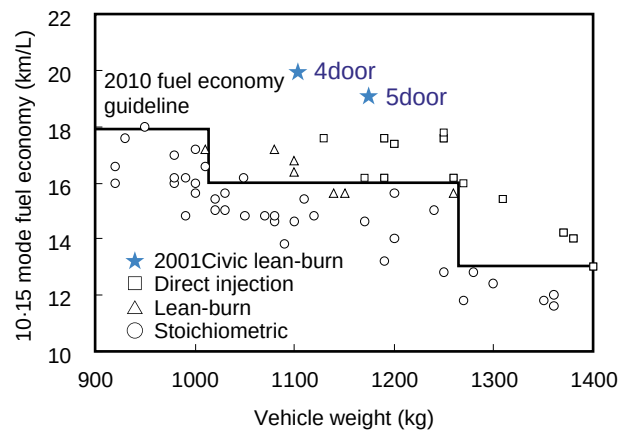


Fig. 6 Relationship between vehicle weight and 10-15 mode fuel economy in production gasoline-fueled vehicles

equipped with a lean-burn engine achieves 10% better fuel economy than a vehicle equipped with a conventional stoichiometric air-fuel ratio engine, while a vehicle equipped with a direct injection (DI) engine achieves an additional 8% improvement in fuel economy. However, a Civic equipped with our newly developed lean-burn engine achieves an 8 to 12% improvement in fuel economy over a vehicle equipped with a DI engine.

4.3.2 Means of improving engine combustion through intake port and combustion chamber design

In order to stabilize and combust the lean air-fuel mixture that flows into the cylinder, it is necessary to create an easily ignitable air-fuel mixture in the vicinity of the spark plug and then rapidly spread the combustion to the surrounding lean mixture. In order to accomplish this, it is necessary to generate a good balance of swirling and tumbling within the cylinder. The swirl intake port that we developed for this engine produces a stronger tumble component than the conventional port shape. As shown in Fig. 7, the shape of the intake port that we developed produces a swirl ratio of 2.6, a tumble ratio of 2.0 and a swirl slope [alpha] of 37 degrees. In experimental results with 1.0L to 1.8L lean-

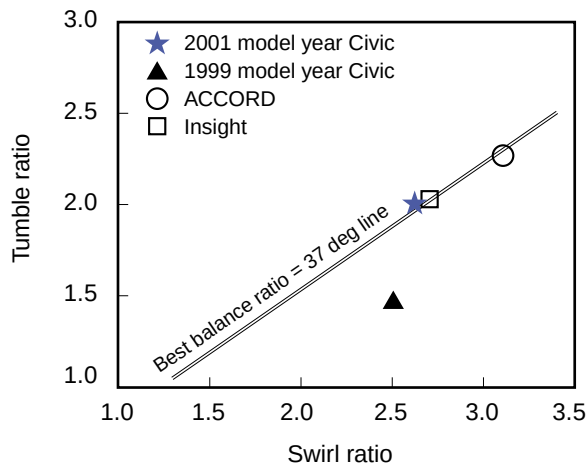


Fig. 7 Relationship between swirl and tumble intensity

burn engines, this swirl slope demonstrated a good correlation with the lean-limit characteristics, and served as a valid metric of intake port shape designs.

Along with flow within the cylinder, the build up of the air-fuel mixture within the cylinder is also an important element of lean combustion. We ran a computer simulation of the distribution of the air-fuel mixture within the cylinder for different types of intake port shapes. The conditions for our calculations were an average air-fuel ratio A/F of 22 (air surplus factor: 1.5), and a fuel injection timing of 40 degrees after the intake TDC. Fig. 8 shows the air-fuel mixture distribution within the cylinder at 40 degrees before compression TDC, which corresponds to the ignition timing in an actual engine. The pictures in Fig. 8 depict horizontal and vertical cross sections in the vicinity of the center electrode of the ignition plug. Compared with the earlier port, the new port builds up the dense air-fuel mixture in the region of the center electrode of the plug. As shown in Fig. 9, good correlation was achieved with our test results for

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

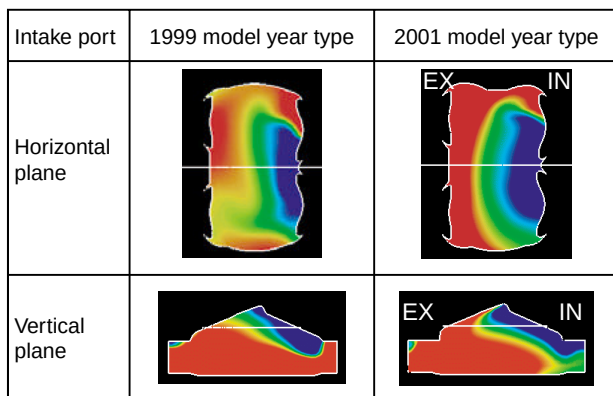


Fig. 8 Effect of intake port shape on in-cylinder A/F distribution

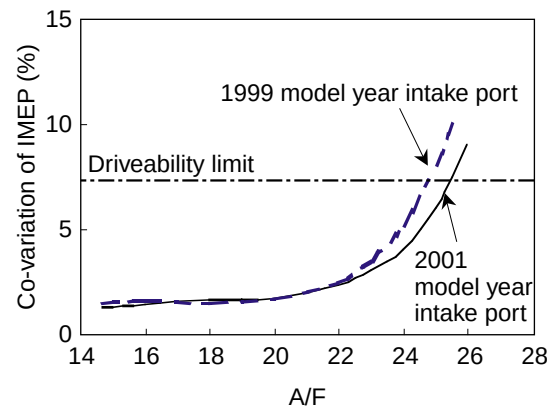


Fig. 9 Comparison of lean-limit characteristics according to intake port shape

improved lean-limit combustion characteristics.

We also ran simulations and tests on the shape of the combustion chamber. Based on these tests, we selected a bowl-shaped piston crown and we optimized the cylinder head clearance to thereby further improve output.

4.3.3 Reducing friction

In order to improve fuel economy, it is not only important to improve combustion, it is also important to reduce friction.

We adopted the following friction reduction technologies:

- Micro-dimple (MD) processing in the pistons.
- Low-tension piston rings.
- Reduction of the crank pin width.

Piston skirt MD processing creates small dimples on the piston skirt surface by shooting roughly spherical ceramic beads (with an average diameter of 40 μ m) at the piston skirt surface at speeds of up to 100 m/s. We were able to confirm that this enhances the oil film retention performance between the piston and the cylinder, resulting in a reduced piston friction loss and an increased seizure load limit.

In addition to these low friction technologies, we adopted other technologies, including a roller cam follower, reducing the load on the valve spring, and using low-viscosity engine oil (0W-20/SJ/GF2). As a result, we were able to reduce friction in the engine as a whole by 12% compared to earlier models.

4.3.4 CVT

The CVT enables driving in the low end fuel consumption ratio (Brake Specific Fuel Consumption: BSFC) range of the engine, which is suitable for improved fuel economy. Honda has been mass producing CVTs for the Civic since 1996. In the new Civic model, we attempted to improve transmission efficiency for the new powertrain by improving the CVT oil pressure control system and reducing side pressure on the belt.

Fig. 10 shows a comparison of the driving range of the newly developed CVT versus the 4AT. The engine operating range is kept in the low rpm range and controlled to operate in the low BSFC range.

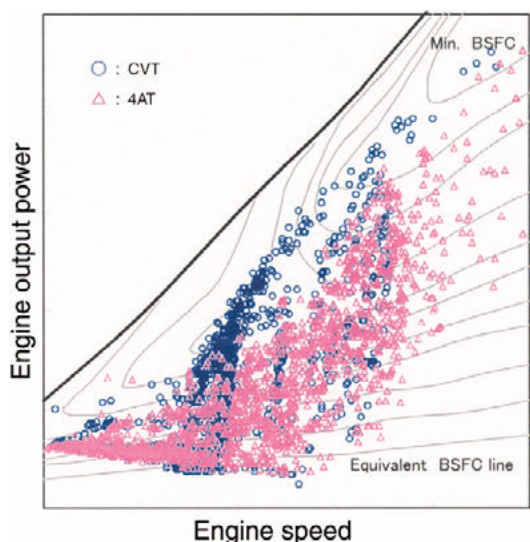


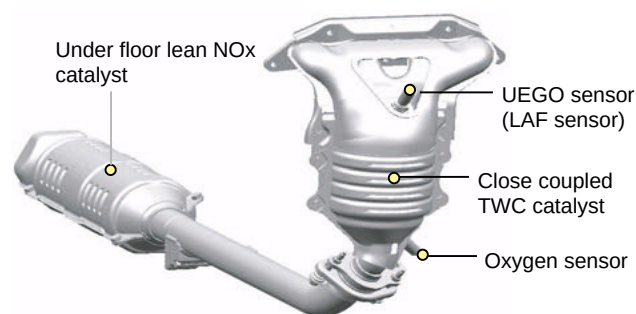
Fig. 10 Comparison of 10-15 mode driving range

4.4 Low emission technology

When we examined the emission system, we decided to set our emission target at J-LEV designation, which means emissions must be no more than 50% of Japan's requirements for the year 2000.

4.4.1 Exhaust system

Fig. 11 shows the exhaust system for this engine. In order to assure conversion performance in the cold and stoichiometric ranges, we positioned a three-way catalyst (TWC) in the close-coupled exhaust manifold; and, in order to assure NO_x reduction in the lean-burn region, we applied an under floor NO_x adsorption reduction type TWC.

Fig. 11 Layout of NO_x adsorption reduction type three way catalyst system

4.4.2 Multi-hole injector

We adopted a four-hole multi-hole type injector for the fuel injector valve. This type of injector reduces the amount of gasoline that adheres to the intake port when the spray is atomized, which resulted in a 14% decrease in hydrocarbons (HC) during a cold start, as shown in Fig. 12. This also allowed us to make the injector valve 33% smaller and 65% lighter.

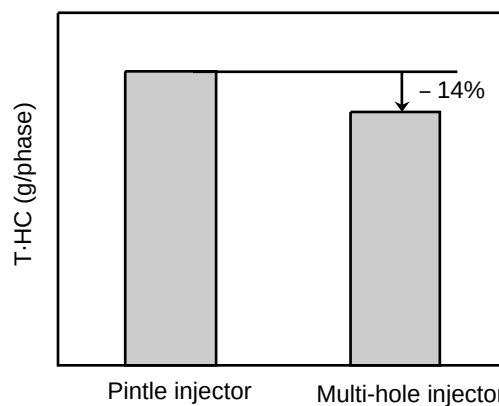


Fig. 12 LA-4 cold transient mode emission

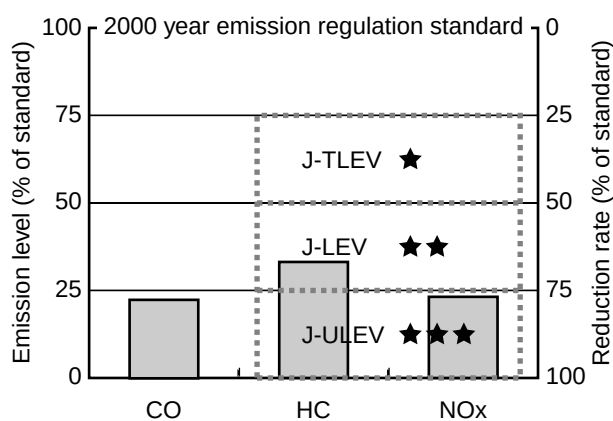


Fig. 13 11 mode emission

4.4.3 Actual emission levels achieved

In addition to the low-emission technologies that we have described up to this point, we also sought to optimize engine control when switching from lean-burn to a stoichiometric air-fuel ratio, and in the EGR range. This allowed us to attain the high fuel economy of 19.4 km/L in the 10-15 mode while also achieving J-LEV designation.

5. Transmission

We established two types of transmission concepts, the CVT and the 4AT. We then developed transmissions with excellent environmental performance (i.e. efficiency) and feeling in accordance with each concept.

In the CVT, the advantages of smooth shifting through hydraulic control of the launch clutch system and modifications of the oil pressure control system contributed to our achievement of the high fuel economy of 19.4 km/L. In addition, by combining this transmission with an engine that provides excellent torque at low speeds, we were able to attain linear characteristics in the power delivery resulting in smooth driving and a tighter acceleration feeling.

The 4AT ranges from 1.4L (EU specifications) to 1.7L/4WD. We were able to improve fuel economy without sacrificing driveability by adding a special linear solenoid for lockup control and a long torsion damper spring system.

5.1 CVT

It is possible to obtain the optimum gear ratio at all times with the CVT due to the continuously variable system of a metallic belt and two pulleys as shown in Fig. 14. Combined with the low loss launch clutch system, the CVT contributes to the high fuel economy of the Civic. We also reviewed each part of the oil pressure control system, further refining how the engine and the transmission work together, which resulted in further improvements in fuel economy.

By adopting a small linear solenoid valve, an oil pressure feedback type pulley pressure control valve, and a 3-way ON/OFF solenoid, we increased the precision of the pulley clamp pressure, and reduced the loss due to the belt-pulley interaction. This, in turn, reduces oil pump loss, thanks to the improved precision of the pulley pressure when shifting, especially in the neutral hydraulic pressure range.

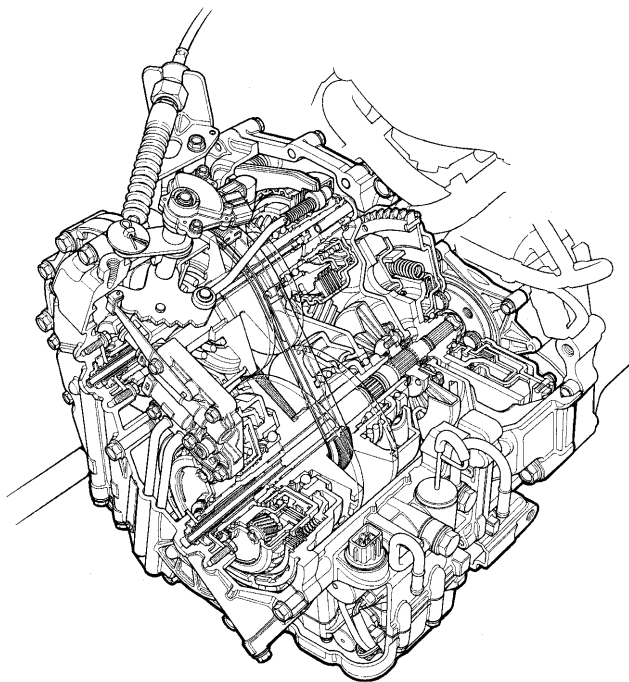


Fig. 14 CVT (Honda MultiMatic)

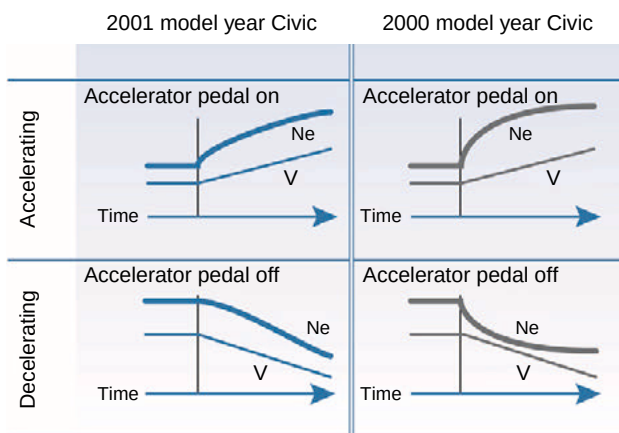


Fig. 15 Improvement of CVT shifting feel (Linear vehicle speed (V) - engine speed (Ne) feel)

We optimized the CVT's ratio transition response, the AMSS (Acceleration Map Select System) response range, and the CVT's ratio transition control by increasing the number of downhill maps for the PROSMATEC (described below). Then, by combining this CVT with a new engine that provides excellent torque at low speeds, we produced the linear response shown in Fig. 15, achieving a feeling that more closely matches the human senses.

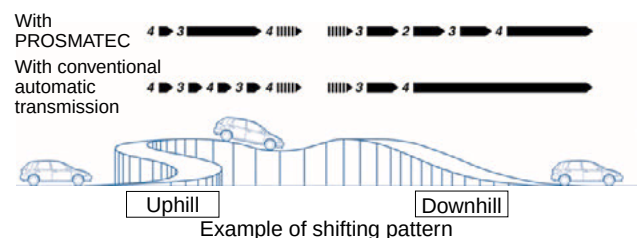
5.2 4AT

We achieved a balance between driveability and fuel economy by incorporating the active L/C system (Active Lock-up Clutch System) in the fully electronically controlled 4-speed AT.

The long torsion damper that we adopted in the lock-up clutch dampens torque fluctuations from the engine, resulting in a reduction in vibration and an expansion of the lock-up region. We were also able to reduce oil pressure fluctuations by adopting a special linear solenoid for the lock-up clutch. In addition, in the acceleration range, the system appropriately controls the lock-up feedback control in accordance with the engine torque and driving conditions, thus improving fuel economy without sacrificing driveability.

5.3 Hill climb and descent

We adopted PROSMATEC in both the CVT and the electronically controlled four-speed AT. As shown in Fig. 16, the PROSMATEC system evaluates the driving conditions on the basis of the throttle opening, the rate of acceleration, etc., and optimizes the shifting timing accordingly. This system provides for smoother driving by suppressing unnecessary shifting when going uphill, and optimally controlling engine braking when going downhill.



PROSMATEC :
PROgressive Shift schedule MAnagement TEChnology

Fig. 16 Image of shifting pattern with PROSMATEC

6. Chassis

6.1 Decreasing the engine room size

For the front suspension, we adopted a newly developed strut system (Fig. 17) that is more compact and beneficial when shortening the nose of the car. Furthermore, as shown in Fig. 18, the modularized engine design allowed us to place the steering gearbox in a higher position that provides more clearance, which in turn allowed us to position the engine farther back. The placement of the steering gearbox

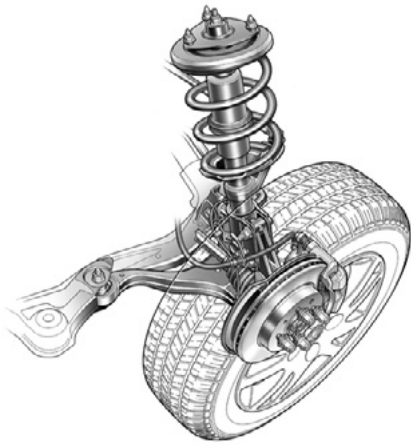


Fig. 17 Front toe control link McPherson strut suspension

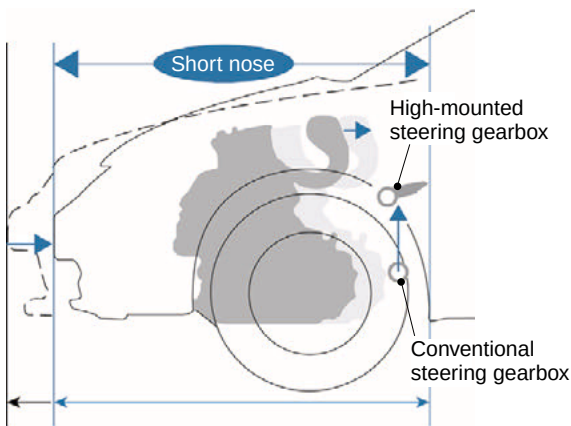


Fig. 18 Short-nose layout

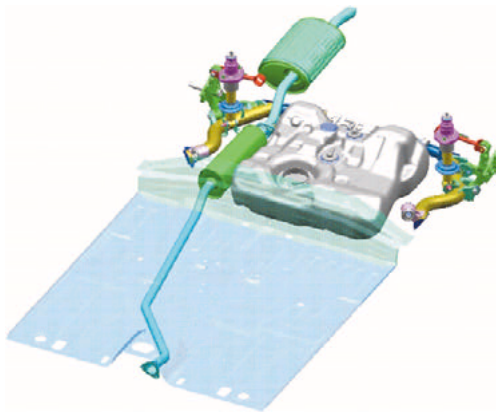


Fig. 19 Flat floor package

also permitted us to smoothly link the left and right side frames, allowing for the efficient absorption of energy from collisions. As a result of all of these modifications, we were able to shorten the nose of the vehicle considerably.

6.2 Rear floor centralized layout for a flat floor

In typical automobile design, the vehicle exhaust system is routed through a center tunnel in the vehicle floor. However, in the new Civic, the exhaust system is positioned

under the right side of the floor, and the prechamber is positioned adjacent to the fuel tank. This placement allowed us to achieve a full flat floor with no floor tunnel. For the 50-liter fuel tank, we adopted a resin mold that afforded us great flexibility in shaping the tank. Furthermore, we also made efficient use of the space under the floor, by, for example, placing the canister behind the fuel tank.

For the rear suspension, we adopted the newly-developed double wishbone suspension shown in Fig. 20. By reducing the front-to-back length of the suspension, compared to the previous Civic model design, and by positioning the rear seat farther back, we were able to enlarge the cabin. In addition, by locating the springs and dampers under the floor, we were able to minimize protrusions into the luggage compartment, resulting in more luggage space.

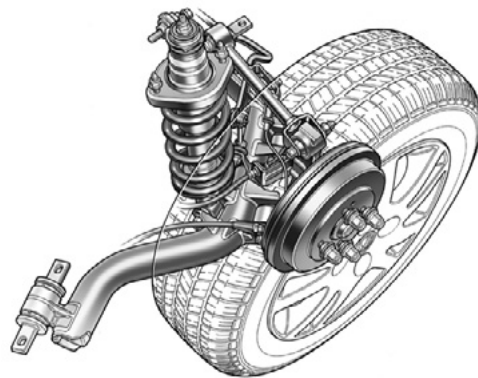


Fig. 20 Rear reactive link double wishbone suspension

6.3 Suspension

6.3.1 Toe control characteristics

As shown in Fig. 21, the long tie rods that originate near the center of the vehicle are the component of the front suspension that allowed us to minimize and straighten the toe angle change throughout the wheel stroke, providing stability at high speeds and linear handling.

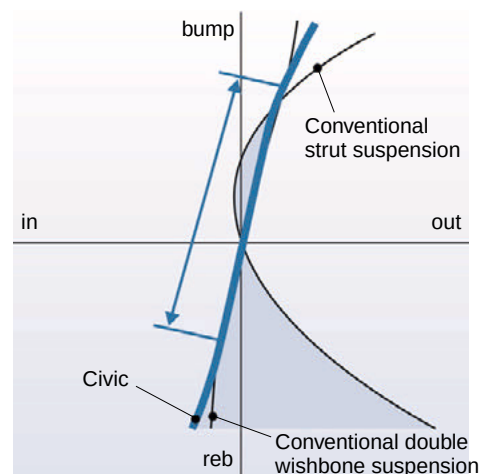


Fig. 21 Straightening the front suspension toe curve

6.3.2 Brake force steering stabilization

We adopted a reactive link that provides toe-in performance, which is effectively brake force steering, during braking. By allowing the flexing of the reactive link when brake force is generated, and as the compliance bushing (which is positioned at an angle) is displaced in direction A, as shown in Fig. 22, the trailing arm gets pulled in the toe-in direction, producing the appropriate toe-in characteristics and providing brake stability.

6.3.3 Rear ground contact rigidity

We were able to attain a high rear suspension rigidity by achieving high rigidity in the rear bushings, hub bearings, and trailing arms, in addition to the high-stiffness body. By making the car solidly rigid against lateral forces and enabling the full performance of the front and rear suspensions, we were able to achieve tightness and stability.

6.3.4. Roll posture

We reviewed the roll center height and the front-to-back balance of the roll theoretical arm length according to the suspension geometry, and assured a front-down roll axis. We were able to optimize the roll posture by linearly increasing the turning pitch angle as the turning lateral acceleration increases during cornering.

6.4 Steering

We decided to use the low power loss Electronic Power Steering (EPS). By adopting the small and lightweight pinion

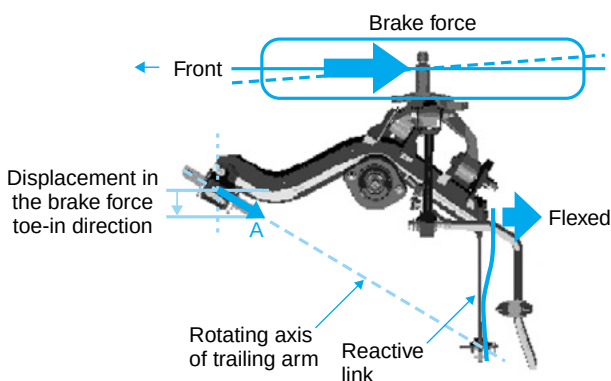


Fig. 22 Characteristic of the rear suspension under braking

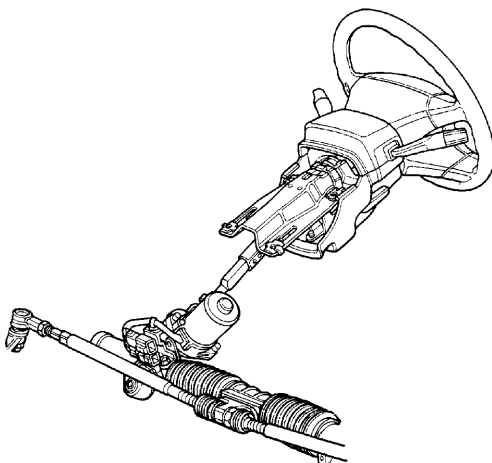


Fig. 23 EPS (Electrical Power Steering)

shaft assist type shown in Fig. 23, we achieved better fuel economy. We set the EPS control with an emphasis on enhancing the feeling of direct control while cornering and a feeling of stability at high speeds. By increasing the steering gear ratio, we achieved quicker handling, and enhanced control at lower speeds. By tuning the front compliance bushing characteristics and by mounting the steering gearbox in a higher position, we refined the aligning torque rigidity. In this way we were able to find a balance between optimizing the change in vehicle posture according to the steering angle at high speeds and achieving a high ratio, allowing us to achieve steering with a linear feel.

6.5 Braking

When we set the brake pedal ratio and the master power characteristics, we were aiming for braking that feels solid at low foot pressures. We made ABS, EBD and brake assist standard equipment on all models. ABS is the four-wheel antilock braking system and EBD is the Electronically controlled Braking force Distribution system; which controls the distribution of the braking force to the front and rear wheels when the brakes are applied, as shown in Fig. 24. Brake assist supplements the foot pressure that is applied to the brake in an emergency braking situation, as shown in Fig. 25.

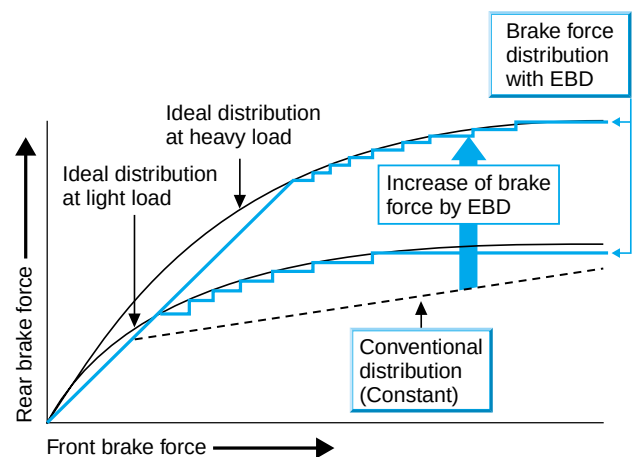


Fig. 24 Characteristic of EBD (Electronic Brake force Distribution)

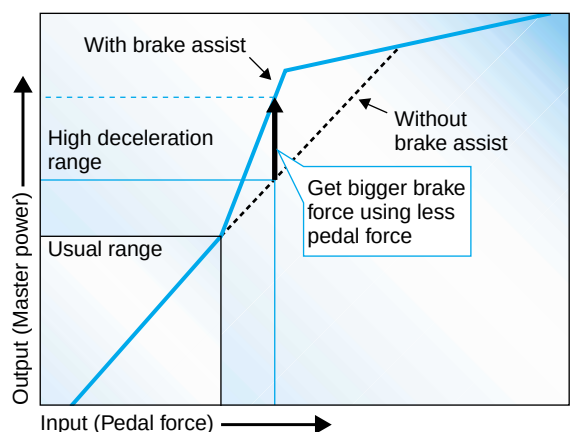


Fig. 25 Characteristic of brake assist

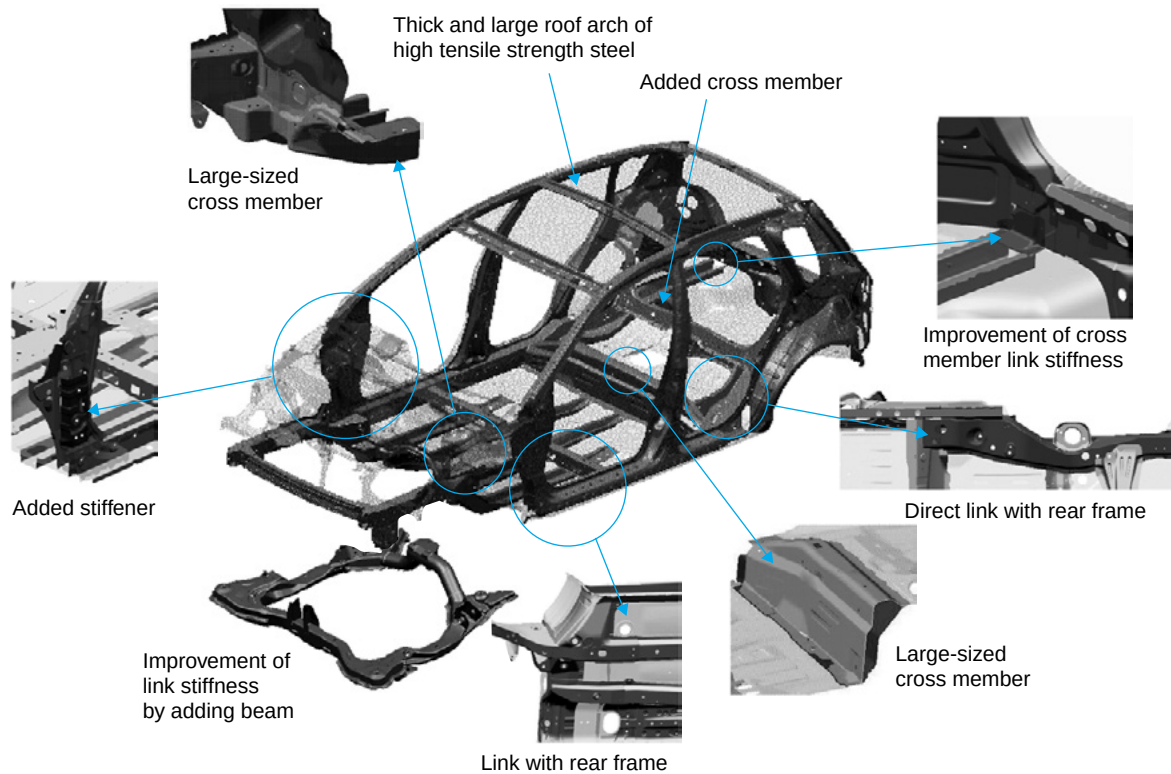


Fig. 26 High-stiffness body construction

7. Body

7.1 High-stiffness body

As shown in Fig. 26, we constructed a strong body framework based on the newly developed platform by enlarging the cross sections of each component, adding stiffeners, strengthening joints, etc. Furthermore, we enhanced the stiffness of the body by reinforcing the bulkhead, reinforcing the damper housing in the front section, reinforcing the housing between the left and right dampers and reinforcing the rear end of the frame, for example. As

shown in Fig. 27, the result of all of this reinforcement was an 80% increase in torsional stiffness compared to earlier models, as well as increases in front lateral stiffness and rear lateral stiffness.

7.2 NV performance

Based on a high-precision body structure that provided high stiffness and high energy absorption, we took steps to remedy the sources of noise and vibration and their transmission paths thoroughly. We were able to effectively reduce transmitted noise without increasing the total weight

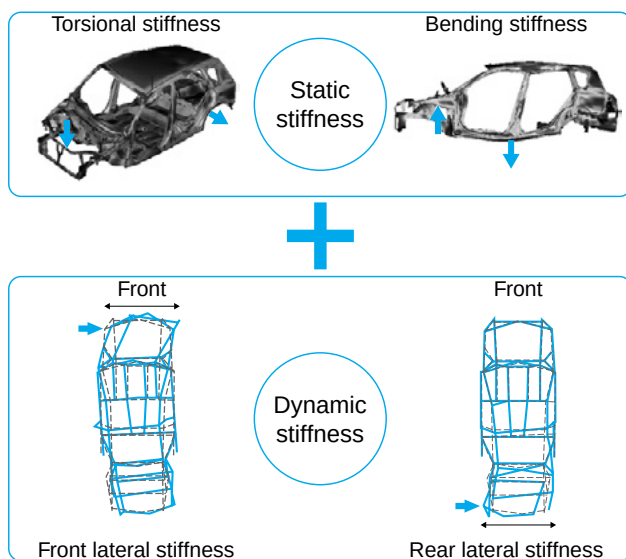


Fig. 27 High-stiffness body analysis

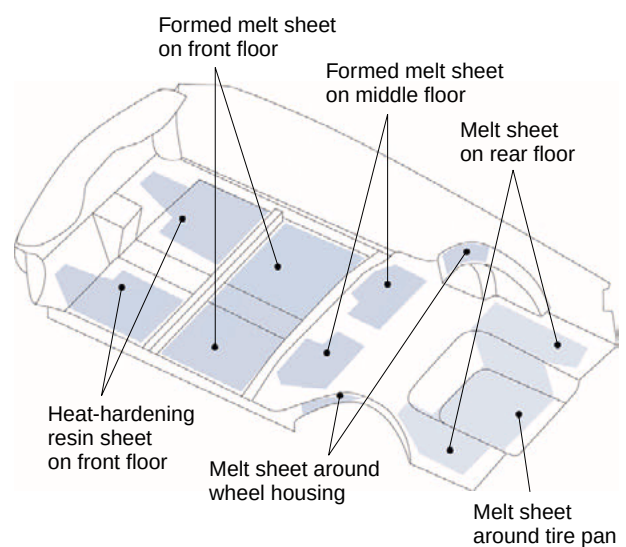


Fig. 28 Application of damping material

of damping material while still expanding the surface area over which the damping material is applied and by using it in combination with sound absorbing material. Furthermore, by effectively using soundproofing and sound absorbing materials elsewhere throughout the vehicle, we were able to greatly reduce road noise and vibration from the engine. Fig. 28 shows the locations where damping material is used, and Fig. 29 shows the locations where soundproofing and sound absorbing materials are used.

As shown in Fig. 30, we placed the engine mounts in the optimal locations on the main inertial axis, and we adopted liquid sealed type mounts for the transmission mount and the engine side mount. We also reduced the transmission of vibration by using aluminum high-rigidity brackets, etc., thereby greatly suppressing vibration from the engine to achieve a more comfortable ride.

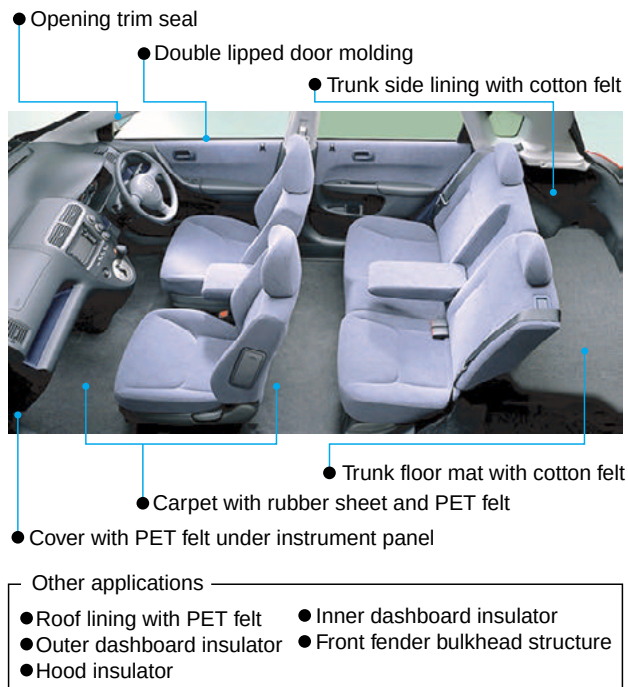


Fig. 29 Application of soundproofing/sound absorbing material

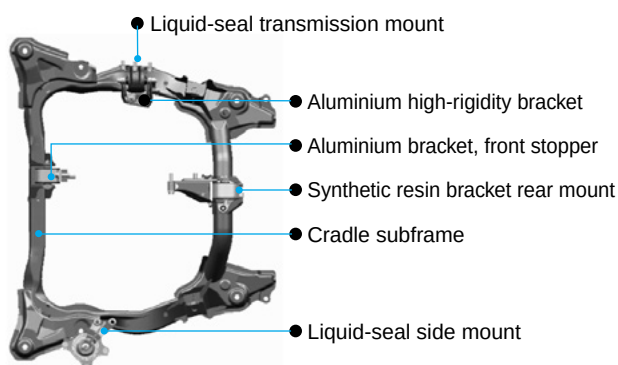


Fig. 30 Layout of engine mounts

7.3 Collision Safety Performance

As shown in Fig. 31, we decided on a front frame structure that effectively distributes and absorbs impact energy through an energy absorbing side-frame combined with a cradle-shaped sub-frame, and a floor construction that distributes impact from the sub-frame in three directions. In addition, by using high tensile strength steel for over 57% of the body components, we were able to produce a high energy absorbing structure.

In full frontal crashes, which tested the impact absorption of the design, as well as in frontal offset crashes, which further tested the protection of the passenger compartment, our design performed well in terms of reducing dummy measured injury assessment reference values and in minimizing deformation of the cabin. As a result, we achieved a body design that met our internal standard for a 55 km/h full frontal crash and a 64 km/h frontal offset crash. The body also passed tests for 55 km/h side crashes and 50 km/h rear-end crashes.

In addition, based on our analysis of collision statistics, we established a target collision of “a 50% frontal offset crash with a passenger vehicle in the two ton class with both vehicles traveling at 50 km/h.” Thanks to the efficient energy absorption and strong body construction, our design cleared this target.

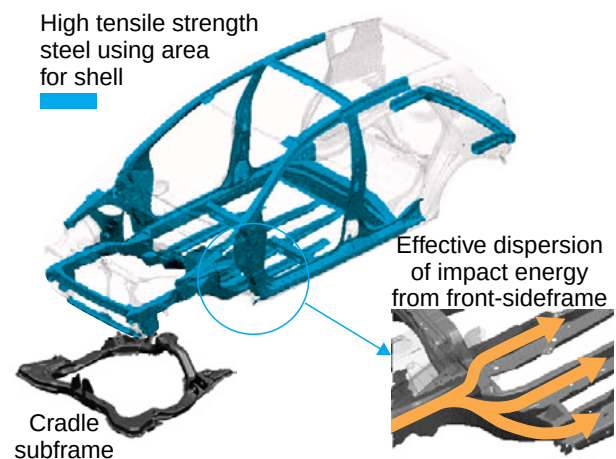


Fig. 31 High-efficiency energy absorbing shell

Table 2 Honda's G-CON technology and NCAP requirements

G-CON: Crash-impact control technology to reduce the occupants' injuries

	55 km/h Frontal crash test	64 km/h Frontal offset crash test
Honda G-CON	●	●
US NCAP	●	Under review
JAPANESE NCAP	●	●
EU NCAP	Under review	●

NCAP: New Car Assessment Program

Table 2 shows a comparison of our own targets and the New Car Assessment Program in various regions. Fig. 32 shows the results of a car-to-car frontal offset crash.

As shown in Fig. 33, the sides of the roof of the passenger compartment and the inside of each pillar contain an impact absorbing material designed to lessen the severity of head injuries by absorbing the impact of a person's head in the event of a collision.



Fig. 32 Car-to-car crash test (Right: Civic)

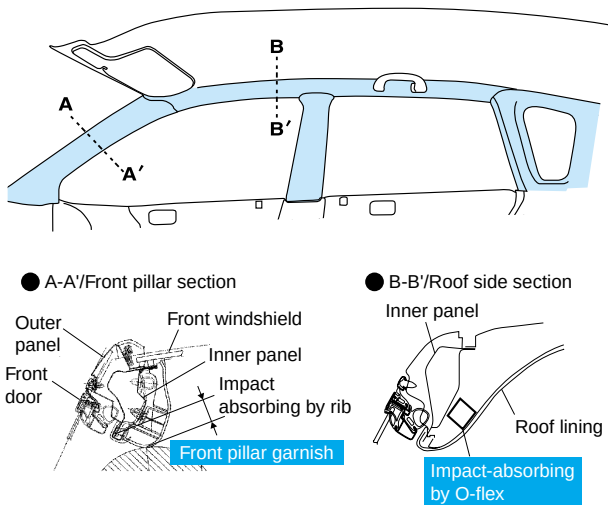


Fig. 33 Reducing the degree of injury to the head

7.4 Pedestrian injury reducing construction

We adopted an impact absorbing construction for the hood, hood hinges, wiper pivots, front fenders, and even the front bumper in order to lessen the severity of not only head injuries in the event of a collision with a pedestrian, but also to expand the area of injury reduction to include the legs.

8. Conclusion

In the course of developing the new Civic, we set high performance targets in terms of comfort, efficiency, and stylishness, all geared towards the concept of a "comfortable life," as well as for collision safety, driving stability, and noise and vibration.

As a result, we were able to achieve the compact size of a small passenger car, best-in-class fuel economy, and comfortable handling. We were also able to achieve a comfortable interior not found in previous passenger cars. Fig. 35 shows the special shape of this vehicle, which



Fig. 35 Appearance of the new Civic

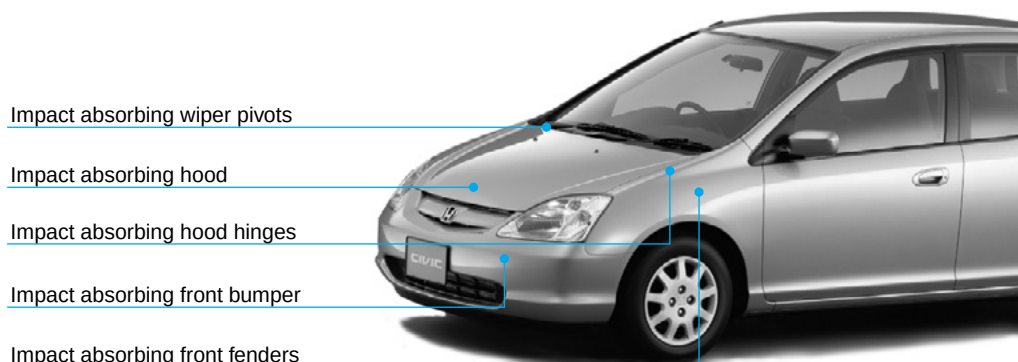


Fig. 34 Pedestrian injury reducing body

characterizes the balance between the compact size and comfort, a shape that says that this is a new design. In addition, the low, flat floor demonstrates the emphasis that we placed on “people” in our design efforts.

Furthermore, the performance targets were achieved in the areas of collision safety performance, driving stability, and noise and vibration.

The completion of this design was made possible through the cooperation of many people, both inside Honda and outside Honda. We wish to express our appreciation for their hard work.

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Development of the New Civic Suspension

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ABSTRACT

New front and rear suspensions have been developed for the new Civic to achieve a highly efficient package layout and good vehicle dynamic performance. The reduced front overhang of 60mm, collision safety target and linear front suspension toe curve are achieved by adopting a strut suspension with a steering gearbox that is mounted 250mm higher than the usual strut suspension. The increased rear luggage room width of 142mm and linear rear suspension toe curve are achieved by adopting an in-wheel double wishbone suspension with a lowered rear damper position. By having these suspension characteristics, the new Civic has enhanced handling and riding comfort performance over the previous Civic.

1. Introduction

Since the development of the original Honda Civic in 1972, Honda's engineers have sought to continue to develop highly efficient package layouts that embody the concept of "Maximum passenger space, minimum mechanical space," while also providing a nimble, sporty ride worthy of the Civic name. The success of these efforts can be measured by the fact that the Civic is held in high regard by customers in 140 countries around the world.

In order to advance these characteristics even further in the new seventh generation Civic, basic development work focused not only on establishing a stylish design and excellent safety and environmental performance, but also on the following two concepts:

- Smart utility:

Maximizing space and facilitating the use of that space

- Smart driving:

Ensuring excellent handling and stability, as well as a smooth ride

In order to realize these goals, new front and rear suspensions were developed that achieved the dual aims of a highly efficient package layout and outstanding handling and stability coupled with a smooth ride.

2. Development aims

Because the Civic is manufactured in at least 12 countries around the world, it has traditionally been necessary during the course of development to consider local parts procurement and the production capabilities of each of the manufacturing centers. However, during the development of these new suspensions, the following two goals, involving elements of product appeal that concern customers directly, were given priority:

- (1) The suspensions should be compact enough to achieve the highly efficient package layout that is demanded in next-generation passenger vehicles. Reducing the height of the front suspension increases the flexibility of the hood design. Furthermore, widening the lateral distance of the suspension while limiting its size in the longitudinal direction would result in a compact design that ensures enough body frame layout space to achieve collision safety performance.

Minimizing the encroachment of the rear suspension on the rear luggage compartment would maximize trunk room, while also providing space for a large-capacity fuel tank and an exhaust prechamber.

- (2) The suspensions should provide excellent handling and stability as well as the comfortable ride that is essential in a next-generation passenger vehicle.

Specifically, both the force steer and the longitudinal compliance characteristics for both the front and rear suspensions were balanced in the course of optimizing the suspension geometry and the spring/damper performance. This was done in order to achieve a degree of handling and stability as well as smoothness that goes beyond what has previously been seen in mass production vehicles.

In order to achieve these goals in the new Civic, a McPherson strut suspension was adopted for the front suspension and a double wishbone suspension was adopted for the rear suspension.

3. Structural features

3.1 Front suspension

As shown in Fig. 1, the basic structure of the front suspension is that of a McPherson strut suspension. However, the steering gearbox is now positioned about 250mm higher

* Tochigi R&D Center

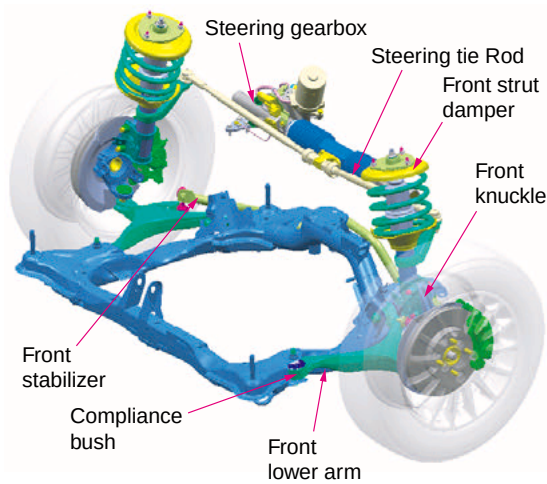


Fig. 1 Front suspension

than in the previous models, where it is roughly the same height as the normal front wheel center. This was done for the two purposes described below.

(1) Reducing the length of the nose

In the case of a typical strut suspension, because the tip of the steering tie rod must be positioned near the rearmost portion of the inside of the wheel in order to provide articulation space, the steering gearbox and the tie rod are positioned at roughly the same height as the center of the wheel.

In addition, The area of the transmission with the maximum outer diameter roughly matches this height. Therefore, if this design, wherein the transmission and the steering box are positioned in a straight line, were used in the new Civic there would be less crush space in the longitudinal direction in regard to a frontal collision. Therefore, if this design were used additional crush space would be needed in the nose of the new Civic.

However, in the case of the new Civic, in which the gearbox is positioned at a higher level, the gearbox moves into the space between the intake manifold and the transmission in the event of a front-end collision. This means that, in the event of a frontal collision, the area of the vehicle at issue can be reduced in the longitudinal direction, allowing for good collision performance. As a result, the new Civic was able to clear strict development standards for both the full frontal and frontal offset collisions, while still allowing the nose to be shortened by 60mm compared to previous Civics, as shown in Fig. 2.

In addition, because the use of struts allowed us to move the dampers farther outside than was possible with the double wishbone suspension used in earlier Civics, it was possible to extend the space between the left and right suspensions by 80mm at the height of the side frame. This, in turn, permitted greater flexibility in the cross-section design of the side frame, which also helped to reach collision performance targets.

(2) Making the toe curve linear

The linearity of the toe curve is one parameter of the suspension characteristics that is extremely important in achieving linear handling. In the case of a typical strut

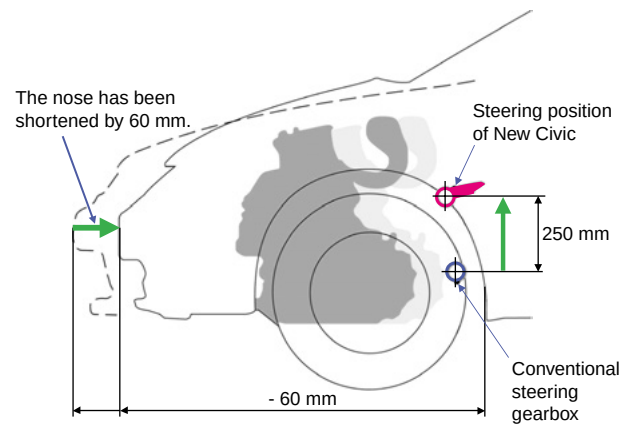


Fig. 2 Position of steering gear box

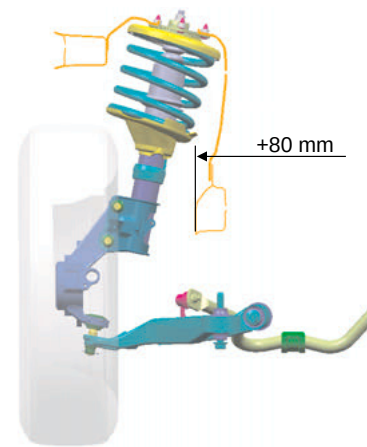


Fig. 3 Front suspension width and side frame

suspension, the ideal method for making the toe curve more linear is to place the lower arm at the same height as the steering gearbox. In a front-wheel drive vehicle, however, the presence of the drive shaft restricts the placement of the lower arm. This inevitably results in the toe change curve having a "C" shape.

The layout in the new Civic, in which the gearbox is placed in a higher position, has a center takeoff construction in which the joint on the gearbox side of the tie rod is centered on the gearbox. By extending the length of the tie rod by about 220mm, it was possible to achieve a toe curve with linearity approaching that of a double wishbone suspension within the range of normal usage (Fig. 4).

In terms of the actual construction, the tie rod arm, which is normally part of the knuckle, was welded to the center of the damper.

More leg room was created for the front seat passengers by moving the lower arm compliance bushing from its position behind the front axle in earlier Civics to a position in front of the front axle. In addition, by setting the bushing characteristics so that there are different spring rates in different directions, it was possible to optimize both the steering characteristics when lateral forces and braking forces are input, as well as the compliance characteristics in the longitudinal direction, balancing a smooth ride with handling and stability.

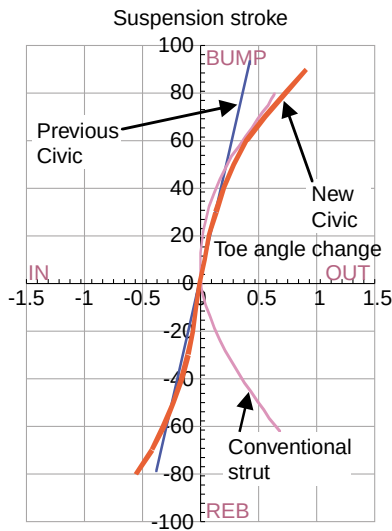


Fig. 4 Front suspension toe curve

In the front damper mount shown in Fig. 5, designing the damping force input to go through the bushing and adopting a construction that directs the spring and bump stopper input loads directly to the body resulted in increased flexibility in setting the bushing characteristics, and made it possible to optimize the bushing spring rate for the damping force input from the damper.

For the front lower arm, a welded construction of stamped components with a closed cross section was adopted in order to reduce the weight while maintaining stiffness.

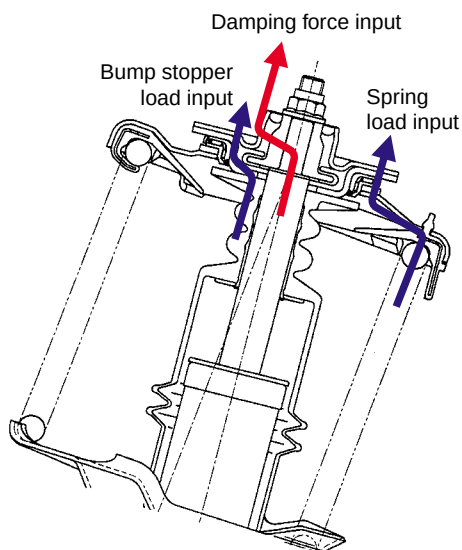


Fig. 5 Front damper mount rubber construction

3.2 Rear suspension

The last three generations of the Civic have utilized a double-wishbone suspension for enhanced handling and stability as well as a comfortable ride. The damper mounts, however, encroach on the passenger compartment, and were a drawback from the standpoint of expanding the trunk space.

In order to retain the handling and stability as well as the

ride of the earlier Civics, the double wishbone suspension is still used in the new Civic, as shown in Fig. 6. However, because it was important to minimize the extent to which the suspension components encroach on the trunk space, considerable effort was devoted to determining how to lower the suspension components and move them as far outside as possible

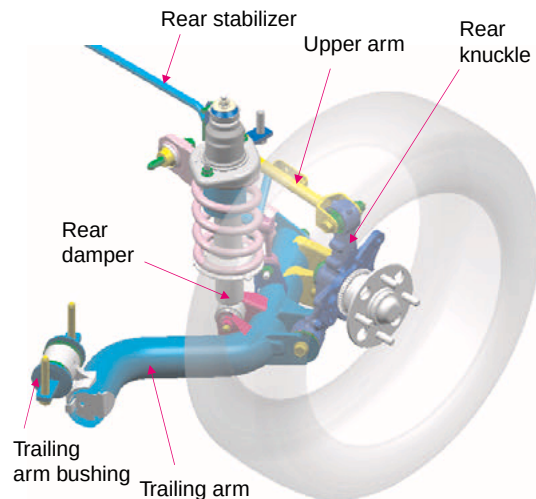


Fig. 6 Rear suspension

By using an in-wheel double-wishbone suspension, in which the upper arm, which was positioned adjacent to the tire in earlier models, is now positioned inside the wheel, it was possible to reduce the overall height of the suspension. In addition, by changing the damper mount base to a deep draw shape and lowering the surface at which it mounts on the body while keeping it roughly perpendicular to the inner face of the tire, it was possible to extend the distance between the damper mounts by 142mm when compared with earlier Civics, as shown in Fig. 7. This resulted in a 40-liter increase in trunk capacity versus earlier models, and also helped to make the trunk space a more useful shape.

The toe curve characteristic when braking force is input is a very important characteristic of a suspension in regards to maintaining stability while a vehicle is being driven. The new rear suspension is constructed so that it allows the toe

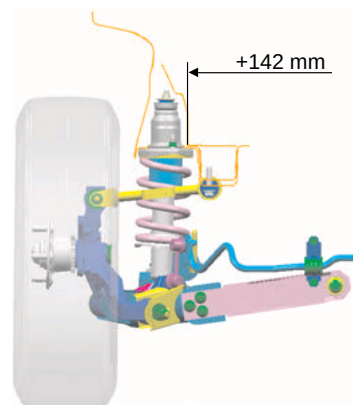


Fig. 7 Rear suspension and rear damper width

control arm to flex longitudinally. The spring rate in the direction of the articulation axis of the bushing at the end of the arm can be set to a softer level, while the characteristics in the perpendicular direction can be simultaneously set to a stiffer level. Doing so provided enough space for the compliance stroke in the longitudinal direction, and, by displacing the end of the trailing arm at an inward angle when braking force is input, also provided stability when the brakes were applied with the tires turned severely in the “toe-in” direction. These concepts are illustrated in Fig. 8.

By changing the mechanical compliance angle from 0.5 degrees to 4.4 degrees as shown in Fig. 9, it was possible to achieve a reduction in harshness, especially when the vehicle drives over a large bump.

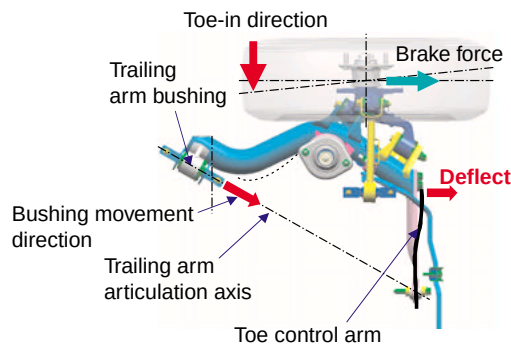


Fig. 8 Rear suspension toe curve with brake force applied

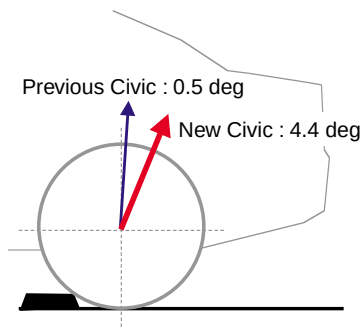


Fig. 9 Mechanical compliance angle

A straight and pipe cross section construction for the upper arm was applied in order to maintain the camber stiffness. This in combination with an optimized bushing spring rate, resulted in a more responsive rear suspension during turns.

Regarding the camber stiffness of the suspension, full consideration was given to the bushing construction (linking the knuckle and the trailing arm), the spring rate, and the articulation axis angle setting when the placement of the bushing axis was determined. As shown in Fig. 10, it was possible to achieve a linear toe curve, while at the same time enhancing the lateral force stiffness by 22% compared to earlier Civics, as shown in Fig. 11.

This construction can also be adapted to the rear suspension of a four-wheel drive vehicle simply by changing the rear knuckle. In addition, the trailing arm itself does not

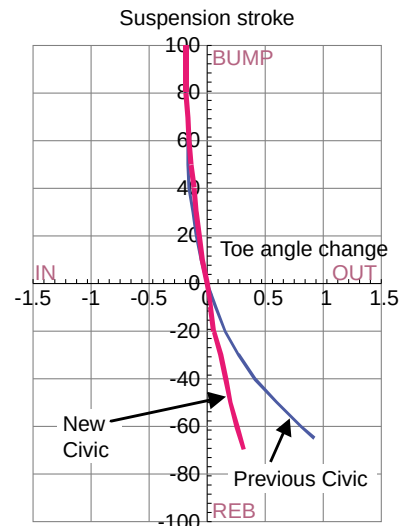


Fig. 10 Rear suspension toe curve



Fig. 11 Suspension lateral stiffness at tire contact point

have a typical press weld construction. Instead, it is constructed by bending pipes, thereby achieving a reduction in weight.

4. Achieved performance

Some representative vehicle characteristics that were achieved by adopting the suspension described in the previous sections are described below.

4.1 Handling and stability

(1) Steering feel

In previous Civics, although the steering had a high response to the initial steering input, turning the steering wheel further produced a response that was less than the initial response, resulting in not as much linearity in the feel of the steering. In the new Civic, by tuning the strut damper stiffness and the characteristics of each bushing, including the steering mount bushing, it was possible to optimize the aligning torque stiffness, resulting in a more linear relationship between the yaw rate change and the steering angle as shown in Fig. 12.

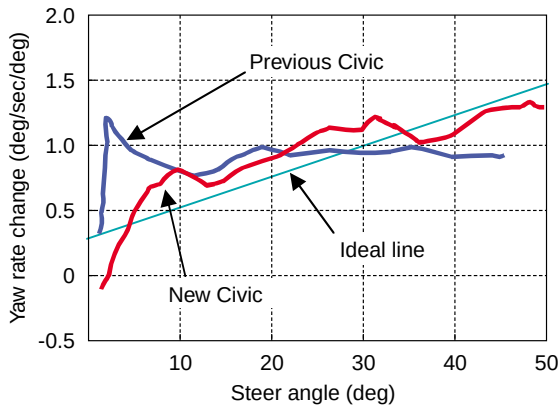


Fig. 12 Steer angle and yaw rate change

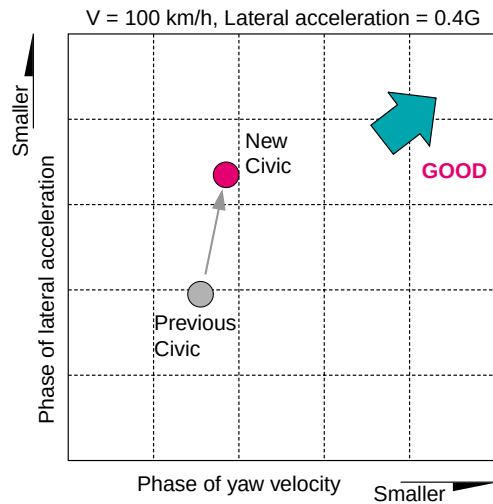


Fig. 13 Yaw rate and lateral acceleration phase

(2) Linear response characteristics

Achieving linear vehicle response requires minimizing the phase delay between the yaw rate and the lateral acceleration during steering. During the development of the new Civic suspension, it was found that by ensuring the camber stiffness of the rear suspension and by controlling the front to rear ratio of the theoretical roll arm length, it was possible to reduce the phase delay between the yaw rate and the lateral acceleration as shown in Fig. 13, and to increase the yaw natural frequency while maintaining an appropriate level of yaw gain, as shown in Fig. 14. As a result, it was possible to achieve linear response and stability during normal driving over a range of steering inputs.

(3) Turning roll axis

By controlling the theoretical roll arm length of the front and rear suspension and revising the setting of the roll center height in order to gradually reduce the pitch angle of the car body in turns as shown in Fig. 15, it was possible to hasten the onset of cornering power of the rear tires, achieving a more comfortable roll feel.

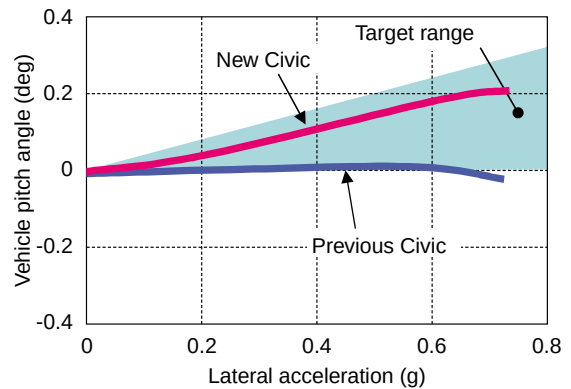


Fig. 15 Lateral acceleration and pitch angle

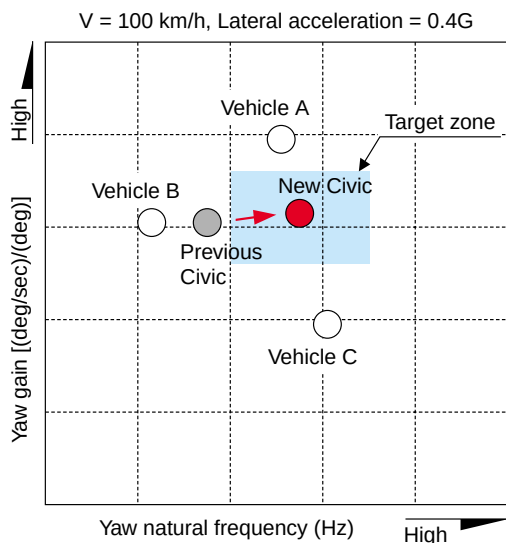


Fig. 14 Steering response

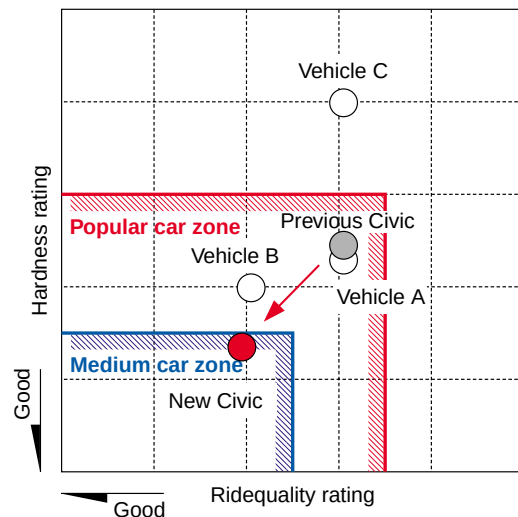


Fig. 16 Ride quality rating and hardness rating

4.2 Ride

Adding the elemental technologies described herein increased the flexibility when setting a smooth vertical movement in the suspension by increasing the margin for the handling range. In addition, by optimizing the compliance bushing spring rate characteristics and enhancing the mechanical compliance angle, it was possible to achieve a ride quality that far surpasses other cars in the same class.

Fig. 16 shows the overall ride quality on the horizontal axis, and the harshness rating (which reflects the ride quality when driving over a bump) on the vertical axis.

5. Conclusions

New front and rear suspensions that meet objectives for the new Civic package layout and vehicle performance have been developed.

By adopting a strut suspension system in the front with the position of the steering gearbox raised by approximately 250mm versus the earlier suspension, and an in-wheel double-wishbone suspension with a toe angle control function in the rear, it was possible to achieve a highly efficient package layout while also achieving outstanding vehicle performance.

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Automation of Mode Driving Operation for Exhaust Gas Measurement

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ABSTRACT

Mode driving is being automated using automatic driving units (robots) with the aim of reducing the number of man-hours needed for the measurement of automobile exhaust gases. Automation using robots offers significant advantages, particularly in high-altitude exhaust gas emissions tests, as it liberates workers from harsh working environments.

This system makes it possible to fully automate all work procedures by constructing automatic cooling fan start/stop systems and automatic hood opening/closing systems, which were issues for full automation with robotic driving under high-altitude conditions. In addition, vehicle speed trace accuracy relative to changes in atmospheric pressure inside the test chamber equivalent to that for driving with a human driver was realized by using separate acceleration parameters for low- and high-altitude conditions and optimizing the control parameters. These changes made it possible to reduce the number of staff and time required for tests and to eliminate tasks in harsh working environments.

1. Introduction

Automobiles are used in a variety of different environments. Consequently, the measurements made to quantify an automobile's impact on the environment are required by regulation to be conducted under a diverse set of modes and environmental conditions. The number of verification tests needed to satisfy these regulations is also increasing. Thus, there is a growing demand for automation of exhaust gas emissions and fuel economy testing in order to reduce the man-hours expended to handle the increasing numbers of these tests.

The demand for automation is of course in part for labor-saving reasons. However, automation is also instrumental in eliminating the need for workers to work in the harsh working conditions involved in working in environmental chambers under such conditions as low temperature, solar radiation and negative pressure and in improving the reproducibility of driving results. Great strides have been made recently in the development of technologies related to automation. The use of an automated driving device (robot) instead of a human being to drive test cars is one extremely effective method of automation. This device is commonly used for vehicle engine endurance testing. It is also used for exhaust gas emissions and fuel economy testing of cars with their air conditioning running. Testing with the air conditioning running takes place in a test room under very harsh working conditions. The temperature is maintained at the very hot temperature of 35°C, with the additional use

of solar radiation devices to simulate the environment of a mid-summer day in California.

The high-altitude exhaust gas emissions test (high-altitude test), conducted at the equivalent of a 1600 m elevation, requires a greater expenditure of man-hours than the low-altitude exhaust gas emissions test (low-altitude test).

The purpose of conventional devices was for the robot to drive in place of a human during driving modes. It was not envisioned that the robot would replace humans when conducting operations other than driving. Consequently, such a device would not provide sufficient advantages for use in high-altitude tests. The reason is that, although humans can easily enter the test room to perform such actions as opening and closing the hood and operating the engine cooling fan during the low-altitude test, this is not the case during high-altitude tests. The robots are not able to perform these operations, and humans cannot enter and leave the test room easily due to the need to acclimatize to the pressure change each time. Thus, there has been no particular benefit in using robots for those tests.

The authors have applied the use of robots to high-altitude testing, and improved the device to fully realize automated testing. This paper introduces the improved device and the robot's high-altitude driving performance.

2. Development aims and target requirements

The high-altitude tests conducted at the equivalent of 1600 m elevation require more man-hours than the low-

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altitude tests. Therefore, we studied methods of replacing human operators with robots in order to improve efficiency of high-altitude test operations and minimize the labor burden. The robots, however, were constructed to replace only the driving function. In the past, the operators of the negative-pressure test room operated the vehicle, turned off the cooling fan, and opened and closed the hood. Therefore, the aim cannot be achieved by robots that simply replace the operators for the driving mode alone. Automating these operations would achieve the objective of raising operational efficiency and reducing the labor burden.

The fundamental requirements came under the following four headings:

- (1) The device is capable of conducting tests in conformance with formal test methods as prescribed by regulation.
- (2) When using the LA4 mode, the robot driving performance is equal in low-altitude and high-altitude testing. In addition, both humans and robots are able to follow the same vehicle speeds.
- (3) Action to avoid danger is to be taken in the case of emergency.
- (4) For practical purposes, the device is generally applicable to all vehicles, not just to certain ones.

The issues that needed to be resolved came under the following four headings:

- (1) Opening and closing the hood and turning the cooling fan on and off in a negative-pressure environment
- (2) Conforming to differences in throttle opening when driving in low-altitude and high-altitude tests
- (3) Setting robot driving performance to be on a par with human performance
- (4) Detecting malfunctions and avoiding danger when malfunctions occur

3. Overview

The conditions were set with the negative-pressure test room at the equivalent of an altitude of 1600 m as the standard. The temperature in the test room is adjusted to within $25^{\circ}\text{C} \pm 2^{\circ}\text{C}$, and the humidity to within $50\% \pm 10\%$. The double roller chassis dynamometer is used.

The control unit for the robot is installed in the control room, with only the mechanical unit installed in the vehicle. This mechanical unit is constructed presupposing use under other environmental conditions, with a heater warming unit to make it capable of trouble-free operation even when left at temperatures as low as -25°C . The mechanical unit is shown in Fig. 1, and installed in a vehicle as shown in Fig. 2.

The previous method of conducting tests with human drivers required two people in the test room and two others in the control room for safety reasons. This was done so that an extra person would be available to help if the driver's physical condition deteriorated in the negative pressure environment. Two people would be present in the control room so that one person could enter the test room to help in case of an emergency, while the other person could operate the equipment. When testing with a robot, the test room is unmanned, and one person is sufficient in the control room because of no possible human emergencies in the test room.



Fig. 1 Mechanical unit



Fig. 2 Mechanical unit with vehicle

It requires some time for human beings to acclimatize their bodies to the pressure when going from the control room into the test room under negative-pressure conditions. Consequently, entering and leaving the negative-pressure test room takes 12 minutes each way. This 12 minutes is required at an altitude of 1600 m, and still higher elevations require correspondingly longer times. This situation is outlined in Fig. 3 and Fig. 4. This requirement for additional time and personnel increased the required man-hours before testing was automated.

A U.S. model ULEV Accord with an automatic transmission was used as the test vehicle.

The matter of selecting a standard for the robot operating accuracy was a subject for discussion. The driving style

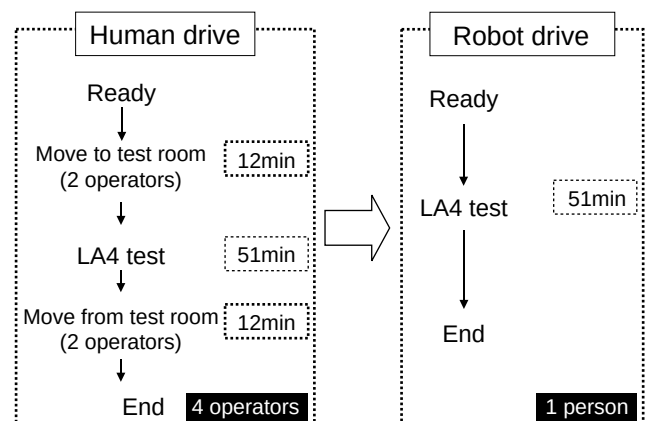


Fig. 3 High altitude test schedule

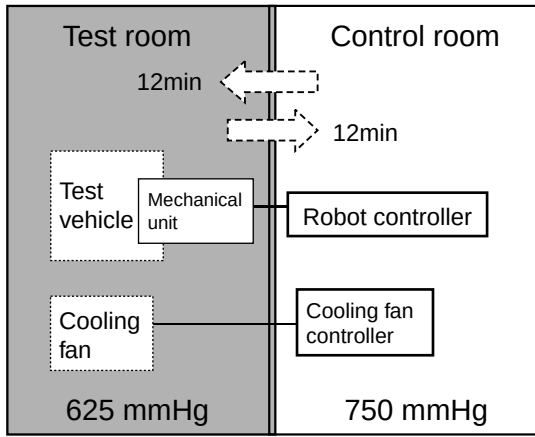


Fig. 4 Test room layout

was eventually based on a driver with three or more years of experience in testing. However, there are individual differences in driving according to the driver. Some drivers follow the benchmark speed exactly while others continuously predict the future and smoothly follow the preceding vehicle. Therefore, an ordinary driving style was selected as the standard.

4. Issue-resolution measures

4.1 Issue 1: Changing of the device

The high-altitude testing requires the engine to be stopped for a 10-minute period (soak), which is an obstacle in the conventional test system. The LA4 mode shown in Fig. 5 has three driving modes: cold transition (CT), cold stabilized (CS), and hot transition (HT). A soak is mandated between the CS and HT modes. At the start of the soak period, the engine is stopped, the hood is closed, and the engine cooling fan is turned off. When the soak ends, the engine is started, the hood is opened, and the engine cooling fan is turned on.

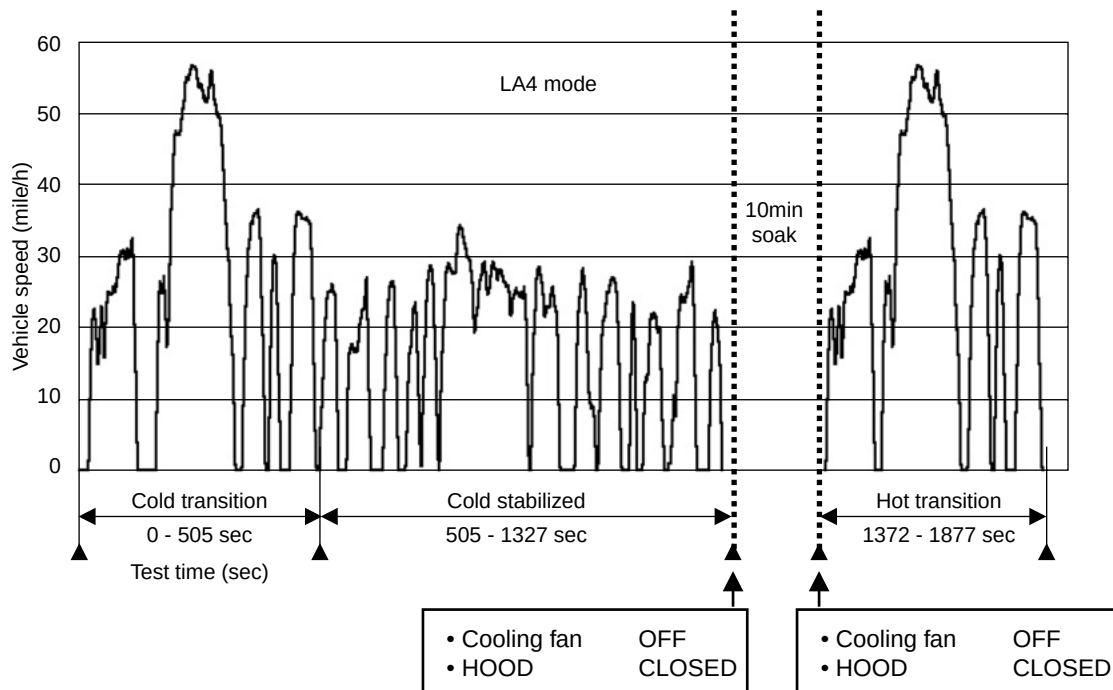


Fig. 5 LA4 test mode

These operations were conventionally carried out by the driver, which means that the high-altitude test as conducted previously could not be performed unmanned.

Fig. 6 shows a comparison between the measurements of exhaust gas emissions with the hood open and closed during the soak period in the LA4 mode. When the hood is open, the NO_x emissions tend to increase. This means that the NO_x emissions values will end up differing during formal tests. This happens because the temperature changes in the catalyst and the exhaust pipe.

An even greater difference than that occurring due to the hood being opened and closed appears in the exhaust gas emissions when the engine cooling fan is not turned off during the soak. This occurs because the engine cooling wind blows directly on the exhaust pipe as far back as the catalyst; thus lowering the catalyst temperature.

As the opened and closed condition of the hood has a significant effect on exhaust gas emissions, full automation by robot cannot be achieved unless this operation is also included as part of the automated test procedure. The

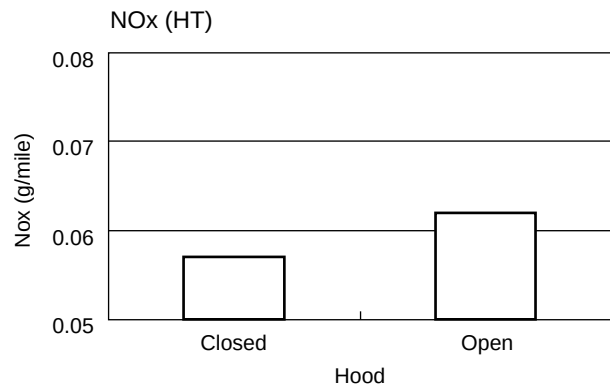


Fig. 6 Comparison of emissions with the hood open and closed (LA4 HT)

required operation is opening and closing the hood during the soak. Therefore, a wire rope was attached to the front edge of the hood, and a reel was installed at the ceiling to reel the rope in and out and thereby raise and lower the hood. This device is shown in Fig. 7. The device was taught the hood open position in advance, and an upper torque limit was set in order to prevent the vehicle itself from being lifted up in the case of a malfunction.

The switch for the engine cooling fan is mounted in the test room. Therefore a branch extension of the switch to the robot was installed to synchronize switching the fan on and off with the robot operation. The device for opening and closing the hood was linked to the robot in the same way.

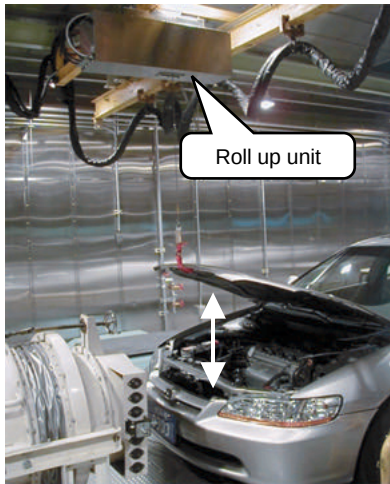


Fig. 7 Automatic hood open-close system

4.2 Issue 2: Differing air densities due to altitude

Differing air densities at low and high altitudes mean that the throttle opening will be significantly different even for identical driving operations. The robot is programmed with a standard acceleration value that is based on the engine output and throttle opening. If the vehicle is accelerated at high altitude using the standard throttle opening for low altitude, the desired acceleration will not be obtained because the lower air density reduces engine output (acceleration). In order to obtain the same acceleration at a high altitude, the throttle must be set to open farther than at low altitudes. Using the standard low-altitude acceleration value at a high altitude will have a negative impact on driving performance and exhaust gas emissions values because efforts to follow the set vehicle speed will be impeded.

It is necessary, therefore, to teach the device the separate throttle openings for low and high altitudes in advance, and to manage the low-altitude and high-altitude data separately. This method eliminated the difference in exhaust gas emissions figures between low and high altitudes that occurred with the robot driving.

Fig. 8 presents comparisons for fuel economy at low altitude and high altitude and fuel economy with human and robot vehicle operation. The discrepancies that arise from the differences when operating at low and high altitudes were studied with reference to differences in fuel economy. However, the differences between results with human drivers and robot drivers were extremely small at both low and high

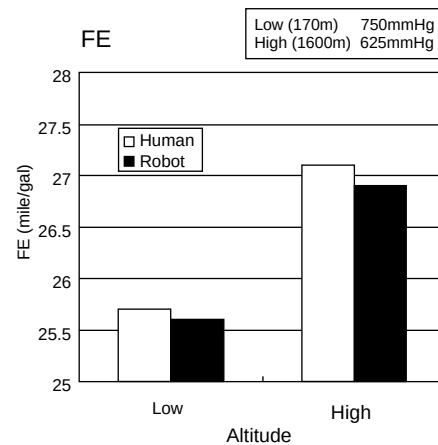


Fig. 8 Comparison of low and high altitude FE for human and robot vehicle operation

altitudes. This confirmed that the device poses no difficulties from the standpoint of practical application.

4.3 Issue 3: Individual variations in following set vehicle speeds due to drivers

Individual variations in following the set vehicle speed occur with different drivers. The regulations prescribe a certain acceptable margin with respect to the standard speed, so that following a speed within that range poses no difficulty. However, variations in vehicle speed trace affect exhaust gas emissions and fuel economy, making it necessary to impose some kind of driving standards. Consequently, the standard selected specifies a driver who has three or more years of experience in testing and the ability to follow the set vehicle speed. Operation of the throttle is an important element in the ability to follow a set vehicle speed. Driver throttle operations can be grouped under three general types.

The standard of a driver who has ordinary speed trace abilities will be defined here as the standard trace. When following the vehicle speed, some drivers will be slow and others will be quick in operating the throttle relative to the deviation between the target vehicle speed and the current target vehicle speed. The former type will be defined here as the slow trace, the latter as the quick trace, and the type in between those two as the standard trace.

We decided to set the robot to have vehicle speed trace abilities equivalent to the standard trace. The speed following control constant for the throttle was adjusted accordingly.

The details of the adjustment are explained here. When the control constant causes a sluggish response to keeping the target speed within the acceptable margin of the vehicle speed range, driving with that control constant constitutes the slow trace setting. Conversely, when the control constant causes a sensitive response that very quickly brings the vehicle to the target speed, driving with that control constant constitutes the quick trace setting. The control constant that lies between those two constitutes the standard trace setting. The standard vehicle speed prescribed by the test method is defined as the trigger speed. Fig. 9 shows a comparison of different vehicle-speed trace performance. This is a comparison of vehicle-speed trace performance within the 55 to 65-second period of LA4 mode initiation. The line

that shows relatively little variation from the trigger speed represents the quick trace. For further comparison of throttle operation, Fig. 10 shows the intake manifold pressure. It is apparent that frequent application of the throttle for the quick trace is a wasteful activity. The slow trace performs future projection for driving with as little throttle application as possible, so there is less waste in throttle operation. Consequently, for the purposes of speed trace performance and throttle operation, the standard trace was set between those two, and it was decided to use the standard trace in the standard control constant for robot operation.

Fig. 11 presents the relationship between vehicle speed trace performance and exhaust gas emissions figures.

Vehicle speed trace performance has little effect on exhaust gas emissions figures. In terms of fuel economy, quick trace causes a 0.6% degeneration, but standard trace produced stable results for both fuel economy and exhaust gas emissions.

The control constants we determined were also used for verification with vehicles other than the Accord. The results showed that the requirement for vehicle speed following was satisfied using the determined control constants.

The following vehicles were confirmed to raise no issues when tested with this control constant:

- (1) NSX with a six-speed transmission (MT and AT)
- (2) Legend with a four-speed transmission (AT)
- (3) S2000 with a six-speed transmission (MT), also verified using the push-button starter
- (4) Civic with a five-speed transmission (MT and AT), also continuously variable transmission (CVT)
- (5) Insight (hybrid vehicle) with a five-speed transmission (MT and AT), also with CVT

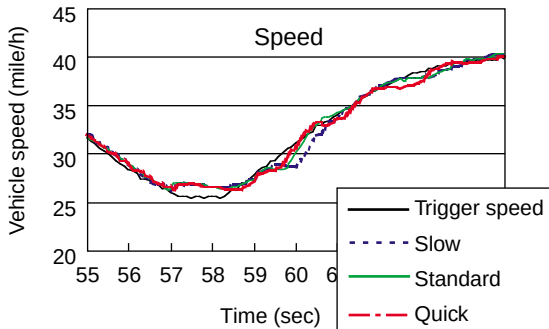


Fig. 9 Vehicle speed trace

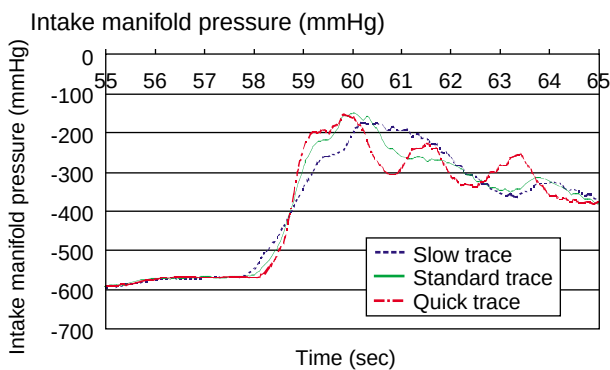


Fig. 10 Intake manifold pressure

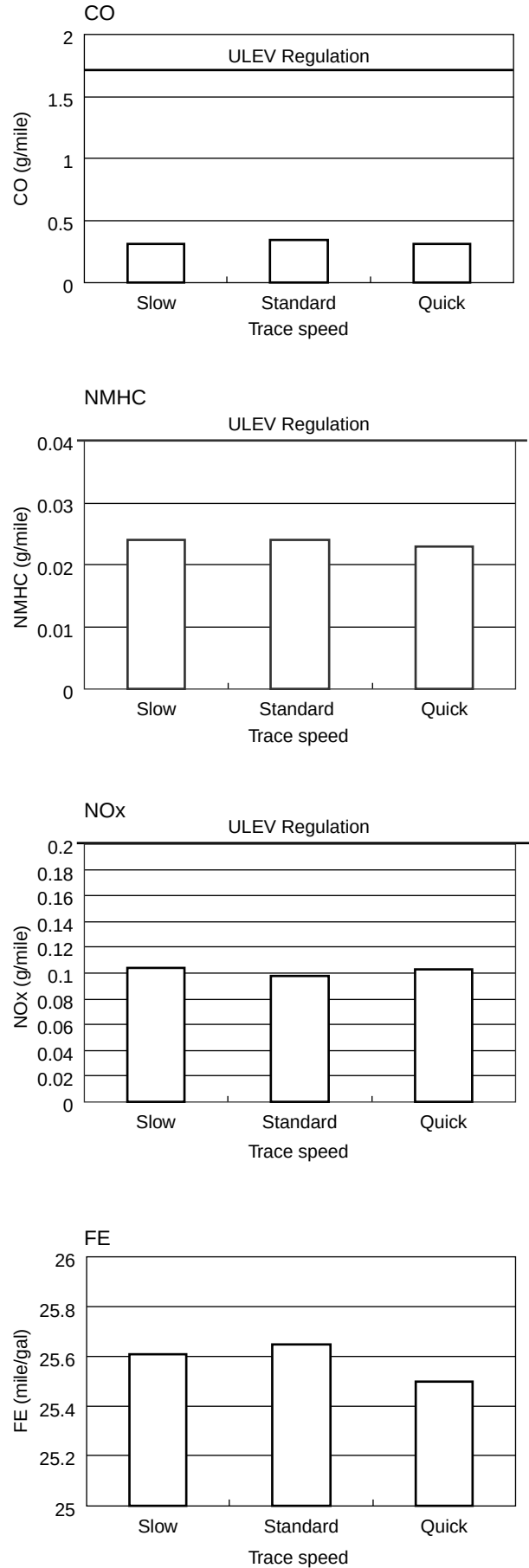


Fig. 11 Comparison of emissions and FE for slow, quick, and standard trace speeds

4.4 Issue 4: Countermeasures for malfunctions

The device is intended for use in unmanned testing in the test room. It is necessary, therefore, to install monitors to detect tire blowouts or other vehicle malfunctions that may occur. Consequently, vehicle position monitoring sensors were installed to monitor vehicle behavior in the vertical and longitudinal directions for malfunctions. Engine speed is also monitored, and malfunctions are identified by the engine stopping or exceeding a set speed. The control room is notified by an alarm when a malfunction occurs, and an emergency stop function is activated to prevent accidents.

5. Effects

5.1 Eliminating highly burdensome work

The use of robots to replace the workers who drive and manage operations in the negative-pressure test room has eliminated the need for those workers to conduct tests in an environment that imposes a heavy work burden. Making the transition to a high altitude in a short time can have a negative effect on the human body. Engine developers have been requesting the elimination of testing that required them to enter the negative-pressure environment of the test room.

5.2 Labor savings

Safety measures for high-altitude testing formerly required the presence of two workers in the negative-pressure test room and two workers in the control room, so that the work standards mandated four workers in all. The use of robot driving, however, has made it possible for a single worker to conduct tests. A single test can now achieve a personnel reduction of three workers.

5.3 Time savings

It previously required a total of 24 minutes for the tester to enter and leave the test room in order to make the transition from low-altitude to high-altitude testing. Elimination of the need for this time has made it possible to cut the time needed by 30%. There was no way to reduce the time required to enter and leave the negative-pressure test room by ingenuity or effort, but automation has made it possible to increase the number of tests conducted in one day.

5.4 Other effects

Advances made in exhaust gas purification performance in recent years have narrowed the difference between concentrations of emissions in the atmosphere and in exhaust gas. A major part of total exhaust gas emissions across the measurement modes is now made up of the highly-concentrated emissions that occur in launches immediately after engine start-up, before the catalyst is activated. Automated testing by robot provides superior reproducibility of driving immediately after engine start-up. This has improved the accuracy of exhaust gas emissions measurement and enhanced the value of the data.

6. Conclusion

The replacement of drivers by a robot for high-altitude

testing and automation of the operations of opening and closing the hood and turning the cooling fan on and off have eliminated the need for human labor in negative-pressure environments. The device deals with driving performance by changing between low-altitude and high-altitude data, and achieved vehicle-speed trace performance by optimizing the control constant. Test results have shown that there is no difference between manned and unmanned testing.

Consequently, the device has enabled the elimination of workers from work that imposes a heavy burden. It provides labor savings and time savings, which has significantly improved operational efficiency. This has also brought improved reproducibility of exhaust gas emissions and fuel economy and enhanced the value of measurement data. For the future, we intend to develop further expanded applications for improving test efficiency and for testing under even harsher environmental conditions.

Acknowledgements

We would like to express our gratitude to the persons concerned at Horiba, Ltd. who gave us their cooperation in the deployment of this device.

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Study on the Generation Mechanism of Backfires

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ABSTRACT

In motorcycle engines, backfires can become an issue with increased cylinder bore size. It has been thought that backfires are caused by an extra ignition at an overlapping top dead center or by backflows into combustion chamber of high temperature exhaust gas during exhaust stroke. As a result of taking pressure measurements inside the combustion chamber, at the inlet and exhaust ports, and of observing the inside of the combustion chamber with a high speed CCD camera, it was found that an extremely slow-burning cycle exists which continuously burns during the exhaust stroke. Further, it was confirmed that the extremely slow-burning cycle caused the fresh air-fuel mixture introduced in the subsequent intake stroke to ignite thus generating backfires. This paper provides an overview of the investigation and describes some of the factors that influence the extremely slow burning cycle.

1. Introduction

Backfires occur as an explosive ignition sound from an engine intake port when the throttle is opened suddenly. This poses concerns in terms of product appeal. However, the present fact is that there has not been any theoretical analysis of the backfire generation mechanism to date.

Up to now, the source of the ignition for backfires had been identified as the ignition occurring during an overlap at top dead center (ignition spark during overlap), or as the high-temperature burned gas flowing back into the combustion chamber due to an exhaust pressure wave from another cylinder. However, there were cases when a backfire occurred even when these factors had been eliminated. This project took pressure measurements in each relevant part during backfires and observed the inside of a cylinder using a high-speed camera. Here we report the results, which show that there are sources of ignitions for backfires other than ignition sparks during overlap, or the backflow of burned gas.

2. Test equipment

2.1 Engine control device

Table 1 presents the main specifications for the engine being tested. This research utilized a 996-cm³ V-type twin-cylinder (longitudinal cylinder arrangement) engine. The tests were conducted primarily on the back cylinder.

Backfires occurred most frequently when the throttle was

Table 1 Major specifications of test engine A

Engine type	4-stroke V-90° twin cylinder
Cooling type	Liquid-cooled
Bore × stroke	98 × 66
Compression ratio	9.4
Valve train	DOHC 4-valve
Maximum power (kW/r/min)	77.3 / 9000
Maximum torque (Nm/r/min)	93.1 / 7000
Ignition timing	15° (BTDC)

opened up suddenly while reducing speed under a no-load condition. Backfire tests of all kinds used to be conducted by manually operating the throttle, but this caused variations in the frequency of occurrence and was also not very precise. For this research, therefore, the device indicated in the diagram at the top of Fig. 1 was fabricated and used for testing.

This device monitors the engine speed, and when revolutions fall below a set value, it triggers a magnetic valve that operates an air cylinder. As shown at the bottom of Fig. 1, this device is made to open the throttle a specified amount (θth) for a specified time (T1). The device repeatedly snaps the throttle with the engine under no load conditions, a specified number of times. The relationship between the engine speed and throttle opening during actual operation is shown at the bottom of Fig. 1.

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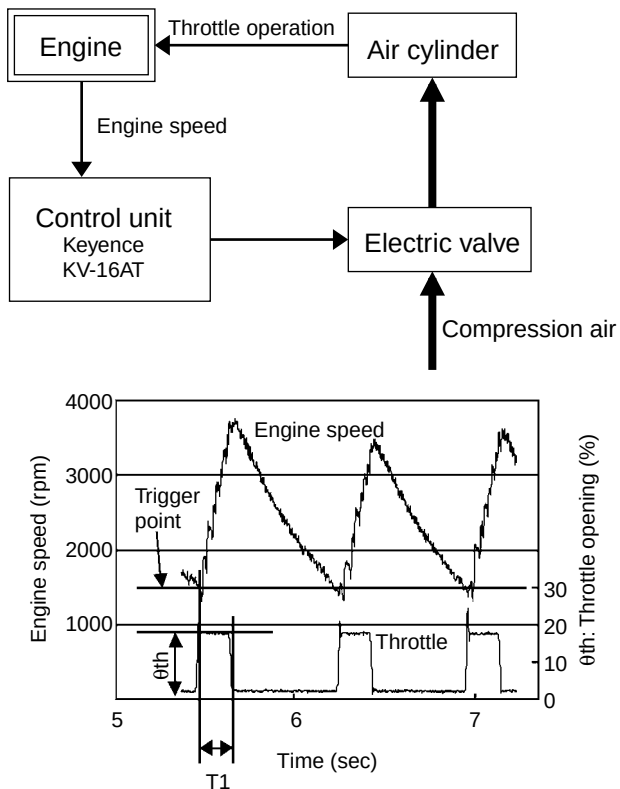


Fig. 1 Backfire test device

2.2 Pressure sensor installation position

Various pressure measurements were conducted in order to clarify the backfire generation mechanism. Fig. 2 shows the names and installation positions of each pressure sensor used. A Kistler 5053C pressure sensor was used to measure combustion pressure, and Toyoda PMS-5 pressure sensors were used to measure intake and exhaust port pressure. In order to measure cylinder pressure accurately in the vicinity of bottom dead center, a Toyoda PMS-5 pressure sensor was also attached to the cylinder wall at a location 35 mm below the deck surface, which corresponds to a 95° crank angle from the top dead center position.

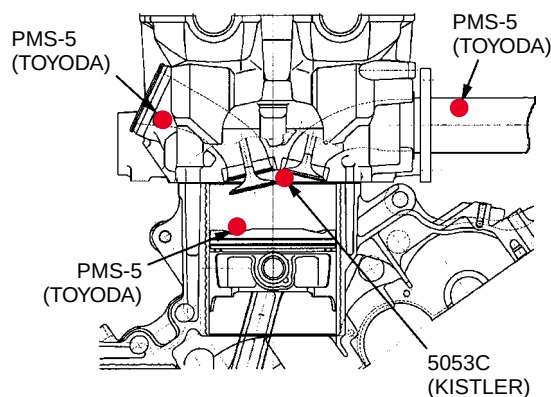


Fig. 2 Installation positions of pressure sensors

3. Backfire generation circumstances

3.1 Conditions for backfire generation

Empirically, it is known that backfires occurred more frequently when the throttle was opened suddenly on an engine running at low speeds. This is thought to be because combustion is unstable when the engine is under a low load. The conditions for backfire can be summarized as follows:

- The engine is a large-bore engine.
- Backfires occur after operation under low load conditions, at low speeds.
- The throttle is operated with a small opening.
- The throttle valve is opened suddenly.

3.2 Conditions most conducive to backfire generation

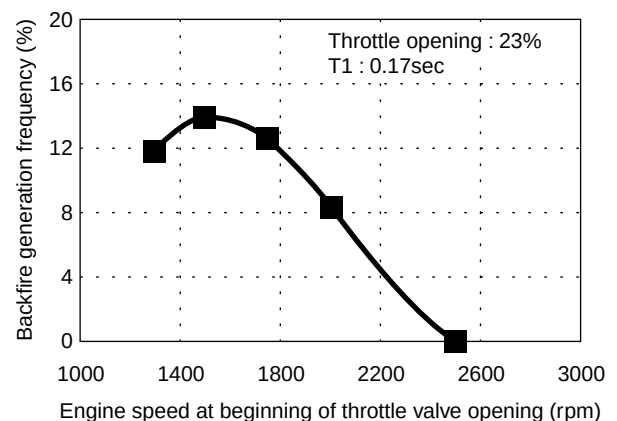
The test engine described in section 2.1 was set to the conditions shown in Table 2, and the conditions under which backfire occurred most frequently were confirmed.

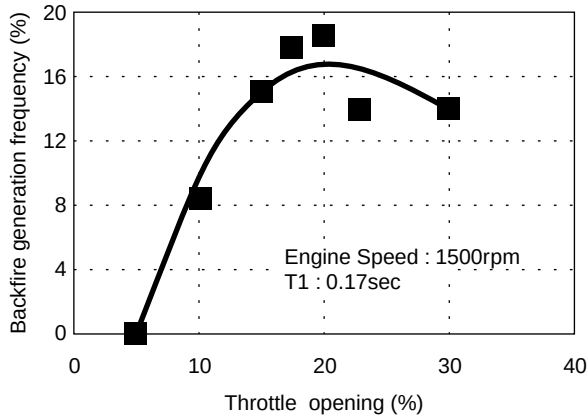
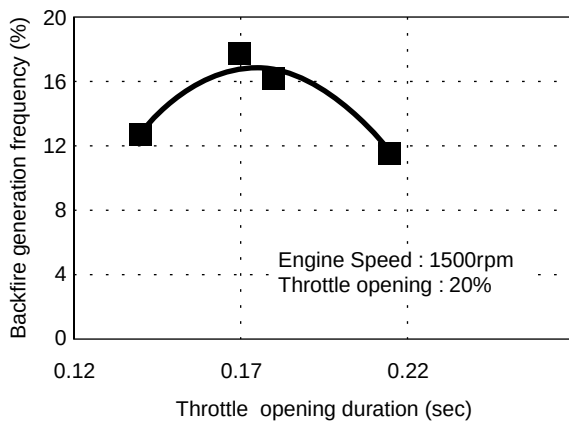
Table 2 Operating conditions

Item	Operating conditions
Engine speed at start of operation (rpm)	1300, 1500, 1700, 2000, 2400
Throttle opening (%)	5, 10, 15, 18, 20, 22, 30
Operating time (sec)	0.14, 0.17, 0.18, 0.20

The conditions were set as follows:

- (1) Backfire generation frequency was measured while changing the engine speed at which the throttle was opened (Fig. 3).
- (2) Selecting the engine speed at which backfires occurred most frequently, backfire generation frequency was measured while changing the throttle opening (Fig. 4).
- (3) Selecting the engine speed and throttle opening at which backfires occurred most frequently, backfire generation frequency was measured while changing the throttle operation time (Fig. 5).

Fig. 3 Backfire generation map
(Changing engine speed)

Fig. 4 Backfire generation map (Changing θ_{th})Fig. 5 Backfire generation map (Changing T_1)

The results of tests conducted under the above conditions showed that backfires occurred most frequently when the test engine was running at 1500 rpm at the time the throttle started to be opened, the throttle was opened 20%, and remained open for 0.17 sec. This test engine was adjusted to idle at a speed of 1200 rpm, so it is apparent that 1500 rpm was slightly higher than its idle speed. We also found that backfire generation frequency changed relatively little when the throttle was opened 15% or more.

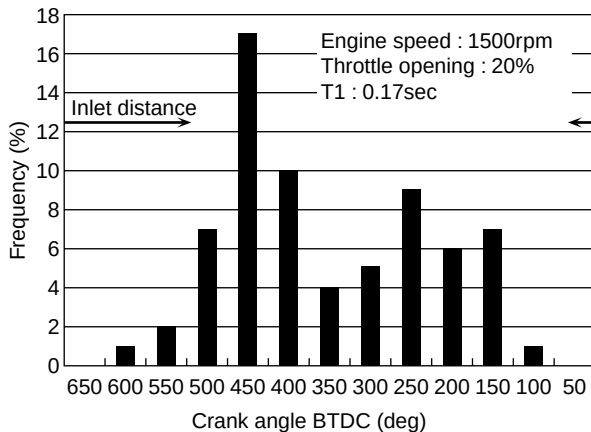


Fig. 6 Throttle opening timing

3.3 Influence of throttle open timing

The relationship between backfire generation frequency and throttle open timing is shown in Fig. 6. It is apparent that backfire frequency is lower when the throttle is opened while the inlet valve is open. This was because the lag in fuel that is generally said to follow the inflow of air became more marked and the air-fuel ratio shifted toward the lean side when the throttle valve was opened while the inlet valve was open, thus making it less likely that backfires would occur.

4. Results of pressure measurements of each part during backfire

4.1 Pressure at the time of backfire generation

The pressure waveforms for each part during backfire occurrence are shown in Fig. 7.

With the throttle fully closed (the blue line in the lower graph), the engine speed decreases (the red line in the lower graph), then the throttle opens in the vicinity of 1500 rpm. Immediately after that, the pressure in the intake port (the green line in the upper graph) and the pressure inside the cylinder (the red line in the upper graph) both rise, and backfiring takes place. The rise in pressure can also be seen with the Kistler pressure sensor, though the waveform peaks are lower because the measurement range is different.

The exhaust port pressure wave (the orange line in the upper graph) shows no change in the vicinity of the overlap at the top dead center immediately before the backfire occurs. This shows that the backfire is not generated by the influence of an exhaust pressure wave from another cylinder, as had previously been thought.

Ordinarily, operation under a no-load condition at a decreasing speed causes a misfiring cycle. In the case of a misfiring cycle, the condition approaches adiabatic compression and adiabatic expansion. Therefore, the pressure in the cylinder at the instant when the exhaust valve

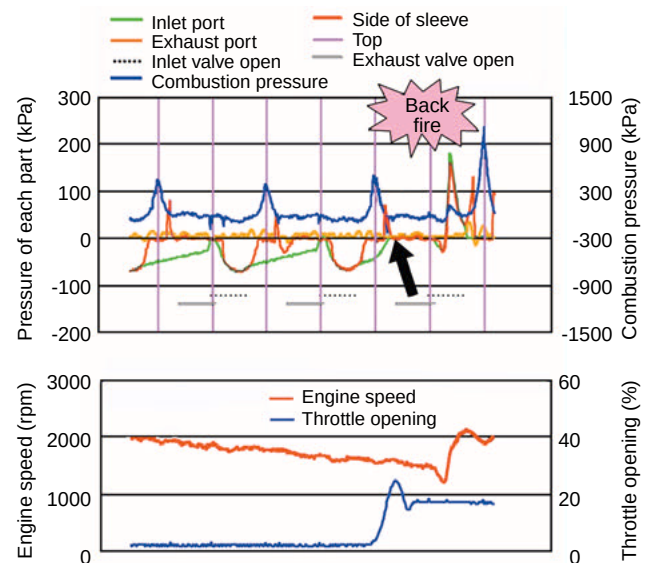


Fig. 7 Pressure of each part when a backfire is generated

opens becomes generally equal to the pressure at the start of the compression stroke before that, as shown in the upper graph in Fig. 8. In the case of an ordinary slow-burning cycle, there will be pressure of 100 kPa or more at the instant when the exhaust valve opens, as shown in the lower graph in Fig. 8.

Consider the cycle before backfire occurs in Fig. 7. This may at first appear to be a misfiring cycle, but the cylinder pressure (section of graph indicated by the arrow) at the instant when the exhaust valve opens is clearly higher than the cylinder pressure at the start of the previous compression stroke, showing that this is not a misfiring cycle. However, the pressure is not as high as it is in a usual slow-burning cycle. In other words, this can be understood as an extremely slow-burning cycle that is close to being a misfiring cycle. For the purposes of this research, we decided to term this kind of cycle where combustion occurs so slowly an extremely slow-burning cycle.

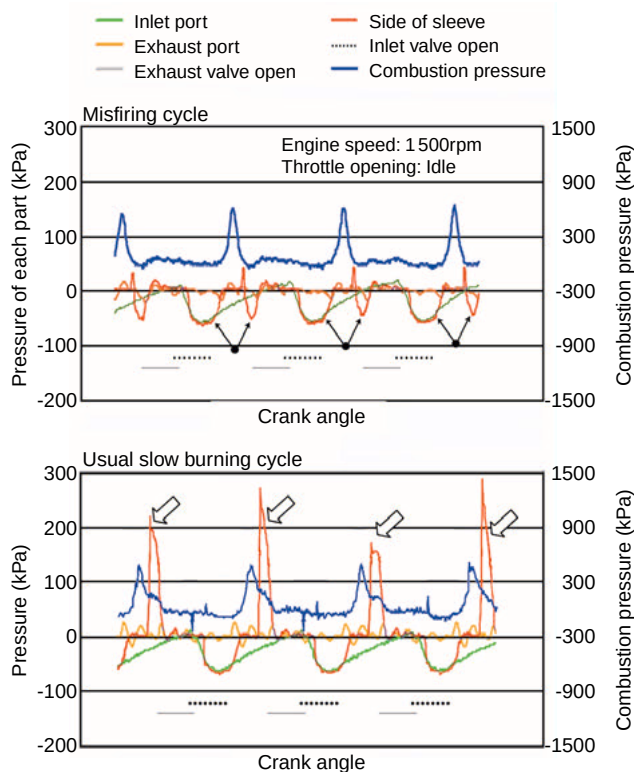


Fig. 8 Pressure of each part during a usual slow-burning cycle and during a cycle when a misfire is generated

4.2 P-V diagram

The P-V diagram for an extremely slow-burning cycle measured using a DS-0928 Ver. 3.58 combustion analyzer made by Ono Sokki Co., Ltd. is shown in Fig. 9. The cylinder pressure rises during the latter part of the intake stroke, and backfiring occurs. Consider the transitional portion of this cycle from the combustion to the exhaust stroke (the circled section of the graph). Immediately before the exhaust stroke, the cylinder pressure does not change even though the exhaust valve is open.

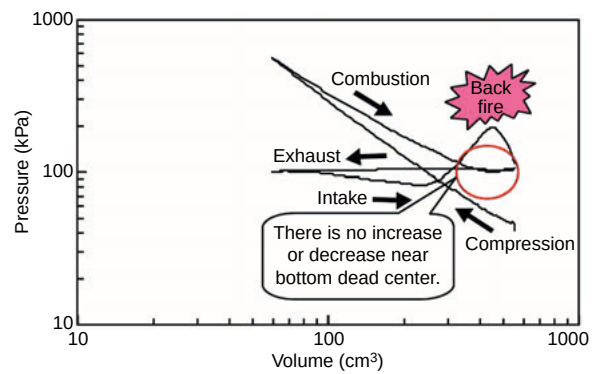


Fig. 9 P-V Diagram (Extremely slow-burning cycle)

Let us compare this again with a usual slow-burning cycle and a misfiring cycle. As shown in the bottom graph in Fig. 10, during a usual slow-burning cycle, the cylinder pressure drops toward atmospheric pressure when the exhaust valve opens because combustion is taking place. Also, as shown in the upper graph in Fig. 10, pressure is negative at the start of compression during the misfiring cycle, so the cylinder pressure is also negative immediately before the exhaust valve opens. When the exhaust valve opens up, the cylinder pressure rises toward atmospheric pressure. The extremely slow-burning cycle does neither of the above.

The extremely slow-burning cycle, therefore, can be described as a cycle in which the cylinder pressure neither increases nor decreases during the time before and after the exhaust valve opens. The reason for this is that it is a cycle

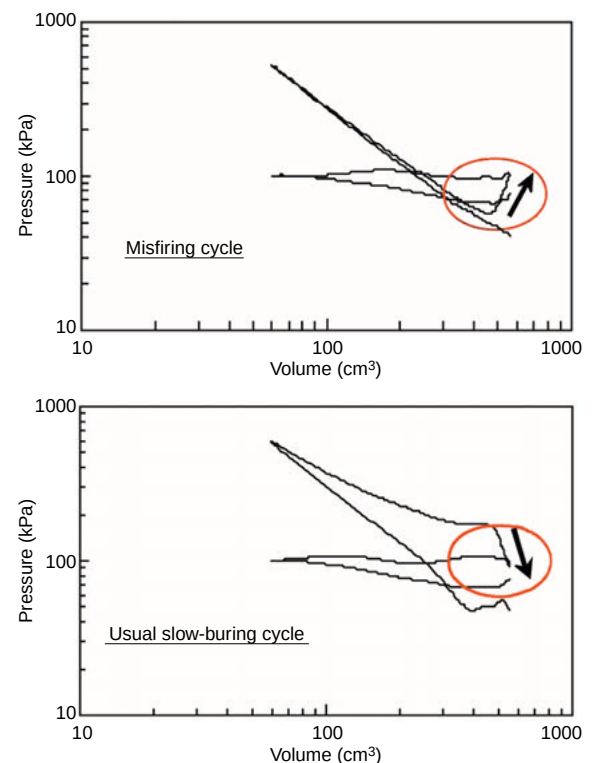


Fig. 10 P-V Diagram (Misfiring and usual slow-burning cycle)

in which the slow burning is made even slower, and combustion continues to take place even when the exhaust valve opens.

The extremely slow-burning cycle can be understood as a phenomenon that occurs during the state of transition from the misfiring cycle to the slow-burning cycle. As shown in Fig. 11, reducing volumetric efficiency generally causes the burning velocity in the usual combustion cycle to gradually grow slower, until ultimately it becomes a misfiring cycle. When there is a transition from a no-load condition at decreasing engine speed to idle, the situation is reversed. In other words, a transition takes place from a misfiring cycle to a slow-burning cycle. At this time, the misfiring cycle does not suddenly become a slow-burning cycle. Rather, the extremely slow-burning cycle can be understood to occur as an intermediate condition between a misfiring cycle and a slow-burning cycle. The rate of heat release in this extremely slow-burning cycle amounted to 140–160 J.

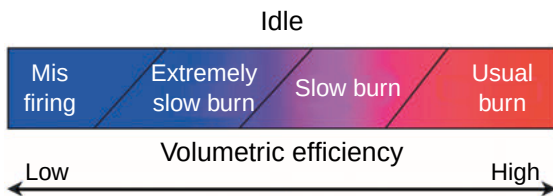


Fig. 11 Location of extremely slow burning cycle

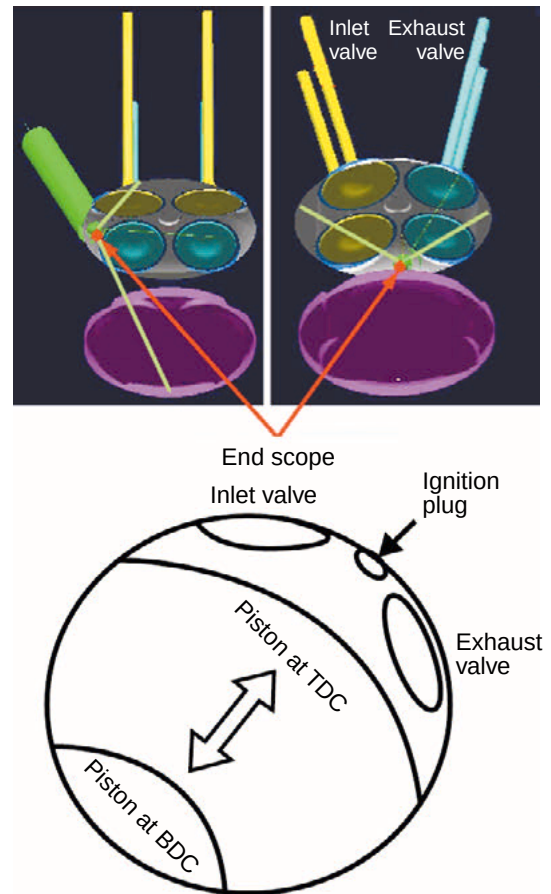


Fig. 12 Observation area in cylinder

5. Observations of the inside of the cylinder

Examination of the cycles immediately preceding the cycle in which backfire occurred showed that these were all extremely slow-burning cycles. This suggested that the extremely slow-burning cycle was the source of ignition for backfires. We hypothesized that an extremely slow-burning cycle has such a slow burning velocity that it does not finish combustion during the combustion stroke, and so its combustion continues into the exhaust stroke. Backfires can be considered to take place when the inlet valve opens during the latter part of the exhaust stroke, and the new air-fuel mixture enters the hot cylinder where it undergoes combustion, so that the backfire occurs during the intake stroke.

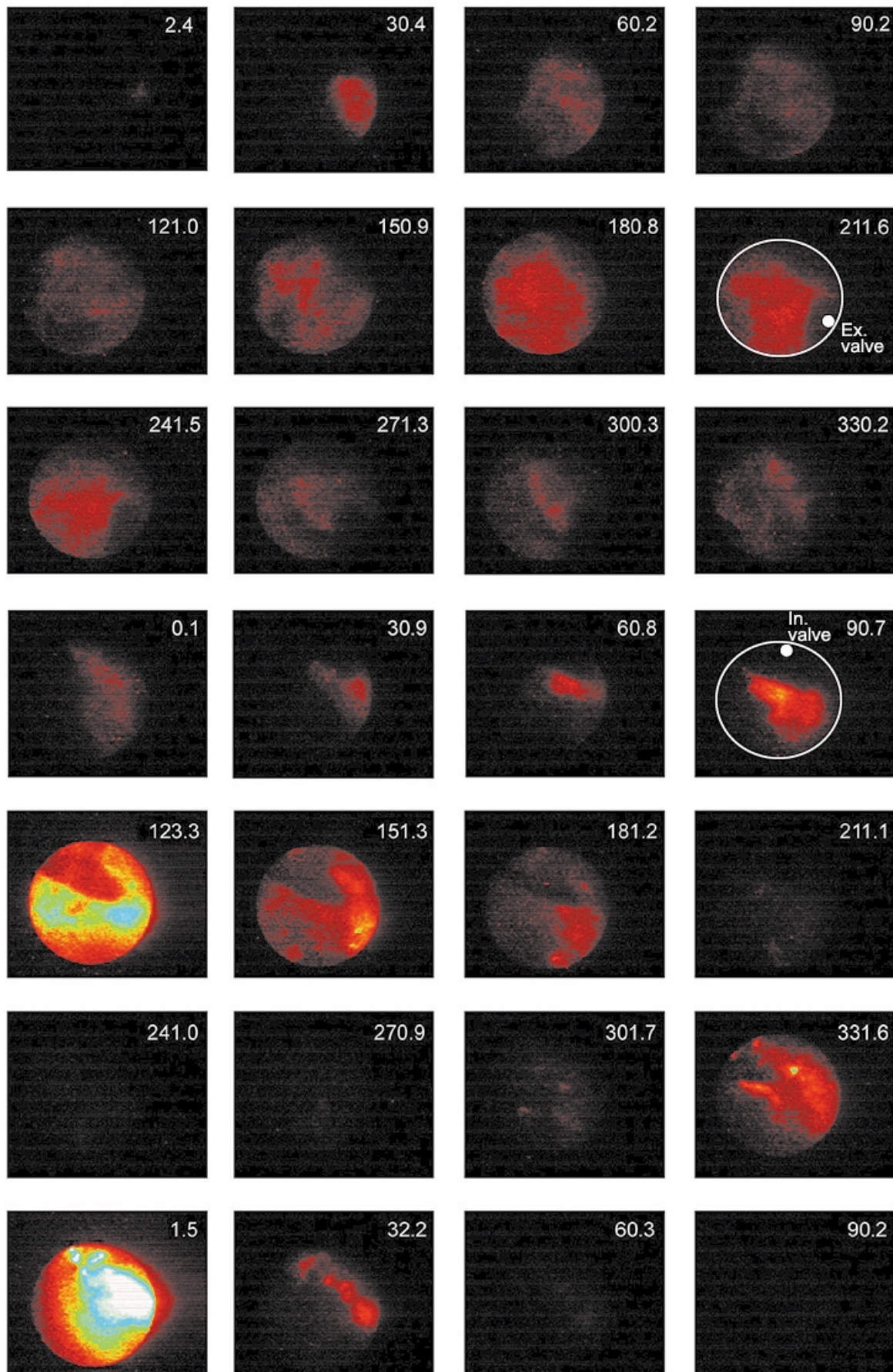
The continuation of combustion during the exhaust stroke can be confirmed by measuring the cylinder temperature difference between cases when extremely slow-burning cycles occur and when they do not. However, it has been difficult to measure such temperatures accurately under the high-temperature, high-pressure conditions inside the cylinder, and moreover within a period of just several milliseconds. Consequently, visual observation inside the cylinder was implemented.

A hole 10 mm in diameter was opened in the side wall of the test engine combustion chamber to mount an endoscope made by AVL List GmbH. The top illustration of Fig. 12 shows the area of observation in the combustion chamber. The sectorial portion of the green lines in this

figure is the observation area. The bottom illustration of Fig. 12 presents a schematic view of the combustion chamber interior as observed through the endoscope.

A C5836-60 image intensifier made by Hamamatsu Photonics K.K. was attached to the rear end of the endoscope, and the images were recorded using a Kodak EKTAPRO HS motion analyzer model 4540, which is a high-speed CCD camera. This was used to photograph the continuous sequence of phenomena occurring from when the extremely slow-burning cycle began to after the backfire took place. The recording rate was 4500 frames per second. Fig. 13 presents images recorded at approximately every 30° of crank angle. The figures inside these images show the crank angle. Although the original images are monochrome, the images in Fig. 13 have been put through a process of coloration based on differences in luminance.

Here, it can be seen how ignition begins from the spark plug at 15° before top dead center, and how the flame then spreads after top dead center. Approximately 130° after top dead center, the exhaust valve starts to open, but this can only be verified on the images at points past 180° after top dead center. After this, combustion continues during the exhaust stroke, or in other words, combustion continues even though the exhaust valve is open. The images show that combustion then continues up to the intake stroke. The inlet valve opens in the vicinity of 30° after top dead center and combustion begins in the area of the inlet valve, and backfiring takes place.



Note: Values in photos indicate the crank angle (deg).

Fig. 13 Photographs of burning in cylinder

It can be seen that intense combustion is occurring especially in the vicinity of 120° after top dead center. Then, the backfiring ends and the transition from the intake to the compression stroke takes place. Next, combustion again takes place during the compression stroke after a point around 300° after top dead center has been passed. A bright burning condition similar to diffusive combustion occurs in the area around top dead center, then combustion stops after top dead center. The ignition timing is 15° before the top dead center, which means that this is not ignited by the spark plug. The image also suggests that ignition is not from the ignition plug.

We reason that this phenomenon occurs as described below. Since backfiring occurs during the intake stroke, the temperature inside the cylinder rises at the end of the intake stroke, when the compression stroke begins. As a result, self-ignition takes place during the compression stroke and combustion begins.

Therefore, observation of conditions inside the cylinder when backfiring occurs led to the determination that the source of ignition for backfires is an extremely slow-burning cycle in which combustion continues even during the exhaust stroke.

6. Relationship between the extremely slow-burning cycle and backfires

The results of the investigation into the relationship between backfire generation frequency and extremely slow-burning cycle occurrence frequency are shown in Fig. 14. The backfire test device described earlier was used to snap the throttle approximately 250 times with the engine under no load. The frequency of backfire generation and the frequency of extremely slow-burning cycles occurring immediately before the throttle was opened were both measured. This test was carried out a total of six times, with the air-fuel ratio being changed across the range of 9.5 - 13.0.

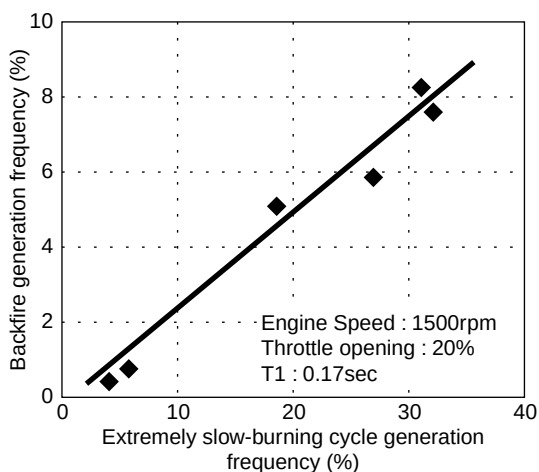


Fig. 14 Relationship between the extremely slow-burning cycle and backfiring

This revealed that a proportional relationship generally exists between the frequency of occurrence of backfires and of extremely slow-burning cycles. The slopes of these straight lines are about 0.25 and 1 or less, which indicates that there are cycles in which backfire generation will not occur even when cycles of extremely slow burning do occur. This is thought to be because the extremely slow-burning cycles, of course, have variations of the combustion itself, and also because the frequency of occurrence varies with the timing of inlet and throttle valve opening.

6.1 Cases of other test engines

The relationship between backfire generation frequency and extremely slow-burning cycle occurrence frequency was confirmed using a different test engine. Table 3 shows the main specifications for that engine, which is designated engine B, while the test engine described up to this time is designated engine A. Fig. 15 presents the relationship between backfires and extremely slow-burning cycles using the same axes as in Fig. 14.

A proportional relationship also exists between backfires and extremely slow-burning cycles in test engine B, and it lies generally along the same straight line as for engine A. The fact that two engines with these widely varying specifications show generally the same tendency can be taken to suggest that similar tendencies are also to be found in other engines.

Table 3 Major specifications of test engine B

Engine type	4-stroke single cylinder
Cooling type	Air-cooled
Bore × stroke	100 × 82
Compression ratio	8.3
Valve train	SOHC 4-valve
Maximum power (kW/rpm)	28.1 / 5750
Maximum torque (Nm/rpm)	52.3 / 4500
Ignition timing	8° (BTDC)

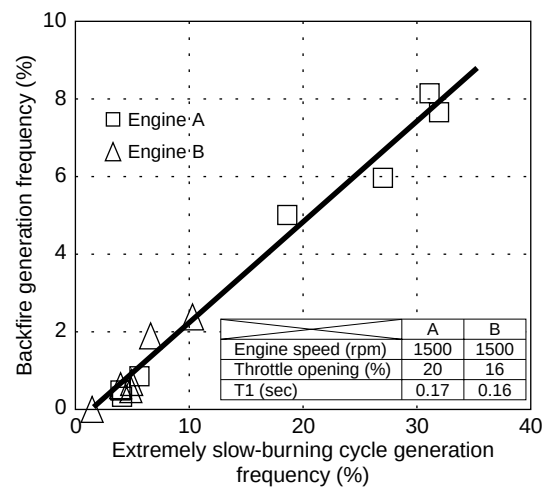


Fig. 15 Relationship between the extremely slow-burning cycle and backfiring with a different test engine

6.2 Factors affecting extremely slow-burning cycles

Factors that affect burning velocity include the air-fuel ratio, compression ratio, squish area, and swirl ratio. Tests were conducted of the relationship between extremely slow-burning cycles and backfires with different values for these four items. The results are shown in Fig. 16. It is apparent that, in general, changes to specifications that increase burning velocity cause a reduction in the frequency of extremely slow-burning cycle occurrence, which results in a corresponding reduction in the frequency of backfire generation. Even in such cases, a proportional relation continued to exist between the frequency of occurrence of backfires and of extremely slow-burning cycles.

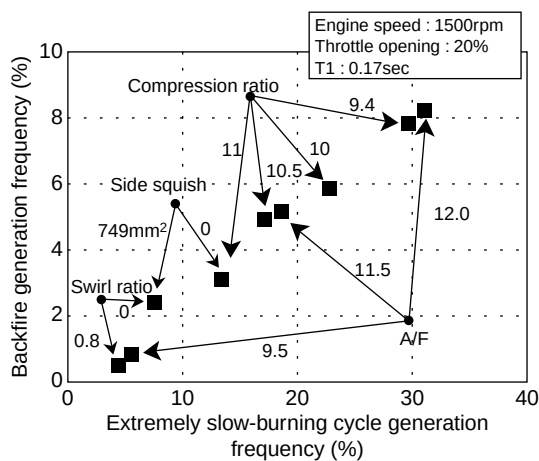


Fig. 16 Factors affecting extremely slow-burning cycles

7. Conclusion

We have analyzed the mechanism of backfire generation, for which no systematic analysis had been previously conducted. As a result, we have determined that other causes for backfire generation exist in addition to those considered in the past. The conclusions reached through this research are summarized below:

- (1) Extremely slow-burning cycles occur while reducing engine speed under a no-load condition, and combustion continues in these cycles even during the exhaust stroke.
- (2) Backfires occur with extremely slow-burning cycles acting as a source of ignition.
- (3) Combustion by self-ignition take place during the compression stroke immediately following backfire generation.
- (4) The occurrence of extremely slow-burning cycles can be reduced by increasing burning velocity.

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Masaya ADACHI

Development of an On-Board System for the Simultaneous Analysis of Tailpipe Exhaust and Ambient Air

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ABSTRACT

An on-board sampling and analysis system has been developed, capable of measuring on-road emissions at extremely low levels. Ambient air quality can also be monitored and compared to tailpipe emission levels in real time during actual roadway driving. This FTIR-based system can measure several chemical species, including HC, NO_x, and CO. Hardware and software were developed to facilitate the monitoring of target species at concentration levels below 1ppm, in the harsh environment of an on-road vehicle. Countermeasures were required to mitigate or eliminate the effects of chemical interferences and vehicle motion. Algorithms were developed to calculate results on a mass basis (g/mi), thereby allowing such data to be acquired in real-world conditions outside a laboratory environment. This system has been used in the development of near-zero-emission vehicle technology; it has allowed instant assessment of vehicle software tuning during actual roadway driving. It is currently being used to determine the true impact of various advanced-technology vehicle classes upon the environment. In this capacity the system is performing a dual role as an on-board analyzer and also as a high-sensitivity ambient air monitoring station.

1. Introduction

1.1 Background

Tailpipe emission rates are commonly determined by operating a vehicle on a chassis dynamometer under strictly controlled, reproducible conditions. For example, the US FTP-75 emission test cycle is used in California to assign emission categories; i.e., LEV, ULEV, etc. However, to achieve a more complete assessment of a vehicle's impact upon the environment, laboratory testing must be augmented by evaluating real-world emission performance. This had previously been attempted by mounting bulky measurement equipment in a vehicle, or by towing such equipment behind in a trailer. The performance of the vehicle was thereby altered, and true real-world emission data could not be collected. Furthermore, instrument calibration and data reduction were complicated, due to the use of several discrete measurement devices, each with its own sample inlet, data format, sampling rate, etc.

A new concept was therefore required to address these issues. To this end, a relatively compact, integrated on-board sampling/analysis system has been developed. This system can measure the concentration of several chemical species from the tailpipe exhaust and ambient air simultaneously, while the vehicle is driven on the highway. In addition, the system can be easily modified to function as a stationary,

highly sensitive air monitoring station.

This system has been used in the development of advanced low-emission vehicle technology, and to confirm the robustness of such technology under varying driving conditions on actual city roadways⁽¹⁾. The data from this on-board analysis system can also be used to develop better estimates of advanced-technology emission system inventories for use in air quality simulation studies.

1.2 Objectives

Advancements in emission control technology have led to the development of vehicles with progressively lower tailpipe emission levels; in some cases, the concentration of criteria pollutants approaches zero. Therefore, the target for this project was set to these "near-zero" concentration levels. These concentrations are similar in magnitude to those encountered in ambient air. The following requirements were placed upon the development of this system:

- Target components = NMHC, NO_x, and CO
- Limit of detection (LOD) near 0.1 ppm, with acceptable accuracy and precision
- Minimal weight burden on vehicle
- Robust and verifiable
- Integrated system with real-time data acquisition and display

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1.3 Choice of analytic technique

Given the requirements above, it was decided to use Fourier Transform Infrared Spectroscopy (FTIR) as the basis for this on-road analyzer. Acquiring data with a single system would facilitate synchronization with engine parameter data (simultaneously recorded on a separate computer). Fieldwork logistics would be simplified because an FTIR requires no daily calibration per se. As an added benefit for future research, FTIR data can be re-processed to reveal concentration data of other components. (This is possible because the spectrum recorded by the FTIR contains concentration information for all infrared-sensitive components.) However, the use of this technique presented some special challenges:

- Spectral interference (primarily from H₂O and CO₂)
- Signal-to-noise ratio affected by external vibration and instrument orientation
- Demanding accuracy and LOD requirements
- Response time

Other single-system approaches that were considered for this project included non-dispersive IR, high-resolution mass spectrometry, and gas chromatography - mass spectrometry. These were rejected due to sensitivity, size, and response time considerations, respectively.

2. Experimental

2.1 System overview

The initial test platform chosen for this project was a prototype Honda ZLEV (Zero Level Emission Vehicle). The

exhaust emissions from this gasoline-powered vehicle are less than 1/10 ULEV levels⁽²⁾. An overview of the sampling and measurement systems is shown in Figure 1. Intake air was sampled from the front of the vehicle, and an exhaust sample was taken via a heated line inserted into the tailpipe. The samples were drawn through a water-removal system by a dual-head pump, and then directly into the gas cells of twin FTIR spectrometers. System power was supplied through a DC-AC inverter from three nickel-metal hydride battery cells. Results were processed by two networked 400 MHz computers, and combined onto a single touch-screen flat panel display.

2.2 FTIR principles

In infrared gas analysis, IR light is passed through a sample in a gas cell. Some of the infrared light is absorbed by the sample and some is passed through. The measurement of these absorbed and transmitted wavelengths of infrared light constitute an infrared spectrum; see Figure 2. Since no two chemical species have the same infrared spectrum, the spectrum serves as a fingerprint for identifying different gas components in the sample. The intensity of the infrared absorption indicates the quantity of the component in the sample.

Unlike dispersive IR spectrometers, an FTIR instrument can acquire an entire spectrum in less than one second; i.e., it can record all frequencies simultaneously in this time. FTIR uses an interferometer with a moving mirror to modulate the light by phase, producing an interferogram. This interferogram is then Fourier transformed, thereby

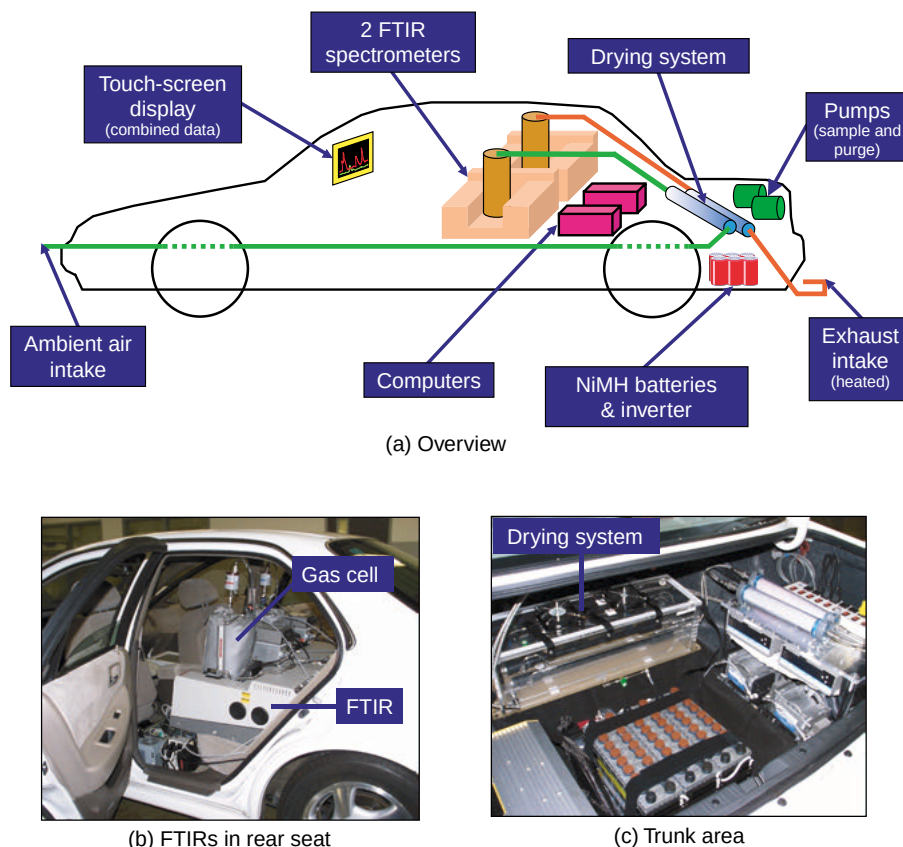


Fig. 1 On-board sampling/analysis system

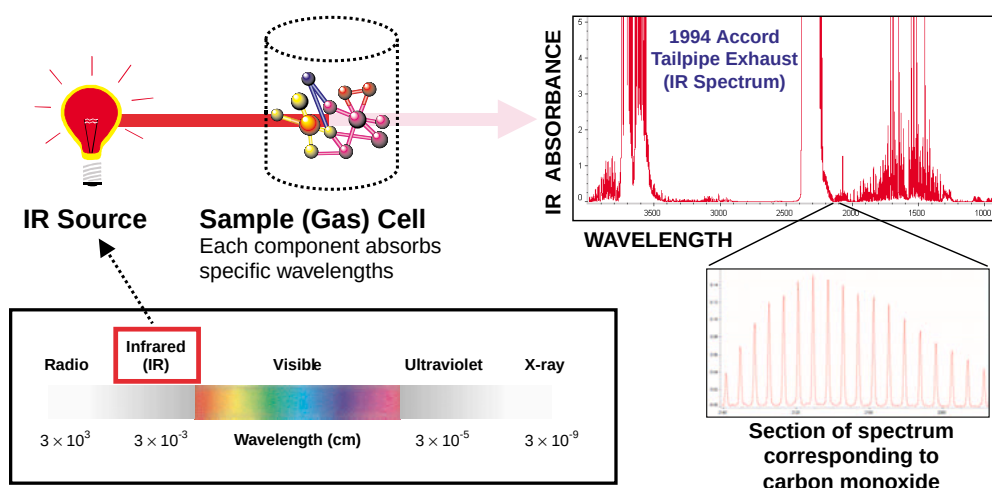


Fig. 2 Principle of infrared spectroscopy

producing the sample infrared spectrum in frequency or wavenumbers (cm^{-1}).

The infrared spectra were analyzed using a “classical” least squares (CLS) pattern recognition algorithm⁽³⁾. This data reduction technique compares spectra of reference gases to sample spectra, determining a degree of fit of the reference gas spectra to the sample spectrum. The degree of fit is then reported as a concentration.

2.3 Gas cell

One of the key components of an FTIR system is the gas cell. The sensitivity of the instrument is directly proportional to the amount of sample that is exposed to the infrared beam. FTIR gas cells typically use the *White* optical design, in which multiple internal reflections occur. The cell is classified based upon the total “path length” of these reflections. For example, the gas cell chosen for this work was less than 50cm in length, yet had a path length of 10m.

Initially, a 2m path length gas cell was evaluated for this application, due to its favorable size and weight characteristics. It quickly became clear that the 2-meter cell would not achieve the target noise level, and therefore the required LOD could not be attained. Acceptable results were

obtained upon switching to a 10-meter cell; Figure 3 shows the improvement in sensitivity. The strong infrared absorption bands from (exhaust) water became a problem with the increase in path length, as discussed later in this paper. The longer sample mixing time resulting from the 10m cell’s larger internal volume was also of some concern, but the final system provided adequate temporal resolution. The 10m cells can clearly be seen in Figure 1b.

2.4 Calibration

The initial calibration of an FTIR gas analyzer is performed similarly to that for a non-dispersive infrared or a chemiluminescent analyzer. In this case, spectra of single-component reference gases (in a nitrogen matrix) were acquired, and the resulting patterns used in a CLS spectral data analysis. The absorbance response of a reference gas coupled with its concentration determined the calibration. A sequence of concentration steps was added in order to calculate a non-linear correction, when necessary.

A different approach was required for the calibration of the non-methane hydrocarbon component. Instead of using reference spectra for the individual hydrocarbon species, it was desirable to use a spectrum of a representative mixture of hydrocarbons as the reference gas in the CLS data analysis. The mixture used to calibrate the FTIR analyzer was a bag sample of engine exhaust from a prototype ULEV sampled during the first 30 seconds after a cold start. A concentration for the non-methane hydrocarbon component in the bag sample was assigned using careful bench FID and gas chromatography analyses.

2.5 Sample treatment

Exhaust gas from gasoline-powered internal combustion engines contains roughly 10 to 15 mole% CO_2 and H_2O . Both pose interference challenges for the FTIR analysis. Dilution was never considered; the resulting sensitivity loss would have been unacceptable for this project. The CO_2 problem was mitigated by the careful choice of analytic regions in the spectra, aided by the 0.5 cm^{-1} spectral resolution. However, the H_2O spectral features proved too pervasive (Figure 4); a similar countermeasure was

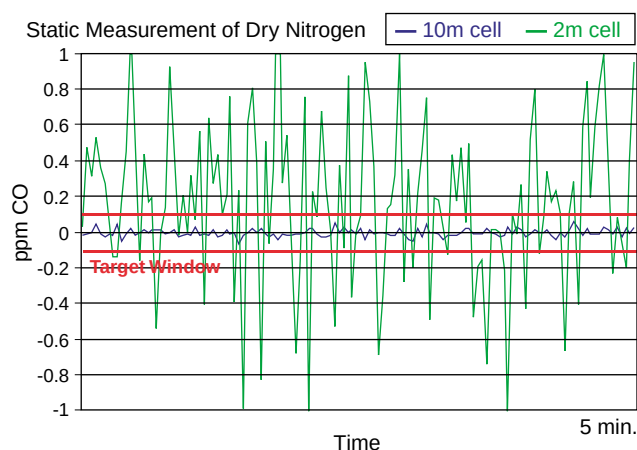


Fig. 3 Effect of gas cell path length upon measurement precision

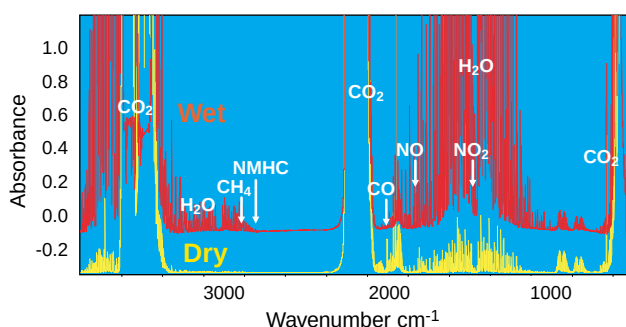


Fig. 4 Effect of water upon exhaust spectrum
Note that water occludes the spectra of many target species

precluded. Consequently, various techniques were tried to remove the water from both the ambient and tailpipe samples. A technique using an ice-chilled aluminum heat exchanger, water trap, and desiccants efficiently removed the water. As expected, NO_2 was unfortunately removed with equal efficiency. Studies with NO , CO , and C_3H_8 (propane) indicated no loss in this water-removal system. Since actual exhaust gas contains virtually no propane, a 15-component hydrocarbon mixture was also tested. Analysis by gas chromatography⁽⁴⁾ revealed that substantial hydrocarbon losses were occurring as early as C_6 , necessitating the search for a different drying technology. (Data are shown later.)

A drying system based on the polymer Nafion[®] was assembled for evaluation. Nafion has a Teflon[®] backbone, with periodic side chains of perfluorated ether terminating in a sulfonic acid group. These acid groups form “ionic channels” that extend through the walls of the Nafion tubing. Each sulfonic acid group can absorb up to 13 molecules of water. Assuming the partial pressure of water in the sample exceeds that external to the Nafion tube, water molecules will travel along the ionic channels and be released outside the tubing. This mechanism is selective for water; the only (potential) exhaust species that could experience some loss are alcohols, ketones, and ammonia.

For this application, multi-tube Nafion dryers manufactured by Perma Pure Incorporated were selected for use, after comparison to a single-tube configuration. This dryer consisted of a bundle of 0.76mm (OD) Nafion tubes contained within an aluminum shell. Fittings near each end of the shell permitted a purge gas to flow countercurrent to the sample; Figure 5. Two dryers in series were used for both of the sampling systems.

The exhaust sample was heated (70°C) from the tailpipe through half of the first dryer. There were two reasons for this: (1) the interaction of liquid water with Nafion is endothermic, and leads to a cascade failure reaction. Therefore the exhaust gas must be kept above the dew point temperature until most of the water has been removed. (2) The initial water absorption and transport by Nafion proceed as a First Order Kinetic reaction, in which absorption becomes more rapid with increasing temperature⁽⁵⁾.

The second dryer for each sample stream was blanketed with ice water. The purpose here was to coax more of the retained water from the sulfonic acid groups, since this

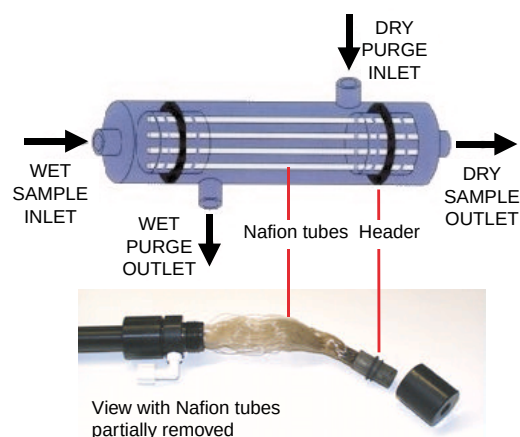


Fig. 5 Concept of Nafion-based sample drying

residual water becomes the limiting factor after the initial First Order reaction. Leckrone, et al, have recently quantified this effect⁽⁶⁾.

The ultimate performance of the Nafion dryers is dependent upon the partial pressure of water in the purge gas. Dry nitrogen was considered for this application, but the weight and safety ramifications of carrying a compressed gas on board a passenger vehicle precluded this choice. Ambient air would normally have an insufficient dew point for this application. However, good results were obtained by pulling ambient air through a specially constructed dehydrating system with a vacuum pump. The resulting negative pressure in the dryer shell lowered the partial pressure of the water in the treated ambient purge gas to an acceptable level. The water content of the exhaust gas sample was decreased to below 100ppm using this technique, equivalent to a 99.9% reduction.

A comparison of the Nafion-based dryer performance with that of previous dryer versions can be seen in Figure 6. The 15-component hydrocarbon mixture was passed through a given drying system at 7 L/min, and collected into Tedlar[®] bags. These bags were subsequently analyzed on a gas chromatograph, and the resulting concentrations compared to the label values. The Nafion system produced acceptable results for all components, with only a slight (4%) loss for a C_9 aromatic.

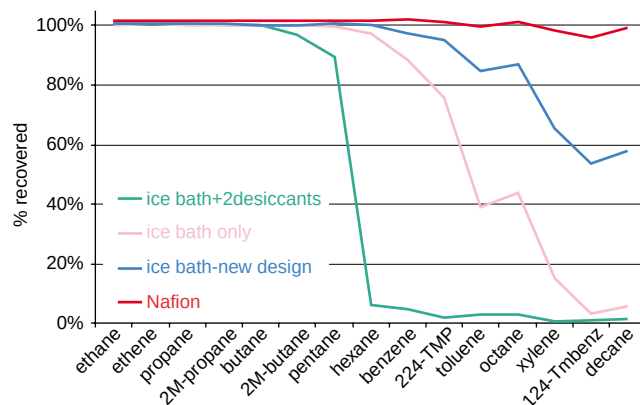


Fig. 6 Comparison of drying system performance, in terms of hydrocarbon recovery. (Gas chromatography data)

A more severe test of the drying system was performed⁽⁷⁾, in which CO₂ (11.5%) and H₂O vapor (10%) were injected into the reference gas streams. As previously noted, these compounds are a potential source of interference for FTIR, and are present in exhaust gas at approximately these concentration levels. In addition to the 15-component hydrocarbon mixture, CO and NO reference gasses were used for this study. The gas chromatography results shown in Figure 7a indicate good recovery for the individual hydrocarbons in the mixture; even when the sample matrix includes CO₂ and H₂O, there was no preferential loss of hydrocarbon species. (The data have been corrected for the effect of dilution, and propane was omitted due to contamination.) Figure 7b indicates the same good recovery performance for CO and NO. It is important to note in this figure that the FTIR system was used to measure the recovery. Therefore, these results also demonstrate the accuracy of the on-board FTIR system, including the ability to measure the full range of hydrocarbon species encountered in vehicle exhaust. This level of accuracy was attained despite the presence of CO₂ and H₂O in the sample gases, indicating that interference from these compounds had successfully been avoided.

2.6 System environment

FTIR instrumentation is basically designed for use on a vibration-free laboratory bench in a controlled environment. Relocating such instrumentation to the interior of a passenger vehicle is challenging, given the unpredictable dynamics resulting from road surface irregularities and vehicle motion. One of the initial concerns was the effect of the vehicle acceleration force vector upon the interferometer (moving mirror) component of the FTIR. Nicolet suggested that this force should be perpendicular to the motion of the mirror; (Figure 8). This configuration did in fact prove to be best, although the mirror was still subjected to coaxial forces during vehicle turning maneuvers.

The main area of concern was the unpredictable mechanical vibration and shock transmitted to the FTIR as the vehicle encountered imperfections in the road surface. An attempt was made to characterize these forces through the use of an accelerometer, but the test results only confirmed our prediction that the vibration frequencies varied widely. Several studies were performed involving dampening systems, changes in FTIR location, etc. The final conclusion was that the FTIRs were best isolated by simply placing them on the rear seat of the vehicle. As an added benefit, placing the FTIRs side-by-side on the rear seat exposed them to an identical environment, and the instrument electronics could be kept within the specified temperature range.

Finally, each FTIR received some internal modifications; refer to Figure 8. For example, the laser and IR source were secured more firmly to prevent movement.

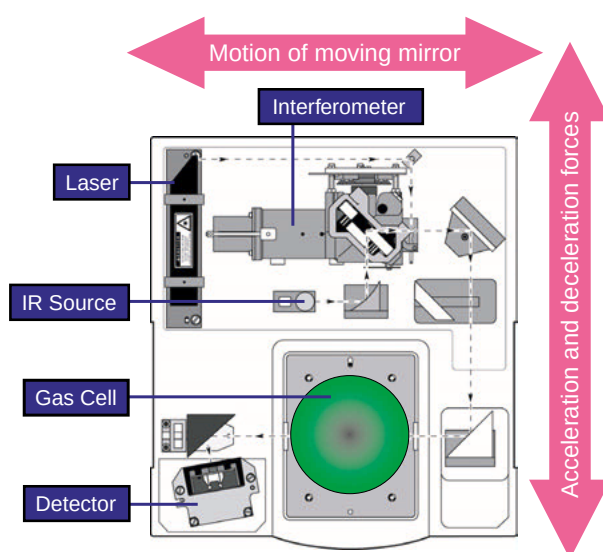


Fig. 8 Overhead view of FTIR interior

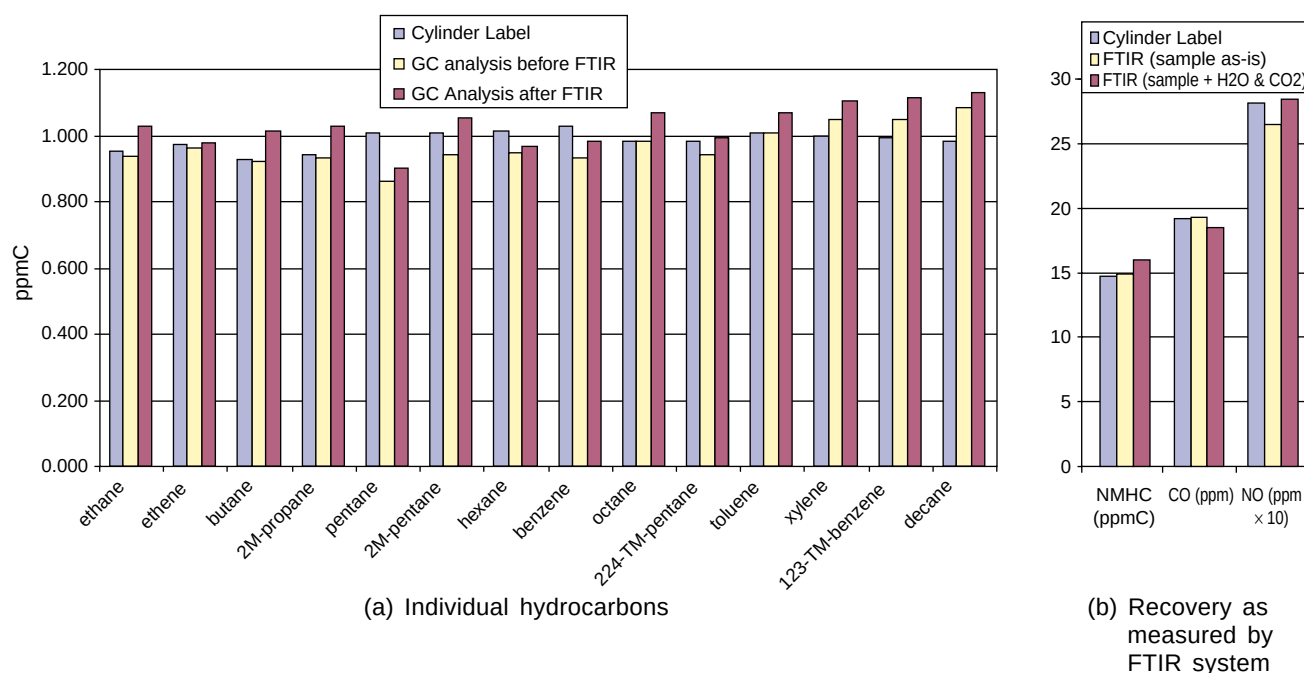


Fig. 7 Component recovery through final version of drying system
Sample injected with CO₂ and H₂O.

3. Results and Discussion

3.1 System specifications

The final system specifications were as follows. Information is shown for a single sampling/analysis system.

Analytic system

FTIR base unit : Nicolet model Protégé™ 460
 Detector : MCT, LN₂ cooled
 Spectral resolution : 0.5 cm⁻¹
 Gas cell : 10m path
 Sample conditions : 70°C, 93kPa (absolute)
 Sample flow rate : 12 L/min (max)
 Scans/sample : 2 complete

Drying system

Nafion dryer : 0.76mm (OD) tube bundle
 Purge gas : dehydrated ambient air under negative pressure

3.2 System validation

In addition to the previously mentioned accuracy confirmation, numerous other validation studies were performed. A typical example can be seen in Figure 9. In this case, sensitive bench instruments in use as ambient air analyzers were used for reference. Actual exhaust gas samples were analyzed in addition to reference gases. In the case of the reference gases, the on-board FTIR results were also compared to the gas cylinder label concentrations. To prepare the exhaust samples, a ZLEV-class vehicle was started under cold conditions, while a slipstream of undiluted exhaust gas was pumped into a large Tedlar bag. As can be seen in Figure 9, the FTIR system compared very favorably with the bench analyzers for the exhaust samples and gas mixtures.

3.3 On-road results

Following an on-road test program in Los Angeles, the system was sent to Japan to evaluate its performance in the Tokyo area. A representative example from the Tokyo testing is shown in Figure 10. Roadway (ambient) pollutant concentrations were found to be roughly similar to those measured in the Los Angeles area. Traffic conditions were typically stop-and-go, alternating with cruise periods.

During the course of testing, severe conditions were sometimes encountered in which the roadway pollutant concentrations of CO and NMHC were an order of magnitude greater than those shown in Figure 10. Long tunnels in the Tokyo area and highly congested traffic near Detroit are examples of these conditions. As can be seen in Figure 11, the ZLEV test platform was unaffected by this severe ambient air quality.

The data shown in Figures 10 and 11 are not smoothed, other than two-scan averaging performed by the FTIRs. This produced an approximate two-second sampling rate for this data set. The detection limit for all target compounds was approximately 0.1 ppm for such on-road testing. For this road test, NO₂ remained below the detection limit for both sample streams; the observed NO_x concentrations resulted from NO alone. In the rare cases where NO_x concentrations over 5 ppm were measured in the ambient sample during road tests, NO₂ was found to contribute approximately 20% to the total NO_x value. This ratio is similar to that expected in diesel vehicle exhaust⁽⁸⁾.

3.4 Off-road measurement of ambient air

As part of a cooperative research program with the University of Riverside, a project was initiated which required the measurement of ambient air quality at various

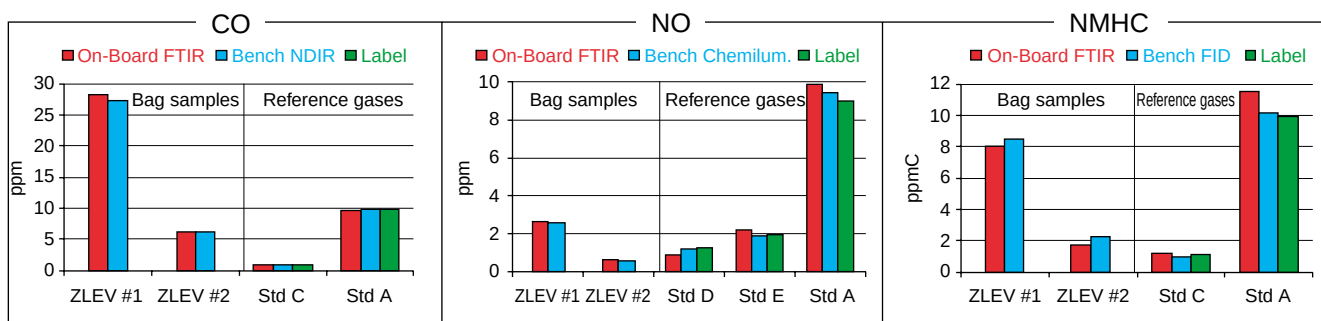


Fig. 9 Comparison of on-board analyzer and bench instrumentation

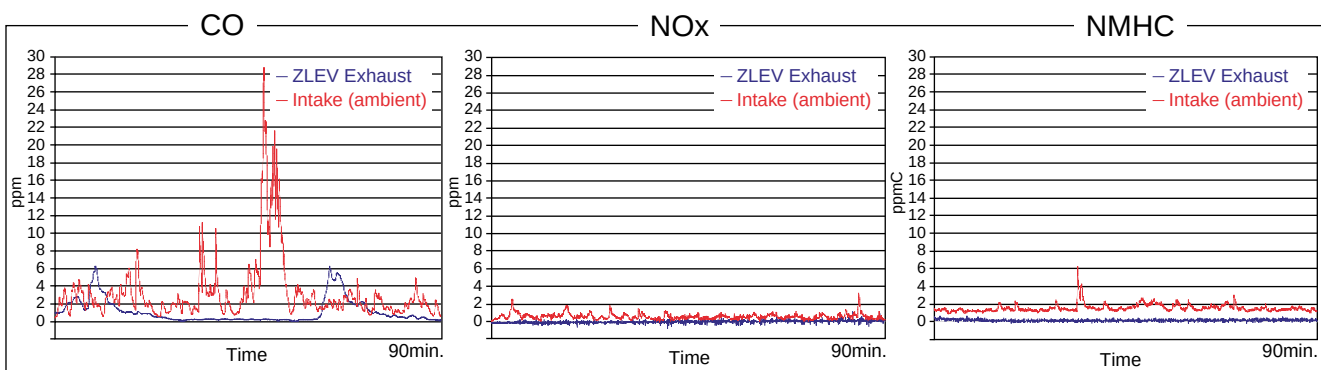


Fig. 10 Tokyo on-road test results. ZLEV with on-board analyzer

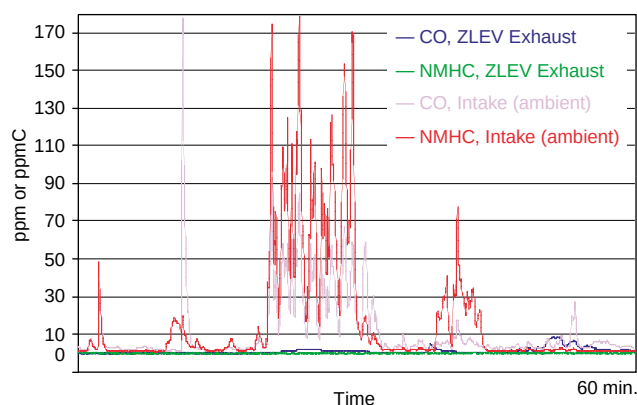


Fig. 11 Detroit on-road test results, severe traffic congestion ZLEV with on-board analyzer

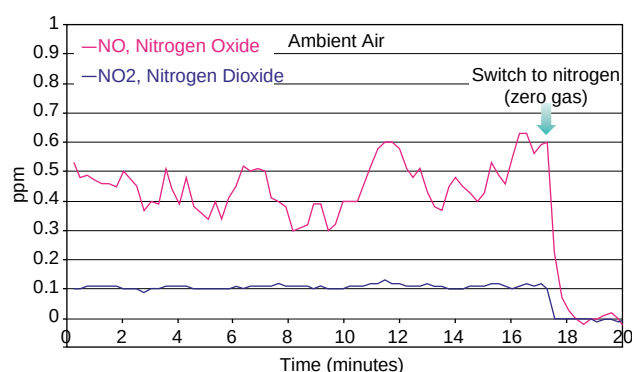


Fig. 12 Real-time ambient air measurement Shopping mall parking area, Azuza, California

sites in the Los Angeles area. The on-board FTIR system was recruited for this work. In this case, the system was optimized for stationary, ambient-only measurement, and was installed into a Honda Odyssey van. The Odyssey's GPS-based navigation system was used to record the location of each test site. The ambient measurements were made in areas away from the roadway, where pollutant concentrations had become equilibrated and/or mixed. This permitted the collected FTIR spectra to be averaged over a longer period; in this case 15-second averages were performed before the data point was recorded. These longer sampling intervals in a non-moving platform allowed for excellent signal-to-noise ratios, with a corresponding decrease in the limit of detection. Figure 12 indicates that even challenging species such as NO_x (NO and NO_2) could be clearly measured in sub-ppm concentrations. (The variance in NO observed in this figure is due to actual changes in ambient concentrations.) Subsequent calibration work performed with reference gases demonstrated that NO could be measured well below 0.1 ppm under these conditions.

3.5 Calculation of mass-based results

All data discussed to this point have been in terms of concentration, (e.g. ppm). If the tailpipe flow rate were known, the concentration data could be converted to mass results, (e.g., grams), by the following equation:

$$M = D \int C_t \times V_t dt$$

Where, for each component:

M = cumulative mass

D = density

C_t = concentration at time t

V_t = volumetric flow rate at time t

In reality, the conversion to mass-based results is somewhat more complex. In addition to the flow rate, sample delay time and instrument response characteristics must be considered. These steps are outlined below.

Flow rate determination. Measuring the tailpipe flow rate on a moving vehicle is a challenge. The concept of add-on flow measurement devices was discarded for this work, due to accuracy and size considerations. Instead, an estimate of the flow rate was determined by using data from the vehicle's ECU (engine control unit) to calculate the fuel usage rate. Examples of data extracted from the ECU for this purpose included fuel injector pulse duration, engine speed, and air/fuel ratio. Given that vehicle constants (such as exhaust system dimensions) and fuel composition were known, a sufficiently accurate estimate of the tailpipe flow rate could be calculated. Comparison with external methods of flow measurement, such as the CO_2 tracer technique, confirmed the accuracy of this method.

Sample delay time. The next step was to align this flow measurement with the FTIR's concentration measurement. The delay time between the sampling point and the instrument was determined experimentally with pulsed injection of reference gases. The previous equation was modified to include this delay time, designated d :

$$M = D \int C_{t+\delta} \times V_t dt$$

Instrument response. The FTIR gas cell has a volume of 2L, and the sample flow rate is about 6 to 12 L/min with this system. Therefore, the concentration data resulting from fast changes in pollutant concentrations will appear somewhat smoothed, due to this relatively slow response time. To address this response characteristic, the data were processed using a "well-mixed flow cell" model; see Figure 13. The final equation shown in this figure was utilized to estimate the actual inlet concentration of the FTIR gas cell prior to mixing. The effect of the delay time and instrument response correction can be seen in Figure 14, in which a very good comparison has been achieved with dynamometer output during an FTP cycle.

Comparison with dynamometer CVS results. As a final validation of the techniques described above, testing was performed at UC-Riverside in which the FTIR system was run concurrent with standard dynamometer methods over various cycles. To ensure that sufficient pollutant concentration were produced to facilitate this comparison, the FTIR system was moved from the ZLEV platform into an LEV Accord. Figure 15 shows the results from the third phase of a Unified test cycle, (similar to an FTP cycle). These data represent the average of three separate runs. The grams/mile of CO , NO_x , and NMHC were in good agreement between the FTIR-based system and the UCR CVS system. Results from CO_2 and CH_4 (not shown) were also good.

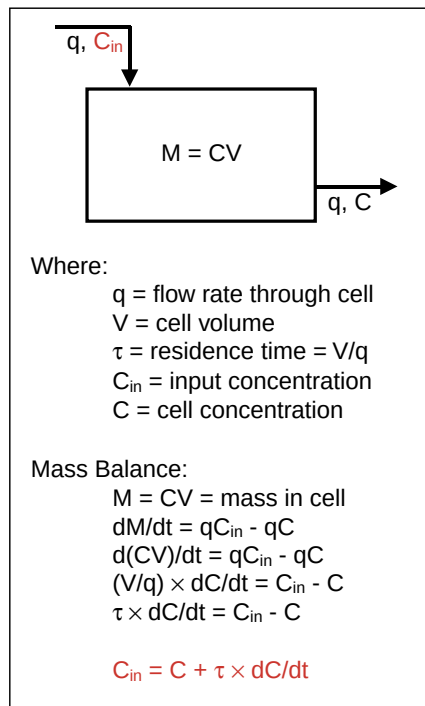


Fig. 13 FTIR gas cell mixing model;
Determination of input concentration

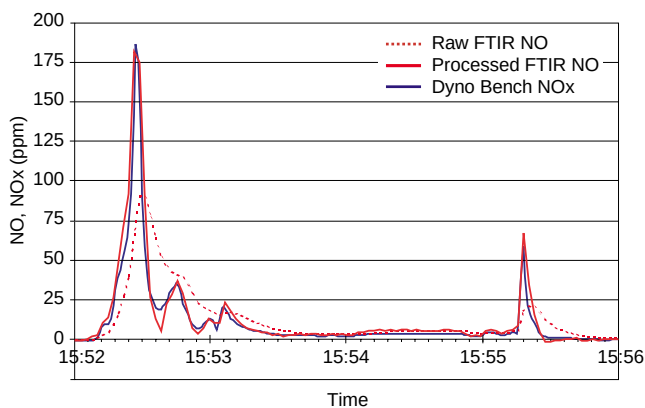


Fig. 14 Comparison of on-board FTIR system with
dynamometer bench analyzer during FTP
Vehicle = ZLEV

This was generally the case for all of the test cycles and phases, although NMHC results were occasionally in relatively poor agreement. The FTIR calibration was subsequently confirmed over the range of concentrations encountered during these tests. Therefore, the deviation was likely attributable to another source, such as CVS contamination.

4. Conclusion

An on-board analyzer system has been developed, capable of measuring exhaust species from advanced-technology vehicles in real time. Accuracy was validated through laboratory studies, and overall performance was confirmed through on-road data collection.

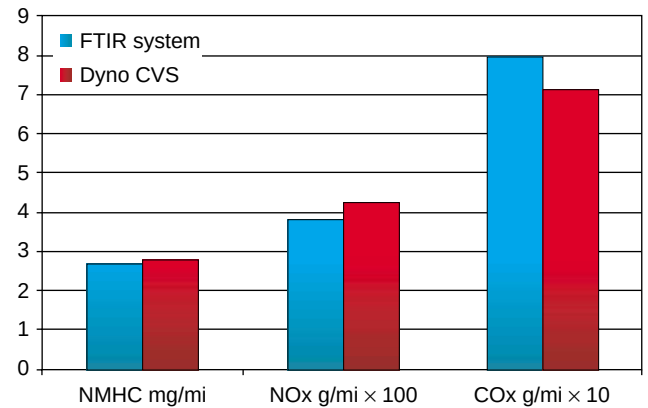


Fig. 15 Mass-based comparison of on-board FTIR
system with dynamometer CVS.
LEV vehicle, 3rd phase of Unified Cycle.

Therefore, it is now possible to measure exhaust and ambient species in units of grams/mile during actual highway driving, with accuracy equivalent to that of dynamometer CVS data. Several chemical species can be measured simultaneously, in addition to the standard criteria pollutants. Some examples of new technologies and techniques developed during this project :

- Various techniques associated with the installation of a complex sampling/analysis system into a passenger vehicle, including electricity management and vibration mitigation.
- Analytic methods that attenuate FTIR spectral interference, including a unique calibration method for NMHC.
- A new sample treatment technique. The water content of exhaust gas can be reduced by 99.9%, with no loss of criteria pollutants.
- Custom software for real-time, simultaneous display of tailpipe and ambient data.
- Algorithms allowing the calculation of mass-based results.

This on-board analyzer is now in use as a development tool for advanced emission-control systems. It has also proven effective as a mobile air quality monitoring station.

Acknowledgements

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Computer-aided Engineering Technique for the Reduction of Exhaust System Noise and Back Pressure

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ABSTRACT

A technique has been established for predicting the effects of changing cold end parts in an exhaust system on engine output and exhaust discharge noise. This technique enables the sharing of such parameters as velocity and pressure due to the application of flow analysis and acoustic analysis. One-dimensional compressible fluid analysis was used to predict engine output, and a value derived by steady flow analysis using the three-dimensional finite volume method (FVM) was introduced for the effective flow area, which is one of the parameters. Further, it was clarified that exhaust discharge noise is determined by the sum of pulsating noise and flow generated noise. The pulsating noise was predicted by the application of acoustic analysis using the three-dimensional finite element method (FEM). For the FEM input boundary conditions, precision verification was carried out using both pressure fluctuation values derived by compressible fluid analysis and measured values from an actual vehicle, while velocity derivable by steady flow analysis was introduced as a parameter for flow-generated noise prediction. As a result, it was possible to predict the variations of engine output caused by changes in the exhaust system with an accuracy of $\pm 2\%$, and to predict exhaust discharge noise within ± 2 dB. This has made it possible to optimize muffler design by implementing three-dimensional computer-aided design (CAD) modeling to evaluate parts performance from the initial layout investigation stages.

1. Introduction

In conventional exhaust development, one of the main issues has been how to determine specifications so that the high-level balance required between the opposing requirements for high performance engine response and high muffling performance is achieved.

In recent years, there has been an increasing demand for the development of computer-aided Engineering (CAE) technology that can evaluate the function and performance of a 3D CAD model in the early stages of development before prototype parts are made.

This report gives details of the calculation method developed to meet the above needs for both acoustic and flow engineering. The method utilizes the finite element method (FEM) and computational fluid dynamics (CFD), etc.

2. Analysis Method

The analysis flow for exhaust noise analysis and engine output analysis is given in Fig. 1. The details of the

procedure are given in later sections. The aim of this project was to bring the accuracy of analysis up to a level where exhaust system performance can be evaluated without having to create prototype parts. The range for changes to parts handled by this project is highlighted by diagonal lines in Figure 1. These parts are positioned after the catalytic converter in the muffler system and are called the “cold end exhaust”.

Accordingly, both the calculated and actual values for the area immediately behind the catalytic converter were used for the physical values of pressure, velocity, temperature, etc., which are the analysis input conditions. Then, by varying the muffler system characteristics it was possible to study what effects the changes have on exhaust discharge noise, and on engine output.

3. Steady Flow Analysis

The general-purpose VECTIS code was used in the 3D steady flow analysis.

In terms of methods for digitization, the FVM (finite volume method) was employed for the flow parts, and the

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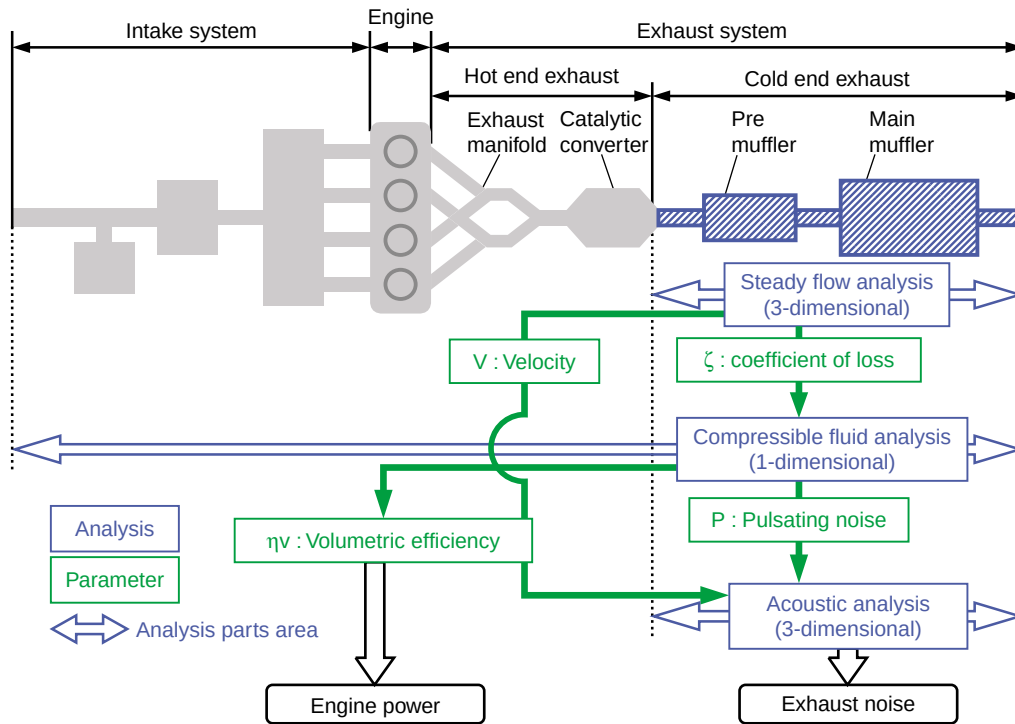


Fig. 1 Flowchart of analysis

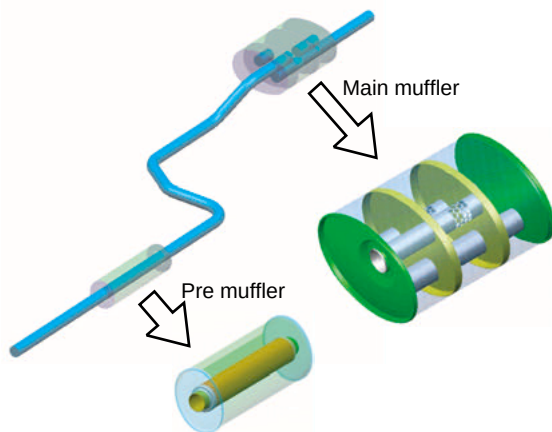


Fig. 2 Steady flow analysis model

SIMPLE method was applied for the velocity and pressure location calculations. The standard $k - \epsilon$ model was used as the turbulence model.

The analysis model used for verifications – an exhaust system with a pre-muffler and main muffler placed along the exhaust pipe – is shown in Fig. 2.

Fig. 3 shows a comparison of the main muffler configurations applied for this verification. Fig. 4 shows a representative FVM mesh model (a) and velocity analysis FVM example (b).

Boundary conditions were set as follows: pressure was provided at the inlet (1 to 6 kPa), the air was released at the outlet, and the temperature was a normal 27°C⁽¹⁾.

$$Q = S \cdot k \sqrt{\frac{2g\Delta P}{\zeta}} \quad (1)$$

Q : steady flow amount
 S : outlet section area
 ΔP : pressure difference
 ζ : loss coefficient
 k : flow rate coefficient

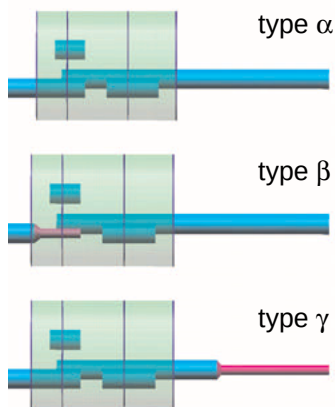


Fig. 3 Comparison model types

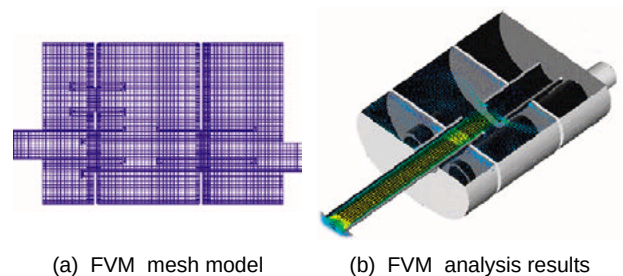


Fig. 4 Example of FVM mesh and analysis results

The following items were kept in mind when improving the accuracy of the steady flow analysis.

- (I) Method for modeling the air release area
- (II) Small shapes (punched holes)
- (III) Method for stabilizing digitalization
- (IV) Dependency of mesh size

Fig. 5 gives the results of comparison between steady flow rate analysis as determined by Equation (1), and the measured values for a flow rate test piece.

There is a correlation between the values, and calculation accuracy shows the favorable result of no more than 2% error.

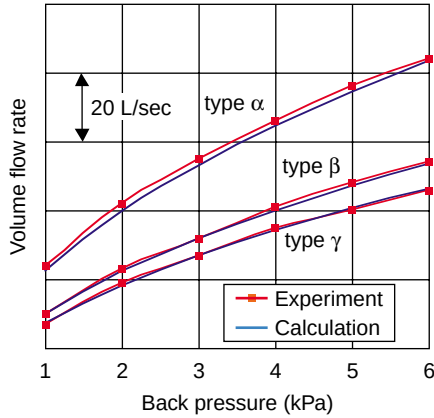


Fig. 5 Comparison of experiment and calculation results

4. Volumetric Efficiency Analysis

1D transient compression flow simulation was used for volumetric efficiency calculations.

Here, flow loss in the muffler can be converted, as shown in Fig. 6, into a single uniform pipe loss. The effective flow area (Z), which is required for volumetric efficiency analysis, was determined using Equation (2) and the flow rate (Q) determined in the aforementioned steady flow analysis.

$$Z = Q \sqrt{\rho / 2 \Delta P} \quad (2)$$

Z : effective flow area
 Q : steady flow amount
 ΔP : pressure difference
 ρ : density

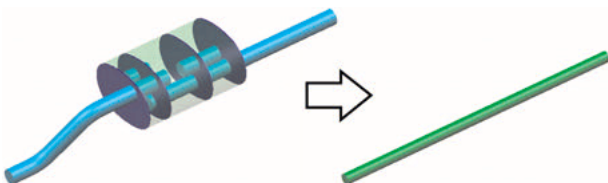


Fig. 6 Conversion of loss coefficients

Also, pipe length and muffler capacity were used for 3D modeling values. Fig. 7 gives an example of one dimensional volumetric efficiency analysis.

$$P_s = 2 \pi \cdot N_e T / (60 \times 1000) \quad (\text{kW}) \quad (3)$$

$$T = k \cdot \eta_i \cdot \eta_m \cdot \eta_v \cdot V_{st} \quad (\text{N} \cdot \text{m}) \quad (4)$$

P_s : engine output
 N_e : rpm
 T : torque
 k : coefficient
 η_i : thermal efficiency
 η_m : mechanical efficiency
 η_v : volumetric efficiency
 V_{st} : cylinder capacity

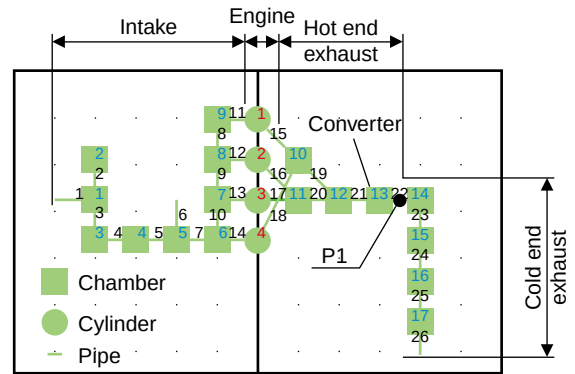


Fig. 7 One dimensional volumetric efficiency model

Fig. 8 shows a comparison between engine output and torque analysis results determined from Equations (3) and (4) and the results of measurements from engine bench testing.

There is a correlation between measured values and calculated values, and error is 2% or less. Thus, analysis accuracy is sufficient.

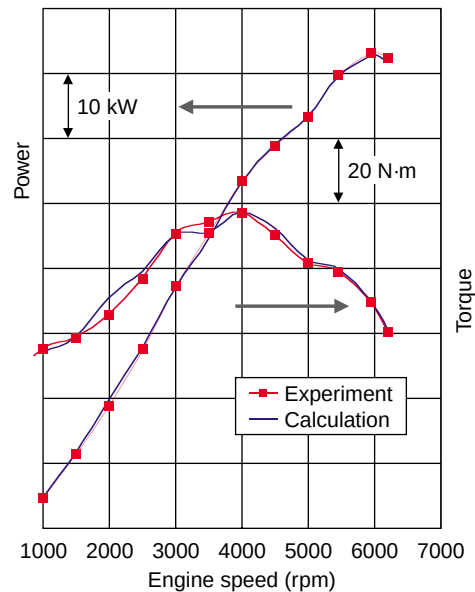


Fig. 8 Comparison of experiment and calculation

5. Exhaust Noise Analysis

5.1 Causes of Exhaust Discharge Noise

Exhaust discharge noise, as shown in Fig. 9, is comprised of pulsating noise (composed of basic order and harmonic components), resonance noise (due to the operation of amplitude within the exhaust pipe), and flow noise (from fluid motion).

A 4-cylinder engine was used in this project, and the effect of changes in muffler characteristics (structure and layout) on exhaust discharge noise pressure was verified through test and calculation results.

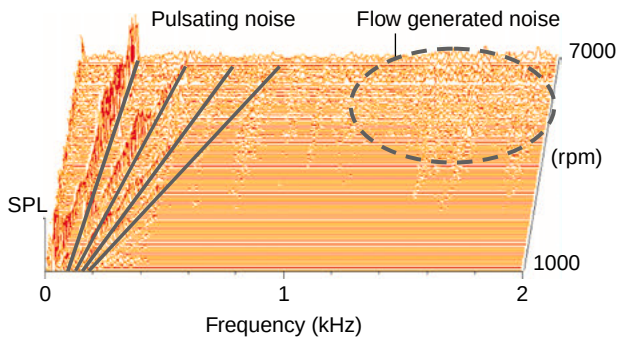


Fig. 9 Exhaust noise components

5.2 Analysis of Attenuation Amount using Acoustic Theory

5.2.1 Attenuation Amount Characteristics

First, verifications were carried out of the acoustic characteristics in a state where exhaust flow factors are eliminated. This was done by comparing actual measurements taken at normal temperature using an acoustic vibration tester and the results of calculations using FEM.

The general-purpose code SYSNOISE was used for the 3D acoustic flow analysis. Fig. 10 shows the analysis model and the corresponding FEM mesh model.

The following items were kept in mind when improving the accuracy of the attenuation analysis.

- (I) Wall surface boundary condition settings
- (II) Sound absorbing material characteristics
- (III) Small shapes (interference hole, gaps with mating parts)
- (IV) Release area boundary conditions
- (V) Dependency of mesh size

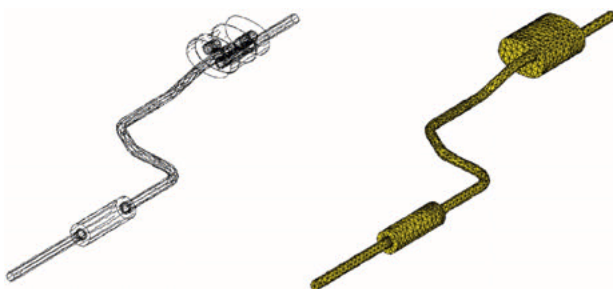


Fig. 10 Analysis model and mesh model

Fig. 11 gives an example of acoustic pressure mode analysis results. Equations (5) and (6) were used for attenuation amount calculations ⁽²⁾.

$$X(p) = H_B(p) \cdot H_F(p) \cdot F(p) \quad (5)$$

$$ATT = 20 \text{LOG} \frac{|F(p)|}{|X(p)|} \quad (6)$$

That is, the response $X(p)$ to acoustic vibration input $F(p)$ could be calculated from the exhaust pipe release end transmission function $H_B(p)$, and the transmission function $H_F(p)$ that is determined by multiple physical property values and structures. $H_B(p)$ was calculated using the boundary element method, and $H_F(p)$ using the finite element method.

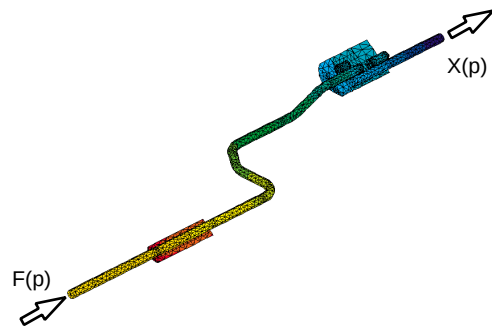


Fig. 11 Acoustic pressure mode (45Hz)

Fig. 12 shows the comparison between the attenuation amount measured using an acoustic vibration tester and the calculated results. The results are extremely accurate, within ± 2 dB, in the 0 to 500 Hz range.

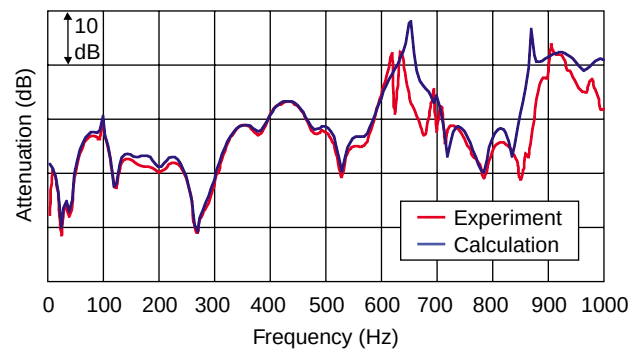
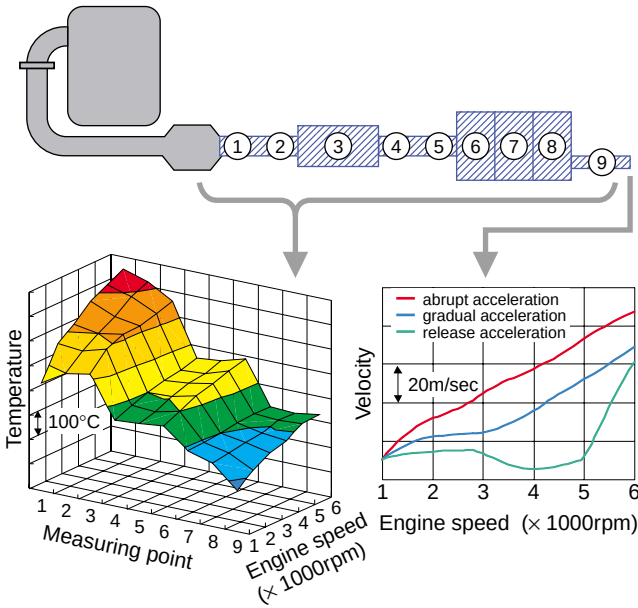


Fig. 12 Comparison of attenuation

5.2.2 Transmission Characteristics with Consideration of Heat and Exhaust Flow

When the engine is operating, the exhaust pipe is under high-pressure exhaust gas flow and high-temperature conditions of use, as shown in Fig. 13. Also, it is necessary to consider velocity and temperature parameters when analyzing pulsating noise.

In order to resolve the transient response problem concerning what kind of effect velocity and temperature have



(a) Temperature of muffler (b) Velocity of tail pipe

Fig. 13 Temperature and velocity measurements

on acoustic characteristics, the authors adopted the following equation of motion (7) ⁽³⁾.

$$[M]\{\ddot{x}(t)\} + [R]\{\dot{x}(t)\} + [K]\{x(t)\} = \{f(\theta^2)\} \quad (7)$$

[M]: rigidity matrix
 [R]: attenuation matrix
 [K]: mass matrix
 {x}: displacement vector
 {f}: vibration force vector

Here the right half of the formula is the velocity function, and the sound velocity and density corresponding to temperature changes are given as variables for each matrix. By introducing the solution of this equation of motion into the attenuation equation, it was possible to calculate the transmission function giving full consideration to velocity and temperature. Verifications were implemented, comparing the above results with measurement results from a vehicle chassis dynamo (Fig. 14).

The patterns used for the accuracy verification are recorded in Table 1.

In Pattern 1, the test vehicle's engine was turned off with the tires rotating in order to minimize temperature dependency. Fig. 15 shows a comparison of transmission function calculation results (2nd order) with changes only to the flow velocity parameter.

In Pattern 2, the test vehicle was in a working engine - throttle off mode in order to minimize the velocity dependency. Fig. 16 shows a comparison of transmission function calculation results (2nd order) with changes only to the temperature parameter.

Extremely high accuracy was obtained for both Patterns 1 and 2. Thus, it was proven that acoustic characteristics under changing temperature or velocity conditions can be determined using Equation (7).

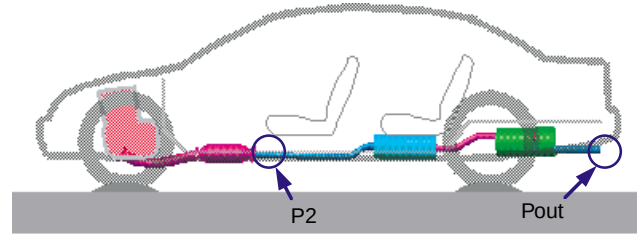


Fig. 14 Chassis dynamo test

Table 1 Comparison patterns

Pattern	Pattern 1	Pattern 2
Vehicle test mode	Motoring	Working engine & throttle off
Calculation condition	Change velocity parameter only	Change temperature parameter only

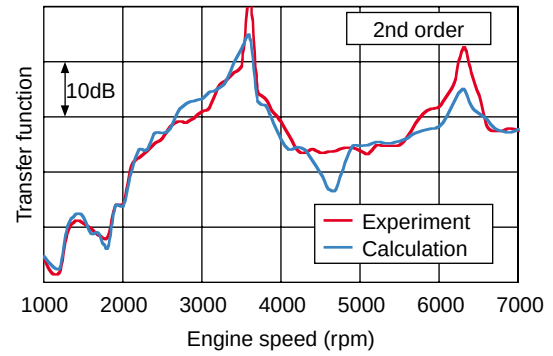


Fig. 15 Comparison of transfer functions

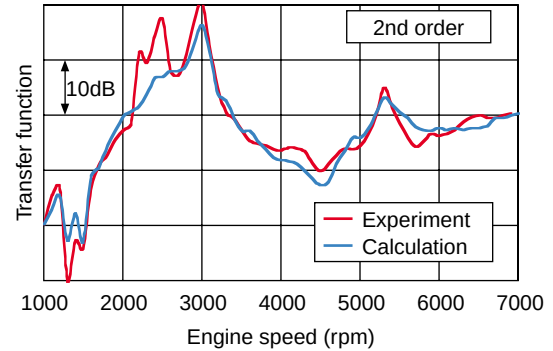


Fig. 16 Comparison of transfer functions

5.3 Pulsating Noise Analysis

In actual exhaust discharge noise analysis, it is necessary to consider both temperature and velocity parameters simultaneously.

A comparative verification of pulsating noise basic order with muffler a used in flow analysis was implemented in the abrupt acceleration mode.

Equation (7) was employed when calculating both temperature and velocity as transient response problems. The results of comparisons with the spectrum calculated from vehicle exhaust noise tests and with 2nd to 8th order pulsating noise are shown in Fig. 17 and Fig. 18, respectively.

The differences in both are greater with the higher frequencies. However, it is possible to obtain sufficiently accurate results when carrying out performance evaluation of parts without making prototype parts.

Using Equations (8) and (9), the exhaust pulsating noise overall value is calculated from two inlet boundary conditions: (1) pressure P_1 , determined in volumetric efficiency analysis; and, (2) pressure P_2 , measured in vehicle chassis testing.

$$X(p_n) = H_{v,T}(p_n) \cdot H_L(p) \cdot F(p_n) \quad (8)$$

$$SPL_n = 20 \log \frac{|X(p_n)|}{P_0} \quad (9)$$

$F(p)$: input sound pressure (exhaust source noise)

$X(p)$: response sound pressure (exhaust discharge noise)

$H_{v,T}(p)$: temperature, velocity transmission function

P_0 : minimum audible sound pressure

n : rpm

$H_L(p)$: distance attenuation

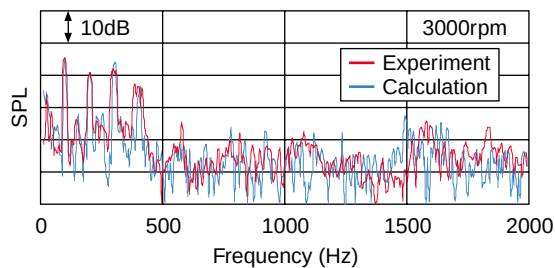


Fig. 17 Comparison of frequency spectrums

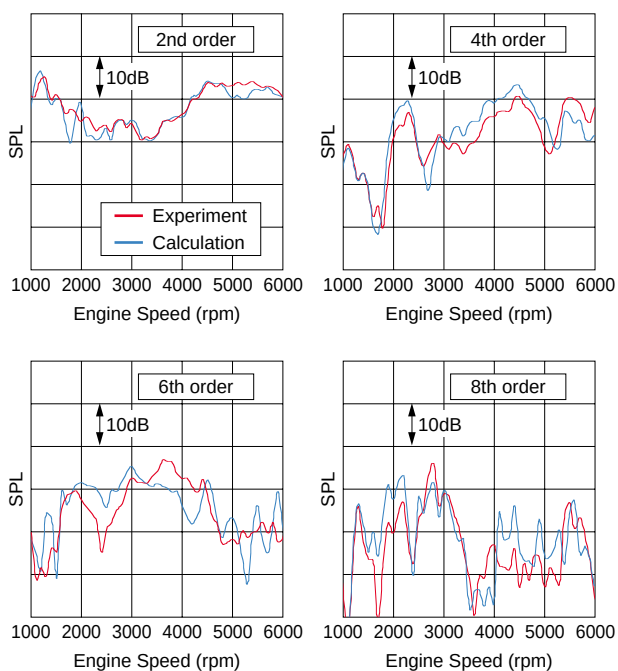


Fig. 18 Comparison of sound pressure levels

Highly accurate analysis results were obtained, with the accuracy of the results for (1) being ± 2 dB and (2) being ± 5 dB, as shown in Fig. 19.

The above are results with a muffler system that considers fully maintaining exhaust pipe diameter and generating a relatively small flow noise. It was thus judged possible to calculate exhaust discharge noise with this sort of exhaust system by adding a spectrum for each engine speed.

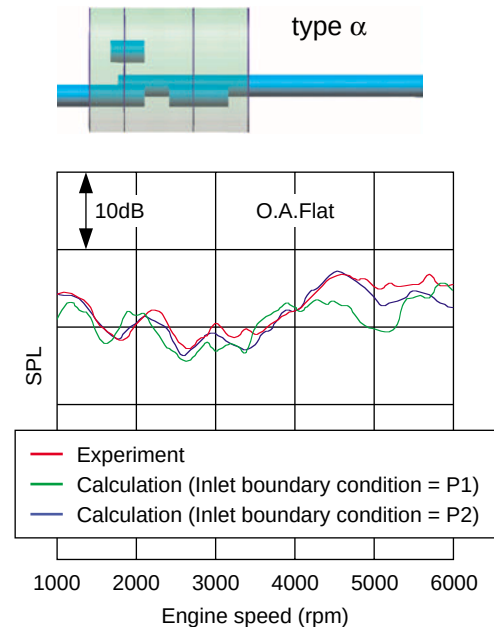


Fig. 19 Comparison of sound pressure levels

5.4 Flow Noise Analysis

With actual exhaust systems (Fig. 24 and 25 show types b and g), it is not uncommon for some portion of the exhaust pipe to be narrow in order to maintain attenuation mainly in the low frequency range. In these cases, it can be seen that the effect of the flow noise on the exhaust discharge noise grows larger as the pipe becomes narrower.

Generally speaking, flow noises are turbulent noises generated from eddies or impacts when high velocity exhaust gas flow passes through the muffler, and jet noises are generated when exhaust gasses are released from the end of the tailpipe. (See Fig. 20.) The frequency of the noise covers a wide range like white noise, but the flow velocity effect is mainly on the high frequency side.

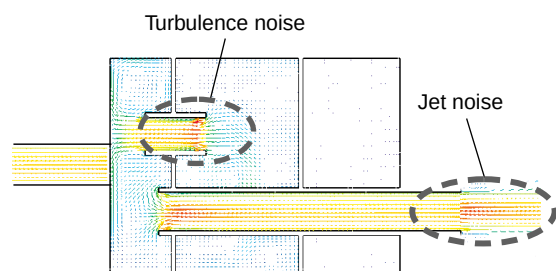


Fig. 20 Exhaust gas flow noise

The authors then made prototypes with narrower internal muffler parts and tailpipes. They then established relationships between engine speed and velocity (Fig. 21) as well as engine speed and exhaust noise (flow noise) without pulsating noise components (Fig. 22) with measurement data, and calculated flow analysis.

From these results it was determined that velocity and flow noise have almost linear characteristics with engine speed. Accordingly, by eliminating the engine speed component, it is possible to judge that velocity and flow noise have a proportional relationship, as expressed in Equation (10) (Fig. 23).

$$SPL = n10\log V + m \quad (10)$$

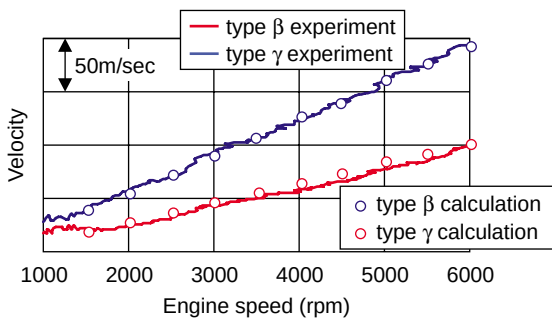


Fig. 21 Relationship between velocity and engine speed

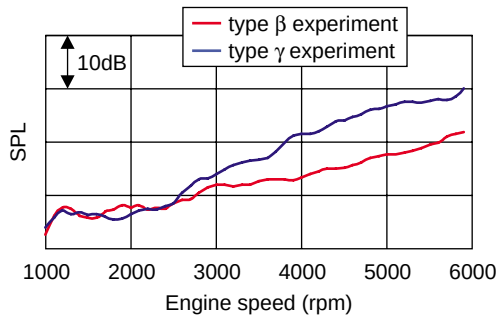


Fig. 22 Relationship between sound pressure level and engine speed

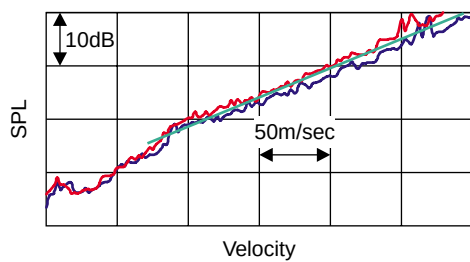


Fig. 23 Relationship between sound pressure level and engine speed

5.5 Exhaust Noise Analysis

Based on the above, exhaust noise can be calculated by adding pulsating noise and flow noise, as shown in Equation (11).

$$SPL = 20\log \frac{|X(p)|}{p_0} + \sum (n10\log V + m) \quad (11)$$

Next, comparing exhaust noise analysis results to exhaust noise measurements for type b and g mufflers as determined from Equation (11), as shown in Fig. 24 and Fig. 25, the authors found that the results were very uniform.

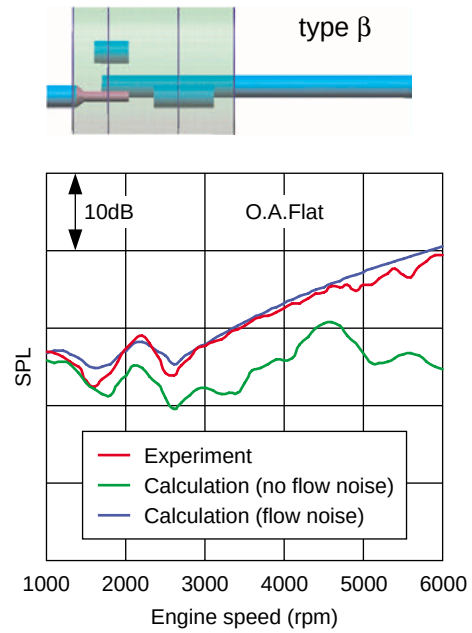


Fig. 24 Comparison of sound pressure levels (type β)

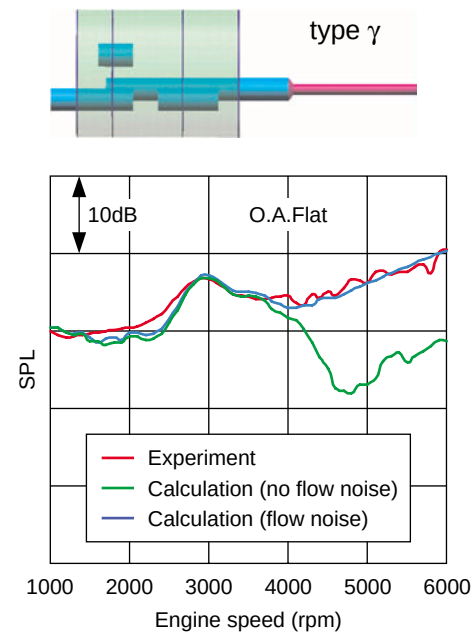


Fig. 25 Comparison of sound pressure levels (type g)

6. Examples of Applications to Muffler Development

As a result of the methods above, the effects that the cold end parts of the exhaust system have on engine output and exhaust discharge noise have become clear. Further, it is now possible to make optimization studies of muffler design based on analysis of this mechanism.

That is to say, in the prevention of flow noise and back pressure rise, etc. the basic task is to reduce velocity. However, in addition it is very important to prevent flow separation ⁽⁴⁾.

Conventionally, it was most effective to make separation visible and take countermeasures to prevent it. However, as shown in Fig. 26, the physical flow phenomena in CFD analysis results are very uniform. Thus, flow noise can be reduced and the flow rate improved by changing the shape of pressure loss areas without actually producing any parts.

Therefore by utilizing the analysis technology described thus far, the authors examined to what degree it would be possible to make the muffler smaller and lighter, and to reduce the exhaust discharge noise.

The assumption here was that the muffler layout within the exhaust system and the back pressure characteristics would remain unchanged.

The results are given in Table 2. To summarize, through the analysis of the mechanisms involved in the exhaust system acoustic characteristics and the fluid characteristics, it is possible to develop a muffler that is small, lightweight, and (as shown in Fig. 27) has a reduced exhaust discharge noise. This technology was applied to the 2001 Civic.

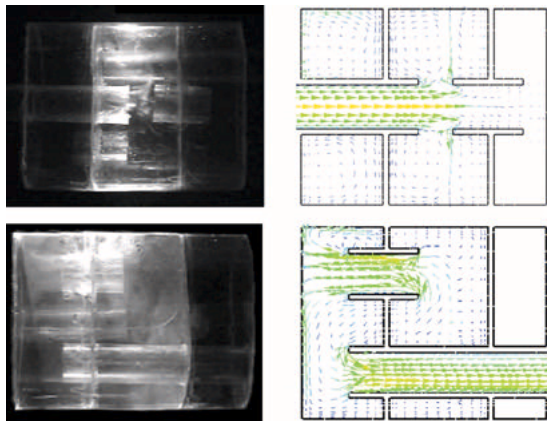


Fig. 26 Comparison of flow visualization and calculation

Table 2 Comparison of exhaust systems

	Base exhaust system	Modified exhaust system
Muffler volume rate	1	0.762
Muffler weight rate	1	0.742
Exhaust noise	Fig.27	

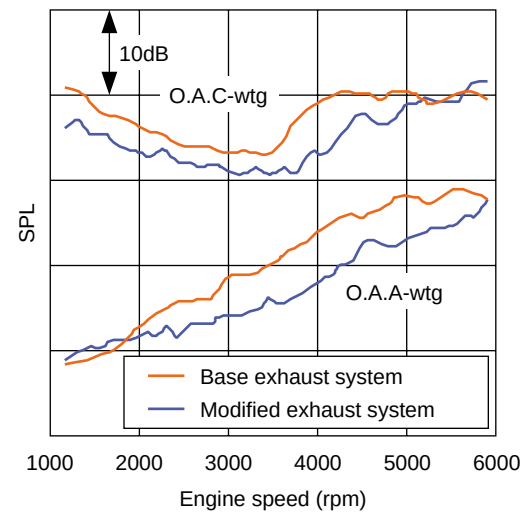


Fig. 27 Comparison of exhaust noise

7. Summary

Aiming for highly accurate predictive technology for exhaust system parts using CAE the following conclusions were reached.

- (1) It is possible to convert even mufflers with complex internal structures into straight pipes with a set diameter, by calculating the loss coefficient from the flow rate determined through 3D steady flow analysis.
- (2) Engine output can be calculated with very high accuracy ($\pm 2\%$) by looking at the muffler as a straight pipe as stated above, and then applying 1D volumetric efficiency analysis.
- (3) In order to improve the accuracy of pulsating noise analysis, it is very important to determine acoustic rigidity, attenuation, and mass matrixes while giving full consideration to temperature parameters, and to define acoustic vibration vectors which introduce velocity.
- (4) It was learned that flow generated noise is composed of turbulence noise generated internally in the muffler, etc., and jet noise generated at the tailpipe outlet. This sound pressure has a proportional relationship to velocity.
- (5) It is possible to calculate exhaust discharge noise with the extremely high accuracy of $\pm 2\text{dB}$, by adding together pulsating noise and flow noise.

Also, it has become possible to develop an exhaust system that has a high-level balance, by analyzing the characteristic factors that affect exhaust discharge noise and engine output.

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Air Quality and Emission Control Strategy Evaluation Software BRIDGE PROGRAM

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ABSTRACT

An original software tool, BRIDGE PROGRAM, has been developed to simplify the evaluation of emission control strategies for improving air quality. It is particularly effective for in-house evaluation of emission inventory reductions and air quality impacts as a result of introducing innovative, low emission technologies. The software uses a Relational Database Management System (RDBMS) to manage and modify the large emission data sets that characterize specific regions. Mobile, stationary, and biogenic emission rates can be scaled up or down, individually or by groups of source categories. The software automatically generates compatible files of gridded emissions for photochemical models used by regulatory agencies. This software has been used to modify emission inventories in regions of the United States and Japan. This paper describes the methodology and utility of the new software.

1. Introduction

There is a growing interest around the world for regions with poor air quality to search for pollution control strategies that will reduce emission inventories and achieve healthful air. These regions, typically urban or influenced by nearby urban emissions, have emission inventories that are dominated by mobile sources. As a result, strict regulations have been directed toward the drastic reduction of mobile emissions, particularly light-duty vehicles (LDV). Using the emission inventory of California's South Coast Air Basin (SCAB) as an example, reductions of roughly 80% total organics gases and 60% oxides of nitrogen are required to attain the National Ambient Air Quality Standard (NAAQS) for ozone. These are enormous reductions that will require implementing a host of clean technology solutions across all industries, not just from the mobile sector.

Honda has responded to the strict regulations by researching and developing many clean technologies, including clean gasoline powered vehicles, alternative fueled vehicles such as natural gas and pure electric vehicles, as well as fuel cell vehicles. Until now, Honda has lacked the internal capability to evaluate the air quality impacts of the most promising clean technologies introductions. It would be advantageous to have the in-house ability to evaluate the emission and air quality benefits of these vehicles before or during the early research phase. This capability would provide insight for determining the most effective direction

for development or future research. This type of early indication of emission benefit could also be valuable for scientific discussions with regulatory agencies or other interested groups.

Mathematical models are commonly used to describe the chemical and physical processes within the atmosphere during the assessment of emission reductions on regional air quality⁽¹⁾. These models require comprehensive meteorological and emissions data that are spatially and temporally allocated. To accurately represent an emission control strategy, baseline emission data should be modified at the disaggregate level.

An effective software system, BRIDGE PROGRAM, has been established to "bridge" the gaps between agency-supplied emissions data, emission control strategies, and advanced photochemical air quality models. The methodology is based on applying control factors to the baseline emission contributions of carbon monoxide (CO), oxides of nitrogen (NOx), sulfur oxides (SOx), total organic gases (TOG), and particulate matter (PM). The emission rates from individual or groups of emitting source classes can be scaled up or down to represent the target emission rates. The hourly gridded emission files required for photochemical models are automatically generated.

The system described in this paper has the structure to activate a sophisticated mechanism for designing emission

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control strategies for not only the transportation industry but for other industries as well. This software tool can assist in finding the technology combinations to clean our air.

2. Description of the Evaluation Model System

The system design for applying a relational database management system (RDBMS) to evaluate air quality emission control strategies is shown in Fig. 1. All of the input and output data flow is conducted through the RDBMS. Air quality data, chemical mechanisms, and baseline emissions of mobile, stationary, and biogenic sources are loaded into the database. Tables are created to store the data and links are established to form relationships. A control strategy Graphical User Interface (GUI) allows novice users to communicate with the database and easily modify any emission source type. The database can be interrogated directly with Structured Query Language (SQL) scripts to extract information and return custom reports for inventory analysis⁽²⁾. The hourly gridded emission files required by regulatory and research photochemical models, specifically the Urban Airshed Model⁽³⁾ (UAM CB-IV) and the California Institute of Technology Model^{(4),(5)} (CIT), are automatically generated.

It was important to select a commercially available RDBMS that could be easily adapted and widely used⁽⁶⁾. Oracle8 was selected although early development was concurrently performed in Microsoft SQL Server. The RDBMS has the capability to easily manipulate, display, and analyze large and diverse data sets.

3. Mechanism to Identify Source Types

Reflecting the introduction of a new technology or a proposed regulatory standard into an emission inventory requires the ability to identify the appropriate source types for modification. The Source Classification Code (SCC) was selected as the mechanism to identify source types. The U.S. Environmental Protection Agency, California Air Resources Board (CARB), and the South Coast Air Quality Management District (SCAQMD) support the SCC usage.

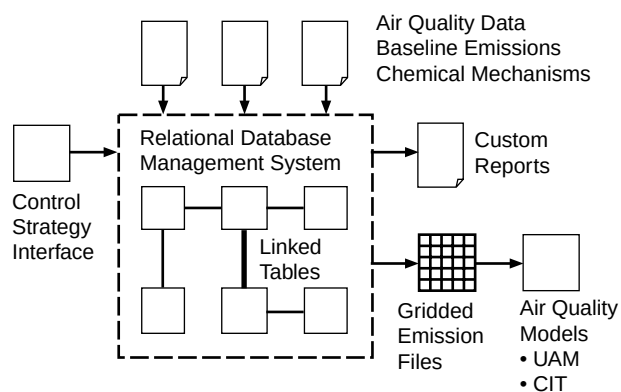


Fig. 1 The system design and data flow

Many countries interested in improving air quality look to these agencies for guidance. The SCC is an 8-digit code divided into four parts (X-XX-XXX-XX), which correspond to the four hierarchical levels of source description as illustrated in Fig. 2. Specifying a full 8-digit code uses all four levels and completely describes a particular source type. Not specifying the deeper levels will lead to increased groupings of industries and eventually will describe the most general categories.

All of the emission entries in the database are obtained from local and government agencies and contain key fields for values of SCC, CO, NO_x, SO_x, TOG, and PM⁽⁷⁾. The SCC provided a logical means to apply control factors across entire industries or specific processes, which realistically simulates the implementation of a new regulation or technology. Several of the tables created in the RDBMS include a common column, like the SCC column. These types of recurring columns weave relationships throughout the RDBMS⁽⁸⁾. Countries or regions that have not yet assembled or are beginning to assemble an emission inventory can adopt the SCC or design a suitable source identification scheme^{(9),(10)}. Consultants or "think tank" groups may be utilized as an alternative means of generating emission inventory estimates in countries where this type of data is not publicly disclosed.

4. Designing an Emission Control Strategy

An emission control strategy is a plan to reduce the emissions from targeted source types in order to attain a desired emission inventory level, typically described in units of tons/day. Usually the inventory goals are set to achieve air quality standards, like ozone attainment. Designing an effective control strategy typically requires an understanding of the relative emission contributions from the various active sources within a region. The powerful tools accompanying a relational database allow the user to query the enormous emission data files. Ad hoc queries can be rapidly scripted

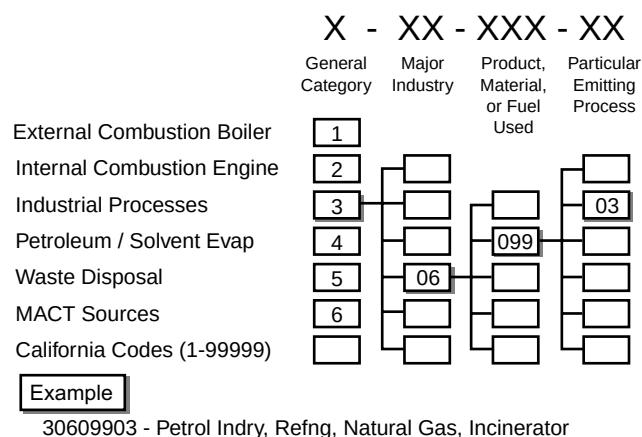


Fig. 2 The structure and tree diagram representation of the eight-digit, four-tier Source Classification Code

and refined for successive interrogation of the database. Information can be extracted to expose leading candidate sources for emission reduction. Knowledge of reactivity and ozone forming potential are insightful for choosing some sources over others but is not mandatory because this information is incorporated into the photochemical models. These mathematical models describe the chemical and physical processes within the atmosphere and predict the pollutant concentrations, which can be plotted and animated for easier comprehension.

A GUI was created to simplify the most common tasks during the process of modifying an emission inventory. GUIs are valuable tools to facilitate communication between the database and non-expert users. Experts can always communicate with the database directly by using SQL. Fig. 3 shows the general GUI for selecting an episode date, defining the spatial and temporal range, and choosing the appropriate format for the gridded emission files as required by air quality models.

The domain size can be customized to represent the region of interest as shown by the GUI in Fig. 4. Offsets are available to align the corner of the inventory data so it coincides with the domain representation of the ensuing photochemical air quality model. This is a particularly useful feature when studying new regions.

The SCC button in Fig. 3 will launch the SCC GUI shown in Fig. 5. The left-hand column of Fig. 5 contains the entire list of over 2000 unique SCCs. This tree diagram can be expanded or collapsed to expose any specific source type. An entire branch of sources within the tree diagram can be selected by marking the box at the most general level. Any source can be deselected by marking it again.

Once a source type has been selected, it is added to a summarized list on the right-hand side of Fig. 5. The sources in this list are available for emission modification. Control factors can be applied to CO, NO_x, SO_x, TOG, and PM emissions. A unity factor represents no change to the baseline emission rate. A factor less than unity represents an emission reduction while factors greater than unity represent emission increases.

The example in the summary list of Fig. 5 shows three different SCCs will be used to reduce emissions from ON-ROAD VEHICLES. Emission rates for CO, NO_x, and TOG will be significantly reduced to reflect a clean fleet technology. This is evident because the emission control factors, or coefficients, have been set to values much less than unity. The control factor values are determined by the relative contribution of the new emission rates compared to the existing baseline fleet emission rates that are stored in the database⁽¹¹⁾⁻⁽¹³⁾. The user must also consider the number of vehicles or market penetration when modifying emissions from ON-ROAD VEHICLES.

After applying control factors to the selected sources the user returns to the general GUI, Fig. 3, and executes the

data manipulation with the query button. The database applies the new emission rates to each spatial location where that specific source type is active within the region. Emission files are automatically gridded into cells as a function of time and space for use in regulatory or research air quality models. Fig. 6 shows emission inventories for Japan's Kanto region and the United States' South Coast Air Basin. Because most of the steps in the process of designing an emission control strategy have been automated, the influence of human error has been minimized. This process of creating control strategies would continue until the results of the emission reductions and air quality simulation successfully met a predetermined criteria.

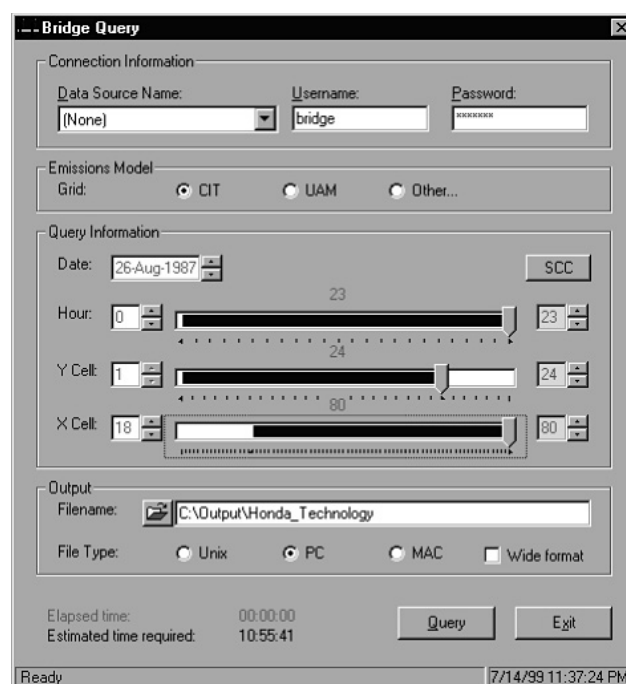


Fig. 3 The general GUI simplifies common tasks

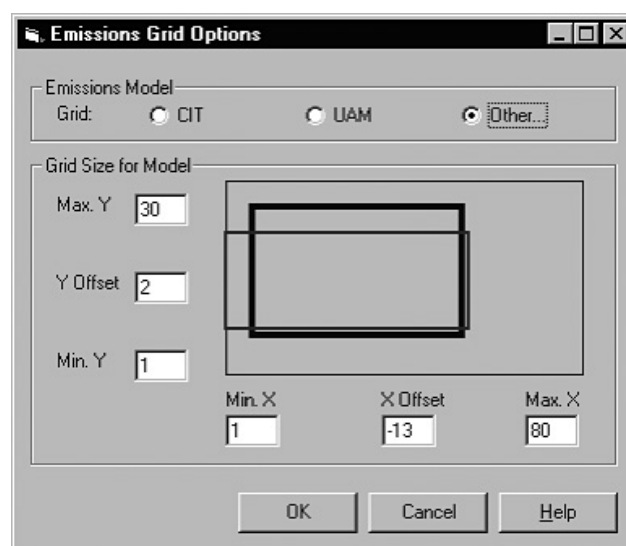


Fig. 4 Flexible domain sizing and inventory offsets

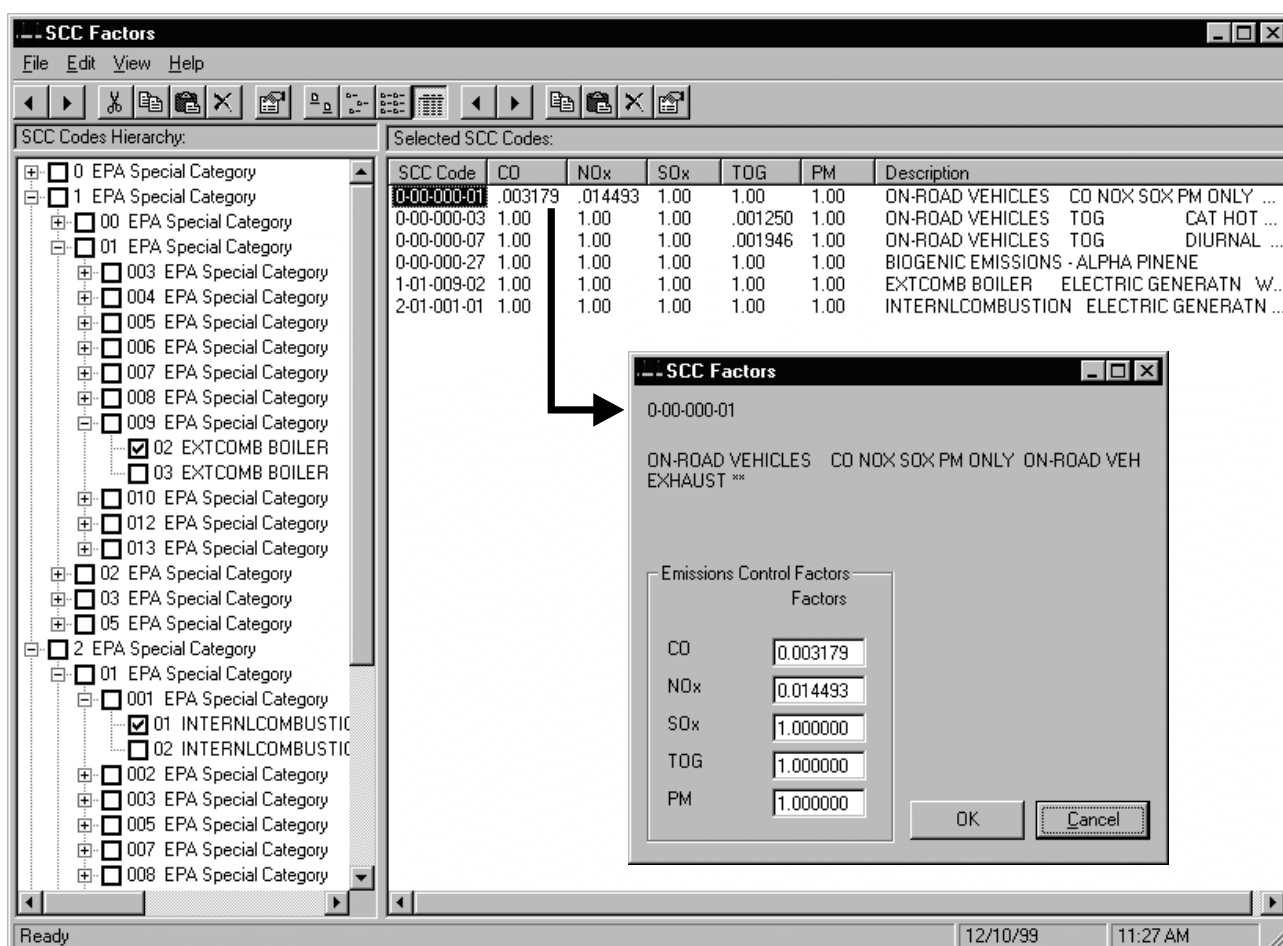


Fig. 5 The SCC GUI allows the user to select individual or groups of SCCs and apply control factors

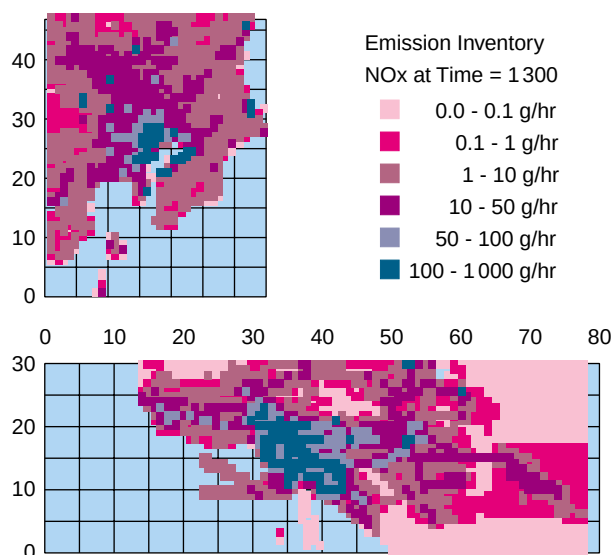


Fig. 6 An example of hourly emission inventories for Japan's Kanto region and U.S. SCAB

5. Conclusion

An original and effective software tool has been developed to simplify the evaluation of emission control strategies for improving air quality. Mobile, stationary, and

biogenic emission inventories are quickly modified to reflect the introduction of innovative, low emission technologies. Control factors are applied to an emitting source class to alter emission contributions of CO, NOx, SOx, TOG and PM. The hourly gridded emission files required for photochemical models are automatically generated. The tool gives Honda the in-house capability to evaluate the emission and air quality benefits of vehicle technologies before or during the early research phase, which can provide insight for determining the most effective direction for research and development.

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Yoshiaki MATSUZONO

Development of an Ultra Flat Torque Converter by Applying Three-dimensional Computational Fluid Dynamics

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ABSTRACT

A method has been created to optimize design parameters using three-dimensional CFD to analyze the internal flow condition of a flat torque converter, and one-dimensional angular momentum theory. This method has been used to design a torus and blade configuration which reduces flow loss, and to optimize the specifications of the stator, which has a great affect on torque capacity characteristics. This has allowed the development of the thinnest ultra flat mass production torque converter in the world without sacrificing torque converter performance.

1. Introduction

The trend toward increased steps in automatic transmissions (AT) and continuously variable transmissions (CVT) with the aim of improving comfort and fuel economy has accelerated in recent years. The development of an ultra flat torque converter that is thin and which delivers high performance is essential to meeting these demands.

Regarding analysis of the internal flow condition of torque converters, actual measurements of internal pressure and flow distributions around the stator⁽¹⁾⁽²⁾ and the results of characteristic predictions using three-dimensional computational fluid dynamics (CFD)⁽³⁾ have been reported thus far. However, in order to develop an ultra flat torque converter, it was necessary to analyze factors causing performance deterioration when the torus section is flattened and the stator is made thinner, and to find a way to shorten the development period for verifying the design parameters.

This study created a method for verifying design parameters that is based on analysis of the internal flow condition of a flat torque converter using three-dimensional viscous CFD and one-dimensional angular momentum theory. This method was then used to design a torus and blade configuration that reduces flow loss and to optimize the flat stator specifications. This made it possible to develop the world's thinnest ultra flat torque converter. A 15% reduction in the axial dimension and a 3% improvement in transfer efficiency compared to the conventional Honda 2-liter engine class torque converter was achieved without sacrificing torque converter performance.

2. Development Issues

When attempting to design a flat torque converter, the main design restrictions are on flattening the torus section and making a thinner stator, which are known to influence torque converter performance as follows.

- (1) Flattening the torus section causes the torque capacity characteristics and the transfer efficiency to drop.
- (2) Making the stator thinner creates torque capacity characteristics with peaks.

Of these, (1) influences the fuel economy, and (2) influences the drivability of a vehicle at launch by creating a discrepancy between the engine torque characteristics and the torque absorption characteristics of the torque converter.

3. Development Targets

The following development targets were set in consideration of the above technical issues and the need to shorten the overall AT axial dimension.

- To achieve torque capacity characteristics and a transfer efficiency equal or superior to the current model.
- To reduce the overall torque converter width by 15% compared to the current model.

To achieve these targets it was necessary to analyze factors causing and quantitatively evaluate changes in torque converter performance caused by flattening the torus section and using a thin stator. Therefore, the following torque converter tests and three-dimensional viscous CFD were performed. The CFD tool used was a general-purpose code,

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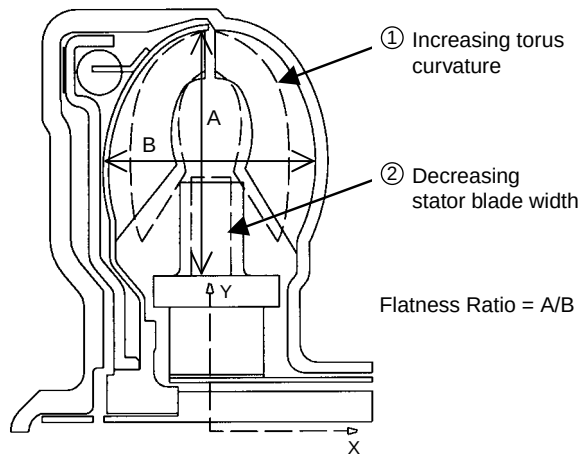


Fig. 1 Restrictions on the design process

which applied the finite volume method for digitizing, the CFD turbulence kinematic model was applied as the standard κ - ϵ model, and the SIMPLE method was used as the analytic algorithm.

4. Effects of Flattening the Torus Section

4.1 Confirmation of Torque Converter Performance

Torque converter performance was confirmed using torque converters having the same design parameters of flow channel area, blade angle, number of blades, bias angle, and blade inlet/outlet radius, with the only difference being the torus section flatness ratio. Table 1 shows the main torque converter specifications and test conditions. Fig. 2 shows the torque converter characteristic measurements. Note that Type A and Type B in Table 1 were designed based on conventional design methods⁽⁴⁾, and that the Type A torque converter is currently used in 2-liter engine class vehicles.

As shown in Fig. 2, the transmission capacity factor τ and the torque ratio K decrease for all speeds. The transfer efficiency was also confirmed to drop by approximately 7% as the torus section flatness ratio decreased from 84% to 75%.

4.2 Analysis of Flow Conditions inside the Torque Converter by Computational Fluid Dynamics (CFD)

It is known that the turbulence kinematic energy κ determined by solving the two-formula κ - ϵ model can be applied to predict separation flow and secondary flow, the factors which degrade flow performance, in other fluid devices. These predictions are accurate to a degree appropriate for practical application. Therefore, the turbulence kinematic energy κ at the speed ratio $e = 0.8$, which occurs frequently while driving, was calculated for the Type A and B torque converters shown in Table 1 in order to analyze factors causing and quantitatively evaluate changes in characteristics due to flattening. The calculation results shown in Fig. 3 confirmed local rises in the turbulence kinematic energy value when the flatness ratio was decreased from 84% to 75%.

Table 1 Torque converter specifications and test conditions

		Type A	Type B
Hydraulic diameter (mm)		232	
Number of blades	Impeller	27	
	Turbine	23	
	Stator	15	
Flattening rate (%)		84	75
Stator blade width (mm)		13.5	
Rotation speed of input shaft (rpm)		2500	
ATF temperature (°C)		85	

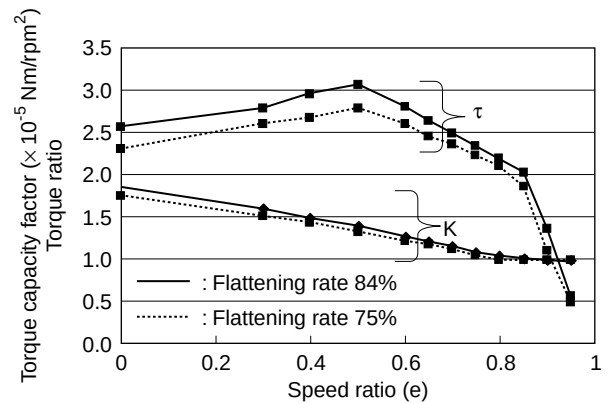


Fig. 2 Affect of flattening rate on torque converter performance

Next, the flow direction and velocity were analyzed for the AA' section of the Type B converter shown in Fig. 3 which exhibited a rise in the turbulence kinematic energy value. The flow visualization in Fig. 4 shows that a secondary flow occurred and the flow velocity dropped on the suction side of the turbine blades where the turbulence kinematic energy value increased.

In addition, Fig. 5 shows the relationship between the length of the fluid circulation passage and the turbulence kinematic energy in order to determine locations where separation and secondary flow increase in the torus section. The horizontal axis represents the ratio between the overall length of the fluid circulation passage and the distance from the pump inlet to some set location. This shows that the turbulence kinematic energy at the turbine inlet, turbine outlet, stator inlet and pump inlet were all higher for the Type B converter than for the Type A converter.

These torque converter tests and numerical calculations confirmed that the factors causing a drop in the torque capacity characteristics and transfer efficiency due to the flattening of the torus section are the secondary flow and separation occurring around each blade inlet on the flattened torus section. In addition, it was confirmed that the value of the turbulence kinematic energy κ could be used for quantitative evaluation of flow loss.

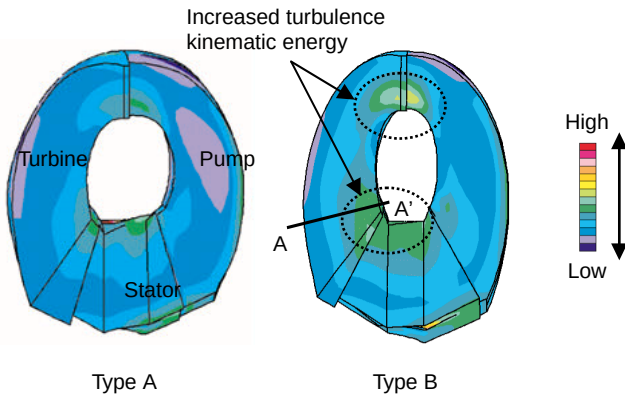


Fig. 3 Calculation results of turbulence kinematic energy

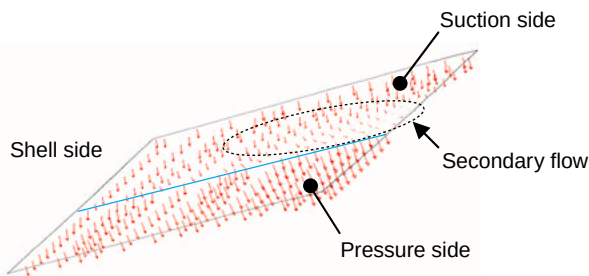


Fig. 4 Flow visualization at section AA'

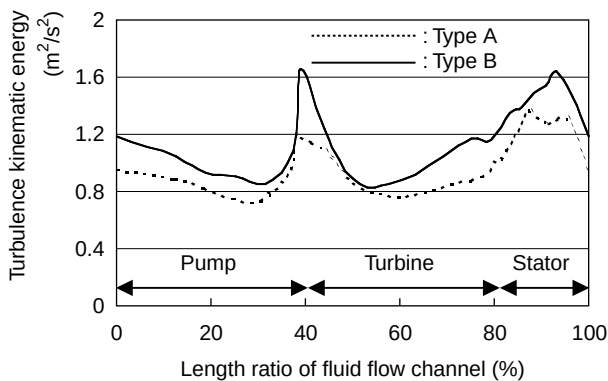


Fig. 5 Relationship between turbulence energy and length ratio of fluid flow channel

5. Effects of Using a Thin Stator

5.1 Confirmation of Torque Converter Performance

Torque converter tests were conducted by varying only the stator blade width based on the Type A converter shown in Table 1. The torque converter specifications and test conditions are shown in Table 2. The Type C stator has the same number of blades as the Type A stator, but the blade width is 2 mm shorter in the axial direction and the blade shape is reduced the same ratio as the blade width reduction ratio. Fig. 6 shows the torque converter characteristic measurements. These results show that reducing the stator blade width causes the torque capacity factor to increase centering around a speed ratio of 0.5 as well as causing the transfer efficiency to decrease.

Table 2 Torque converter specifications and test conditions

	Type A	Type C
Hydraulic diameter (mm)	232	
Flattening rate (%)	84	
Stator blade width (mm)	13.5	11.5
Rotation speed of input shaft (rpm)	2500	
ATF temperature (°C)	85	

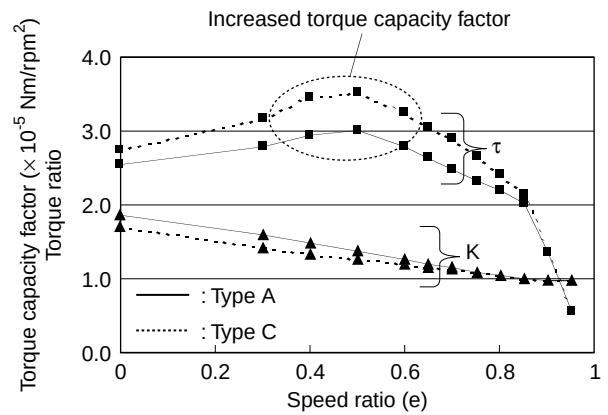


Fig. 6 Affect of decreasing the stator blade's width on torque converter performance

5.2 Analysis of Flow Conditions around the Stator by Computational Fluid Dynamics (CFD)

The flow conditions around the stator were analyzed at the speed ratio of 0.5, which showed the greatest change in torque capacity for the Type A and Type C stators in Table 2 using the three-dimensional CFD described in section 4.2 above. The numerical calculation method is the same as that described in section 4.2. Fig. 8 and Fig. 9 show the calculation results for the flow conditions around the stator. Fig. 8 shows the turbulence kinematic energy and Fig. 9 shows the velocity vectors for the plane corresponding to the center between the stator shell and core as shown in Fig. 7.

Fig. 8 and Fig. 9 show that compared to the Type A stator, the increase in the turbulence kinematic energy is larger at the Type C stator suction face, and separation also occurs at the Type C stator suction side. Furthermore, all flow from the outlet of the Type C stator was confirmed to be offset from the designed stator blade angle. The one-dimensional angular momentum theory⁽⁴⁾ also clearly showed

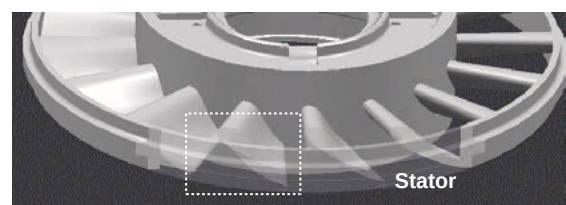


Fig. 7 Visualization area

that this change in stator outlet flow angle applied a rotational load to the pump, which resulted in a drop in transfer efficiency.

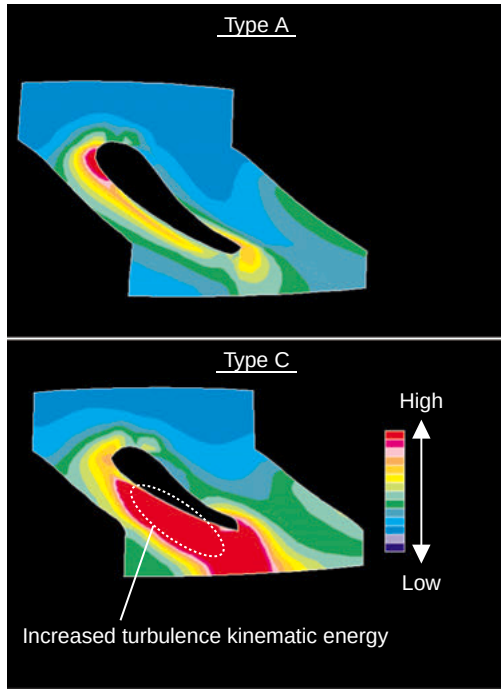


Fig. 8 Calculation results of turbulence kinematic energy

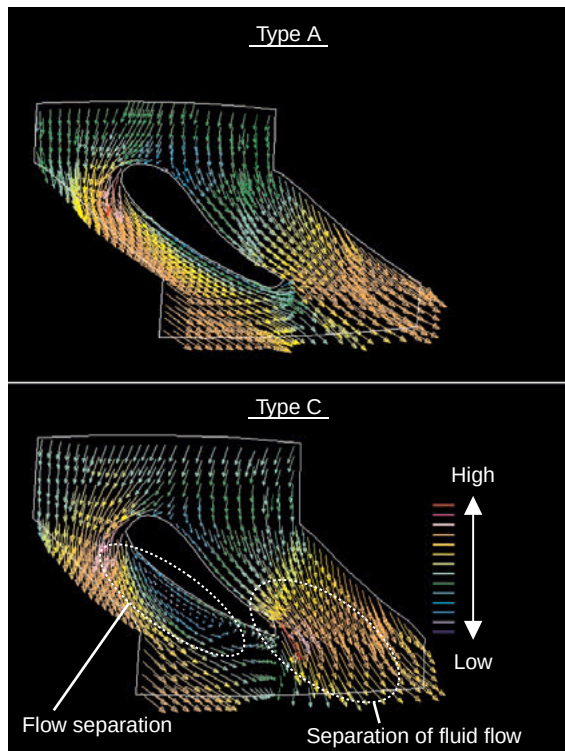


Fig. 9 Calculation results of fluid velocity

The above torque converter tests and numerical analysis confirmed that the change in outlet flow angle induced by separation occurring at the stator suction face was the main factor causing peaked torque capacity characteristics and lower transfer efficiency when a thin stator is applied.

Furthermore, the following method was incorporated to set the optimal torque converter parameters in a short period of time.

6. Method for Optimizing Design Parameters

When actually designing a torque converter, a large number of design parameters including the flow channel diameter, flow channel area, blade angle, bias angle and number of blades must be verified in combination. In consideration of matrix generation and calculation time, verifying all the design parameters using three-dimensional CFD would require enormous development time. A prediction method that reflects changes in the internal flow conditions due to the application of a flat converter is essential. Therefore, one-dimensional angular momentum theory was used as a basis to further improve the accuracy of the practical simulation method,⁽⁵⁾⁽⁶⁾ which is currently used in the design stage at Honda.

In this simple performance prediction method, the torque converter internal loss is divided into impact loss and friction loss. The secondary flow, separation and other loss in the flow channel due to the flattening of the torus section is defined as the friction loss L_f , and expressed by equation (1) below. Here, γ is the specific weight, Q is the flow rate, and h is the head loss.

$$L_f = \gamma \cdot Q \cdot h \quad (1)$$

h is further broken down as follows.

$$h = \sum_{i=1}^3 h_i \quad (2)$$

$$h_i = \zeta_i \cdot \left\{ \frac{1}{2g} \right\} \cdot \left\{ \frac{C}{\cos(\alpha_{i2})} \right\} \quad (3)$$

Here, ζ_i is the loss coefficient, g is the acceleration of gravity, C is the circumferential velocity, α is the blade angle, and the super- and subscripts 1, 2 and 3 indicate the pump, turbine and stator, respectively.

Focusing on the loss coefficient ζ_i , which was formerly defined based on past test data, three-dimensional CFD was performed for each designed torus section. ζ_i was then corrected based on the calculation results from the torque generated by the pump and turbine. Fig. 10 shows the design process incorporating this method.

7. Application Example

The knowledge obtained from this analysis and the ultra flat torque converter developed using the design process shown in Fig. 10 are described below. The flatness ratio of the newly developed ultra flat torque converter shown in Fig. 11 was set at 65% in consideration of torque converter performance and the target for reducing the overall torque converter width.

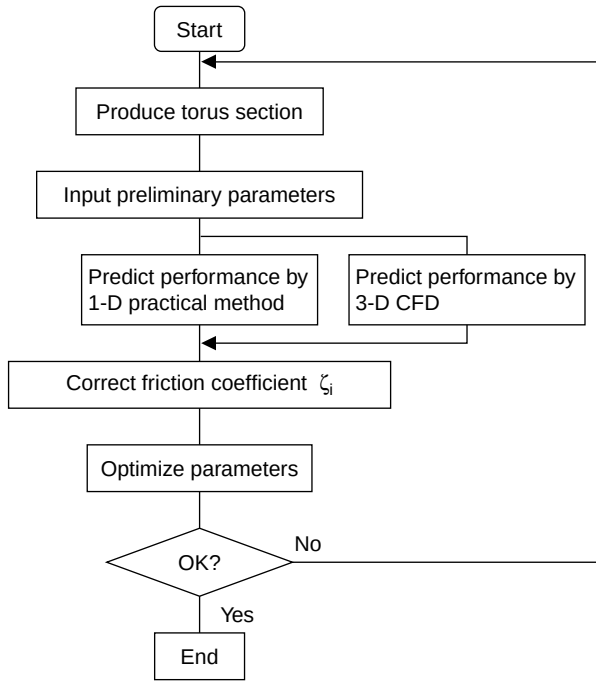


Fig. 10 Summary of design process



Fig. 11 Side view of developed torque converter

7.1 Torus Section and Blade Shape Design

The pump and turbine blade angle, bias angle, number of blades, and flow channel area were optimized with the aim of reducing flow loss. In order to illustrate the effects of this optimization, Fig. 12 shows a comparison of turbulence kinematic energy with the Type A converter currently used in the 2-liter engine class shown previously in Fig. 5. Note that the calculation conditions are the same as for Fig. 5. Fig. 12 clearly shows that the turbulence kinematic energy value and flow loss for the newly designed torus section decreased over the entire flow channel compared to the current model.

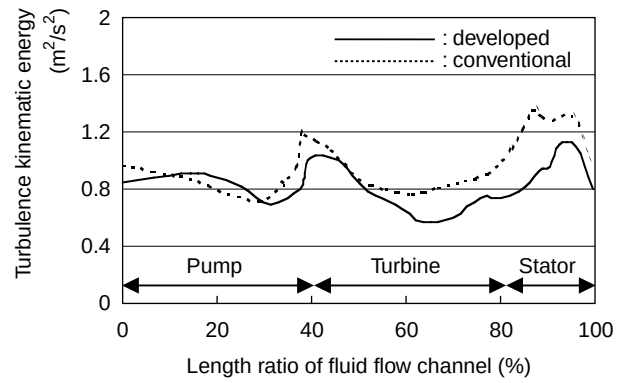


Fig. 12 Relationship between turbulence energy and length ratio of fluid flow channel

7.2 Optimization of the Stator

The number of stator blades, inlet/outlet angle and blade shape were newly designed as shown in Fig. 13 to suppress fluid slip caused by separation occurring at the thin stator suction face. Optimizing the stator suppressed fluid slip as shown in Fig. 14 and resulted in flat torque capacity characteristics.

In addition, torque converter characteristic measurements confirmed that the torque capacity characteristics for the developed torque converter are equal to those of the current model, and the transfer efficiency improved by 3%.

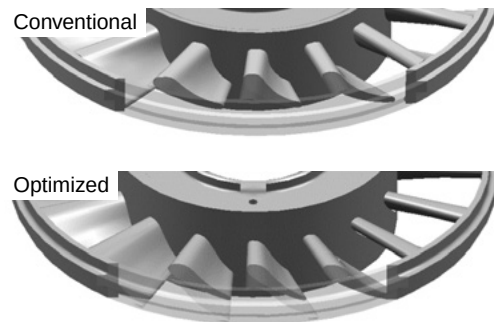


Fig. 13 Comparison between conventional and optimized stator blades

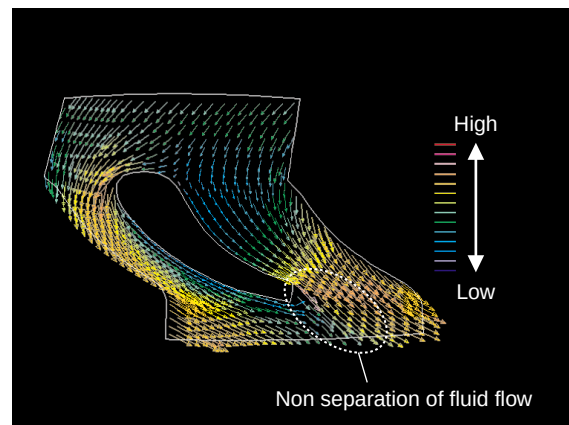


Fig. 14 Calculation results of flow velocity around optimized stator

8. Conclusion

The following items were clarified through torque converter tests and three-dimensional computational fluid dynamics. Furthermore, the axial dimension was shortened by 15% compared to the conventional Honda 2-liter engine class torque converter without sacrificing torque converter performance. This was achieved by creating a method for optimizing design parameters based on one-dimensional angular momentum theory.

- (1) The drop in the torque capacity characteristics and transfer efficiency due to flattening of the torus section is caused by secondary flow and separation occurring inside the flattened torus. Quantitative evaluation by the value of the turbulence kinematic energy κ is possible.
- (2) The peak torque capacity characteristics and the drop in transfer efficiency due to the application of a thin stator are mainly caused by changes in the outlet flow angle induced by separation occurring at the stator suction face.

Finally, the authors would like to express their deep thanks to Mr. Akira Kitajima, Mr. Kiyoshi Saitoh and other related parties at Yutaka Giken Inc., and to Mr. Akira Tsuboi, Mr. Satoshi Kawamura and other related parties at FCC Inc. for their participation in the joint development of this torque converter.

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Application of Helical Gear Tooth-root Bending Stress Simulation for Automotive Transmissions

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ABSTRACT

This technique was adapted from Professor Aizoh Kubo's program, which is capable of simulating tooth-root bending stress in helical gears, while taking into account the effects of misalignment and tooth surface configuration. A technique was created that is capable of more accurately simulating in a short time helical gear tooth-root bending stress in an actual automotive transmission during the first stages of design. The resulting tool was named GRANPLI. Misalignment in actual transmissions was measured using original gap sensors for a drive gear and driven gear behavior at low revolution speeds, and a misalignment map was created with respect to the loads. Particular attention was paid to the position of the primary reduction gear with respect to shaft orientation, the phase angle of the final reduction gear and the differential case, and so on. The appropriateness of calculation results obtained by GRANPLI was confirmed through comparison with data from measurements by strain gauges attached to a gear in an actual automotive transmission.

1. Introduction

There is an increasing demand for ever quieter and, at the same time, lighter and more compact automotive transmissions (referred to as transmissions hereafter) to improve fuel economy.

In addition, there is also call for the development of vehicles heavier than ordinary sedan passenger cars, exemplified by the sports utility vehicle (SUV), giving rise to higher loads on helical gears and other transmission components.

Inspections of the surfaces of helical gear teeth after testing the fatigue durability of a transmission under a constant load showed evidence in many cases of one-side contact in the meshed state. It is therefore necessary to have a gear strength prediction method for the final reduction gear, where the load is highest, that considers the effects of tooth surface configuration and misalignment.

Ever-shorter development periods also require a tool that can refine the design of helical gears from the first stages of the design process.

The authors carried out an analysis of the mechanism during helical gear meshing and proceeded with research on mesh fluctuation analysis of helical gears for automotive transmissions.

2. Conventional Helical Gear Strength Prediction

Conventional evaluation methods of tooth-root bending

stress in the helical gears of transmissions include the JSME, AGMA, BS, ISO, JGMA, DIN and other formulas. However, these formulas all share the basic concept for calculating bending strength⁽¹⁾.

When predicting the helical gear stress using these formulas, it is not possible to take misalignment or tooth surface configuration into account. Misalignment is the error in parallelism that occurs while the gear is in operation. It is a major factor contributing to bending stress in a transmission, because there are a wide variety of changes caused by a number of factors including the load transmission direction, the rigidity and backlash of the transmission case, bearings, gear shafts, and helical gears. Although it is therefore important to fully understand misalignment, it is very difficult to predict in the early stages of the design process when carrying out layout studies.

As a result, a lot of time is consumed in establishing and implementing many cases as criteria for judging the calculated results.

Although a method utilizing the finite element method (FEM) has recently been proposed to consider misalignment as an input condition, it requires a great deal of calculation time. Thus, it cannot be called the optimal method for use in the early stages of design.

Therefore, an alternative and more suitable method for predicting misalignment from the early stages of design was developed. This method applies a map for misalignment in regard to load under various conditions imposed by different layouts and component parts. This technique uses the program developed by Professor Aizoh Kubo of Kyoto

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University (referred to as the Kubo program hereafter). It allows misalignment to be considered when predicting the tooth-root bending stress. This new analytical tool is called "GRANPLI" (GeAR ANALyzing Program for Loading Influence).

3. Development of GRANPLI

3.1 Introduction of the Kubo Program

The Kubo program adopted in GRANPLI was modified in cooperation with Professor Kubo and his associates. This program can determine the various stresses generated in the gears according to the amount of rotational delay of drive gear to driven gear and allows for misalignment and tooth surface configuration. This allows the program to be applied to many different specifications for tooth height in both drive and driven helical gears and for meshing conditions that change in the tooth-width direction of helical gears used in automotive transmissions.

The Kubo program uses a function based on actual test results and the theories of Kubo, ⁽²⁾⁻⁽⁴⁾ Umezawa, ^{(2), (5)-(8)} and others for both tooth meshing and the tooth surface load distribution that accompanies it. The moment image method and the Aida and Terauchi formula ⁽⁹⁾ are also used in the bending stress calculations for the helical gear tooth root.

3.2 Program Modifications

The integral equation that expresses the load distribution on the single contact line of a helical gear and that forms the basis for this program is given in equation (1).

$$\int_0^{L(t)} K_b(x, \xi) p(\xi) d\xi + K_c(x) p(x) = \delta_0(x) = \{\Delta - e_{(x)}\} \cos \beta_g \quad (1)$$

- t : relative rotational angle on the gear
 $L(t)$: length of the contact line at location t
 x : coordinate of deflection measurement on the contact line
 ξ : coordinate of the point of application on the contact line
 $K_b(x, \xi)$: function of the effect of tooth shearing stress deflection for one pair of meshing teeth (referred to as the deflection effect function hereafter)
 $K_c(x)$: function of the effect of approach due to tooth contact for one pair of meshing teeth
 $p(\xi)$: distributed load
 $\delta_0(x)$: deflection of a pair of teeth at point x
 Δ : driven gear's delay angle in relation to the drive gear converted to a distance on the action line perpendicular to the shaft
 $e_{(x)}$: Compound error at point x of one pair of meshing teeth perpendicular to the shaft
 β_g : torsion angle on the base cylinder

The deflection effect function (K_b) is expressed by equation (2), ⁽⁶⁾ which is based on actual test results and a limit rack calculation analysis by the differential method according to Umezawa.

Deflection Effect Function:

$$K_b(x, y, \xi, \eta) = U \frac{v(r)}{\sqrt{F(x-\xi)} \sqrt{G(y-\eta)}} \sqrt{F(x)} \sqrt{F(\xi)} \sqrt{G(y)} \sqrt{G(\eta)} \quad (2)$$

$$\bar{r} = \sqrt{\bar{x}^2 + \bar{y}^2}$$

- x : coordinate of the deflection measurement in the tooth widthwise direction
 ξ : coordinate of concentrated load in the tooth widthwise direction
 y : coordinate of deflection measurement in the tooth height-wise direction
 η : coordinate of the point of load application in the tooth height-wise direction
 U : absolute value of deflection
 $v(r)$: characteristic function of deflection
 $F(x)$: deflection characteristics at the point of load application in the tooth width-wise direction
 $G(x)$: deflection characteristics at the point of load application in the tooth height-wise direction
 $\bar{x}$: Values with a bar across the top are standardized for tooth height ($\bar{x} = x/h$, where h is the total tooth height)

Corrections have been made to both the values and the formulas with the Kubo program. The corrections were made so that the formulas and values match the various gear damage examples for various machines. The authors have limited themselves to relatively narrow tooth-width gears for automotive applications and have further revised the coefficients according to FEM analysis and tests of the following items.

- (1) The compensation coefficient for a low pressure-angle gear of 20° or less used in transmissions and not verified by the measurements carried out by Professor Umezawa and associates⁽⁶⁾.
- (2) The coefficient expressing the rigidity of the tooth tips in narrow tooth-width helical gears.

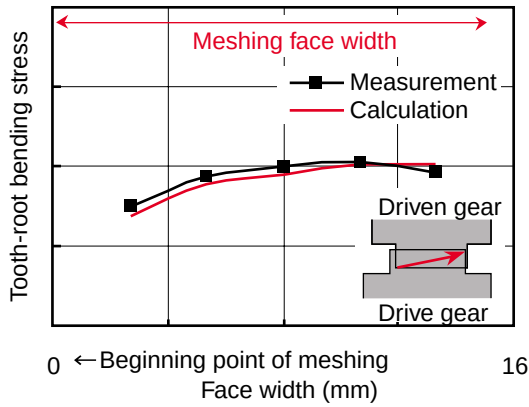
A verification of these coefficients was carried out by measuring the tooth-root bending stress while varying the gear meshing width. A high-rigidity gear unit test that makes it possible to ignore misalignment was used.

The specifications of the gears used in these tests are given in Table 1. Fig. 1 gives the measured and calculated results for the drive gear with a layout having a gear meshing face width of (a) 15mm and one of (b) 11mm.

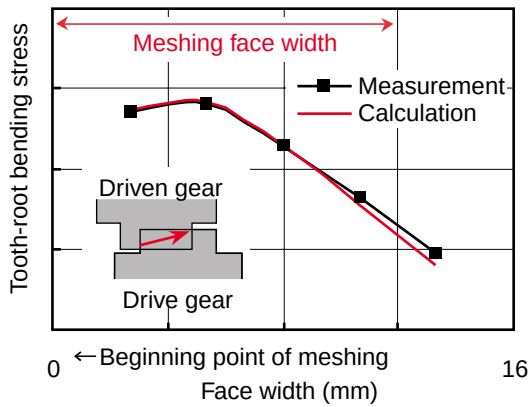
There is good agreement between the measured and calculated results, and it was thus confirmed that the set coefficient values were appropriate.

Table 1 Specification of gears used in experiments

	Drive	Driven
Normal module	2.00	
Standard pressure angle (°)	17.5	
Number of teeth	30	30
Face width (mm)	16.0	15.0
Helix angle (°)	36.0	



(a) Meshing face width: 15mm



(b) Meshing face width: 11mm

Fig.1 Calculation results of tooth-root bending stress distribution

3.3 Creation of the Misalignment Map

3.3.1 Measurement Conditions for the Misalignment in Transmissions

It is necessary to be able to predict misalignment in transmissions so that this program can be used effectively during transmission development.

This report presents a method for creating a misalignment map from measurements taken on a 5-speed manual Giacosa type transmission used in front-wheel-drive vehicles developed by Honda.

The reduction mechanism in this transmission comprises a helical gear and a conventional differential case as shown in Fig. 2. The structure of this differential case has open areas, so the rigidity is not constant throughout the case. This rigidity difference results in a variation in the misalignment of the final reduction gear according to its angular position.

In order to fully understand this misalignment, the behavior of the final reduction gear at a very low rotational speed was measured by using a test device that absorbs the load with a torque controller as shown in Fig. 3.

The specifications of the gears used for this test are given in Table 2.

3.3.2 Misalignment Measurements

Misalignment measurements were taken using a method that measures gear displacement with eddy-

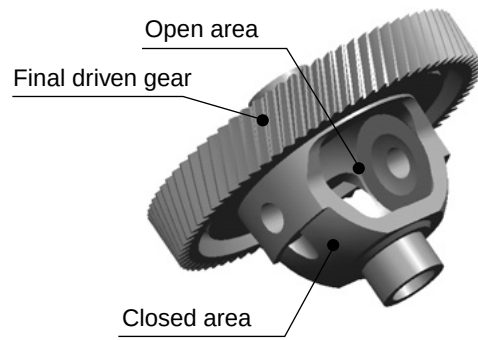


Fig. 2 Conventional differential case

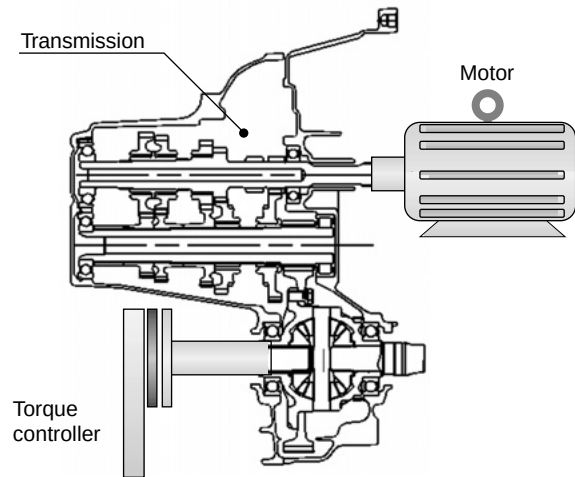


Fig. 3 Concept drawing of test equipment

Table 2 Specification of gears used in experiments

	Drive	Driven
Normal module	2.25	
Standard pressure angle (°)	17.5	
Number of teeth	18	79
Face width (mm)	37.5	33.0
Helix angle (°)	29.0	

current gap sensors.

Measurements were also taken so that the relative misalignment of the unit incorporating the differential case, bearings, shaft and gears could be evaluated.

The gap sensors and final reduction gear used in the gear test were positioned as shown in Fig. 4.

It was not possible with this arrangement to measure from the tip surface of the final reduction drive gear, so the behavior inside the shaft was measured.

The displacement was measured in two places – on the mesh lead side in the axial direction of the shaft, and on the mesh trail side. The relative cylindrical displacement is taken as the misalignment.

In order to understand the deflection components of the final reduction driven gear, measurements of the displacement on the gear tip surface were taken,

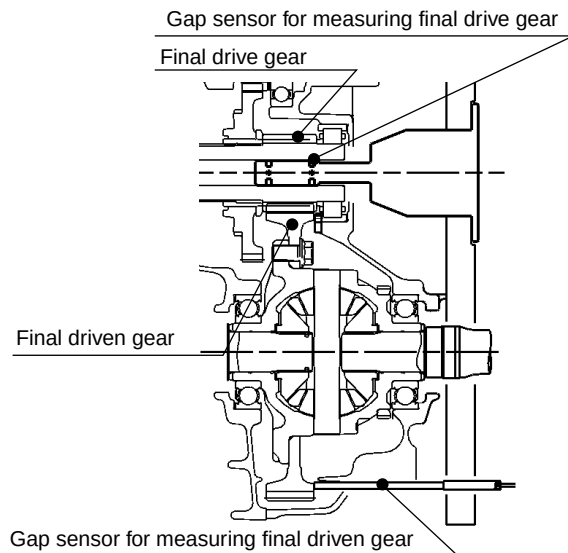


Fig. 4 Detail view of final gears

concentrating at a number of locations around the circumference and particularly near the mesh area.

These misalignment measurements enabled displacement data to be obtained for both the final reduction drive and driven gears, as shown in Fig. 5.

The spikes in the waveform for the final drive gear are marks that made it possible to identify the angular position of the differential case.

3.3.3 Misalignment Coordinate System

The misalignment coordinate system is based on Professor Kubo's definition shown in Fig. 6.

When the drive gear shaft moves off center with the driven gear shaft as the reference, the starting side shaft edge meshing surface directional component, when viewed from the meshing edge of the drive gear shaft, is AL. The component perpendicular to AL is AN. However, the direction away from the driven gear shaft is the positive direction.^{(10), (11)}

Let us now consider the effect that AL and AN have on meshing. When a point on a theoretical tooth surface reaches the plane of action due to the rotation of the gear, the distance from that point – according to the line of action – that the corresponding point on an actual tooth surface reaches on

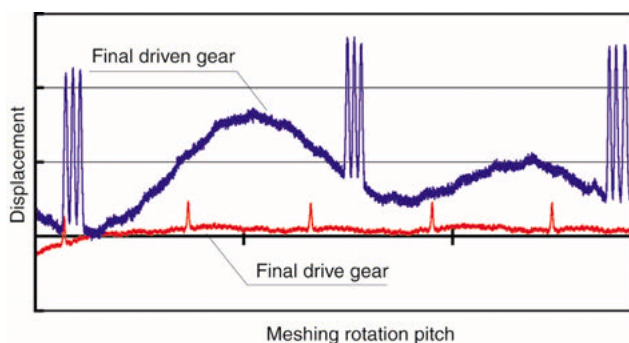


Fig. 5 Displacement by meshing rotation pitch

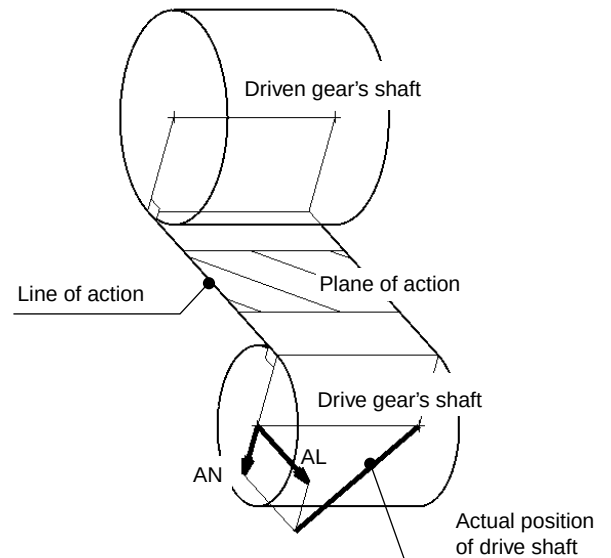


Fig. 6 Definition of misalignment

the plane of action determines the error value. If the vertical axis is set at this error value of said point of the plane of action, a single curved surface is created for the error of the entire surface. The curved surface determined from the plane of action is the error surface of the tooth shape. Calculation of the error surface according to the final reduction gear specifications given in Table 2 result in the error surfaces shown in Fig. 7. The effect of AL on the error surface was large, while the effect of AN on the error surface was small.⁽¹¹⁾ Fig. 7(a) is the curved error surface when AL is 50 μ m, and (b) is the error surface when AN is 50 μ m. However, these values were calculated for a tooth shape having no surface modifications.

Let us now consider the actual behavior of the meshing tooth surface. A change in the value of AL means that the meshing tooth surface is inclined in the tooth widthwise direction. This behavior promotes single-side contact. However, a change in the value of AN means that, although the meshing tooth surface is kept nearly parallel, this behavior reduces the meshing surface area. Considering this from the viewpoint of strength, it is considered that AN is a

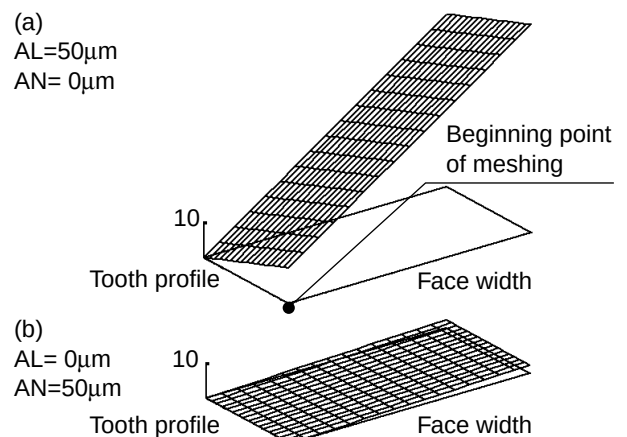


Fig. 7 Error surface

parameter that can be ignored to a reasonable degree in the early stages of design.

3.3.4 Misalignment of the Final Reduction Drive Gear

With the final reduction drive gear, the primary reduction gears and the final reduction gear are simultaneously in mesh. Consequently, differences in the load direction due to the proportion of each transmitted load, the support distance between the shaft bearings and each gear's location have different effects on gear displacement.

When measuring the final reduction drive gear displacement, tests were therefore focused on instances when the primary reduction gears were the 1st, 3rd and 5th gears. The specifications of the gears used in these tests are given in Table 3.

Fig. 8 shows the layout of the primary reduction gears and final reduction gear in the transmission.

Fig. 9 shows the measured and FEM-calculated results of behavior in the shaft when each primary reduction gear was engaged and transmitting power.

The results show the relative change in location of the mesh lead position when the mesh trail side is taken as the reference point of origin, in the tooth width. The measurements were taken with fixed conditions on the reduction drive gear for use in comparisons. Good agreement between the measured and FEM results confirmed the validity of the figures.

Finally, Fig. 10 shows the change in AL, which greatly affects the error surface, with respect to the torque when first gear was engaged and transmitting power.

Table 3 Specification of gears used in experiments

	1st gear	3rd gear	5th gear
Normal module	2.00	1.75	1.75
Standard pressure angle (°)	17.5	17.5	17.5
Gear ratio	3.266	1.212	0.738
Helix angle (°)	33.4	36.0	36.0

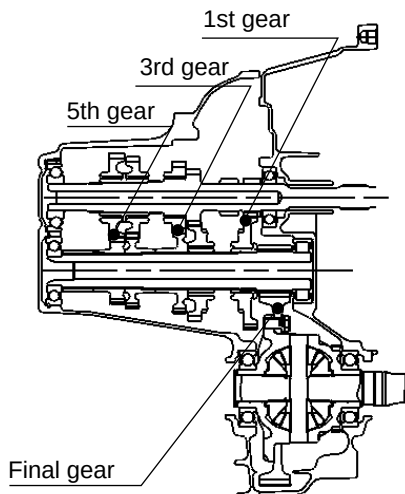


Fig. 8 Position of the primary reduction gears

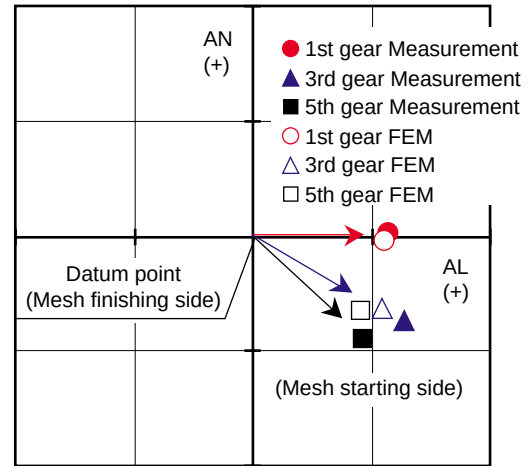


Fig. 9 Final drive gear position resulting in displacement

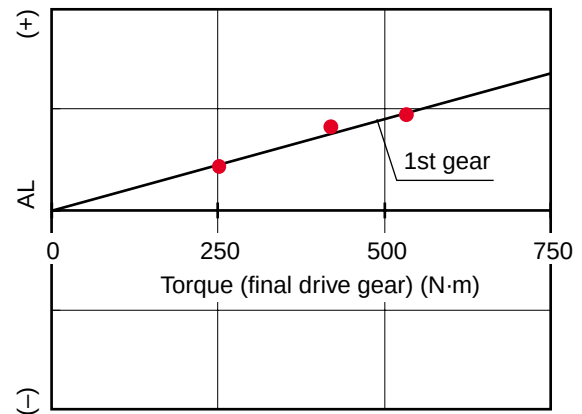


Fig. 10 Misalignment map (Final drive gear)

It can be seen from this graph that the misalignment changed in direct proportion to the torque over the range measured in these tests. This graph is applied as the misalignment map of the final reduction drive gear.

3.3.5 Final Reduction Driven Gear Misalignment Map

Fig. 11 shows the change in displacement of the final reduction driven gear tip surface with the changing structural rigidity of the differential case as the gear rotates. The origin represents the meshing location.

It can be seen from these plots that the steepest gradient was immediately before and after meshing. It was also confirmed that the gear tip surface displacement differed according to whether the open or closed portion of the differential case was meshing.

Fig. 12 shows the gear tip displacement as AL with respect to the rotational position of the differential case. The 0° angle occurs when the center of the open area in the differential case is adjacent to the point of meshing, with the positive angle in the direction of case rotation. As the case has a symmetrical structure, the range for the +/- 90° is shown.

It can be seen from Fig. 12 that the misalignment is so great that there is a change from the positive to the negative according to the rotational phase. Further, the maximum

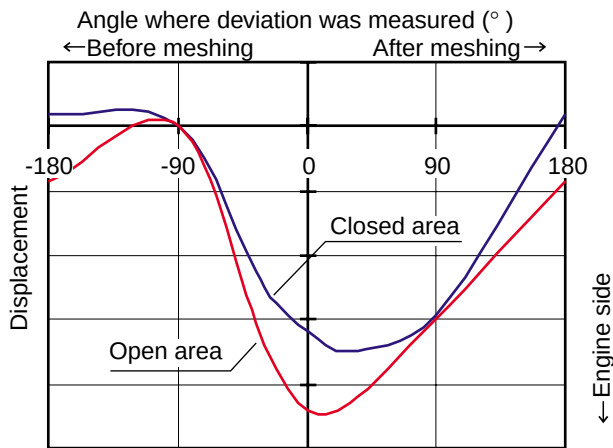


Fig. 11 Final driven gear displacement

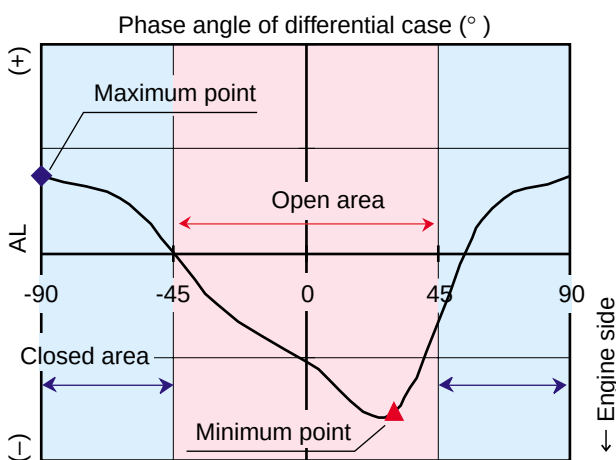


Fig. 12 Misalignment of the final driven gear according to the phase angle of the differential case

misalignment occurs when the closed area is adjacent to the point of meshing, and the minimum misalignment occurs when the open area is adjacent to the point of meshing.

Finally, Fig. 13 shows the change in AL with the torque, creating a misalignment map for final reduction driven gear.

It can be seen from Fig. 13 that even the misalignment in the final reduction driven gear changed in direct proportion to the transmitted torque in the range tested. The misalignment likewise varied according to the rotational angle of the differential case.

3.3.6 Relative Misalignment of the Final Reduction Gear

Combining the two misalignment maps given in Figs. 10 and 13 makes it possible to create a relative misalignment map for the final reduction gear as shown in Fig. 14.

4. Verification of the Validity of GRANPLI

4.1 Outline of the GRANPLI Verification Method

The validity of applying GRANPLI was verified by comparing calculated and measured values for the tooth-root bending stress. The stress measurements were taken at the same time as the misalignment measurements. The

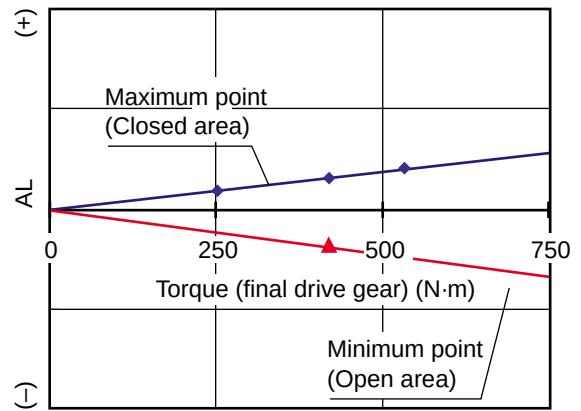


Fig. 13 Misalignment map (Final driven gear)

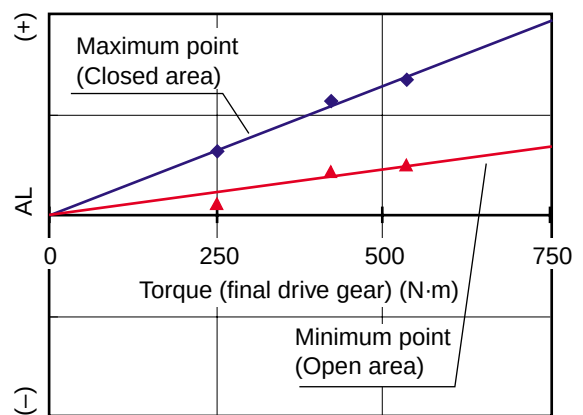


Fig. 14 Misalignment map (Final gear)

calculations by GRANPLI used the values for AL that were obtained from the misalignment map and the actual measured shape of the tooth surface.

4.2 Tooth-root Bending Stress Measurements

The tooth-root bending stress was measured on the pulled side of the tooth by strain gauges attached to the tooth root of the meshing tooth surface. Three strain gauges were attached at equal intervals as shown in Fig. 15 along the flank of a tooth at the location of the critical section in the meshing tooth surface tooth height direction as determined

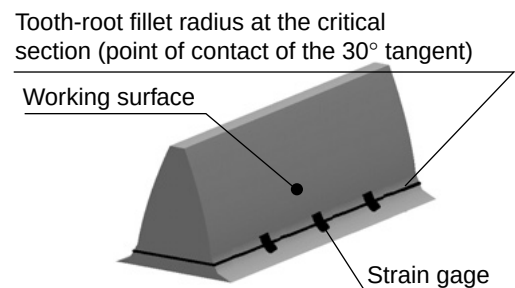


Fig. 15 Strain gage attachment points on final drive gear

by H. Hofer's 30° tangent method. The 0.2mm long gauges were set perpendicular to the tooth.

The measured data for the tooth-root bending stress is given in Fig. 16 along with the corresponding misalignment measurements recorded simultaneously.

It is possible to identify the phase angle of the differential case during the tooth-root bending stress measurements due to the relationship of the position of the wave forms along the rotational path.

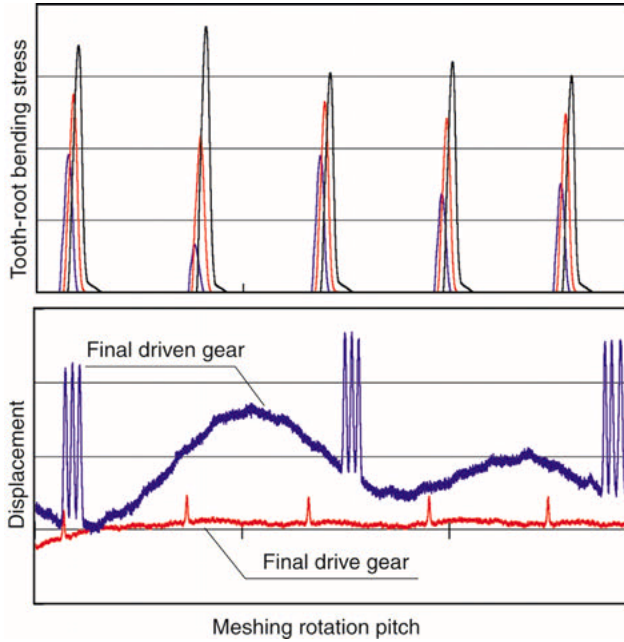


Fig. 16 Displacement and bending stress at tooth-root according to meshing rotation pitch

4.3 Calculation Conditions

The calculations for verification were carried out under the following conditions.

The calculations for misalignment applied the value at the torque obtained when measuring the tooth-root bending stress that was deduced from the misalignment map. The calculations were performed even when there was no misalignment for comparative purposes

As it is necessary to perform calculations for surface configurations under the same conditions as used for the measurements, the measured root surface configuration data shown in Fig. 17 was used: (a) final reduction drive gear data and (b) final reduction driven gear data.

4.4 Calculation Results

Fig. 18 shows the bending stress distribution on the drive side of the tooth root plotted against the tooth face width. Fig. 18(a) shows the greatest misalignment, while Fig. 18(b) shows the least misalignment.

Although calculations for the tooth-root bending stress show an equal distribution of the meshing range across the tooth face width when misalignment is not taken into consideration, the measured stress distribution is such that

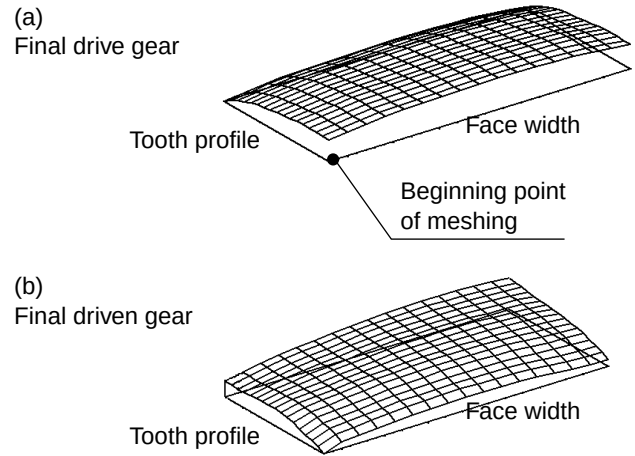
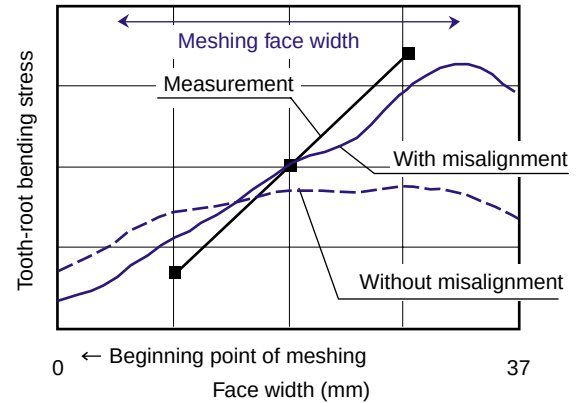
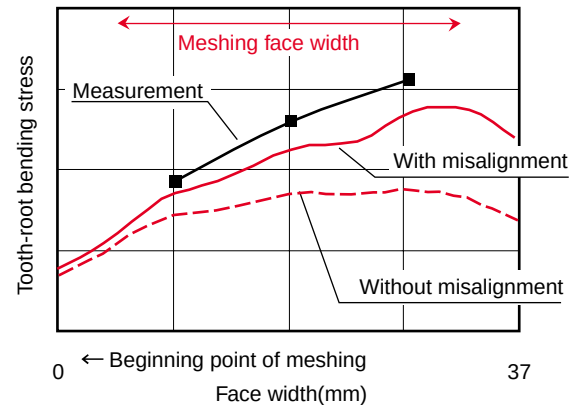


Fig. 17 Modified tooth flank of test gear



(a) Maximum point (Closed area)



(b) Minimum point (Open area)

Fig. 18 Calculation results of tooth-root bending stress distribution

the greatest tooth-root bending stress is generated on the mesh finish side. However when full consideration is given to the amount of misalignment obtained from the misalignment map, the calculated stress distribution also showed the greatest tooth-root bending stress is generated on the mesh finish side. Thus, results confirmed that accurate repeatability was possible.

The slope on both of the relevant misalignment maps is

reflected in the tooth-root bending stress distribution in respect to the open and closed areas of the differential case.

It was accordingly confirmed that the misalignment maps are appropriate for use.

4.5 Applications for Misalignment Maps

The applicability of the misalignment maps was next evaluated. The slope of the plot on the misalignment maps changed according to such conditions as the actual rigidity and backlash in the transmission. Creating a misalignment map makes it possible to apply these data to transmissions with different layouts.

The results were next verified by carrying out tests on the final reduction gear of a different transmission, the Giacosa type 4-speed automatic transmission. The final reduction driven gear of this Giacosa system is the same as that used in the preliminary study and has the same rigidity. Thus, a similar misalignment map could be used (Fig. 13). However, the final reduction drive gear is different in terms of the shaft layout and location of each primary reduction gear, so the misalignment map was modified to that shown in Fig. 19, with the relative misalignment map for the final reduction gear as shown in Fig. 20.

The GRANPLI calculation results from this misalignment map were compared with the tooth-root bending stress measurement results for the final reduction

driven gear. Fig. 21(a) shows the results when the misalignment was the greatest, and Fig. 21(b) shows them when it was the least.

It was confirmed that the calculated results for stress distribution by using the misalignment map could also be applied to transmissions with different layouts, since it was possible to accurately simulate the measured data.

5. Summary

The Kubo program, which allows for misalignment and tooth surface configuration in gear tooth-root bending stress prediction, was modified in conjunction with Prof. Kubo and associates so that the program could be applied in the early stages of automotive gear design. Misalignment has a large effect on the tooth-root bending stress. Thus, a map of misalignment in relation to the applied load was created by measuring gear behavior. This misalignment map makes it possible to more accurately predict gear tooth-root bending stresses in a short time in the early stages of design. This tool was named "GRANPLI."

The validity of the data calculated with GRANPLI was confirmed by comparing with measured tooth-root bending stress data.

The slope of the line on the misalignment map changes under a variety of conditions, but by using the most

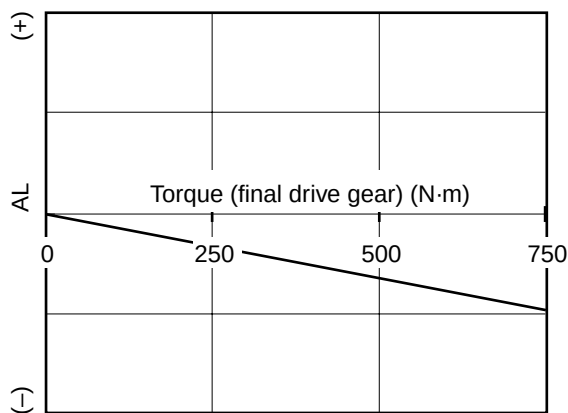


Fig. 19 Misalignment map (Final drive gear)

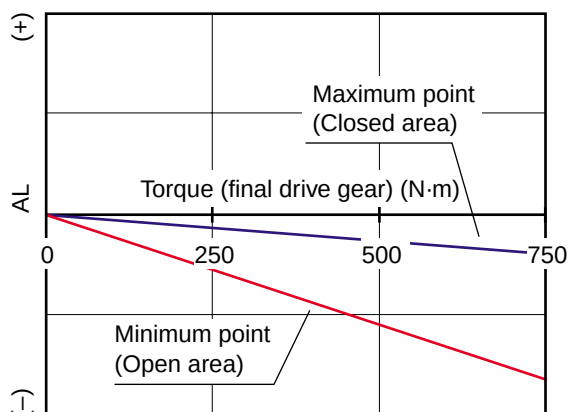
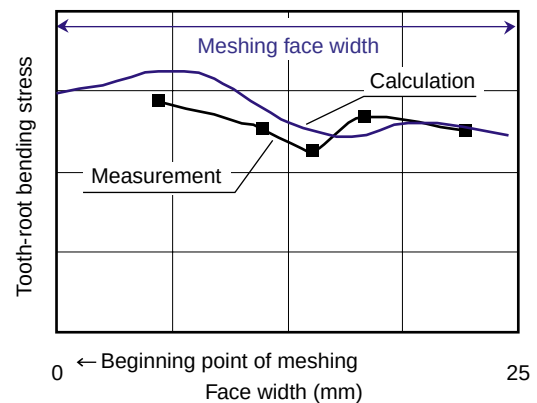
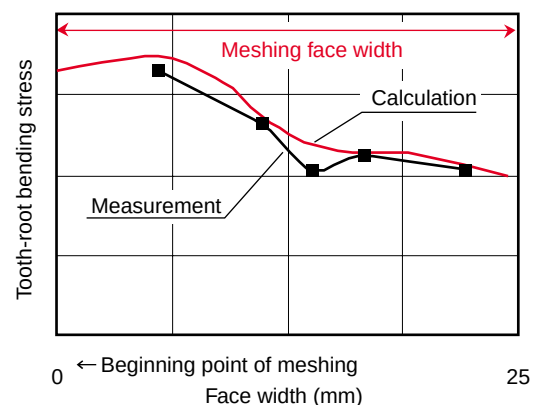


Fig. 20 Misalignment map (Final gear)



(a) Maximum point (Closed area)



(b) Minimum point (Open area)

Fig. 21 Calculation results of tooth-root bending stress distribution

appropriate misalignment map to suit the situation, this method can be applied to transmissions having different layouts.

This new method will be applied to create gear specifications in the early stages of the transmission design and to define early refinements to tooth surface configuration.

6. Acknowledgements

The authors thank Professor Aizoh Kubo of the Department of Precision Engineering at Kyoto University, and all of those in his research group, for their valuable assistance throughout this study, particularly with the modification of the program.

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Enhancements in Vehicle Stability and Steerability with Slip Control

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ABSTRACT

To enhance both vehicle stability and steerability, vehicle dynamic parameters such as vehicle side slip angle, road friction coefficient, and tire side forces were precisely estimated in real time; and, utilizing four wheel slip control, a logic for optimizing vehicle dynamics was formulated. Tests on ice- and snow-covered roads demonstrated the effectiveness of the control, and confirmed the validity of the logic.

1. Introduction

Honda's Vehicle Stability Assist (VSA), which was introduced to the market in 1998 ⁽¹⁾, was developed based on front wheel brake control with the purpose of enhancing stability in FF (front-engine-front drive) vehicles. Since then the system has been further refined. However, in order to further enhance steerability and to apply VSA to vehicles other than FF vehicles, technology needs to be developed that can control slip at all 4 wheels. The goal of this research project was to enhance the performance of the VSA. Therefore, we looked at the issues involved and how to achieve this goal, and later we verified the developments made with actual vehicles. The vehicle dynamic controls applied are slip control, utilizing 4-wheel hydraulic brake actuators, and engine torque control. The test vehicle used was a FR (front engine – rear drive) vehicle, and the test course was a snow and ice covered track, where the control is particularly effective.

2. Research Goals

VSA is based on front wheel slip control, and is a stability control system that applies the appropriate braking to the outside front wheel when the vehicle's yaw rate exceeds the nominal yaw rate to restrain excessive moment. In this research project, the team worked to take VSA a step further, to achieve enhanced handling particularly during avoidance maneuvers, as shown in Fig. 1, by securing not only stability but also securing tire cornering force on various road surfaces to the greatest degree possible.

That is, the basic goal of this research is to achieve both of the following:

- (1) Enhanced stability (rear tire cornering force)
- (2) Enhanced steerability (steering effectiveness)

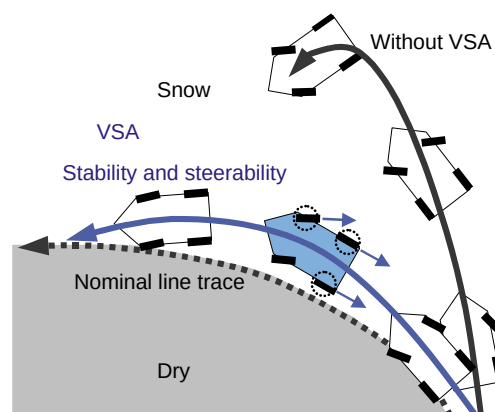


Fig. 1 Image of improved VSA

3. Research Issues

When a vehicle is cornering, the tire side forces on all 4 wheels determine the lateral acceleration and yaw moment. These side forces increase proportionally to tire side slip angle, but at a certain angle they become saturated. This grip limit differs according to road surface, as shown in Fig. 2. On icy or snowy roads the limit is reached between 1° and 5°, after which the vehicle becomes unstable. The process from cornering force critical conditions to instability, and the relationship between vehicle operation force and tire side slip angle are shown in Fig. 3. As you will come to see, in order to accurately understand vehicle dynamics, it is necessary to obtain the tire force from the graph of lateral force and side slip angle characteristics using the tire side slip angle that is converted from the estimated vehicle side slip angle. Also, it is necessary to estimate road surface friction coefficients in order to respond to various road

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surfaces. Furthermore, over-steer (OS) and under-steer (US) are judged from the estimated 4-wheel side forces, the vehicle's yaw moment and side slip angle are corrected with the optimal 4-wheel slip control, so that both stability and steerability can be achieved. From the above, the research issues can be summarized as the following three items.

- (1) Develop vehicle side slip angle and tire side slip angle estimation logic (β estimation logic)
- (2) Develop logic for estimating the friction coefficient for the friction between the tire and the road surface (μ estimation logic)
- (3) Develop optimal 4-wheel slip control logic (λ control logic)

Hereafter, the vehicle side slip angle is referred to as β , the friction coefficient between the tires and the road surface as μ , and tire slip rate as λ .

4. Basic Estimation Logic for Tire Slip Angle α and Vehicle Side Slip Angle β

Target estimation accuracy for tire side slip angle α is

$\pm 0.5^\circ$, making it possible to distinguish the 1° to 2° where side forces are saturated on ice. The basic technology's system configuration is shown in Fig. 4. The sensors include those for steering angle, longitudinal acceleration, lateral acceleration, yaw rate, and wheel velocity. Fig. 5 shows the basic logic configuration. The total logic is comprised of the following four types of logic – α and β estimation logic, μ estimation logic, tire characteristics data map logic, and β control logic (4-wheel λ control). The method for estimating vehicle side slip angle β is a logic that can estimate forces on each tire by combining the conventional two tire vehicle model (above spring) and non-linear tire model (below spring). Furthermore, in order to avoid integration noise, the total of the estimated side forces for all 4 wheels is fed back, rather than the lateral acceleration sensor values, and those estimated side forces are read from the tire data map. Tire data is expressed as three characteristics,

- (1) Side slip angle / side force
- (2) Slip rate / side force
- (3) Slip rate / braking and traction force

and is applied according to the estimated μ .

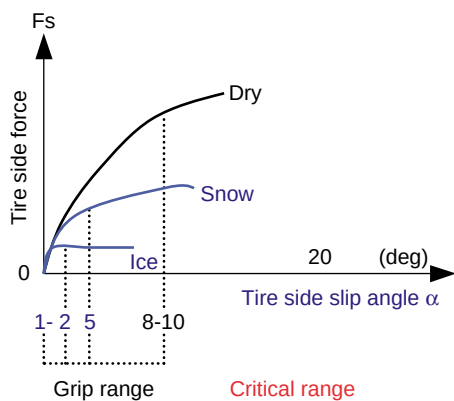


Fig. 2 Tire side force characteristics

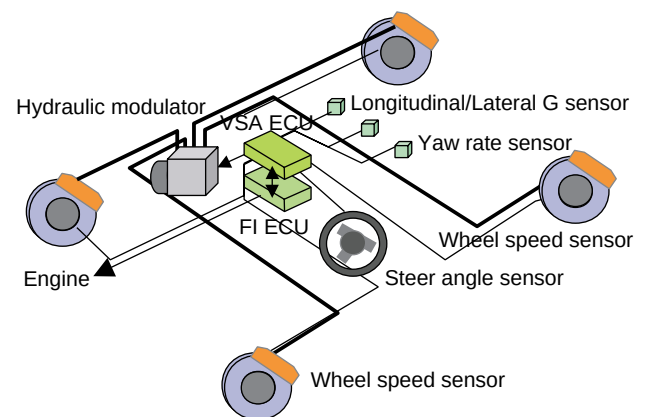


Fig. 4 System configuration

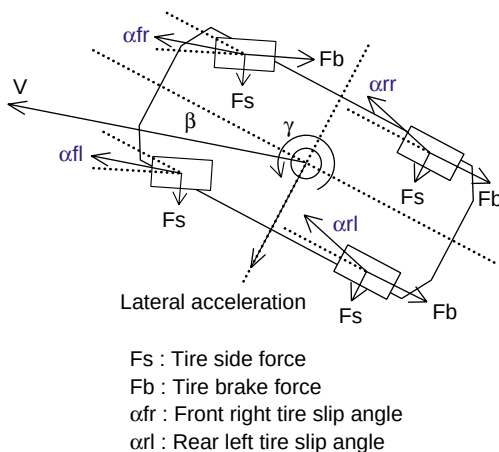
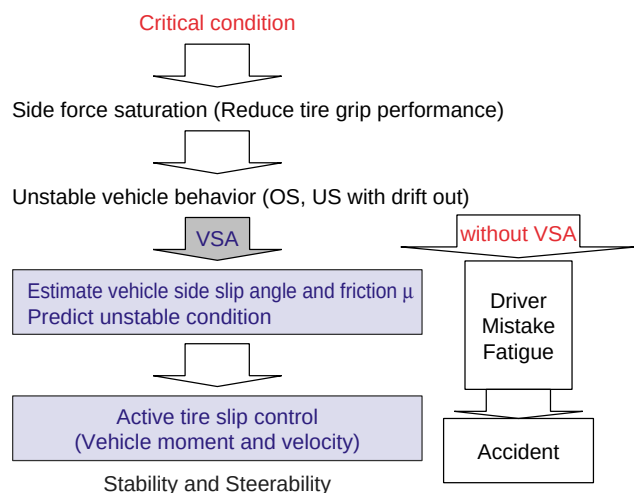


Fig. 3 Vehicle dynamics and stabilizing flow



6. Vehicle Side Slip Angle β and Road Surface μ Estimation Vehicle Data

The results of actual vehicle tests using the two types of logic outlined above are shown below. Fig. 7 shows the β estimation results during slalom course driving (40km/h) on an icy road. It was confirmed that the estimation results obtained were almost identical to those of the vehicle side slip angle sensor. Fig. 8 shows the μ estimation results when performing a slalom maneuver between dry asphalt and a low μ (wet basalt) surfaces at 40 km/h. It was confirmed that μ correction is carried out in real time when entering the low μ area. Wheel velocity data shows control dynamics.

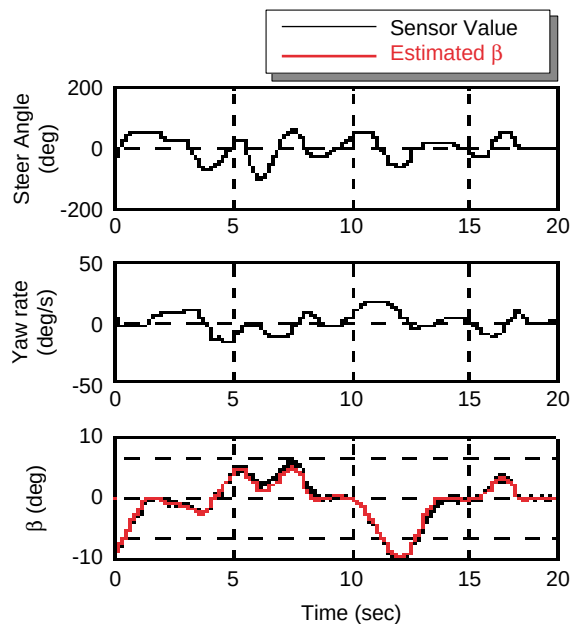


Fig. 7 β estimation test data

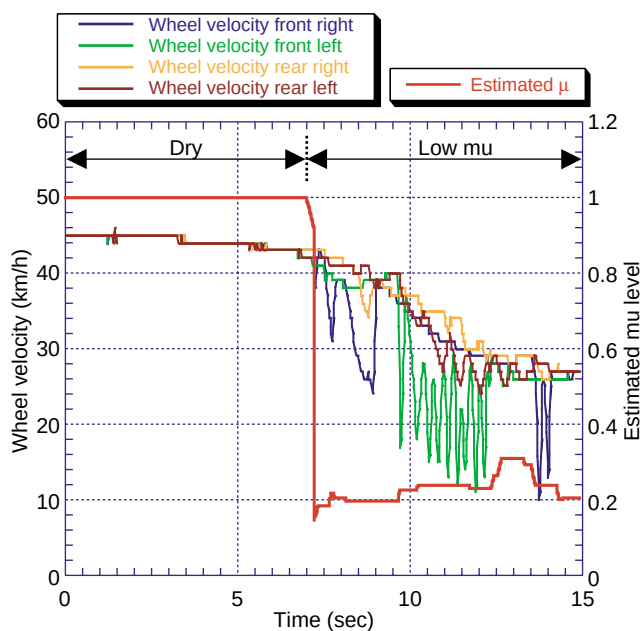


Fig. 8 μ estimation test data

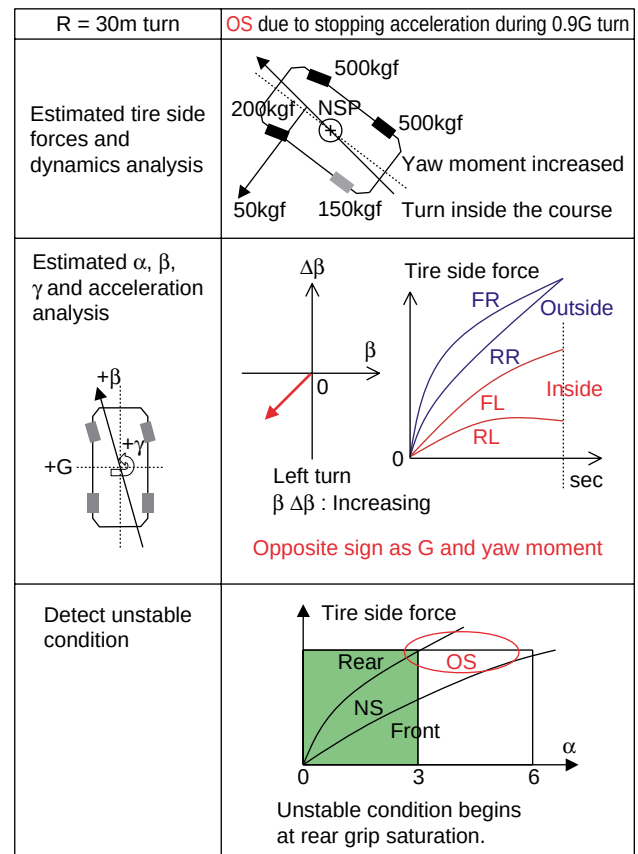


Fig. 9 Stability analysis

7. Tire Side Force Estimation and Stability Judgments

Fig. 9 shows the results of analysis of OS using estimated tire side slip angle α , vehicle side slip angle β , yaw rate, lateral acceleration, and tire side forces. According to these results, we found that rear tire side forces are saturated from the onset of OS. Calculating the absolute value of αr (rear wheel side slip angle) and the differential value $\Delta \alpha r$, then comparing them with each limit (judgment value), makes it possible to make an early prediction of critical conditions.

For reference, here is a brief discussion of US. In comparison with OS, which is excess moment due to a drop in rear tire grip, almost all US is a deviation from the target trace line due to excess speed. Quantifying the amount of deviation is difficult with only the current sensors and logic. This is because there is no method for accurately knowing what the target driving line should be.

Also, with conventional US control, the control amount is set by a comparison between the nominal yaw rate at the velocity estimated from the steering angle, and the sensor's yaw rate. However, even if yaw moment control is achieved with this method, it is not possible to return to the target driving trace line. Therefore, this research project is limited to OS control, explaining the control to achieve stability and steerability, and comparing it to conventional yaw rate control. As for US control, it is not thought that current performance can be enhanced with simple vehicle unit control, and thus it is left as a future issue.

8. Optimum Control of Vehicle Dynamics

The optimum control, this project's third issue, is discussed here.

The process for analyzing OS and spin during cornering on icy and snowy roads can be divided into four stages as shown in Fig. 10.

- (1) Rear tire side slip angle αr exceeds limit value.
- (2) Rear tire side forces are saturated.
- (3) Vehicle side slip angle and yaw moment increase.
- (4) Grip is lost and consequentially so is steerability, rendering turning impossible.

As can be understood from this process, if OS can be predicted in step (1), then by repeating both counter-moment control and deceleration control in steps (2) and

(3), vehicle stability can be maintained, steering is effective, and it is possible to recover the motion state. This is outlined in Fig. 11.

That is, stability uses front outside wheel brake control, and steerability uses the brake control of the two rear wheels to return to the grip range. This control flow is shown in Fig. 12.

The innovation of this research lies in the fact that, in addition to stability control, steerability control is also achieved simultaneously. That is, while leaving the lower limit side forces required to maintain rear tire stability, optimum control (referred to as β control hereafter) is used to obtain maximum deceleration. Tire side slip angle α and road surface μ estimation logic are indispensable in achieving this control.

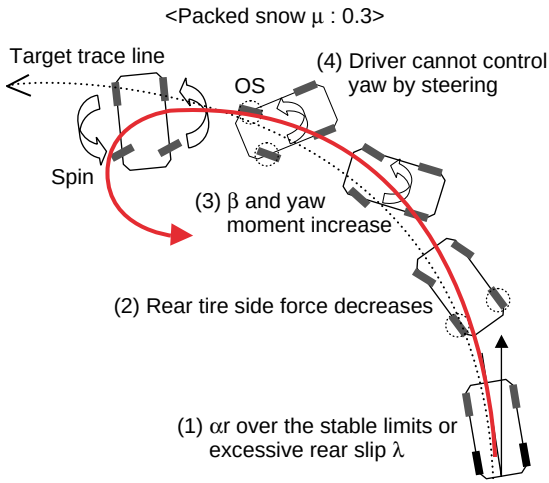


Fig. 10 Over steer dynamics analysis

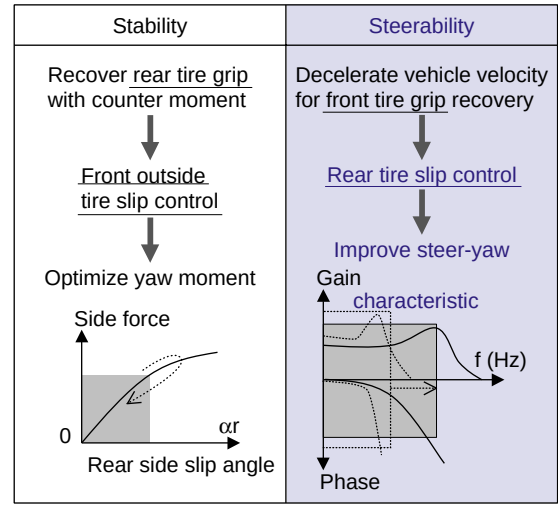


Fig. 11 Optimum control concept

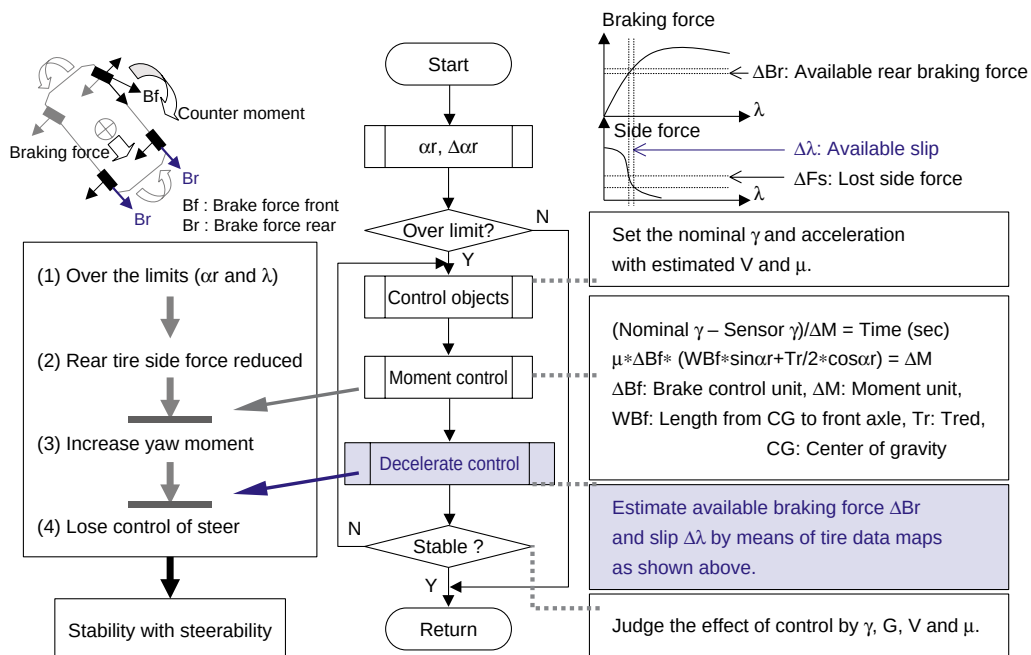


Fig. 12 Basic control logic

9. Vehicle Control Test Data

Driving tests were implemented in three modes, slalom, J-turn, and winding, on both compact snow (μ = approx. 0.4) and ice (μ = approx. 0.15). Due to limits on the length of this paper, only the data for the slalom and the J-turn on compact snow are included here (Fig. 13, Fig. 15: β control, and Fig. 14, Fig. 16: conventional yaw rate control). The graphs show time series data for steering angle (red: left turning is +), estimated vehicle side slip angle β (blue), and yaw rate (green: left turning is +).

When comparing β control to conventional yaw control, it is clear both β control stability and steerability are

enhanced. With β control, yaw rate and side slip angle restraint performance is good, and steering angle movement (driver input) is stable. The driver can feel a secure cornering force. This effect matches the first targeted effect of enhanced stability. In comparison, conventional yaw rate control quickly subdues excess yaw movement itself, but the ability to maintain a stable position (side slip angle) is missing. That is, grip force when turning fluctuates and steering correction is large and frequent.

In other driving modes as well, β control has better stable line trace performance. In particular, β control effectiveness is greatest with a low μ and at high vehicle speeds, that is, the closer to the limit, the bigger the effects of β control.

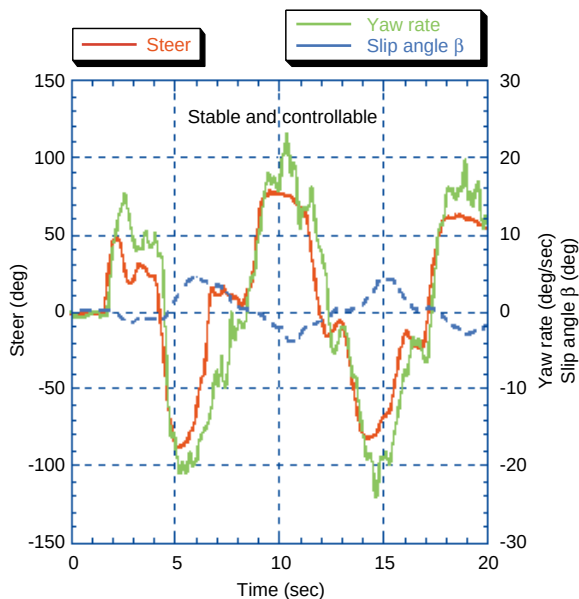


Fig. 13 Slalom test on snow (β control)

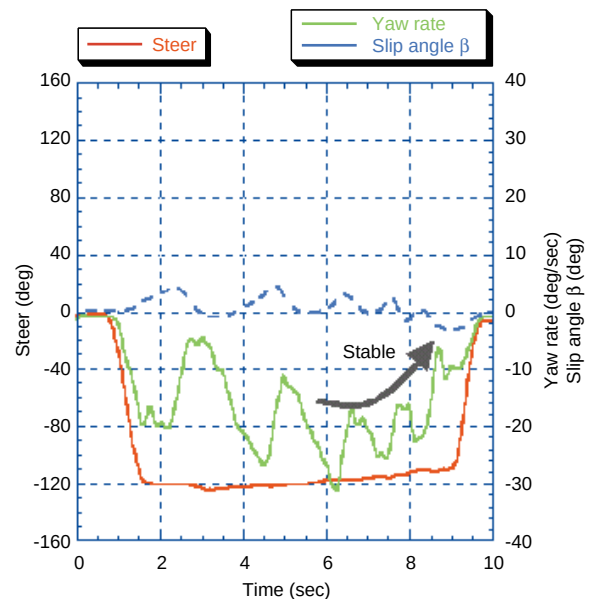


Fig. 15 J turn test on snow (β control)

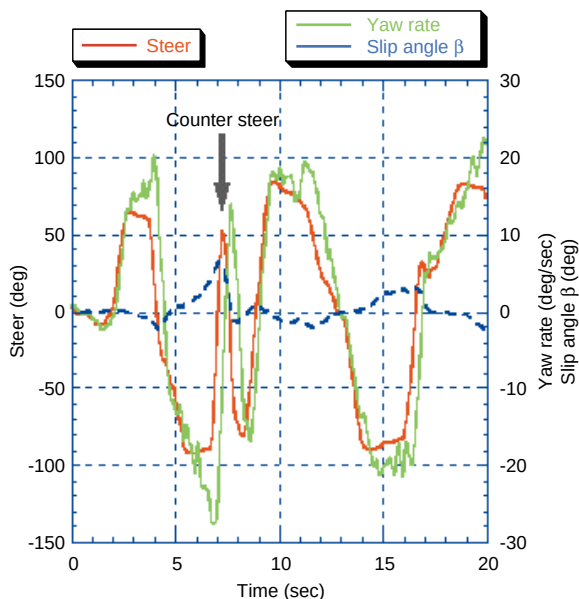


Fig. 14 Slalom test on snow (Yaw rate control)

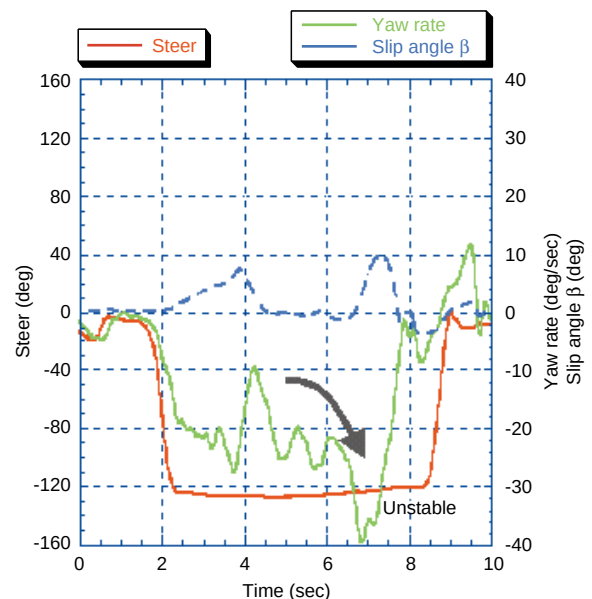


Fig. 16 J turn test on snow (Yaw rate control)

10. Control Effects and Evaluation Method

The results of testing on ice and snow covered roads are examined. First, the evaluation method will be discussed.

When evaluating non-linear dynamics, $\beta - \Delta\beta$ phase flat Lissajous should be considered, but as shown in Fig. 17, we must divide the dynamics into two types – steering angle and yaw rate, and steering angle and side slip angle – and examine them closely. The reason for this is that vehicle dynamics control makes it possible to achieve both stability and steerability by supporting good input/output characteristics in relation with the driver. That is, it is

necessary to study whether or not vehicle yaw rate and side slip angle response process for driver steering input is stable and feeding back the amounts and directions which correspond to the predictions. Also, if these results are good, then the characteristics should be good in terms of $\beta - \Delta\beta$ phase, too.

(1) Steerability Evaluation

As long as the steering angle / yaw rate flat trace stays, as a whole, in a narrow band range, time phase delay is small and over-steer and steer lag are reduced (steerability). Furthermore, if the trace path passes close to the point of origin, control is certain, correction by

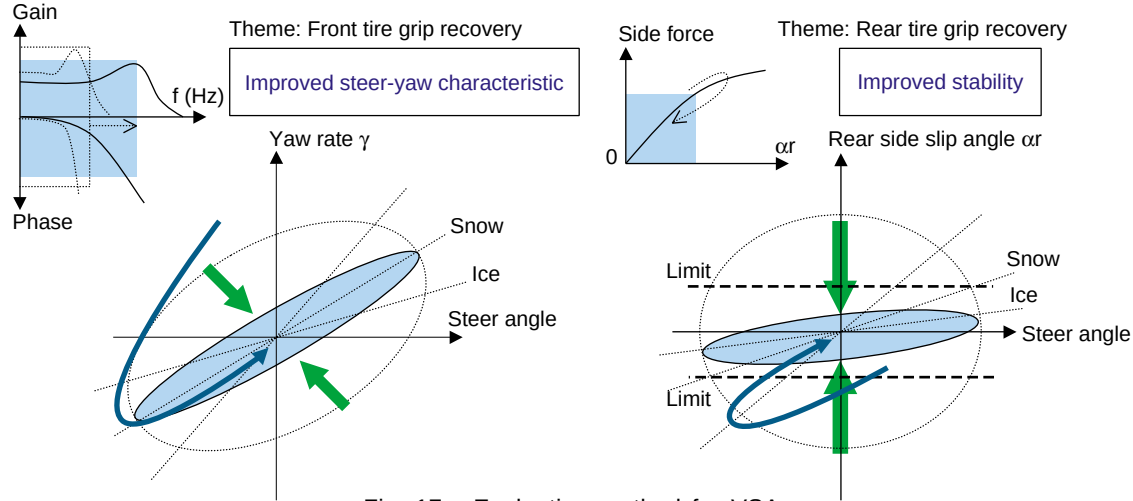


Fig. 17 Evaluation method for VSA

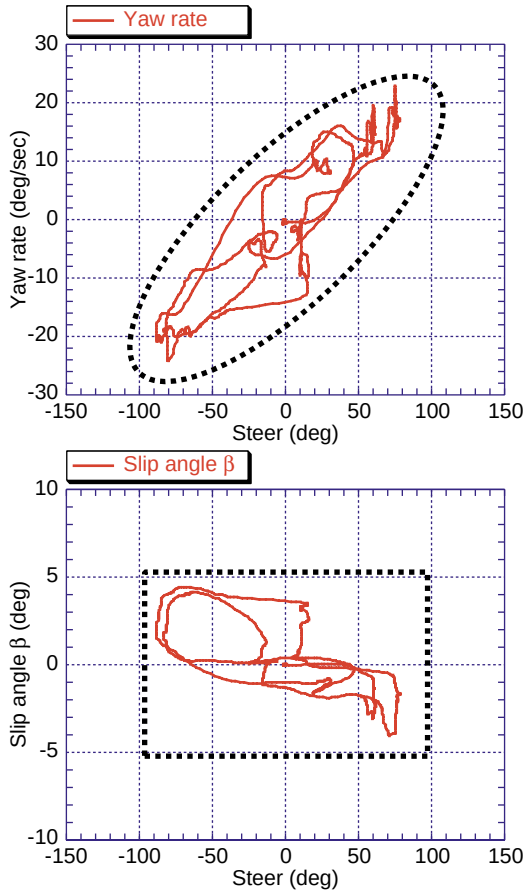


Fig. 18 Slalom evaluation on snow (β control)

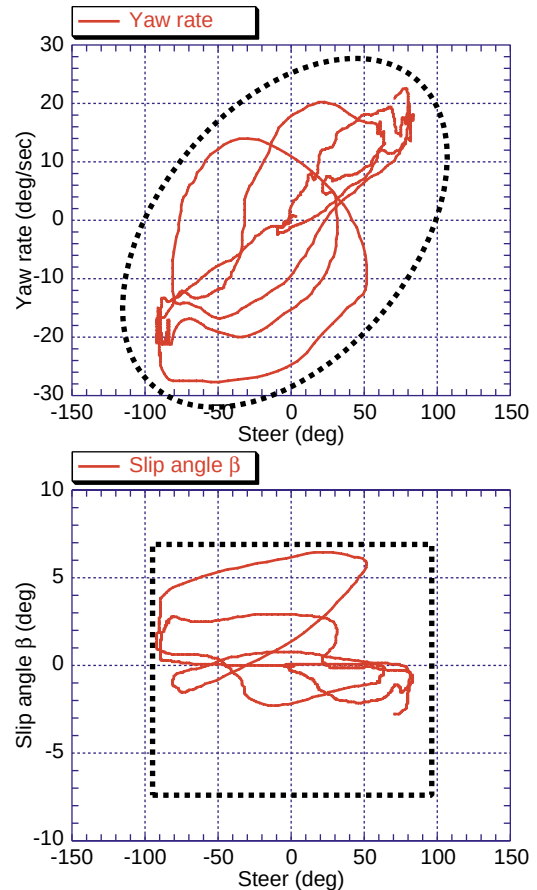


Fig. 19 Slalom evaluation on snow (Yaw rate control)

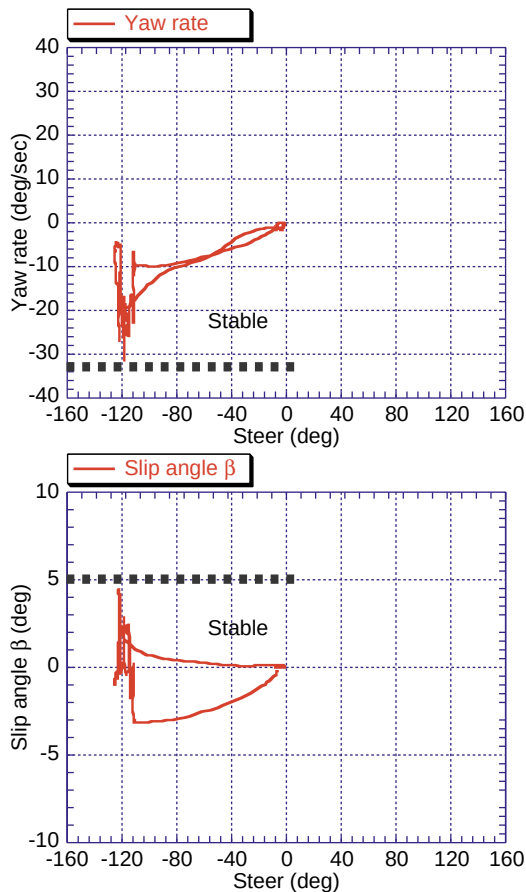
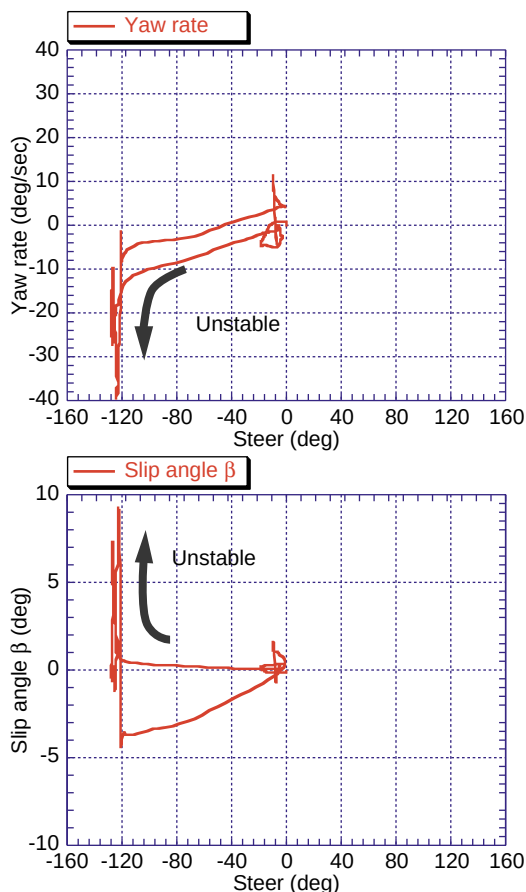
Fig. 20 J turn evaluation on snow (β control)

Fig. 21 J turn evaluation on snow (Yaw rate control)

the driver is small, and stability can be maintained.

(2) Stability Evaluation

As long as the steering angle / side slip angle flat trace is kept below the rear side slip angle α limit value, yaw rate movement is stabilized (stability). Furthermore, if the trace path passes close to the point of origin, control is certain, and there is little counter-steer by the driver. It is possible to reduce divergence in slalom driving.

Next, the test data for the compact snow slalom and J-turn test modes are discussed. (Fig. 18, Fig. 20: β control; Fig. 19, Fig. 21: conventional yaw rate control.)

β control has the following qualities over yaw rate control.

- (1) Concerning steering and yaw characteristics, time phase lag is small, and control is reliable, so steerability is high.
- (2) Concerning steering and side slip angle characteristics, excessive side slip angle is reduced and stability is high.

11. Conclusions

The goal of this research project was to achieve both of the following:

- (1) Enhance stability
- (2) Enhance steerability

Based on the vehicle control effects shown in the section above, "handling that is stable and is faithful to the steering" is achieved. Thus, basically, the validity of the optimum 4-wheel slip control logic has been proven.

The following three concrete results have been achieved:

- (A) Creation of a basic logic for accurately estimating stable side slip angle (vehicle, tires)
- (B) Creation of a basic logic for a practical, real-time, tire / road surface μ estimation logic
- (C) Creation of a basic method for optimum slip control in order to obtain both stability and steerability.

We will continue development in the fields of stability and steerability technology as we work toward practical applications of this technology.

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Study of the Dynamic Characteristics of the Suspension as Related to Brake Judder Reduction

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ABSTRACT

The reduction of brake judder requires refined tuning of the dynamic characteristics in the suspension, which as an oscillatory system has a great influence on brake judder. Here, a component model of the suspension is used to calculate its vibration mode and frequency characteristics. This makes it possible to conduct a comparative examination of the influence of the suspension geometry (the main factor in brake judder), bushing characteristics, parts weight, and other such factors by means of simulation. As a result, the values predicted by calculation have been found to correspond closely with the actual measurements, and compliance steer has been confirmed to play a significant role in reducing brake judder.

1. Introduction

The source of brake judder is known to be excitation caused by braking torque fluctuation resulting from uneven wear and rusting of the disk rotor. Countermeasures include reducing disk rotor run-out and improving the grinding characteristics of the friction material, but these efforts still have not eliminated brake judder.

Whereas the source of excitation for the phenomenon is the brakes, the resonance system is the suspension and the transfer system includes the body and steering system. Thus, many parts are involved with the phenomenon and the mechanism by which it occurs is quite complex. Therefore, experimental analysis approaches come to a stop at understanding the phenomenon and are difficult to translate into guidelines for further development, which has made finding a solution to the issue difficult.

Looking at the suspension, which is a resonance system, the dominant mode when brake judder occurs is known to be the mode in which the tires move in a longitudinal manner called compliance vibration of the suspension.

This compliance plays an important role in improving, for example, ride comfort, but it also has the conflicting characteristic that this first order vibration is the main cause of steering shimmy and brake judder vibration.

The focus of this study is to clarify why compliance vibration of the suspension excites the steering system. This issue was approached through vibration testing using actual vehicles and parameter studies using finite element method simulations of the placement of suspension mounting points and the bushing characteristics. The results confirmed that reducing suspension transfer sensitivity and reducing input to the steering system are also effective at reducing brake judder in an actual vehicle.

2. Understanding the Phenomenon

The circumferential acceleration on the steering wheel and the acceleration at various suspension points were measured during actual cruising and frequency analysis was performed to understand the suspension behavior when brake judder occurs. The results showed that the largest component occurs around approximately 15 Hz. Fig. 1 shows a typical example of the time history data and frequency analysis results for the steering wheel.

Fig. 2 shows the vibration mode extracted at the peak gain frequency in order to confirm the suspension behavior at this time. This figure shows that longitudinal vibration of the suspension, that is to say compliance vibration, is the dominant behavior when brake judder occurs.

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The various suspension and brake parts are excited by compliance vibration, and this inertia force mainly occurs as a moment around the kingpin shaft. Axial force is generated on the tie rod as reaction force to support this inertia moment, and this axial force passes through the steering gear box and becomes rotational vibration of the steering wheel.

These test results confirmed that braking torque fluctuation is conveyed through this type of transfer path and becomes steering vibration.

3. Analysis by Simulation

The results of the above running tests clarified that compliance vibration of the suspension has a large influence as a resonance system. Therefore, it was necessary to create a simulation model capable of reproducing these phenomena.

The modeled phenomena were the weight of each part comprising the suspension and brake, the bushing characteristics of the body mounting points, the rigidity of the body mounting points, and the position of each point comprising the suspension link mechanism. Fig. 3 shows the calculation model created from this information.

Note that with the exception of the bushings comprising

each device, the influence of elastic deformation of the suspension parts was considered to be minute, so the model was created using rigid body elements. In addition, part weights were modeled by the dispersed location of concentrated mass at the representative points of each part. This also made it possible to easily express inertia moments that had been difficult to obtain thus far.

This model was used to calculate the vibration mode during compliance vibration and the transfer characteristics from the tire ground contact point to the rack end by performing frequency response analysis using the general-purpose finite element method solver NASTRAN. Relative evaluation with respect to changes in each influencing factor is possible by performing comparative investigations of these results.

4. Example of Application to Development

An example of application to actual development is shown below. Here, the two countermeasure proposals A and B were investigated with respect to the base specifications. The proposal A specifications reduce the spring rate and improve the damping factor of the compliance bushing with respect to the base specifications. The proposal

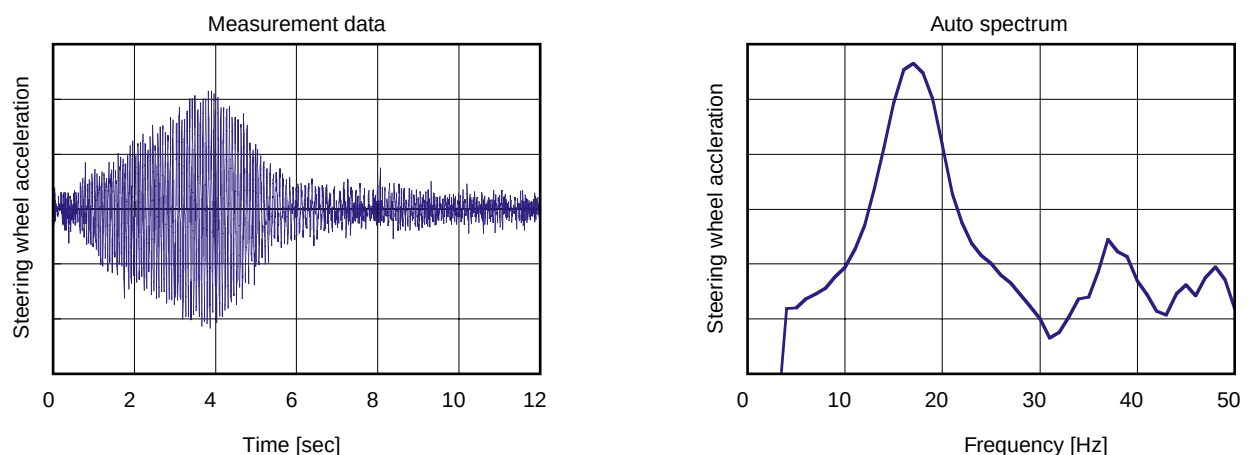


Fig. 1 Example for steering wheel acceleration

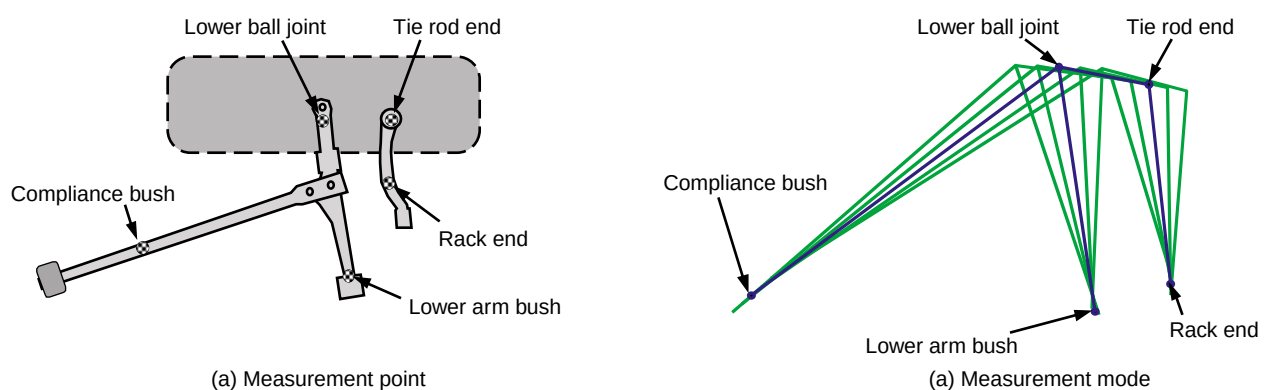


Fig. 2 Compliance vibration: experimental mode

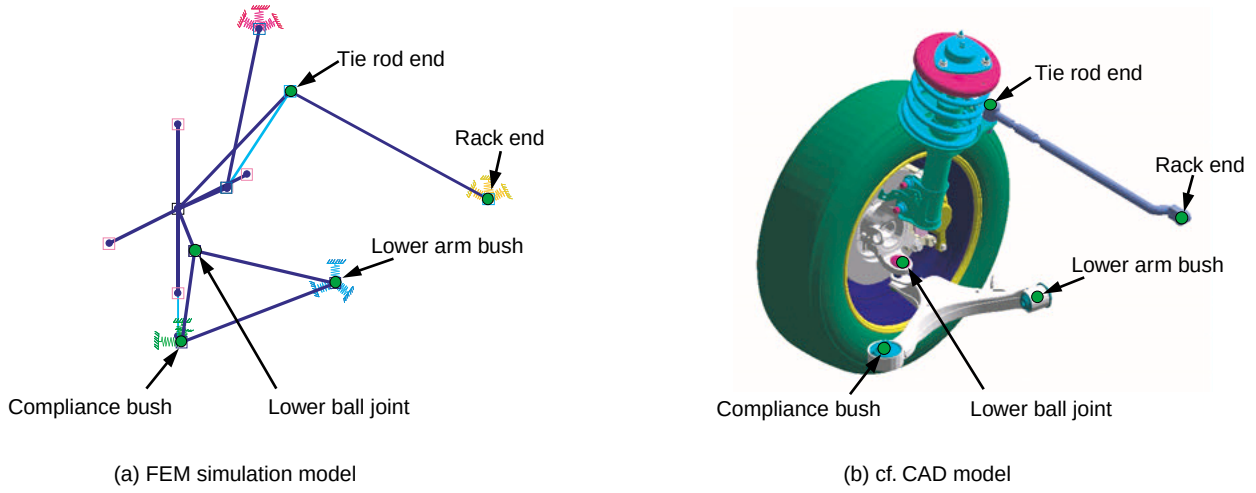


Fig. 3 Suspension component model

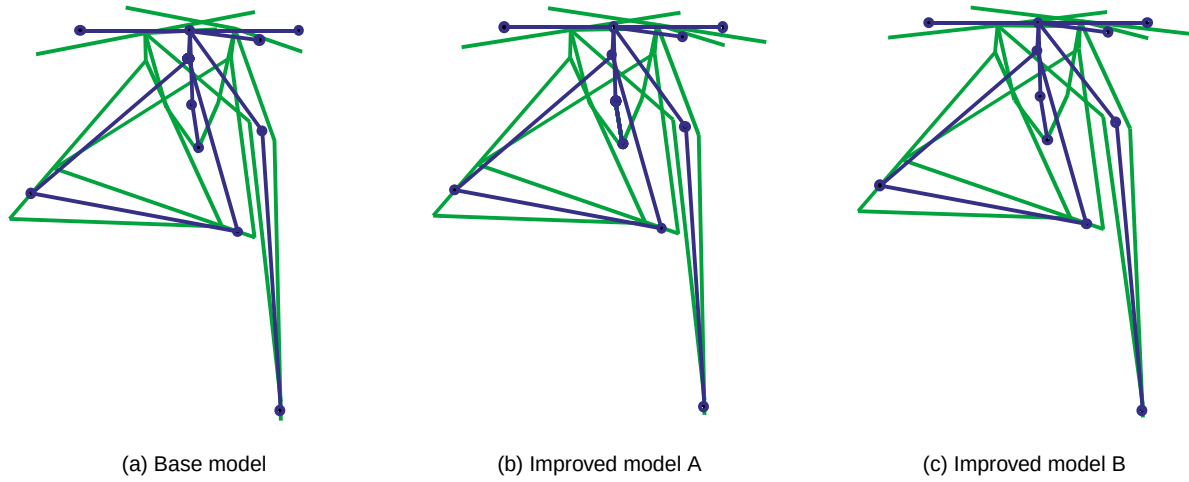


Fig. 4 Simulation result: natural frequency mode

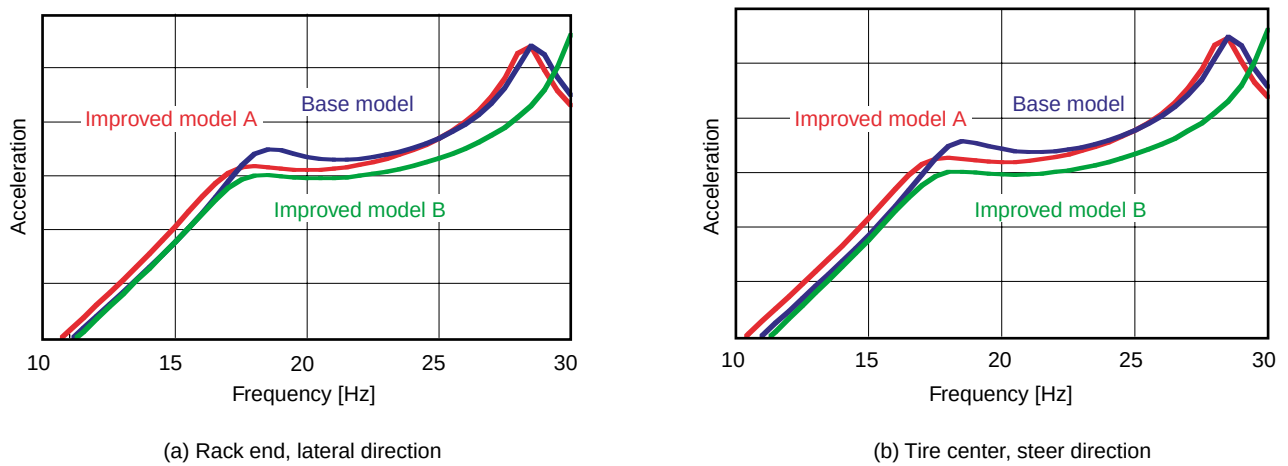


Fig. 5 Simulation result: auto spectrum

B specifications build on the proposal A specs by changing the rack end position to the rear. Fig. 4 and Fig. 5 show the calculation results for each of these specifications.

Proposal A further improves the vibration energy scattering effects due to the compliance bushing by changing the bushing characteristics, and is effective at reducing the peak gain.

In addition, examining the vibration modes for both proposals A and B shows that rotational movement in the tire toe direction, or the so-called compliance steering, due to longitudinal vibration decreases in stages with the addition of each countermeasure item compared to the base specifications. This can also be confirmed from the decrease in the steering direction gain at the tire center in Fig. 5(b).

The moment occurring around the kingpin shaft is generated by the inertia force of the tire, wheel and incidental parts that perform rotational movement, and is a major factor in exciting vibration. Directly reducing this moment is extremely effective in reducing the transfer sensitivity of the suspension.

These results confirmed that each of the countermeasures were improvements over the base specifications, so these improved specifications were verified by actual vehicle tests.

Fig. 6 compares the results of actual measurements and simulations of steering wheel acceleration in the rotational direction. This figure shows an excellent proportional relationship between the calculations and actual measurements, thus confirming the validity of this method.

5. Analysis of the Degree of Contribution of Each Factor

Most of the various suspension-related specifications have an influence on multiple vehicle performances, so it is necessary to set characteristics that achieve a balance between each performance. Accordingly, it is extremely important to know the degree of influence that each factor has on the respective performances.

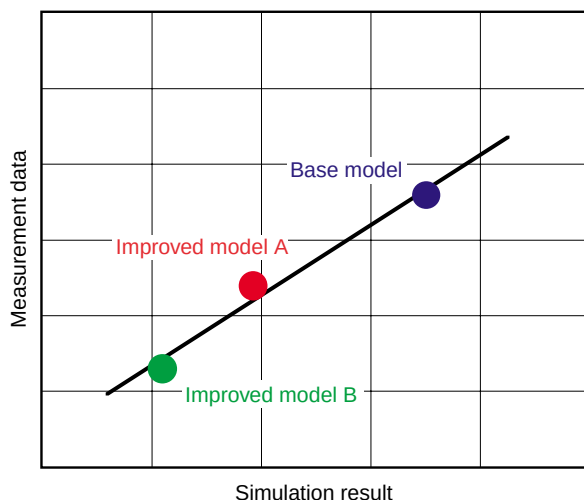


Fig. 6 Acceleration comparison: simulation result and measurement data

Therefore, parameter studies were performed for a number of characteristics that were thought to have a large influence on brake judder, and the degree of contribution was analyzed for each factor. Fig. 7 shows the transfer characteristics from the tire ground contact point to the rack end and the steering direction gain at the tire center as the change in characteristics when each parameter is changed. Here, the calculation results for the base specifications are used as the reference.

These results show that the influence of the compliance bushing characteristics is the largest among the investigated factors. As described in section 4, this is not merely the effect of improved vibration energy dissipation due to a reduced spring rate and improved damping factor; reduced compliance steering due to the spring rate change also has a significant effect. The results of each of these studies indicate a close relationship between the lateral fluctuations in the rack end and compliance steering, and also show a large contribution toward the transfer sensitivity of the suspension.

The results of this study showed that the compliance bushing characteristics also have a large influence on changes in the vibration mode. However, at the stage where each factor can be changed freely, the position of the suspension mounting points has the greatest influence.

However, the suspension geometry has basic characteristics that affect many performances, and investigations are repeated numerous times from the concept stage, so there are many cases where changes are difficult once the specifications have been decided. Therefore, it is important to optimize compliance steering as one of the basic characteristics from the initial development stages.

6. Conclusion

The mechanisms when brake judder occurs were analyzed, and theoretical investigations were carried out using a finite element model which reproduces these phenomenon. The following results were obtained.

- (1) Brake judder can be lessened by reducing the transfer sensitivity during compliance vibration of the suspension which is a vibration resonance system.
- (2) The suspension geometry and the compliance bush characteristics have the greatest influence on the transfer sensitivity during compliance vibration.
- (3) Compliance steering plays an important role as an influence on the vibration mode.
- (4) The simulation method proposed by this study was confirmed to be effective even in actual development.

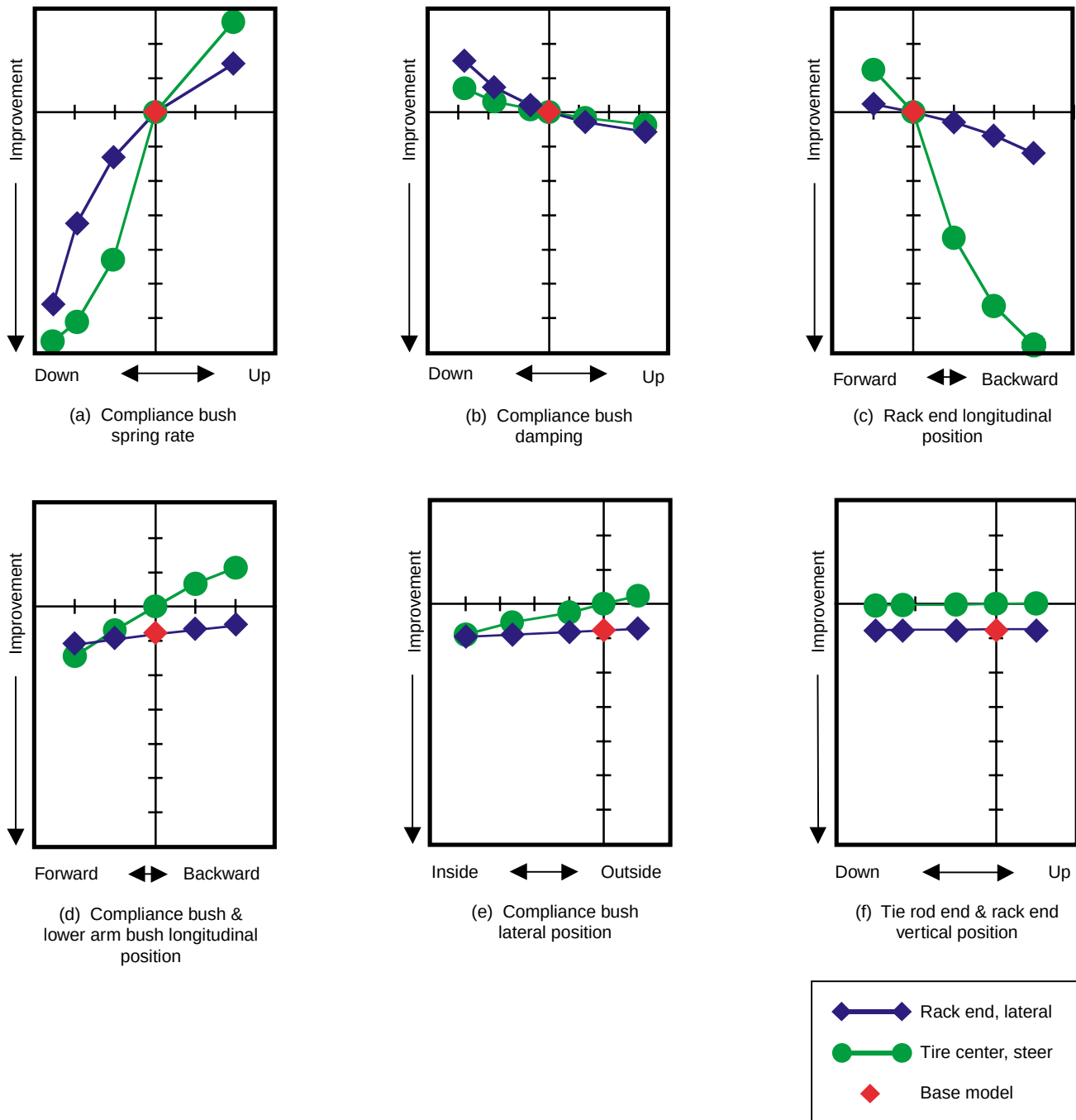


Fig. 7 Parameter study result

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Tetsuya YAMAGUCHI



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Enhancement of Cup Seal Performance in Brake Wheel Cylinders by Hyper Elastic Analysis

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Takeshi ITO*

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ABSTRACT

The shape of the cup seal that is installed in the wheel cylinder of drum brakes was examined using FEA hyper elastic analysis. The cup seal material was changed from the often used SBR to EPDM resulting in enhanced heat and wear resistance. Next, a prototype cup seal was made in a determined shape based on the results of the investigations. Various durability tests were carried out with the results confirming both enhanced durability and appropriate sliding resistance.

1. Introduction

Within the brake system of passenger cars, lip shaped packing normally referred to as a cup seal is used to seal parts that are exposed to high hydraulic pressures and sliding such as the wheel cylinder (Fig. 1).

Cup seals are mainly composed of synthetic rubber materials, and seal by completely contacting the cylinder surface (sliding surface) when fluid pressure is applied within the wheel cylinder.

In addition, there are various specifications for cup seals according to the shape and surface condition of the piston on which the cup seal is mounted and the cylinder against which the cup seal slides in order to satisfy the various functions required of brake wheel cylinders.

Thus far, the cup seal shape has generally been determined by repeatedly creating prototypes based on empirical guidelines, and then performing various tests and confirming durability. However, there are still many development issues such as the inability to observe conditions inside the cylinder during operation and the inability to measure rubber strain.

Therefore, this study conducted investigations to optimize the shape and enhance durability by hyper elastic analysis using finite element analysis (FEA).

Excellent results were obtained in durability tests using a prototype cup seal, and these results are introduced here.

2. Functions required of cup seals and materials

The cup seal in a wheel cylinder is the output portion for the hydraulic pressure inside the drum cylinder, and high durability is required. "Low strain" is generally known to be important in enhancing the durability of rubber materials.

On the other hand, it is also known from experience that there are a number of points which must be taken into consideration in terms of functions. For example, the seepage phenomenon is thought to occur when the cup seal becomes worn due to stick slip with the piston, causing brake fluid to seep around the seal. Therefore, slipping between the cup seal and the piston leads to seepage and wear, and reduces durability. The sectional shape of part A shown in Fig. 2 is formed by cutting, and is called the cut lip. It is known that an acute angle at the end of the cut lip facilitates stick slip and that the high but not-constant sliding resistance exhibited

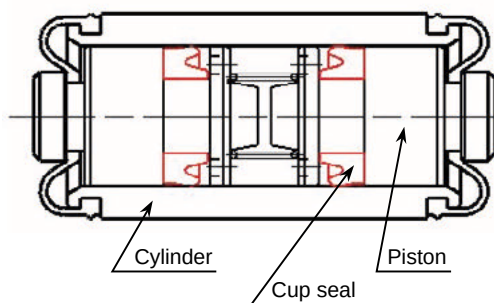


Fig. 1 Wheel cylinder

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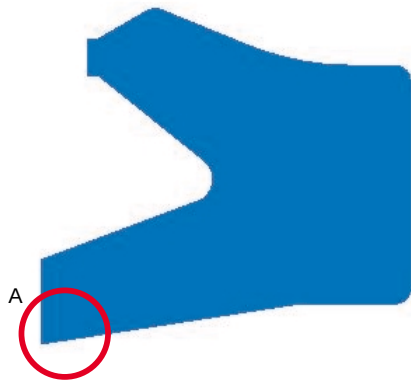


Fig. 2 Cup seal

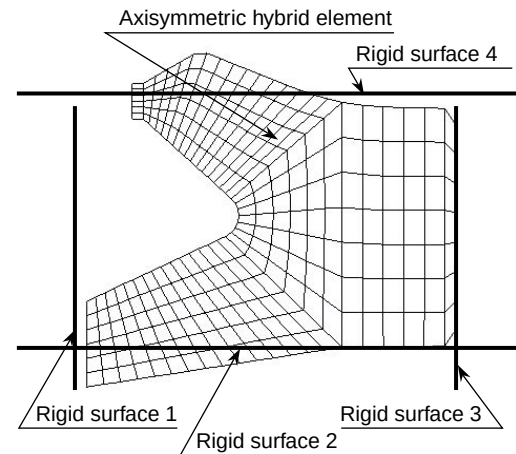


Fig. 3 FEA model

in tests results in unstable sliding resistance.

In addition, when external pressure which exceeds the frictional force is applied, relative sliding occurs between the cup seal and piston and between the cup seal and wheel cylinder, so the proper sliding resistance values must be ascertained. The wheel cylinder side constantly slides during operation, so a low sliding resistance is thought to be better. On the other hand, a high sliding resistance is desired for the piston side to prevent relative sliding when stick slip and rapid pressure changes occur.⁽¹⁾

The cup seal materials are generally rubber-like elastic solids collectively referred to as elastomers. These elastomers deform and produce strain when subjected to a load, but also exhibit non-compressive characteristics. That is to say, they are elastics but the contact portions exhibit fluid behavior. In addition, rubber materials generally experience deformation known as compression set due to macromolecular flow when subjected to stress for long periods of time.⁽²⁾

Styrene-butadiene rubber (SBR) is often used as the cup seal material. SBR is resistant to brake fluid and has excellent low temperature performance, but its physical properties can deteriorate at high temperatures (120°C or more). Therefore, this study investigated an ethylene propylene diene monomer (EPDM). EPDM features superior heat resistance, with little change in physical properties even at temperatures of 150°C or more. Furthermore, it exhibits the same low temperature performance as SBR down to -40°C, and has wear resistance superior to SBR.

3. Modeling the cup seal

The model used in this study requires advanced calculation techniques called nonlinear analysis. The cup seal is made of a hyper elastic material that experiences significant deformation which results in contact, so the correlation was expected to be poor. Therefore, the cup seal was modeled in two dimensions using axisymmetric hybrid elements as shown in Fig. 3 instead of three dimensions in order to simplify the analysis parameters as much as possible. The material characteristics were obtained from biaxial tensile tests on a test piece using the Mooney-Rivlin fixed number that corresponds to Young's Modulus for metal.

The piston and cylinder are extremely rigid compared to the cup seal, so these are modeled as rigid surfaces, and contact elements were located on the contact surface of the cup seal and each rigid surface. The coefficients of friction were calculated using the sliding resistance obtained from tests, and were $\mu = 0.135$ for the piston and $\mu = 0.205$ for the cylinder.

The condition with the cup seal assembled into the wheel cylinder is expressed by the forced displacement of the rigid surfaces in each step.

The cup seal is completely unconstrained in the initial state, so there is no balance of forces and the condition is unstable. Therefore, this study began the calculations from the condition where rigid surface 1 is already in contact with the cup seal.

A pressure load was applied to the end of the cup seal that contacts the brake fluid as shown in Fig. 4 to reproduce the deformed condition under pressure.

The sliding resistance was obtained from the fulcrum reaction force produced on the rigid surfaces corresponding to the piston and cylinder sides.

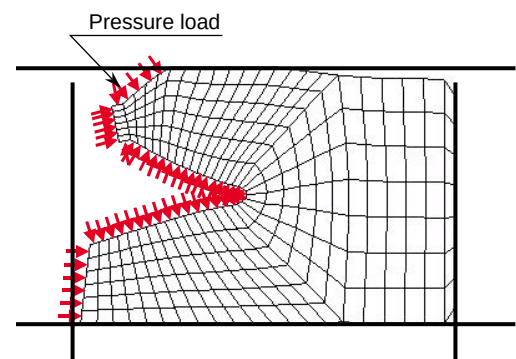


Fig. 4 Boundary condition of pressure load

4. Analysis results

Fig. 5 shows the analysis results for the stress produced on the cup seal, and Fig. 6 shows the analysis and test results for the sliding resistance produced on the piston and cylinder.

As shown in Fig. 5(b), stress was confirmed to concentrate in part B when the cup seal was assembled into the wheel cylinder. In addition, analysis showed that the sliding resistance is smaller on the piston side than on the cylinder side as shown in Fig. 6. This same tendency was also confirmed in tests.

The sliding resistance was compared through finite element analysis and tests for a number of shapes as shown in Fig. 7. These results are shown in Fig. 8. The analysis and test results correlate fairly well, thus confirming that this analysis method can be utilized for shape investigations.

5. Investigation to optimize the shape

Based on the above analysis results, investigations were carried out using finite element analysis to achieve an optimum cup seal shape that satisfies the following target performances.

- (1) Enhanced durability by reducing strain
 - There should be little strain when assembled into the wheel cylinder.
 - Stress should not concentrate locally.

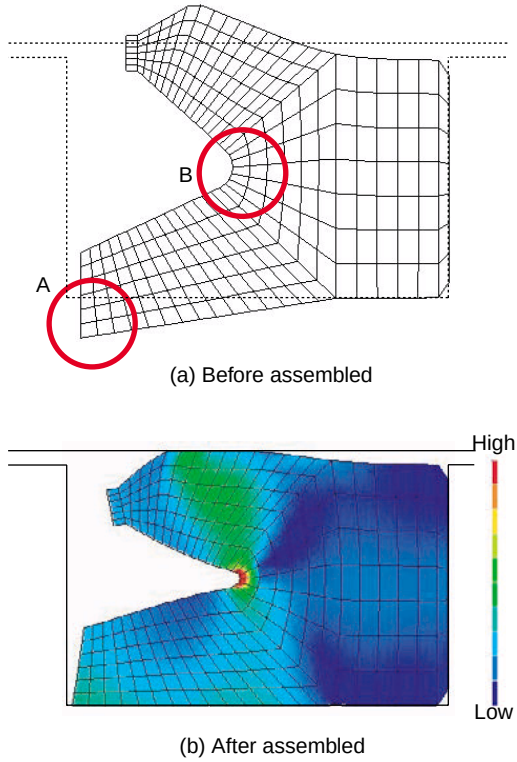


Fig. 5 FEA model of the current design and stress calculation results

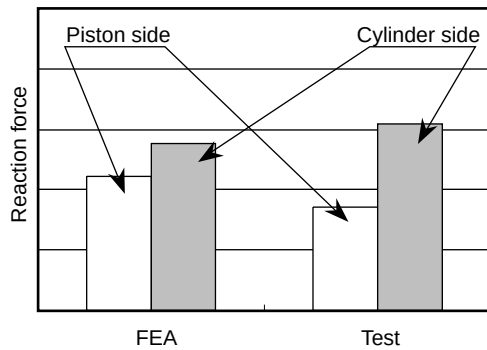


Fig. 6 Calculation result and test results of reaction force

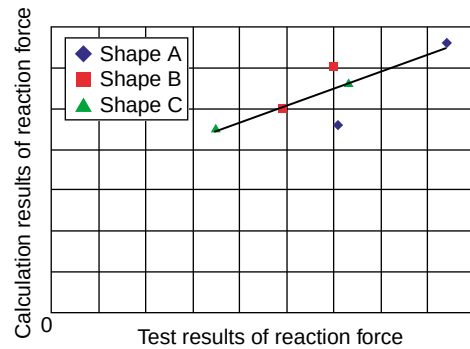


Fig. 8 Correlation of reaction force

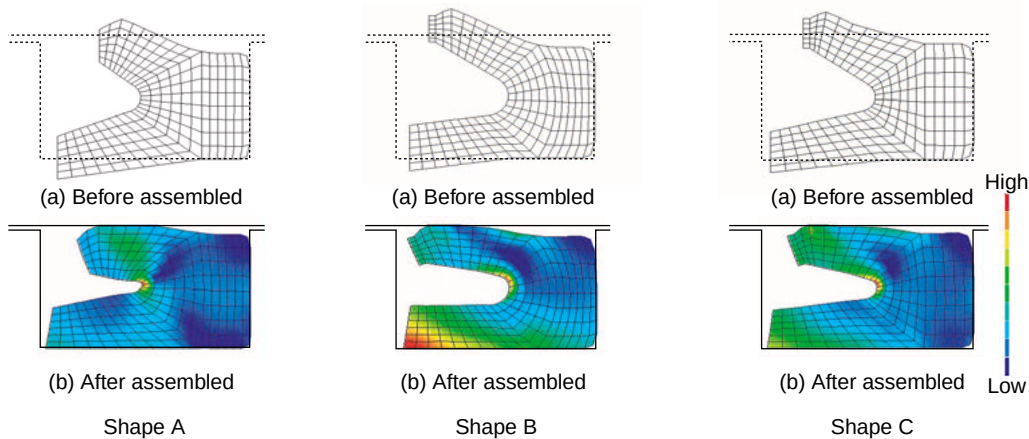


Fig. 7 FEA model of other designs and stress calculation results

(2) Appropriate sliding resistance

- Stick slip should not occur.
- The sliding resistance on the piston side should be larger than on the cylinder side.

5.1 Enhanced durability by reducing strain

When the radius of curvature is increased to reduce the strain when assembled into the wheel cylinder and disperse the stress of part B, the sliding resistance was found to drop and other required performances also could not be satisfied. Various shapes with different lip widths, for example, were investigated, and a shape joined by a large and small radius of curvature such as that shown in Fig. 9 was found to be the most effective for distributing stress.

5.2 Appropriate sliding resistance

An obtuse angle shape called a molded lip was adopted for the shape of part A as shown in Fig. 9(a) to adjust sliding resistance instability. In addition, the sliding resistance on the piston side must be relatively large, so the interference on the inner diameter of the cup seal was increased within the range allowed by the assembly performance. Conversely, the shape was changed to reduce the cup seal surface pressure against the cylinder in order to reduce the sliding resistance on the cylinder side. The analysis results confirmed that these changes achieved the target performance for sliding resistance as shown in Fig. 10. Prototypes were then manufactured based on these analysis results, and measurements performed using these prototypes showed the same tendencies.

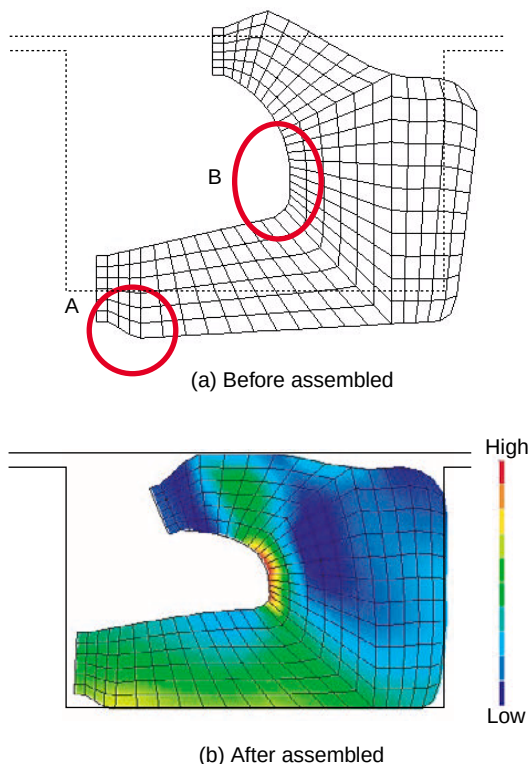


Fig. 9 FEA model of new design and stress calculation results

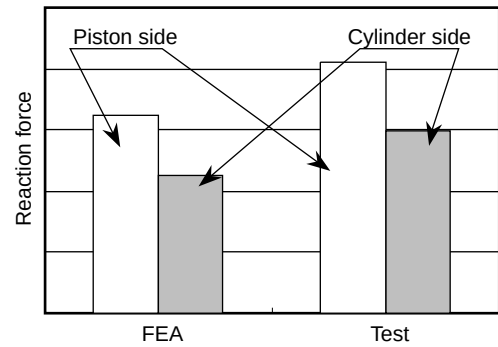


Fig. 10 Calculation and test results for reaction force

6. Confirmation of durability using prototypes

Prototypes were manufactured using the shape obtained through the above investigations, and various durability tests were carried out to compare the prototypes with the current product.

Fig. 11 shows the results of high temperature operation durability tests carried out at a test environment temperature of 120°C. These results show interference is more consistent from the initial state until the end of the test than for the current product, indicating that durability was enhanced.

Fig. 12 shows the results of bench durability tests in which the cup seal was left assembled into the wheel cylinder at a high temperature (120°C). Like the high temperature operation durability test, there was also less change in the

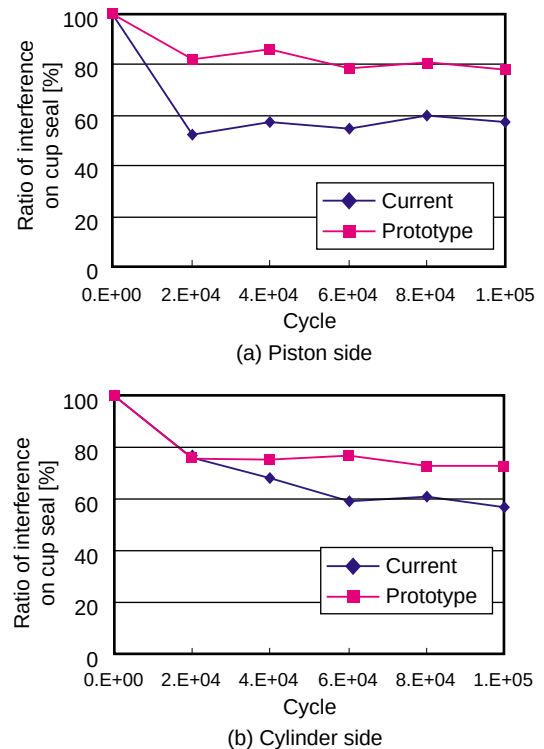
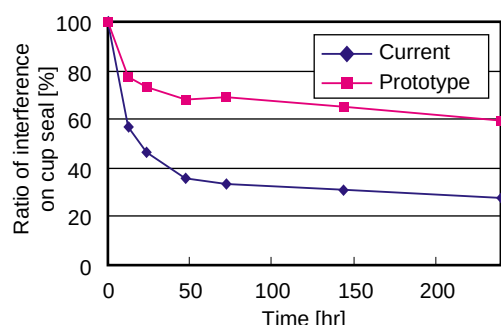
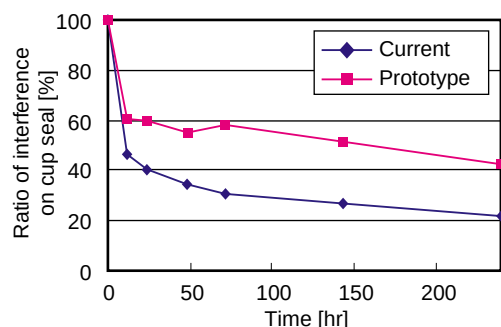


Fig. 11 Cup seal interference degradation during the high temperature cycle endurance test



(a) Piston side



(b) Cylinder side

Fig. 12 Cup seal interference degradation during the high temperature exposure test

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- (2) Toshio Fujiyoshi: Plastic Processing Technological Handbook, Tokyo, Nikkan Kougyo Shinbunsha, p.143-153, 1995

interference when left for long periods, indicating that durability was enhanced.

These results confirmed that the cup seal investigated using hyper elastic analysis achieved the target performances in terms of both enhanced durability and appropriate sliding resistance.

7. Conclusion

A cup seal structure that satisfies the following target performances required by the wheel cylinders of drum brakes was investigated by hyper elastic analysis using finite element analysis.

(1) Enhanced durability by reducing strain

(2) Appropriate sliding resistance

Changing the materials from SBR to EPDM enhanced resistance to both heat and wear.

A prototype was manufactured based on the above analysis results and various durability tests were carried out. As a result, the developed prototype was confirmed to achieve the target performances.

Therefore, the proposed analysis method was found to be effective for investigating the cup seal structure.

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Research of Control Countermeasures for 3D Uneven Solar Radiation with Automatic Air Conditioning Systems for Automobiles

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ABSTRACT

A new automobile air conditioning system was developed that is capable of maintaining passenger comfort under changing and uneven solar radiation conditions. This project differentiated the solar radiation energy that is absorbed by the human body affecting passengers directly, and energy that is absorbed by the car body and radiated to the passengers by means of the cabin air. Values for these two were then derived for every position of the sun. Based on the results, a map was created that enables the air conditioning system to change the coefficient adjusting for the influence of solar radiation. A control system has been developed that calculates the position of the sun relative to the automobile by means of a Global Positioning System, and provides only as much cooling as is needed to offset the amount of solar radiation energy absorbed by the passengers according to the position of the sun. An automobile with this air conditioning system installed was subjected to field study tests, which confirmed that passengers felt a constant level of comfort even when the direction of the sunlight changed with respect to the car body.

1. Introduction

An air conditioning system for an automobile is unlike that for a building in that a considerable part of the heat input from outside is composed of radiation energy from the sun. Consequently, automatic air conditioning systems are equipped with sunbeam sensors in order to control the outlet air temperature and airflow rate according to the intensity of the solar radiation. The conditions in an automobile are such that the solar radiation may hit the passengers directly. In such cases, the direct solar radiation energy that contacts the human body will vary according to such factors as the position of the sun and the location of the passenger. Therefore, it is not sufficient to simply control the cabin temperature in order to maintain uniform comfort.

The control of uneven solar radiation can be accomplished by such means as 2-dimensional sunbeam sensors⁽¹⁾⁽²⁾ and the use of two sunbeam sensors. However, it is not possible to compensate for the effects of uneven solar radiation throughout the cooling region, which changes according to the complex interaction of the sun, the automobile, and the human body. A method has already been developed for calculating the position of the sun from the location of the vehicle, and then using the relationship of this position with solar radiation intensity to control the autoair conditioning system with respect to uneven solar radiation.⁽³⁾ However, there has been no case as yet where the actual control amount was determined.

This paper describes our study of an auto air conditioning system that is capable of controlling the atmosphere

comfortably throughout the area of uneven solar radiation by analyzing the heat conveyed to the vehicle and passengers from solar radiation energy.

2. Analysis of comfort factors

2.1 Factors in comfort

One of the factors in automobile passenger comfort is the thermal environment of the cabin, which is determined by the three factors of heat transfer (convection, radiation, and conduction). These factors are different during the day and at night, during different seasons, and so on. At night, for example, the heat transfer caused by the temperature difference between the cabin and the outside environment, together with the movement of heat caused by ventilation, determine the thermal environment. During the day, the radiation energy from the sun becomes an additional factor that has a significant affect.

Apart from these factors of comfort that determine the thermal environment, there is also the factor of preference. According to standards such as ISO 7730, differences due to individual preferences are distributed over a range like that shown in Fig. 1.

Passenger comfort is ordinarily determined by the combination of these factors. However, the cabin of an automobile is a confined space, and the locations of the passengers are more or less fixed. The glass window area is large relative to the cabin volume, so direct sunlight on the passengers causes them to feel hot. This has a greater impact than the other factors.

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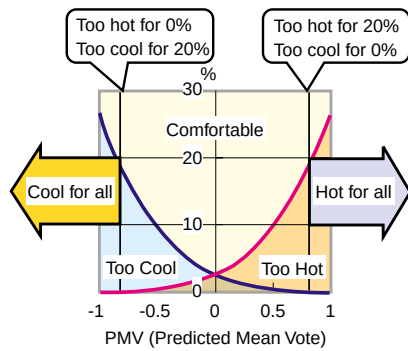


Fig. 1 Preference Distribution chart

2.2 Heat input to passengers

When there is little sunlight, the air temperature inside the car cabin is determined by the heat transmitted from outside, ventilation, air conditioning, and so on. When there is a lot of sunlight, the radiation energy from the sun becomes a larger factor, and, there is little movement of heat due to heat transfer, especially when the ambient temperature is close to the controlled cabin temperature. At such times, therefore, this radiant energy makes up the greater part of the heat input from outside the cabin.

Fig. 2 shows the route of heat input from solar radiation. This can be separated into two factors. One is the solar radiation that enters through the window and raises the temperature of interior materials and so on, thus raising the temperature of the air inside the cabin to thereby reach the passengers. The other factor is the radiation that strikes the passengers directly through the window and raises the temperature of their clothing and skin. With the former, the position of the sun makes relatively little difference, and it is easy to create a constant environment by using sunbeam sensors and so on to correct the cabin temperature. This is equivalent to the situation during a summer day when the solar altitude is high and the passengers are contacted directly by almost no sunlight.

In the case of the latter factor, the radiation energy will differ according to the relative positions of the passengers and windows, and so it will be different for every seat. Since the passenger is subject to a great deal of heat, the range of the difference made by the position of the sun is also large. This is equivalent to all the times apart from the daytime during the summer when the solar altitude is low.

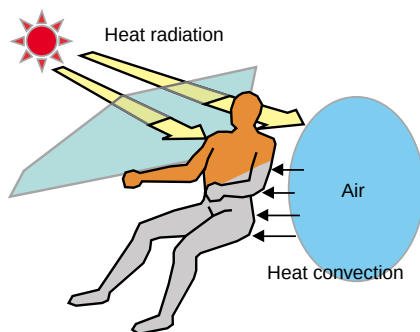


Fig. 2 The routes of solar radiation reaching passengers

3. Current status of and issues concerning solar radiation control

3.1 Current status of solar radiation control

Auto air conditioning systems ordinarily calculate the target air output temperature T_{ao} from the ambient temperature, cabin temperature, setting temperature, amount of sunlight, and so on using formula (1). The air output temperature is adjusted by using the heater unit to change the ratio of cool and hot air. This controls the thermal environment inside the cabin.

$$T_{ao} = F(T_{set}, T_{am}, T_r, T_{sun}) \quad (1)$$

T_{set} : Setting temperature

T_{am} : Temperature from ambient sensor

T_r : Temperature from cabin sensor

T_{sun} : Amount of sunlight

As to the radiation from the sun, the air output temperature is reduced in accordance with the amount of radiation energy detected by the sunbeam sensor so that the cabin area environment is maintained at a constant level.

The sunbeam sensor is installed in the cabin near the driver seat in a location that will tend to remain out of shadows. It is most commonly placed inside the front windshield above the instrument panel and toward the front. The sunbeam sensor gain is determined by the sensitivity of the sensor and the amount of attenuation from the front windshield that comes between the sensor and the sun. The amount of sunlight detected at this location represents the radiation energy from the sun.

3.2 Issues concerning solar radiation control

The amount of solar radiation that reaches the sunbeam sensor varies with the angle of incidence. The passengers, meanwhile, may be in a shadow or they may be bathed in sunlight depending on their positions relative to the windows. Therefore, depending upon the position of the sun, the amount of radiation detected by the sunbeam sensor may not necessarily coincide with the amount of radiation energy contacting the passengers. In order to confirm whether the actual output from the sunbeam sensor can represent the radiation energy for each position of the sun, we studied the relationship between the solar radiation striking the sunbeam sensor and the solar radiation striking the passengers in an ordinary sedan.

Fig. 3 shows the relationship between the solar radiation on the passengers and the vehicle with the sun at an altitude of 50° . The parts visible through the windows are the parts exposed to solar radiation. When the solar radiation strikes from the front, both the passengers and the sunbeam sensor are contacted by it. When the solar radiation is from the side, the sunbeam sensor and the passengers who are on the same side of the car as the sun are contacted by it, while the passengers on the opposite side are in a shadow. When the solar radiation is from the rear, the passengers in the front seat are in a shadow but the sunbeam sensor is contacted by it. This illustrates how the solar radiation contacting the passengers and the sunbeam sensor do not necessarily coincide. In the case of solar radiation from the side, there

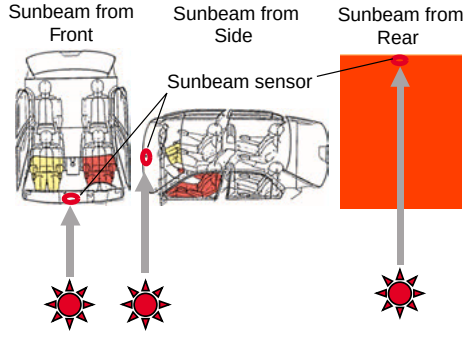


Fig. 3 Comparison of solar radiation on sunbeam sensor and on passengers

will also be a difference between passengers in the right and left seats. Therefore, simple air conditioning control based only on the sunbeam sensor cannot provide comfortable air conditioning to all passengers.

3.3 The effect of uneven solar radiation caused by the sun's position

As described above, the heat balance of the passengers is disrupted when the amount of heat received by their bodies and the amount of solar correction differ, so that passengers may feel hot or cold depending on the direction of the solar radiation. This type of problem must be resolved by making the amount of sunbeam sensor compensation for solar radiation and the amount of radiation energy contacting the passengers coincide.

Fig. 4 shows the position of the sun in relation to the vehicle, as it will be used below.

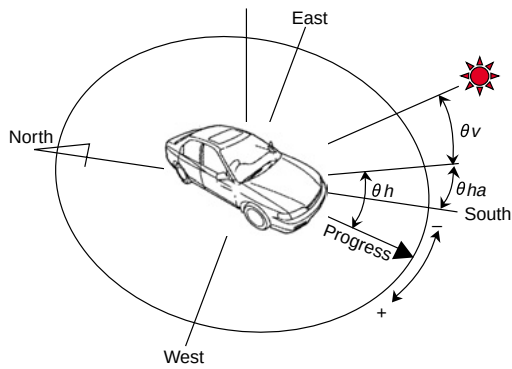


Fig. 4 Description of sun position

θ_h : Relative solar azimuth in relation to the progress of the car

θ_v : Solar altitude

θ_{ha} : Absolute solar azimuth in relation to the South

The actual location where the amount of solar radiation is detected and the locations and conditions of the passengers are different. We verified this quantitatively by comparing the relationships of the sunbeam sensor output and the amount of radiation energy contacting the passengers with respect to the position of the sun. In this way, we verified whether temperature corrections made due to the output of the sunbeam sensor were able to compensate for the amount of heat received by the passengers from solar radiation.

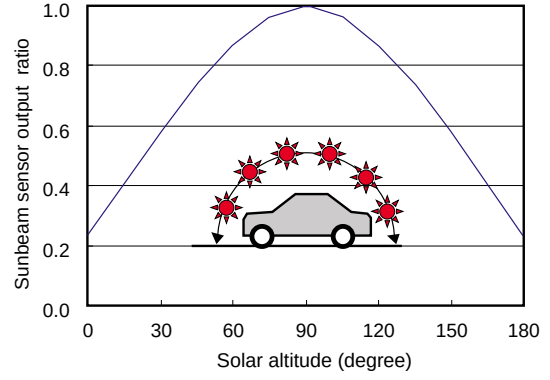


Fig. 5 Sunbeam sensor output ratio

The output of a sunbeam sensor unit varies with the solar altitude, depending upon the directivity of the sensor, as shown in Fig. 5.

The sunbeam sensor unit does not possess directivity with respect to direction, but since the sensor is installed inside the front windshield, the amount of sunlight reaching the sensor changes with the angle of incidence on the glass. The sunbeam sensor output changes that are actually used for control are shown in Fig. 6 with respect to solar altitude and azimuth.

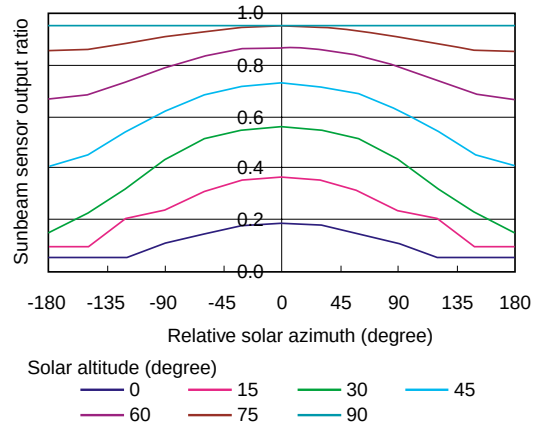


Fig. 6 Sunbeam sensor output ratio in car

The maximum output occurs when the sun is positioned at the front center of the car. The farther the sun's position is to the side of the car, the smaller the angle of incidence on the glass becomes. As a result, the output diminishes and the correction is reduced.

In order to verify the solar correction that is actually required, we determined the relationship between the position of the sun and the radiation energy absorbed by the passengers in the automobile by the routes shown in Fig. 2, using an ordinary sedan as the model.

First, we determined the relationship between the solar altitude and the solar radiation energy absorbed by the cabin air from the sunlight entering through the window glass. This was used as a component that affects the cabin air temperature. The relationship, organized in terms of solar altitude, is shown in Fig. 7.

The radiation energy absorbed by the cabin air varies mainly with the solar altitude. It is clear that the amount of solar radiation energy absorbed increases as the altitude of the sun increases.

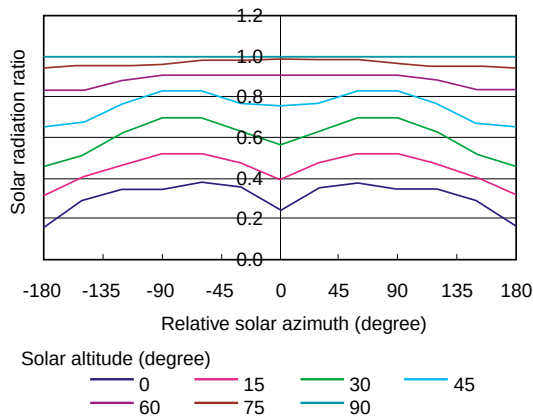


Fig. 7 Solar radiation affecting interior

Next, we determined how much of the radiation energy that passes through the windows and strikes the passengers contacts the passenger in the seat on the left side. The relationship with the position of the sun is as shown in Fig. 8. With high solar altitudes, the solar radiation received is strongest when the sun is on the passenger side. With low solar altitudes, however, the peak in solar radiation occurs instead when the sun is on the opposite side. The difference in radiation energy contact varies greatly particularly with the azimuth when the altitude is low.

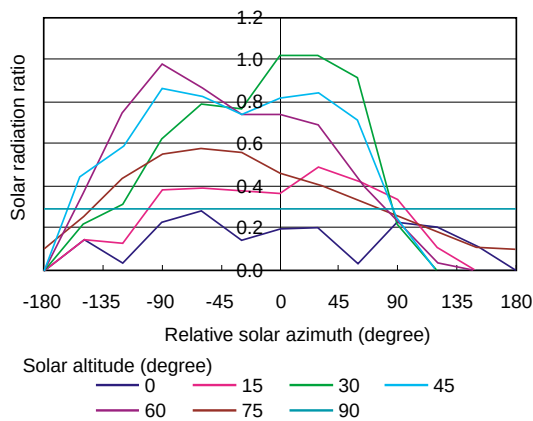


Fig. 8 Direct solar radiation reaching left passenger

The sum of these two amounts of heat is the radiation energy from the sun that contacts the passenger. This was arranged by the azimuth of the sun for the representative altitudes of 30° and 60° and compared with the sunbeam sensor output direction characteristics. It is apparent from the results shown in Fig. 9 and Fig. 10 that the amount of correction made by the sunbeam sensor is insufficient for the entire front area with respect to the actual radiation-energy received at a solar altitude of 30°. At a solar altitude of 60°, the correction is excessive in the rear and on the right, and insufficient on the left. Consequently, it was apparent that there are many areas where the solar correction does not work properly.

3.4 Methods for eliminating the effects of uneven solar radiation

Temperature and airflow rates were sought experimentally to determine how much the outlet air

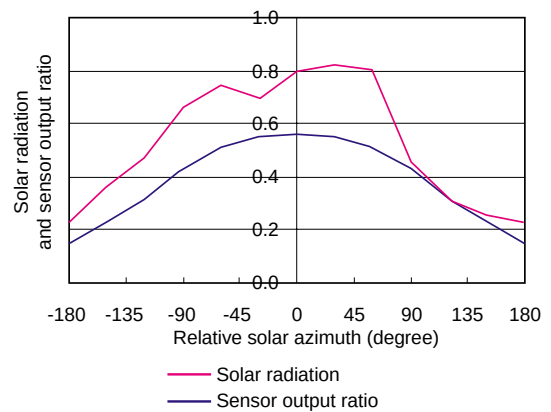


Fig. 9 Output ratio of the solar sensor and the heat ratio of the left passenger (Solar altitude 30°)

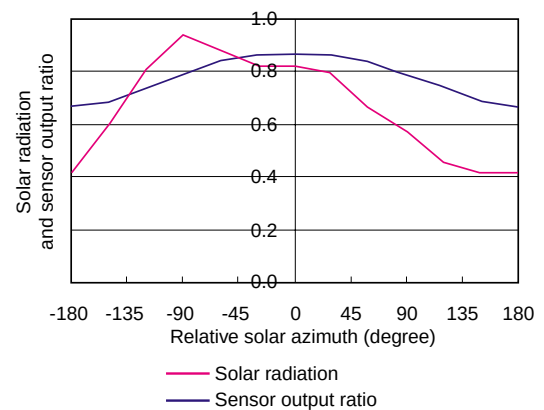


Fig. 10 Output ratio of the solar sensor and the heat ratio of the left passenger (Solar altitude 60°)

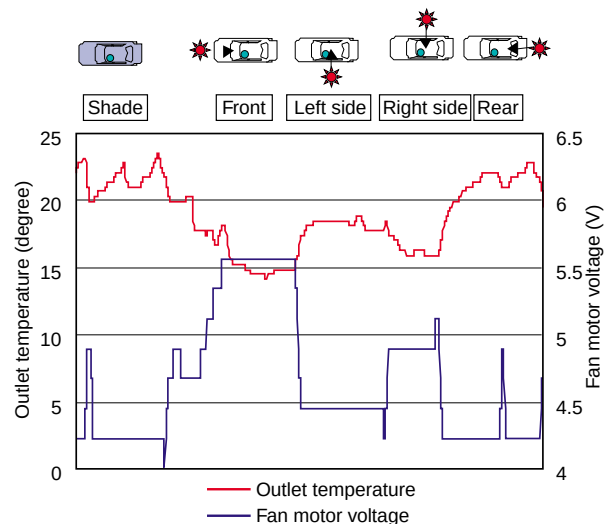


Fig. 11 Outlet temperature producing the same feeling

temperature should be lowered in order that passengers contacted by solar radiation will be at the same cooling level as those who are not.

The outlet air temperature was adjusted in a static vehicle to provide the same comfort to passengers in an actual car under different conditions as shown in Fig. 3. A comparison was made of the adjustments with sunlight from the front (maximum solar radiation condition), from the rear (when

the car body is contacted by solar radiation but the passengers are not), with sunlight from the left and right sides (when there is a difference between the left and the right), and with the car in a shadow (no solar radiation).

As shown in Fig. 11, there are conditions under which adjustment of the airflow rate and outlet air temperature can provide the same passenger comfort when the car is exposed to solar radiation and when it is not.

4. Control with respect to uneven solar radiation

4.1 Basic concept

The amount of radiation energy contacting a passenger changes in a complex manner with the solar altitude and azimuth of the sun. When the amount of correction for solar radiation that is detected by the sunbeam sensor changes with the position of the sun so that it coincides with the amount of radiation energy contacting the passenger, then it is possible to achieve a constant feeling of comfort regardless of the position of the sun.

This requires detection of the sun's position and a map that correlates the sunbeam sensor output with the radiation energy contacting the human body.

4.2 Design of the control algorithm

The formula for calculating the target outlet air temperature T_{ao} utilizes formula (1), with the factor T_{sun} defined as a function of the solar altitude and relative solar azimuth (θ_v and θ_h). The sunbeam sensor directivity is standardized to a solar intensity at a normal surface. This is multiplied by the change in radiation energy contacting the passenger so that the amount of correction equals the change in solar radiation energy contacting the passenger. The correction coefficient for solar radiation, K_{sun} , is derived from formula (2).

$$K_{sun3D} = \frac{F \{F_h(\theta_v, \theta_h), F_r(\theta_v, \theta_h)\}}{F_s(\theta_v, \theta_h)} \quad (2)$$

$F_h(\theta_v, \theta_h)$: Coefficient of radiation energy directly contacting passenger

$F_r(\theta_v, \theta_h)$: Coefficient of radiation energy absorbed by cabin air

$F_s(\theta_v, \theta_h)$: Coefficient of sunbeam sensor directivity

4.3 Detection of solar position

There are various methods for detecting the position of the sun, such as the use of a 3D sunbeam sensor, calculating the position from a GPS signal, and so on. The system used to confirm operation while cruising derived the solar position from GPS signals, which can be obtained relatively simply and with high accuracy.

The latitude and longitude of the current position and the present time were obtained from the GPS signal. The altitude of the sun θ_v and the solar azimuth θ_{ha} in relation to the meridian were then calculated from the geometric relationship of the positions of the earth and the sun.

The GPS signal is further used to derive the direction of vehicle travel. This direction is then used with the absolute solar azimuth θ_{ha} to calculate the azimuth θ_h relative to the direction the vehicle is headed.

5. Results of verification on an actual vehicle

5.1 Configuration of test vehicle

A test vehicle was prepared to verify whether results from control of uneven solar radiation could be obtained using the 3D method on an actual vehicle.

An ordinary sedan was used as the base for the test vehicle. The sunbeam sensor was simply installed horizontally on the roof so that it would not be affected by the front windshield.

The solar altitude and azimuth were calculated using the signal from a GPS antenna input into a computer. The signal was sent to the auto air conditioning system ECU. The heater unit used to adjust the outlet air temperature was modified so that the left and right temperatures could be controlled separately. Outlet air temperature sensors were installed and their outputs were returned as feedback so that outlet temperature control would be accurate. The system was made capable of monitoring the outlet air temperature. The auto air conditioning system target air output temperature T_{ao} from formula (1) was set up separately for the left and right seats. Mapping was established so that the coefficient of correction for solar radiation could be used to control the gain of the sunbeam sensor according to the solar altitude and azimuth. The auto air conditioning system ECU was linked to a personal computer to allow monitoring of all sensor values as well as the solar altitude and azimuth.

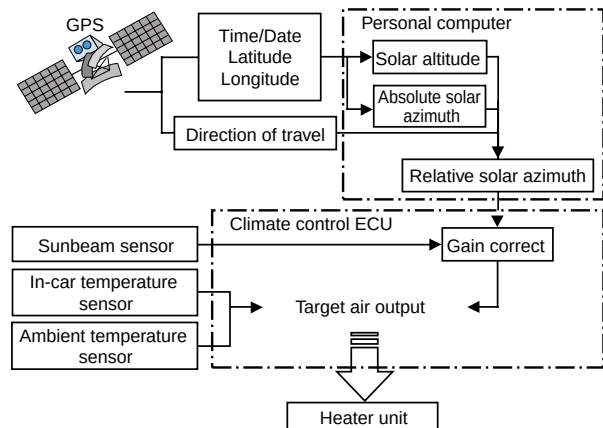


Fig. 12 Control flow chart

5.2 Test method

The test was conducted by a field study. Calculations were performed in advance using the date and time, latitude, and longitude to determine the angle at which solar radiation could be expected to strike at a particular time.

Examples of the relationship between solar altitude, azimuth, and time are shown in Table 1.

Using these data, the vehicle body was oriented to the direction for which verification was desired, the vehicle cruised in that direction, and passenger comfort was verified.

The results were evaluated for each seat according to the ratings given for passenger comfort on the scale shown in Table 2.

5.3 Test results

Fig. 13 shows the results from verification of outlet air temperature control and passenger comfort performed in

Table 1 Test location and solar angle

Time	Solar altitude (°)	Absolute solar azimuth (°)
6:00	4.1	87.1
7:00	15.9	77.7
8:00	27.3	67.2
9:00	37.8	54.4
10:00	46.5	37.8
11:00	52.0	16.4
12:00	52.8	-8.3
13:00	48.7	-31.2
14:00	40.9	-49.4
15:00	31.0	-63.3
16:00	19.8	-74.4
17:00	8.0	-84.0

Longitude	140°E
Latitude	37°N
Date	09/19/00

Table 2 PWM scale

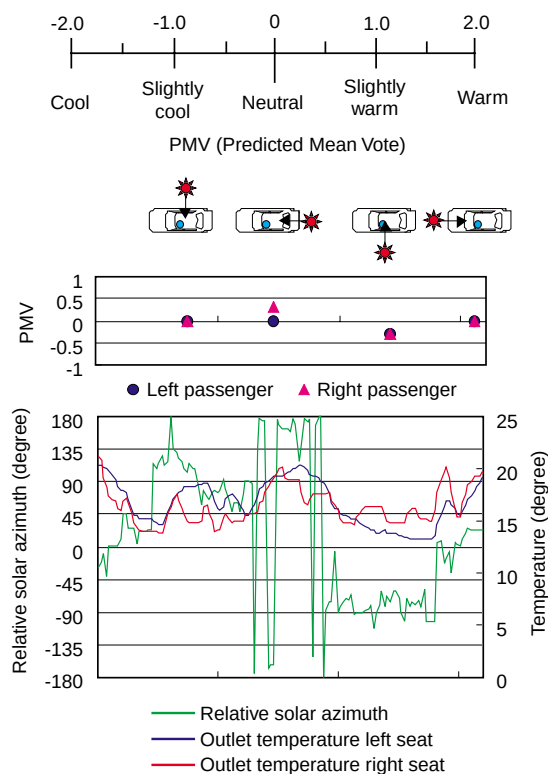


Fig. 13 Outlet temperature and feeling of passengers

each seat of a vehicle for each direction.

In the case of solar radiation entering from the right side, there is solar radiation directly contacting the passenger in the right seat. The outlet air temperature is therefore lowered on the right side, maintaining passenger comfort on both the left and right. Next, in the case of solar radiation from the rear, no solar radiation contacts either of the seats. The outlet air temperature is therefore raised. When the solar radiation comes from the left side, the outlet air temperature at the left seat is lowered to cancel out the heat from the uneven solar radiation on the left side. With solar radiation coming from the front, the passengers receive the most solar radiation. The outlet air temperature is therefore lowered and it is set to the same level on both the left and right.

These results show that matching the amount of radiation

energy with the amount of correction for solar radiation makes it possible to lower the outlet air temperature appropriately even when solar radiation contacts the passenger's body directly. We confirmed, therefore, that it is possible to maintain passenger comfort even when the position of the sun changes and there are changes in the unevenness of solar radiation.

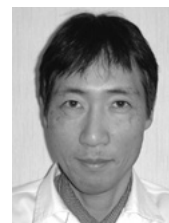
6. Conclusion

- (1) We focused our attention on the heat energy that results from solar radiation contacting passengers in an automobile. We divided this into the solar radiation that reaches the passenger directly through the windows, and that which is absorbed by the car body and reaches the passenger through the air that lies between. We determined both of these for each position of the sun.
- (2) The correction for solar radiation contacting the sunbeam sensor, which depends on sunbeam sensor directivity and the angle of incidence on the glass, was compared with the amount of solar radiation contacting the passengers with the sun at different positions. This demonstrated that the auto air conditioning system does not correct for solar radiation appropriately based on the position of the sun.
- (3) The results from analysis of the solar radiation energy contacting passengers were used to create a map that allows appropriate correction for solar radiation by the auto air conditioning system so as to lower the outlet air temperature according to the heat contacting the passenger.
- (4) A test vehicle was prepared with the capability for changing the outlet air temperature and airflow rate at each seat by means of a map. The results from verification during field tests confirmed that this method could achieve air conditioning control that would be comfortable day and night regardless of the position of the sun with respect to the vehicle.

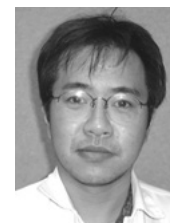
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Application of Finite Element Analysis Using 3D Auto-mesher for Transmission Development

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ABSTRACT

In order to effectively utilize three-dimensional models, the quality of a 3D auto meshing function linked with a 3D CAD system and 3D models was improved. This was achieved with the aim of performing finite element analysis on, without modification of, detail models including detailed fillets, draft angles and other complex shapes which can be applied to die manufacturing. As a result, 3D detail models with a data volume in excess of 100 MB could be converted to CAE models by auto meshing, allowing designers to perform finite element analysis at the transmission design site.

1. Introduction

With the shift to 3D design tools, even cast parts such as the engine and transmission are now designed mainly using detail models including detailed fillets, draft angles and other complex shapes (hereafter “3D detail models”) which can be applied to die manufacturing. Design using these 3D detail models formerly required more process time than 2D drawing design due to the problems of workstation calculation time, insufficient 3D CAD system operation functions, a lack of modeling skills on the part of designers and other reasons. However, workstation performance, 3D CAD system functions, and designers’ skills are improving, and process time problems are gradually decreasing.

The main advantages of 3D design include being able to check interference with other parts using a 3D layout, to check actual shapes through rapid prototyping, and application of 3D detail models to die manufacturing, etc. However, for designers these merits have not offset the increase in process time occurring with the shift from 2D to 3D design.

Therefore, allowing designers themselves to perform finite element analysis directly using 3D detail models was considered as one merit of 3D design for designers.

Based on this background, tetrahedron element auto-mesher functions were improved to allow meshing of a large 3D detail transmission case model which was formerly considered difficult for an auto-mesher. The results showed that finite element analysis can be performed by designers even for large 3D detail models.

2. Aim of 3D Auto-mesher Improvement

For designers to perform finite element analysis, they must learn how to use finite element analysis software, and the 3D model data must be converted from the 3D design tool to that analysis software. The task of learning how to use these many types of software programs is a large burden for designers.

Therefore, if the finite element analysis functions that are a part of Honda’s standard 3D CAD system can be used for analysis, this would have the following merits with respect to the above two requirements. i) The basic operations and user interface are shared, so the finite element analysis functions can be easily learned. ii) Data conversion is not necessary.

For these reasons, using the 3D CAD system equipped with finite element analysis functions that is currently used in-house was thought to be an efficient way to introduce finite element analysis to designers.

For designers to perform finite element analysis on 3D detail models, the auto-mesher function that is a standard function of the 3D CAD system needed to be improved. In addition, changes were also necessary to enable designers to perform finite element analysis in a short time on the same 3D CAD system while performing design work with 3D models. (See Fig. 1.)

2.1 Conventional CAE Models

In the past, performing finite element analysis on 3D detail models required special 3D models for finite element

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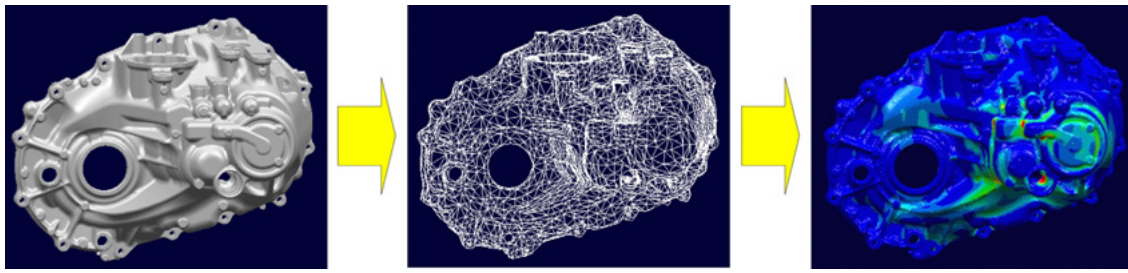


Fig. 1 Finite element analysis of a 3D detail model using the 3D auto-mesher

analysis which omitted detailed fillets, minute surfaces, minute holes and other detailed shapes. This was mainly for the following two reasons: i) It was difficult for the auto-mesher to generate a mesh following detailed shapes. ii) After mesh generation, the number of nodes and elements of the resulting CAE model exceeded the workstation calculation capacity.

In addition, in many cases even with a small number of nodes and elements, hexahedron elements with high accuracy of calculation results had to be used, or meshing had to be performed manually using shell elements in some parts to obtain appropriate results in a short process time. This was mainly done to effectively utilize scarce workstation system resources and to obtain accurate results.

For reference, a full-time CAE staff member sometimes required over 200 hours to create a CAE model of the cylinder block and other engine parts. In addition, decisions such as whether ribs created to disperse stress should be simplified or eliminated required considerable know-how.

2.2 Finite Element Analysis Performed by Designers

For reasons given in the previous section, finite element analysis was generally performed by full-time CAE staff members rather than designers, from the viewpoint of process time. In addition, designers generally followed a process flow in which shapes were determined based on intuition and experience, prototypes were manufactured, and design specifications were determined from actual measurement test results.

However, auto-mesher performance has improved in recent years, and it has become possible to generate meshes using the auto-mesher for palm-sized parts having few small fillets or other minute surfaces and then to perform calculations using this data. It is actually becoming possible for designers to perform finite element analysis for small parts during the design work. There are even some parts for which actual measurement tests are omitted and specifications are determined using only the results of finite element analysis.

The merits of designers performing finite element analysis by themselves include: i) Test results can be simulated at the design site even without the actual part, so the validity of ideas can be confirmed in real time. ii) The surrounding layout, product cost, die configuration and other limitations for that part are easily available, so countermeasures can be formulated in an efficient manner.

iii) Designers can confirm their own design ideas repeatedly in a short time, leading to improvements in their design sense.

The transmission case, which is one of the large parts introduced here, is comprised overall of curved surfaces and fillets; it has almost no flat surfaces, which made auto-meshing difficult thus far. In addition, manual meshing not only took time, but there was also the possibility of analysis being performed which omitted the parts where the maximum stress occurred. Thus, fillet removal and other shape simplifications required considerable know-how. Therefore, finite element analysis was almost never performed for the transmission case due to problems related to analysis time and accuracy. On the other hand, an auto-mesher capable of applying finite element analysis even to large, complex parts such as the transmission case was greatly desired, as this would make it possible to perform accurate finite element analysis in a short time.

2.3 Auto-mesher Direction

Hardware has made remarkable advances in recent years, with speeds increasing and prices falling. Workstations now feature performance on a level formerly only available in mainframe computers.

The conventional approach for creating CAE models was to i) construct individual know-how including the removal of minute surfaces and fillets, ii) perform time-consuming manual meshing, and iii) reduce the numbers of nodes and elements. However, in consideration of these hardware advances, it is clearly more efficient to mesh a 3D detail model with the auto-mesher, create a CAE model and then perform finite element analysis.

In addition, if auto-meshing is possible for 3D detail models, then not only the finite element analysis solver accuracy, but also the 3D detail model shape accuracy and the shape accuracy of the CAE model after meshing can also be taken into account when discussing accuracy of analysis results. Thus, accuracy of results can be pursued from various angles.

Based on the above, the three functions demanded of the auto-mesher are as follows. i) Auto-meshing of 3D detail models should be possible. ii) It should be possible to control the numbers of nodes and elements to optimize the calculation time and overall calculation accuracy. iii) Definition of local detailed element sizes should be possible to improve the accuracy of specific parts.

3. 3D Auto-mesher Improvements

In order for designers to perform finite element analysis during transmission development, the functions required of this 3D auto-mesher are the following two items.

- i) Reliable mesh generation should be possible (= efficient use of time).
- ii) Element quality should be good (= ensuring accurate analysis results).

The specific targets for these requirements when improving 3D auto-mesher functions are the following two items.

With respect to all 3D detail models,

- i) The mesh success rate should be 90% or more.
- ii) The length ratio should be 10 or less.

(The mesh success rate and length ratio are described below.)

3.1 Specific Evaluation Methods and Conditions

In order to achieve these targets, specific methods were determined for evaluating the mesh success rate and length ratio.

3.1.1 Evaluation Methods

- i) Mesh success rate

Using the total number of times meshing was performed as the denominator, the ratio of the number of times a mesh was obtained which satisfied the element quality standards of the finite element analysis solver integrated into the 3D CAD system was defined as the “mesh success rate” as shown in equation (1).

$$MS = VM / MT \quad (1)$$

Here, MS is the mesh success rate, VM is the number of times a calculable mesh was obtained, and MT is the number of times meshing was performed.

- ii) Length ratio

The worst length ratio value was checked to ensure calculation results accuracy.

Here, “length ratio” is the parameter generally referred to as “aspect ratio.” (See Fig. 2.)

In addition to the length ratio, other element quality items such as the Jacobian and shape factor were also checked at the same time to see whether they could be accommodated by the finite element analysis solver.

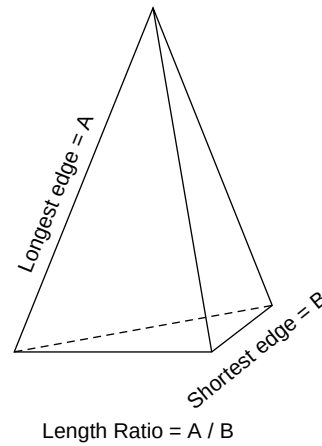


Fig. 2 Length ratio

Of the various element quality standards, the length ratio was used as the standard for the following three reasons. i) When checking the element quality of the generated mesh, the length ratio was the only item that frequently exceeded the finite element analysis solver standard value. ii) The length ratio directly influences the calculation results accuracy. iii) The length ratio is one of the element quality items that is appropriate as an index for ensuring calculation results accuracy.

The calculation fails if even a single element in the mesh exceeds the element quality standard values. It is said that elements with a length ratio of 15 or more should generally be avoided.

3.1.2 Evaluation Conditions

- i) Evaluation model

Five 3D detail models in existence at the time evaluation started were arbitrarily selected as evaluation models.

Fig. 3 shows some of these models. The numbers in the figures indicate the size of that model.

- ii) Meshing conditions

At this point, tests were conducted using a combination of the two meshing parameters of size and absolute sag. (See Fig. 4 and Fig. 5.)

These two parameters were set so that there were approximately 150,000 nodes (approximately 80,000 to 90,000 elements) after meshing the 3D detail model. This



Fig. 3 Evaluation models

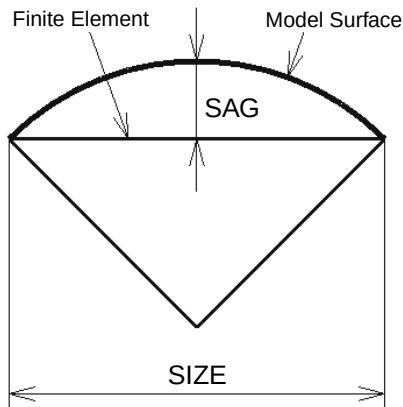


Fig. 4 Size & sag

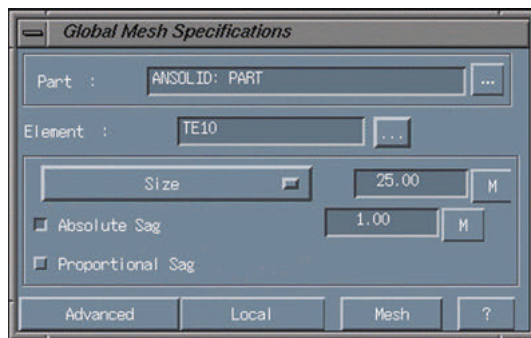


Fig. 5 Meshing parameter settings window

figure of 150,000 nodes is the number of nodes that can be, for the most part, reliably calculated with the 4 GB temporary storage area that can be used by the design workstations currently used for development.

In addition, the same combination of parameters (size and absolute sag) set for the initial evaluation were also used for subsequent evaluations.

3.2 Evaluation Results

The 3D auto-mesher was evaluated in accordance with the above evaluation method and conditions.

3.2.1 Step 1 Test

- Aim: Detailed investigation of functions
- Results: The average mesh success rate for five models was approximately 15%.

This mesh success rate of 15% means that when meshing is performed ten times, the number of times a mesh that can

be used for calculations is obtained is less than two times. (See Table 1.) For designers who must carry out numerous tasks in a limited time, a mesh success rate of 15% makes the meshing work itself a burden.

Therefore, it was judged that this auto-mesher version could not be used for finite element analysis by designers without modification.

• Analysis of meshing results:

Analysis of the meshing failures (hereafter “meshing errors”) which accounted for 85% in the Step 1 evaluation showed that these meshing errors can be classified into two main types: i) Errors rooted in the 3D detail model, and ii) Errors rooted in the meshing program.

i) Errors rooted in the 3D detail model

The parts of 3D detail models that cause meshing errors are the minute portions which are no larger than the standard 3D CAD system tolerance (0.1 mm). Typical examples include the overlapped face and twisted face shown in Fig. 6.

These minute portions are circled in red when they become meshing errors, but the shapes are automatically created by the 3D CAD system, and these portions are extremely difficult to find before meshing starts.

Therefore, the quality of the 3D detail model itself must be improved by enhancing the modeling functions of the 3D CAD system in order to completely avoid errors rooted in the 3D detail model.

In addition to this fundamental solution, errors rooted in 3D detail models can also be avoided by the following two measures.

- Changing the mesher program – The mesher program can be optimized to ignore low quality portions of the model as appropriate.
- Modifying the 3D model – Errors can be avoided by having designers modify portions where problems occur as

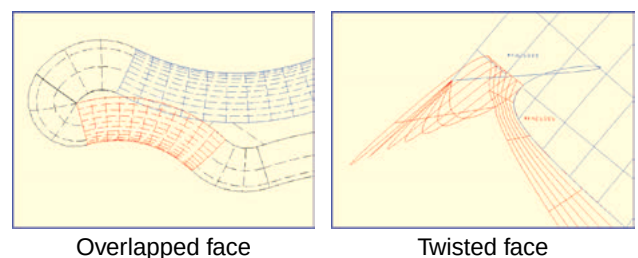


Fig. 6 Problematic geometry examples

Table 1 Step 1 result

Model	A	B	C	D	E	Total
No. of meshing trials	44	42	18	38	12	154
No. of successful meshes	5	2	0	9	7	23
Mesh success rate	11.36%	4.76%	0.00%	23.68%	58.33%	14.94%
Meshing failure details						
3D model quality	28	6	18	17	4	73
Element quality	8	32	0	10	1	51
Others	3	2	0	2	0	7

indicated by the red circles. This method is intertwined with 3D modeling, so it is omitted here.

ii) Errors rooted in the meshing program

Most of these errors are length ratio (element quality) problems. Even when meshing is completed normally, if there are elements with a length ratio value of 50 or more, the data cannot be used for calculations. To solve these length ratio problems, the mesher program must be improved to prevent inferior quality elements from being generated.

In addition, there are also program errors where the mesher program itself causes an error and meshing is completed abnormally.

• Measures for the next version:

Based on the Step 1 results, the following three changes were made to the mesher program.

- The program was changed so that low quality portions on 3D detail models can be ignored.
- Length ratio values were improved.
- The program was debugged.

3.2.2 Step 2 Test

• Aim: To confirm the degree of mesher improvement target achievement.

• Results: A mesh success rate of 88% was achieved, which is a vast improvement. (See Table 2.)

The greatest change in this step was that the occurrence of length ratios of the standard value 50 or more was significantly reduced from 33% to 2% of the total number of meshings.

However, another problem occurred that elements with negative Jacobian values were generated. Negative Jacobian values are an impossible condition for finite element analysis calculations.

• Measures for the next version:

Based on the Step 2 results, the mesher program was changed to prevent elements having negative Jacobian values from being generated.

3.2.3 Step 3 Test

• Aim: To confirm the degree of mesher improvement target achievement.

• Results: No elements with negative Jacobian values were generated, but the mesh success rate dropped from 88% to 86%.

In addition, the meshing error rate due to length ratio also increased from 2% to 5%. (See Table 3.)

3.3 Evaluation Summary

Based on the above results, the improvement in mesher functions by Steps 1 to 3 is as follows. (See Fig. 7.)

• The average mesh success rate for the five models rose from 15% to 86%.

This improvement is broken down as follows:

• The meshing error rate due to a length ratio of 50 or more decreased from 33% to 5% (average value of the five models).

• The meshing error rate due to 3D detail model quality decreased from 47% to 8% (average value of the five models).

Thus, it can be said that the target was for the most part achieved.

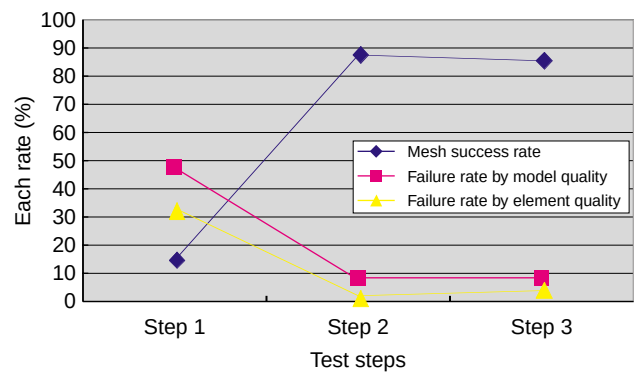


Fig. 7 Transition of results (5 model average)

Table 2 Step 2 result

Model	A	B	C	D	E	Total
No. of meshing trials	44	42	18	38	12	154
No. of successful meshes	37	41	10	36	11	135
Mesh success rate	84.09%	97.62%	55.56%	94.74%	91.67%	87.66%
Meshing failure details						
3D model quality	7	1	3	2	0	13
Element quality	0	0	2	0	1	3
Others	0	0	3	0	0	3

Table 3 Step 3 result

Model	A	B	C	D	E	Total
No. of meshing trials	44	42	18	38	12	154
No. of successful meshes	38	38	11	33	12	132
Mesh success rate	86.36%	90.48%	61.11%	86.84%	100.00%	85.71%
Meshing failure details						
3D model quality	6	0	2	5	0	13
Element quality	0	4	3	0	0	7
Others	0	0	2	0	0	2

4. Introduction of Actual Stress Analysis Examples

This section introduces some actual examples of unit part stress analysis carried out by designers using this auto-mesher.

Fig. 8 shows some actual stress analysis examples. The models used in these analyses are 3D detail models with absolutely no fillet elimination or model modification performed for finite element analysis.

These analyses use parabolic tetrahedron elements, and have approximately 150,000 nodes (approximately 90,000 elements).

It used to take a full-time staff member approximately 80 hours just to perform manual meshing (no fillets, approximately 2000 nodes) for the transmission case. However, boundary conditions can currently be added on a workstation in approximately 10 minutes, and calculations including auto-meshing can be completed in approximately 45 minutes. For reference, the time required for meshing alone is 7 or 8 minutes out of the approximately 45 minutes of calculation time.

Fig. 9 shows a comparison of finite element analysis results for the model shown in Fig. 1 and the results of stress measurement using a strain gauge at each measurement point. These comparison results indicate that finite element analysis is effective for determining design specifications in terms of both trends and absolute values.

5. Conclusion

Based on the above,

- i) Auto-meshing is now possible for 3D detail models; this is something which used to be called impossible.
- ii) Finite element analysis can be performed by designers, and is being utilized at actual design sites.
- iii) The introduction of finite element analysis by designers has reduced the number of design changes and actual measurement tests.

6. Afterword

Finite element analysis was formerly requested of full-time analysis staff, and several days to several weeks were required to receive the results. However, finite element analysis of 3D detail models can now be finished in

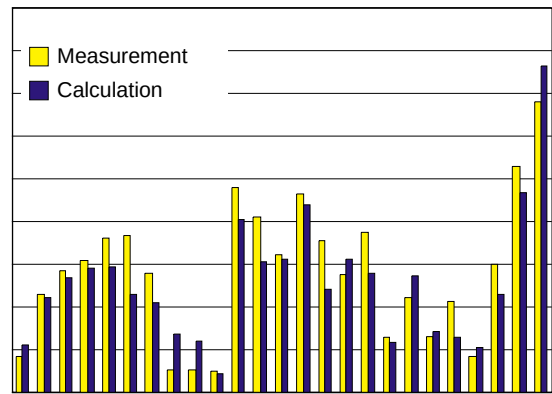


Fig. 9 Comparison of measurement and calculation results

approximately one hour using the improved auto-mesher function. Finite element analysis using the 3D auto-mesher dramatically reduces wasted time on the part of designers waiting for analysis results, and is becoming a useful tool that allows design specifications to be investigated in a short time.

The number of actual measurement tests has decreased and development efficiency is improving steadily by having designers determine specifications with high accuracy using finite element analysis.

7. Acknowledgements

The authors wish to express their heartfelt thanks to Dasseau Systems Inc. (France) for understanding the authors' concepts, working with the authors to set specific target values, and actively supporting improvements in 3D auto-mesher functions so that designers can perform finite element analysis.

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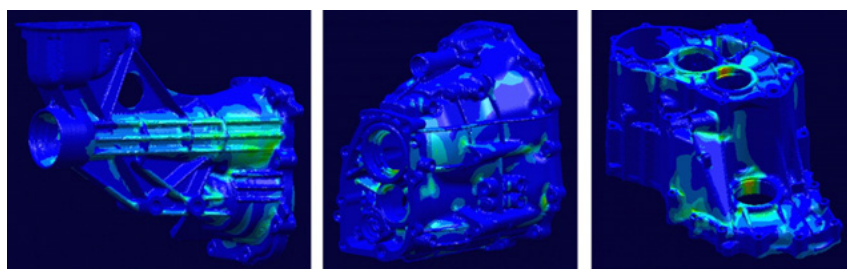


Fig. 8 Finite element analysis examples of 3D detail models

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Development of Metal Pushing V-Belt Stress Simulation for Continuously Variable Transmission

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ABSTRACT

This paper describes the development of a new method to evaluate dynamic behavior and stress for durability simulations of metal pushing CVT V-belts undergoing high rotational speeds. The explicit finite element method typically used in crash analysis is adopted to simulate the high rotational speed range. A finite element model of the complete metal pushing V-belt including the layered bands is constructed where the torque is transmitted by the friction of the contact surface. In the simulation, the rotational speed gradually increases from zero to the target speed. As a result, it is possible to predict the belt stress in the high speed range and as this stress correlated with the test results, belt durability can, for the first time, be judged. To make this simulation more accessible to designers, customized pre- and post-analyzing tools were developed that enable the quick creation of new belt models.

1. Introduction

As demands for environmentally conscious automobiles have grown over the years, various companies have focused on the continuously variable transmission (CVT) as a transmission technology that can simultaneously achieve performance and fuel economy. CVT performance is characterized by quietness, smooth acceleration without shift shocks, and fuel economy superior to that of automatic transmissions achieved by maintaining a rotational speed range with high combustion efficiency through integrated control with the engine. Therefore, as the need for expanded transmission capacity continues to increase for the CVT, it is essential that the durability of the metal belt that forms the framework of the system be increased. Further, the demanded maximum rotational speed is also increasing. However, the transmission mechanism of metal belts is based on contact and friction transmission, so an analytical approach is difficult and vast amounts of development man-hours have been devoted to repeated test confirmation.⁽¹⁾

This paper first describes the development of a simulation method that is capable of predicting metal belt behavior and stress near the maximum rotational speed, and that the strength can be determined from a durability SN curve. Customized pre- and post-processing tools have also been developed so that this technology can be widely used by designers, and these details are also reported.

2. Development Aims

Fig. 1 shows the main section of the CVT used in this development. As shown in this figure, the driving torque is shifted and transmitted by a metal belt from the driving pulley to the driven pulley. In addition, the CVT mechanism shifts the ratio by moving a pair of opposing movable pulleys in the axial direction, so there is some misalignment that cannot be canceled, and the amplitude of this misalignment is

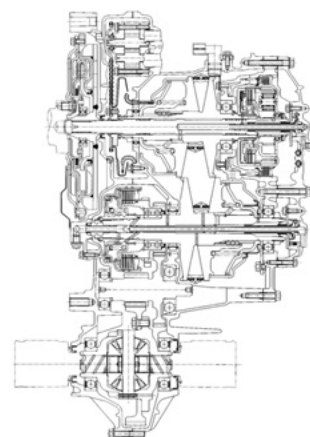


Fig. 1 CVT main section

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determined by the distance between pulley axles and the operating ratio.

The metal belt used was made by Van Doornes Transmission Inc., and was comprised of approximately 428 elements and two rows of 12 rings. (See Fig. 2.) Each ring layer exhibits relative sliding behavior at ratios other than 1.0, and the innermost ring also exhibits relative sliding between elements. The origin of failures at high rotational speeds is presumed to be the inner surface of the innermost rings.



Fig. 2 Metal pushing V-belt

Based on these characteristics, the development aims were set as follows:

- (1) calculating the stress of the innermost ring at the maximum rotational speed;
- (2) creating an SN diagram during high-speed durability testing;
- (3) investigating the influence of pulley rigidity on belt durability; and,
- (4) developing simulation technology that is easy to use.

3. Simulation Development

3.1 Selection of a solver

Metal belts transmit torque by the compressive force between the elements and the tensile force difference of the rings.⁽²⁾ This tensile force difference is produced by the friction force between the rings and the elements mandating that the relative behavior of these components be accurately determined. All metal belt components must be precisely modeled in order to determine the stress of the innermost ring. This modeling method must also take into account the movement and behavior of the individual elements and the ring layers.

The solver employed as the solution method that meets these demands is a general-purpose non-linear finite element method which is classified as a crash analysis code. Most crash analysis codes emphasize calculation efficiency when solving dynamic contact problems. However, a solver with a kinematics contact that does not sacrifice calculation accuracy, even in a kinematics sense, was adopted.

3.2 Simulation method

Fig. 3 shows the metal belt model in the condition prior to analysis. Each ring layer is defined as an elastic body, and the elements are treated as rigid bodies in order to place emphasis on reproducing the detailed shape. The pulley model should normally be treated as an elastic body, but a rigid body was used to reduce the calculation time.

In addition, the perfectly circular shape prior to analysis was used as the zero stress condition for the metal belt.

The analytic process was set up as if actually mounting a metal belt on a CVT and operating the engine. First, the distance between pulley axles is adjusted. Then the belt is clamped by the driving and driven pulley side pressure and the driving pulley is accelerated to the limit rotational speed in a manner so as to prevent belt slipping. Next, all operating conditions are held constant and the calculation process is complete at the point when the ring stress reaches steady state. This method requires a lot of time for analysis, but is considered necessary in order to accurately reproduce the irreversibility of contact and friction.⁽³⁾⁽⁴⁾ (See Fig. 4.)

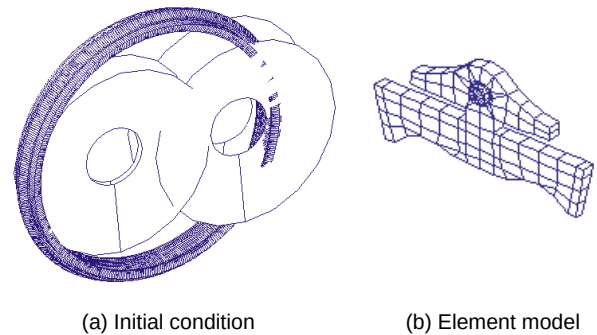


Fig. 3 Metal pushing V-belt model

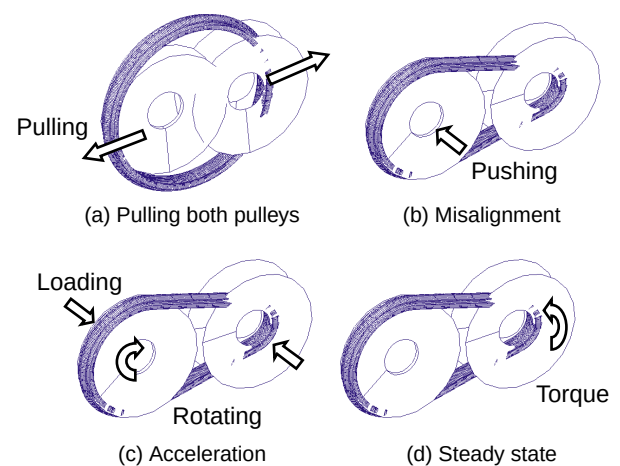


Fig. 4 Analysis procedure

In addition, Fig. 5 shows the relationship between the simulation input conditions and output results. Of these variables, the parameter for the coefficient of friction at each contact surface must be determined through correlation with the measurement results.

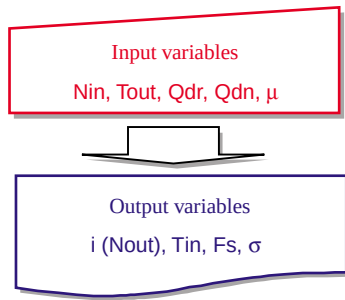


Fig. 5 Input and output variables of analysis

3.3 Verification of accuracy

Fig. 6 shows a comparison of the results obtained by this simulation method and measurement results in order to verify the accuracy. The measurements used a metal belt with only one layer of rings. The figure shows the signal from a strain gauge affixed to the outer circumference of the rings.⁽⁵⁾ The measurement conditions were 1000 rpm and no torque load. These results show that the ring stress is accurately reproduced, and these coefficients of friction at each part were used without modification in simulations at the maximum rotational speed.⁽⁶⁾

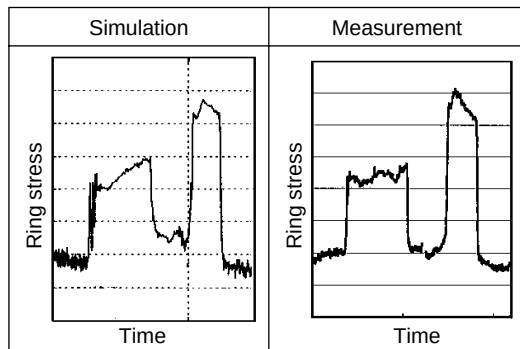


Fig. 6 Comparison of results

3.4 Innermost ring stress

Fig. 7 shows a time history waveform of the innermost ring stress when this simulation method is applied to CVT operating conditions at the maximum rotational speed. The calculation conditions were an engine rotational speed of 6000 rpm, a ratio of 0.61, and transmission torque of 143 Nm.

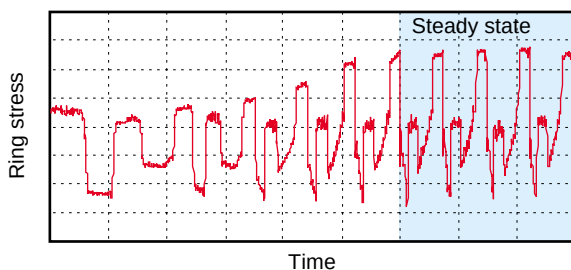


Fig. 7 Time history of ring stress

As shown by this stress waveform, when the operating conditions reach a steady state, the stress waveform also repeats nearly the same waveform, which indicates that a steady state can be reproduced at the maximum rotational speed.

In addition, this simulation method also takes into account belt misalignment. A difference can be seen between the left and right ring stress waveform. (See Fig. 8.) The rings have a crown shape to suppress sliding to the sides between ring layers caused by misalignment, and a stress distribution is produced across the rings when a tensile load is applied. Misalignment was found to further promote this stress distribution and to increase the stress on the edges of the rings.

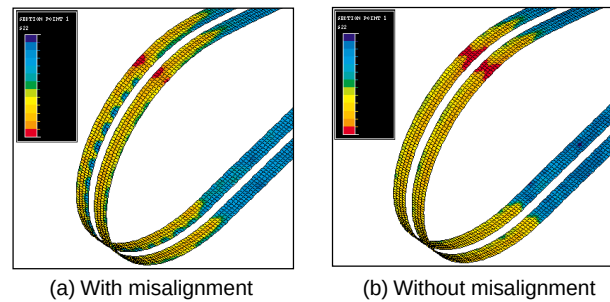


Fig. 8 Influence of belt misalignment

4. Belt Durability Judgement

The CVT operating conditions are prescribed by the ratio, transmission torque, engine rotational speed and both pulley side pressures. That is to say, each of these parameters influences the ring stress waveform, and it is important to operate in a stress state that does not exceed the ring fatigue life.

4.1 SN diagram during high-speed durability testing

Therefore, the ratio was fixed, and two levels were selected for the transmission torque, engine rotational speed and pulley side pressure. Then, the belt life to failure was measured in order to investigate belt durability near the maximum rotational speed. In all cases the cause of failure was assumed to be fatigue failure originating at the inner surface of the innermost ring.

This simulation method was applied to obtain the innermost ring stress corresponding to these operating conditions. Fig. 9 shows the resulting SN diagram during the high-speed durability test. The correlation coefficient of 0.92, indicated a high degree of correlation.

The above results indicated for the first time that belt durability near the maximum rotational speed can be judged according to the stress amplitude on the inside of the innermost ring.

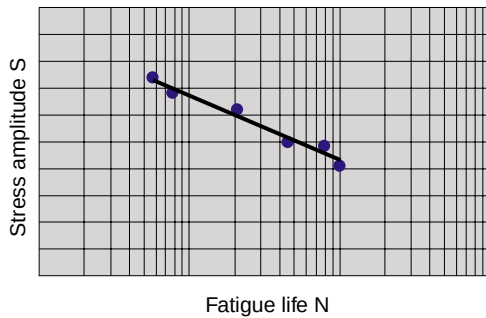


Fig. 9 V-belt SN curve

5. Enhancing Belt Durability

The most effective method for enhancing belt durability is to investigate the influence of the metal belt specifications on the ring stress. However, as previously mentioned the metal belt used was made by Van Doornes Transmission Inc., and it would be difficult to change the specifications. Therefore, the decision was made to examine pulley rigidity, which has a direct influence on belt behavior.

5.1 Influence of pulley rigidity

Fig. 10 shows the analysis model used to investigate the influence of pulley rigidity. The driving side and driven side were both modeled as movable pulleys, with equivalent rigidity. Symmetrical conditions are used to shorten the calculation time, with belt misalignment not taken into account. In other words, only the influence of the pulley deflection on belt stress was investigated.

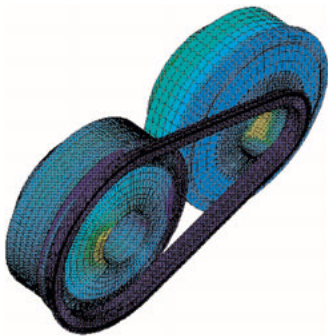


Fig. 10 Flexible pulley model

In addition, Fig. 11 shows the amount of pulley deflection when this simulation method is applied in terms of tip displacement.

Pulley deflection increases toward the tip, being greater for the driving side which turns the belt on the outside. In addition, the belt load changes according to the area where the belt turns around the pulley, resulting in asymmetrical displacement as shown in Fig. 11.

Furthermore, pulley width is widened in the area where the belt turns, and narrowed in the area where the belt does not turn around the pulley.

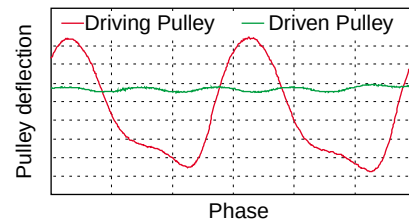


Fig. 11 Pulley deflection

As a result, the load acting on the V-surface of the element increases at the pulley entrance and exit as shown in Fig. 12.⁽⁷⁾ From the perspective of the metal belt, this indicates difficulty in both entering and leaving the pulley, which is hard on the elements.

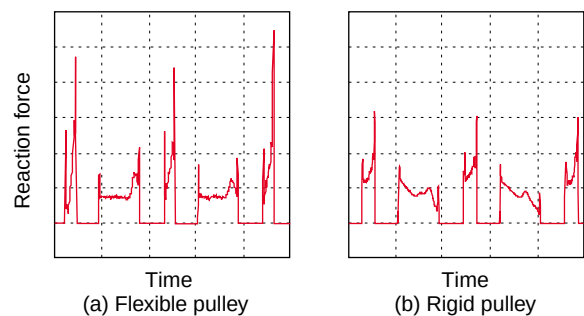


Fig. 12 Reaction force on V-surface

Fig. 13 shows the stress waveform for the innermost layer ring at this time. The influence of pulley deflection on the ring stress can be clearly seen in the turning area of the driving pulley. The stress amplitude also becomes slightly larger as the pulley rigidity decreases.

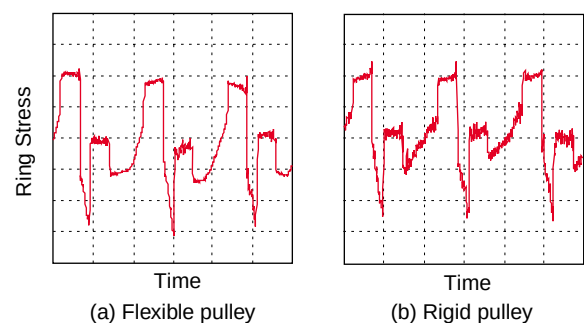


Fig. 13 Influence of pulley flexibility

This is thought to be because the pulley deflection changes the ratio slightly from the top ratio to the OD ratio side, causing the ring bending stress to increase.

5.2 Pulley rigidity balance

As seen from the CVT section, the metal belt is clasped from the right and left by a pair of pulleys having different rigidities.

Normally the amount of pulley deflection on the pulley shaft side is less than that on the movable pulley side, resulting in belt misalignment changes from the dotted line to the solid line as shown in Fig. 14.

Misalignment increases the tip stress across the width of the ring. In order to improve belt durability the design should ensure that the right and left pulley deflections are balanced.

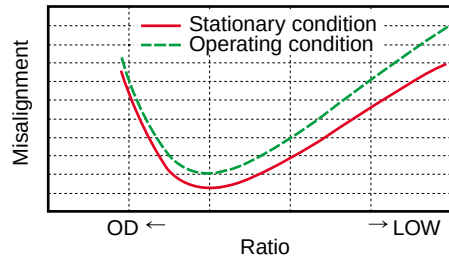


Fig. 14 Influence caused by belt misalignment

6. Metal Belt Analysis System

A dimension input type analysis system, such as that shown in Fig. 15, is an effective means for enabling widespread use by designers of the simulation method developed thus far. This system focuses on the fact that the metal belt element shape topology does not change significantly, and it alters the basic element shape according to the values input by the designer. Therefore, the boundary conditions and contact surface definitions needed for analysis can be automatically set within the system, allowing a metal belt analysis model to be constructed in a short period of time.

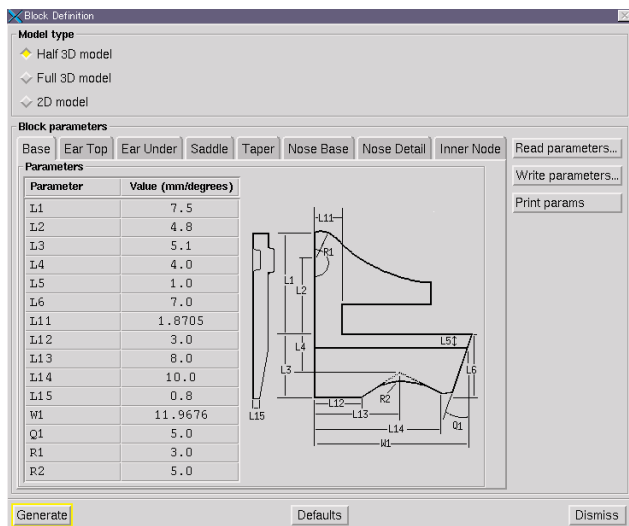


Fig. 15 CVT belt system

7. Conclusion

Simulation technology that determines the stress of the innermost ring at high rotational speeds was developed in order to predict metal belt durability. As a result, the following information was obtained.

- (1) Belt misalignment generates a stress distribution that is skewed in the ring width direction, thus increasing the stress at the ring edges.
- (2) Using the stress amplitude on the inside of the innermost ring as the vertical axis of a SN diagram during high-speed durability tests revealed a correlation with durable life.
- (3) The V-surface load acting on the elements due to pulley rigidity-induced deflection was found to increase markedly at the pulley entrance and exit. Furthermore, increased misalignment due to the left and right pulley rigidity balance must be taken into account in order to enhance belt durability.
- (4) A dimension input type analysis system was established to allow the implementation of the developed simulation technology by designers. This system makes it possible to construct metal belt models in a short time.

8. Acknowledgment

As the need for expanded CVT transmission capacity continues to increase in the future, this technology for judging metal belt durability occupies an important position in determining the basic CVT framework. In the future, the authors intend to further refine accuracy by identifying coefficients of friction, and press forward with the construction of simulation technologies from a designer's viewpoint.

In closing, the authors would like to express their deepest appreciation to related parties at Hibbitt, Karlsson & Sorensen, Inc. for their enormous cooperation over many years in developing solvers and reducing calculation times.

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Development of a New Software for Objective Evaluation of Interior Sound of Sports Cars

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ABSTRACT

An evaluation method was developed for the interior sound of sports cars. This method was derived from the precise subjective evaluations of 12 sports cars and consists of objective parameters. Further, it was found that there is only a slight difference in perception of sporty sound between Austrians and Japanese. As a result of this research, software for use in development of sound quality was developed and will be useful in the development of sports cars with optimum interior sound.

1. Introduction

The character of interior sound is an important element in enhancing the commercial attractiveness of a sports car. Sports cars have high-quality exteriors, powerful engines, and highly responsive suspensions. Furthermore, the engine sound heard in a cabin during acceleration or during a down-shift conveys information to the driver about the vehicle's response. Therefore, the engine sound during acceleration, for example, must communicate vehicle performance and overall vehicle quality. Given this situation, it is inefficient to attempt development of a sporty sound in the absence of concrete numerical criteria. An enormous number of man-hours have been expended on trial-and-error development efforts involving the fabrication of many prototypes and creation of subjective evaluative criteria for the character of interior sound. The use of objective parameters to express the character of interior sound is essential in the logical development of a sporty sound. If this were possible, it would subsequently be possible to determine target interior sound candidates on a sporty sound map during the development of a new vehicle after considering relative positioning with competitor vehicles and our concept for the new vehicle. As a result, it would be possible to determine the target sound after listening to the target sound candidates created according to their positioning on the map. The adoption of this kind of sound character index for interior sound would not only allow determination of a target interior sound, but it would also enable significant reductions in the man-hours

expended in new-car development overall. Very little research, however, makes reference to the character of interior sound of sports cars.^{(1)–(3)}

This paper reports on a method for creating an index that uses objective parameters to express the interior sound character of a sports car, and presents examples of the practical application of that method. It will also describe the difference in how sports car interior sound character is perceived by Austrians and Japanese. Further, as we developed software to enhance the usefulness of these research results, we will provide an example of its practical application.

2. Method for the subjective evaluation of sports car interior sound character

Fig. 1 shows the flow used to approach the creation of an interior sound character index. The left side of this flow chart represents subjective evaluation, while the right side shows objective analysis.⁽⁴⁾⁽⁵⁾ A comparative analysis of the subjective evaluation scores and the objective parameters for each vehicle is then performed using regression analysis and other methods to generate the sporty sound index. This is a standard method for representing sensory quantities as parameters that can be measured using instruments.

The first thing that must be considered when conducting subjective evaluation is the proper selection of test subjects who match the items being evaluated. Statistical methods were used to select the test subjects in order to obtain highly

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** AVL List GmbH

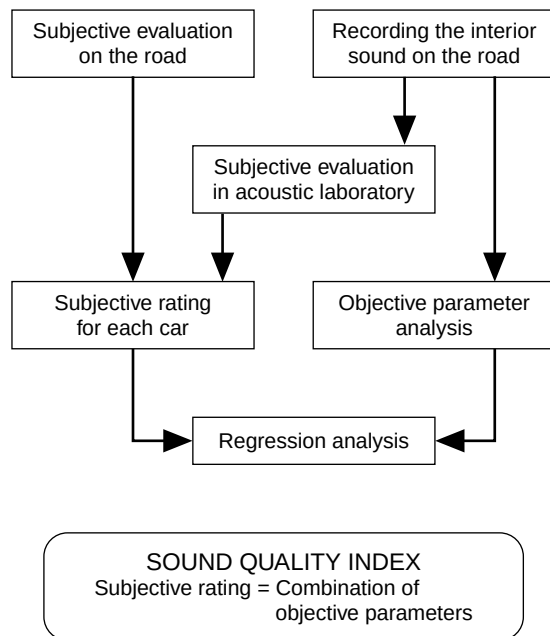


Fig. 1 Overview of approach to create a sound index

accurate evaluation scores that would be usable for comparison with objective quantities. Austrian test subject candidates were presented with questions to identify which of 23 vehicle-related items, such as interior and exterior, specifications, cost, performance, and so on, they considered important. The results were used to select test subjects who were considered capable of evaluating sports car sound. As shown in Fig. 2, the results can be divided into five main clusters⁽⁶⁾⁽⁷⁾ according to their attributes. The test subjects classified under clusters 2 and 3 place importance on the interior sound in sports cars and high-end vehicles (exclusive cars). Therefore, 31 people were selected from these clusters as the final test subjects.

The subjective evaluations were carried out through two tests, an on-road evaluation conducted using actual cars on the road, and an auditory evaluation conducted in an acoustic laboratory by playing back sounds recorded on the road. These tests used the sounds from twelve well-known types of sports cars with different engine types and displacements, as shown in Table 1.

Table 1 List of sports cars tested

No.	Engine type	Displacement [liter]
1	L4	2.0
2	L4	2.0
3	L4 + SC	2.3
4	L6	2.8
5	L4	2.0
6	L4	1.9
7	F6	3.4
8	V6	2.0
9	F6	2.5
10	V8	3.5
11	L4	2.0
12	V8	3.6

3. Subjective on-road evaluations

3.1 Method

For the on-road evaluations, one expert driver operated the vehicles while the test subjects sat in the passenger seat and concentrated on evaluating the interior sound. The tests were conducted on a level road with a dry asphalt surface, and the driving conditions were as follows:

2nd gear Wide open throttle acceleration
 3rd gear Wide open throttle acceleration
 Top gear Constant 150 km/h

No. of People		66	54	68	39	42	total 269 male
		Cluster of driver stereotypes					
		1	2	3	4	5	
Factor Analysis of vehicle main attributes	1 Prestige/Sound						
	2 Performance						
	3 Convenience						
	4 Family						
	5 Noise Quality						
	6 Cost						
Important Attributes	Performance						
	Family						
	Cost						
Unimportant Attributes	Prestige						
Driver Stereotypes	Family Man						

high importance

low importance

Fig. 2 Driver stereotypes

The question posed in the on-road tests was, “How sporty do you consider this interior sound to be for this class of sports car?” The determination was made by assigning a score ranging from 1 point (not at all) to 20 points (completely sporty in character).

3.2 Results

Fig. 3 presents the average scores from 22 test subjects under the three types of driving conditions. The scores for each car diverge widely at wide open throttle (WOT) and the ranking of each car is distinctly defined, indicating that the test subjects were able to differentiate each car from the others. By contrast, the scores from the 150-kph constant speed test were low and they were closer together. This shows that it is more difficult for the test subjects to evaluate sportiness at a constant 150 kph than at WOT.

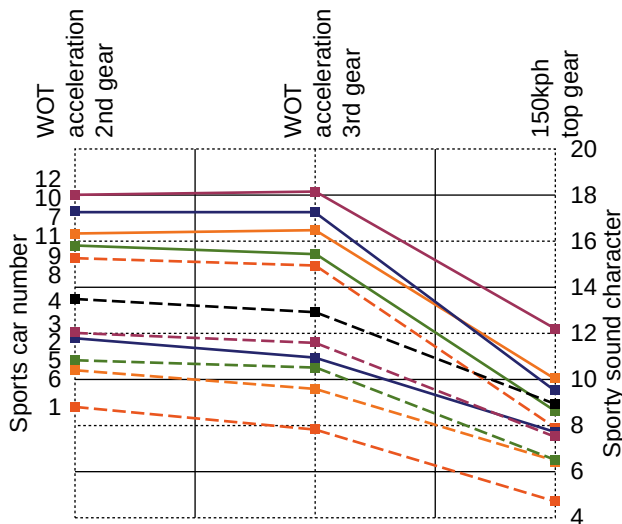


Fig. 3 Results of subjective interior road evaluations of sports cars

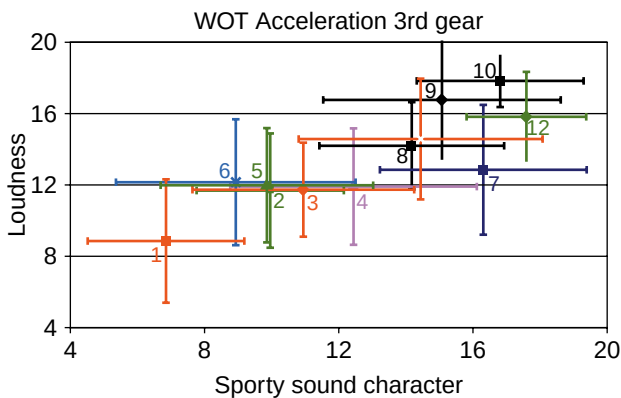


Fig. 4 Results of a subjective interior road evaluation of sports cars at WOT 3rd gear (average judgement of 22 subjects, plus spread of judgements for each car)

Tests were also conducted using multiple types of sensory measurements, with test subjects evaluating loudness and sportiness simultaneously. The results plotted on a two-dimensional map along an axis of score values are presented in Fig. 4. Subjective evaluations were conducted by 22 test subjects under the three driving conditions described above. The results obtained for WOT in third gear are shown as typical. Fig. 4 shows the average scores of all test subjects for each vehicle together with their scatter. Examination of the arrangement of average scores shows that loudness and sportiness appear to be positively correlated. However, there is a very large variance in the scores, exceeding the difference between the average scores, so the results are lacking in accuracy. Thus, the relationship between sportiness and loudness will be further examined, in a later section, by auditory tests conducted in an acoustic laboratory using sample sounds and original sounds tuned to the same sound level.

4. Subjective evaluations by auditory tests

4.1 Method

Recordings were made of interior sounds under the same conditions as for the subjective evaluations on the road. A dummy head microphone was placed in the passenger seat and the same expert driver as in the on-road evaluations operated the test vehicle while the interior sound was recorded. The HMS III made by HEAD Acoustics GmbH, which is capable of digital recording, was used as the dummy head recording system.

The sounds recorded in this way were used to conduct sound evaluations in an acoustic laboratory as shown in Fig. 5. The playback signals are divided into two parts, the low-frequency components of 100 Hz or lower, and the high-frequency components of 100 Hz or higher. The high-frequency sounds are played back through headphones. The low-frequency sounds are played back through a subwoofer so that the test subject can feel the sound through the body while hearing through the ears. The subjective evaluations were carried out using subjective evaluation software made by AVL List GmbH, and the scoring was done using a finer scale (from 1 to 50 points) than in the on-road tests.

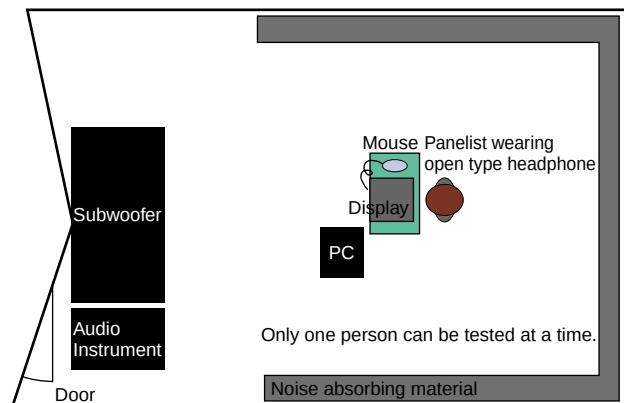


Fig. 5 Layout of psychoacoustic laboratory

4.2 Results

The results from tests on the correlation between loudness and sportiness described in section 3.2 are presented in Fig. 6. This compares the scores for sportiness from auditory tests using samples of the original sound and sounds adjusted to be the same loudness. The data for each car are arranged from left to right in ascending order of engine speed. The original sound from sports car 2 and the sample of the same loudness produced subjective evaluation scores that are completely opposite. This shows that there are cases when sportiness and loudness are not positively correlated. It became clear, therefore, that volume and other parameters of loudness alone cannot represent sportiness.

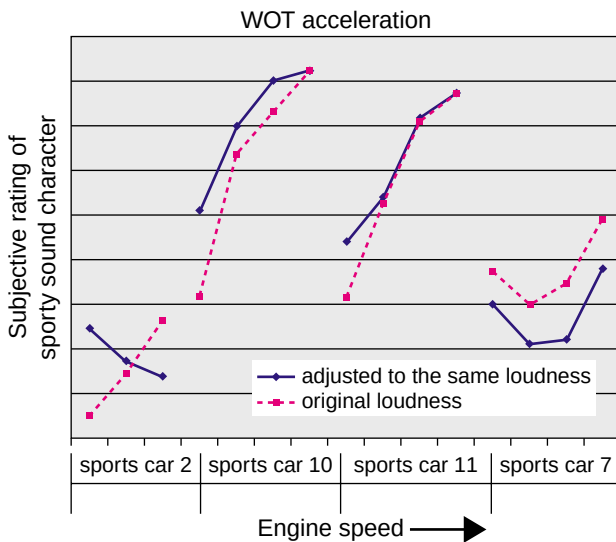


Fig. 6 Comparison of subjective rating between original sounds and sounds adjusted to be the same loudness

5. Comparison of subjective evaluations in the acoustic laboratory and on the road

In order to demonstrate that the sounds played back in the acoustic laboratory are equivalent to the impressions of interior sounds heard on the road, it is necessary to compare the auditory test scores with the on-road scores.

Fig. 7 presents the results from a comparison of the scores from auditory testing in the laboratory and from on-road evaluations. The $\triangle$ symbols indicate comparisons of raw, unadjusted scores. Here the correlation factor is 0.82, which is not very high. This is because the test subjects made their judgments while experiencing the brand image, visual sensations, the stimulation of vibration and acceleration, and so on as well as the sound when they were on the road. Consequently, a comparison was made of the auditory testing and on-road results with consideration of acceleration performance as well as sound.

When regression analysis of the auditory test scores together with the weight/power ratio for each car was

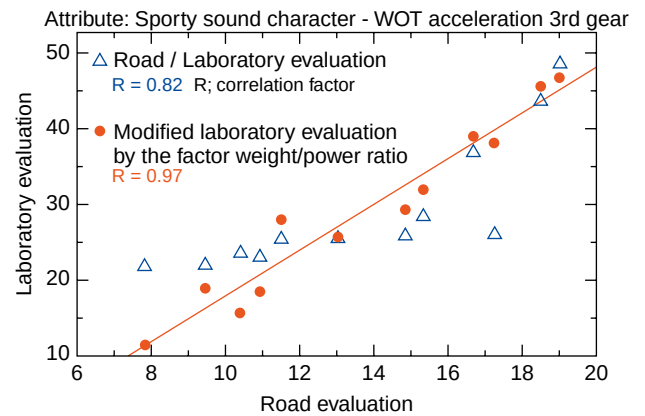


Fig. 7 Comparison of subjective laboratory and road evaluation results for 12 sports cars

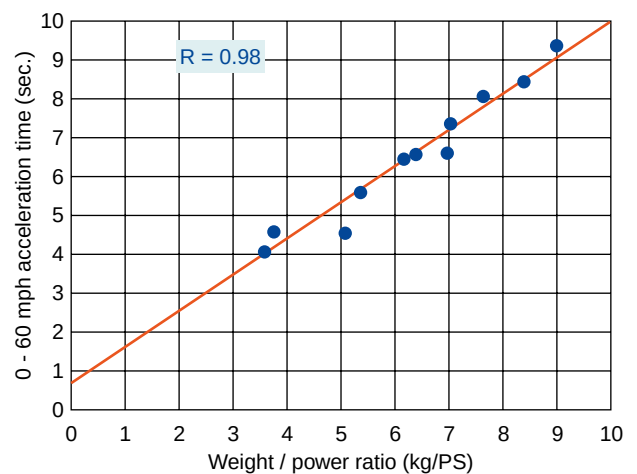


Fig. 8 Comparison between weight/power ratio and 0-60 mile per hour acceleration time

performed, the correlation factor rose to 0.97, as shown by the $\bullet$ symbols. This is an excellent correlation. As seen in Fig. 8, the weight/power ratio is a quantity that is very closely linked to acceleration time. It is apparent, therefore, that the score for sporty sound on the road is formed from evaluation of the interior sound alone and with the evaluation of acceleration. In other words, the auditory testing scores represent the scoring results from on-road evaluation of the interior sound only.

6. Objective representation of interior sound character

Regression analysis of subjective evaluation scores and psychoacoustic measures and other objective parameters was performed in order to determine sportiness, the level of sporty interior sound.⁽⁴⁾⁽⁷⁾ This revealed the involvement of the so-called mid-frequency range. However, it was difficult to obtain a correlation with existing objective parameters sufficient to use this as an index.

As shown in Fig. 9, therefore, we introduced an order component of engine speed to serve as an element of sound

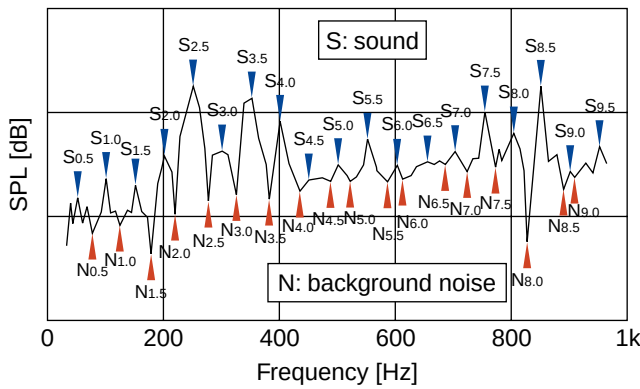


Fig. 9 Definition of sound and noise

character that positively affects sportiness, and background noise other than the order component to serve as an element of sound character that negatively affects the sound character. We then defined the order to noise ratio (ONR) as a new parameter that follows objective procedures to represent the sportiness perceived subjectively using these elements. The ONR is a quantity expressing the extent to which the sound exceeds the background noise. Our research then focused on determining the optimum order domain and weighting.

As a final result we obtained the below expression, which is an objective representation of sportiness:

$$\text{Sportiness} = A \cdot (\text{ONR}) + B \cdot (\text{TP}) + C \cdot (\text{ESP}) - D$$

A, B, C, D: Constants

TP: Tonality

ESP: Engine speed increase rate

Fig. 10 shows the correlation between sportiness on the basis of objective parameters and the subjective scores for each vehicle from the acoustic laboratory. The correlation factor is 0.97, which indicates that there is an extremely favorable correlation.

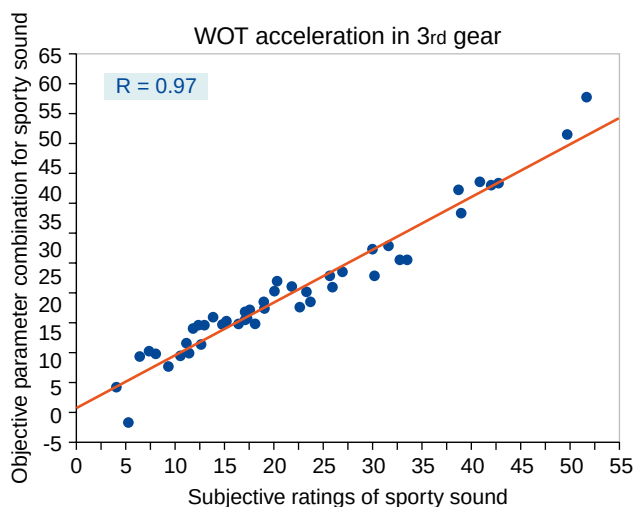


Fig. 10 Subjective rating of sporty sound character compared to objective evaluation results

7. Differences between Austrian and Japanese perceptions with respect to sports car interior sound

The discussion up to this point has dealt with the perceptions of Austrians with respect to interior sound. Japanese subjects were then tested using the same procedures as for Austrians in order to investigate whether there is any difference in the Austrian and Japanese perception of sporty sound. The 32 Japanese test subjects who were selected, however, are people who already own sports cars.

As can be seen in Fig. 11, the scores given each vehicle for interior sporty sound by the Japanese and Austrian subjects showed a correlation factor of 0.98, which is an extremely high correlation. This indicates that the two groups perceive and evaluate sporty sound in a similar manner. A previous study has shown that Europeans and Americans are more tolerant of low-frequency sounds than Japanese, and less tolerant of high-frequency sounds. On first glance, our results may appear to contradict that study.⁽⁴⁾ However, the results of tests with Austrians described in section 6 determined that the frequency range showing a high correlation with sporty sound was the mid-frequency range. Thus, it can be concluded that subject impressions of the mid-frequency range supercede those of the low-frequency range in determining sporty sound, which was why no major differences appeared in the scores from these two groups of subjects.

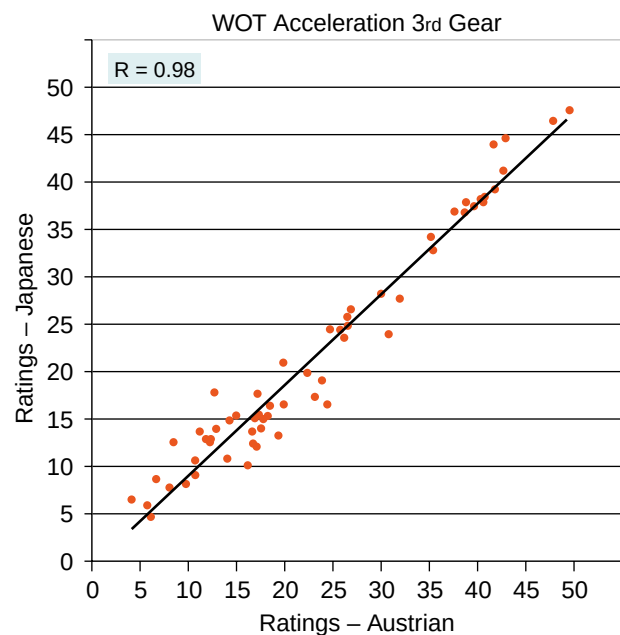


Fig. 11 Comparison of subjective ratings between Austrian and Japanese

8. Sporty sound quality software explanation and application example

As discussed in section 6, it has become possible to use objective parameters, or in other words linear combinations of measurable quantities, to represent the sportiness that is one of the sensory attributes of sports car interior sound character. Consequently, we have developed Sporty Sound Quality Software (SSQS) as a tool that can be applied in sports car development. SSQS not only functions to calculate sportiness, but also provides both database functions and simulation of sportiness using the physical quantities of interior sound. Its basic structure is shown in Fig. 12.

This software was used to improve the interior sound in a certain sports car with a 4-cylinder, 2-liter engine as is shown in Fig. 13. The mid-frequency range of the interior sound contains a component order of engine speed. The vehicle's sportiness increases as this sound becomes larger relative to the background noise. Simply increasing the component order, however, will also end up increasing the loudness itself. Therefore we reduced the low-frequency

component, which has a weaker correlation with sportiness. As a result, not only did this reduce loudness, but we were also able to enhance sportiness.

9. Conclusion

This research yielded the following results regarding the interior sound character of sports cars:

- (1) A method for highly accurate subjective evaluation consisting solely of interior sound alone was developed. The ability to quantify the effect of acceleration on the results of subjective evaluation on the road was also established.
- (2) An expression of sportiness that uses objective parameters to express the character of interior sound in terms of the results from subjective evaluation of interior sportiness was established.
- (3) It was determined that there is almost no difference between Austrians and Japanese in the perception of sportiness.
- (4) SSQS software was developed to enable evaluation and simulation of sportiness as described above for use in automobile development.

We hope that this method proves useful in the creation of future products that will give sports car users greater enjoyment of the sense of sportiness.

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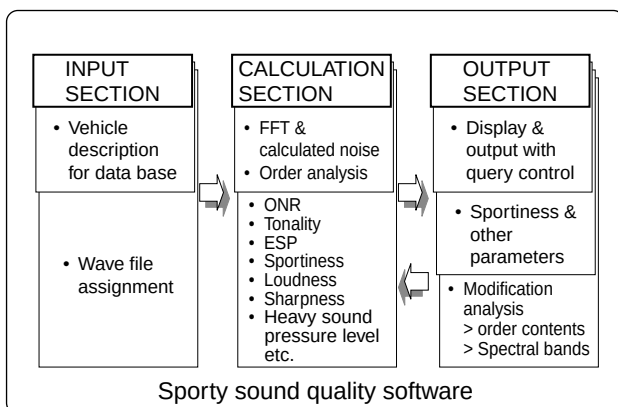
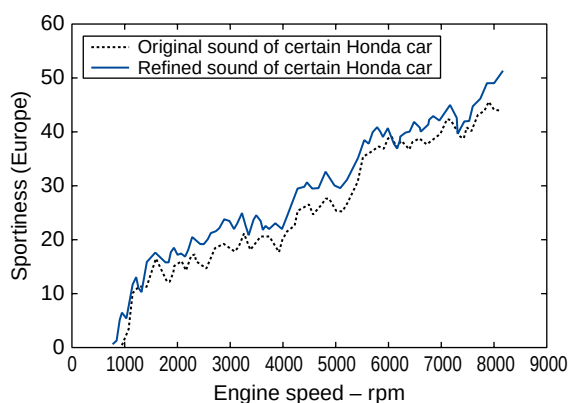
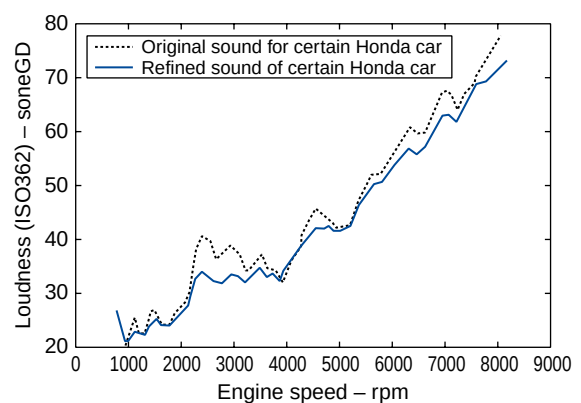


Fig. 12 Schematic flowchart of SSQS



(a) Sportiness



(b) Loudness

Fig. 13 Refinement of sportiness

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A Method for Measuring the Excitation Forces Inside Engines

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ABSTRACT

This paper presents a method for directly measuring the excitation forces exerted on the main bearing journal from the crankshaft. A method was developed for obtaining the amplitude and direction of the constantly changing excitation forces, which had not previously been accomplished. The excitation force direction and absolute value are obtained when the relative error between the strain during actual operation obtained from multiple strain gauges affixed to the inner surface of the main bearing journal and the strain estimated from calibration curves is at a minimum.

1. Introduction

The excitation forces acting in a powertrain, comprising the engine and the transmission, and particularly those excitation forces in the engine exerted by the crankshaft on the main bearing journals during operation, have an important effect on crankshaft rumble, and are therefore critical parameters.

There has previously been no method for directly measuring the magnitude and direction of the excitation forces. Although there has been considerable research leading to various proposals that have focused on indirect measurement methods such as determining the magnitude or direction from vibration measurements or identification using transmission coefficients, direct measurement is still desirable, particularly from the point of view of accuracy.

When measuring directly, it has been conventional wisdom to assume that the direction of the excitation force is fixed, and the value calculated from the calibration curve for a bolt or strain gauge is used as the excitation force. However, in actual running engine, the magnitude and direction of an excitation force is constantly changing. A method for directly measuring the magnitude and direction of an excitation force has now been developed.

2. Method for Measuring Excitation Forces

2.1 Outline of the Measurement Method

The target of this measurement method is to determine the excitation forces acting on the main bearing journals that are generated by the crankshaft during engine operation. The excitation forces are determined by obtaining values through first calibrating the relationship between the output of four strain gauges located on an inner surface of each main

bearing journal and the load at each crankshaft angle. Then the excitation forces are determined using the calibrated values and the strain gauge output that is obtained during engine combustion.

2.2 Test Engine

A Civic 4-cylinder engine was used. The major specifications for this engine are presented in Table 1.

2.3 Measurement Procedure

The following procedure was implemented for measuring the excitation forces:

(1) Calibration Test

Strain measurements were taken with a constant radial load applied to the main bearing journals.

(2) Calibration Curve Calculation

A calibration curve was created for each strain gauge, the horizontal axis being the calibration load on the main bearing journal, and the vertical axis being the calibration sensitivity vector.

(3) Strain Measurements during Engine Operation

Simultaneous measurements of the strain on each main

Table 1 Test engine specifications

Type	Water-cooled, in-line four-cylinder
Valvetrain	SOHC 16 valves
Displacement	1493 cc
Bore/Stroke	75.0 mm / 84.5 mm
Compression ratio	9.4 : 1
Maximum power	105 PS / 6400 rpm
Maximum torque	13.6 kgm / 4500 rpm

* Tochigi R&D Center

bearing journal were taken while the engine was operating.

(4) Excitation Force Calculation

The direction and absolute value of each excitation force was calculated from the relevant calibration curves.

2.4 Calibration Test Method

The test engine block was mounted with a jig on a tensile/compression tester as shown in Figs. 1 and 2, and a static load was applied to the main bearing journals via a load cell and calibration tool.

Strain gauges were attached, as shown in Fig. 3, at four positions around each main bearing journal 45° from the journal mating surface in consideration of the load frequency, direction change, sensitivity, etc.

Each main bearing metal was machined as shown in Fig. 4 to provide pockets for mounting the strain gauges.

The effect of heat on the measurement was eliminated by using a strain gauge with temperature characteristics appropriate for use with aluminum, and by using the 1-gauge, 3-wire connection method.

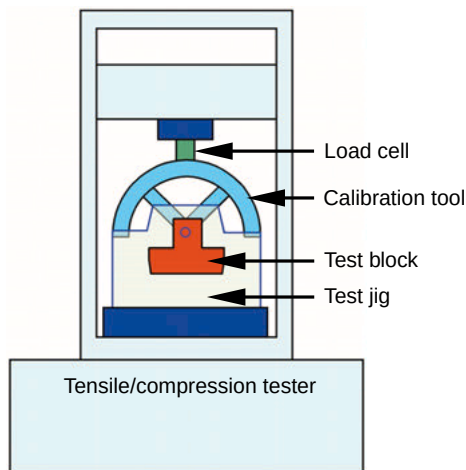


Fig. 1 Calibration test device

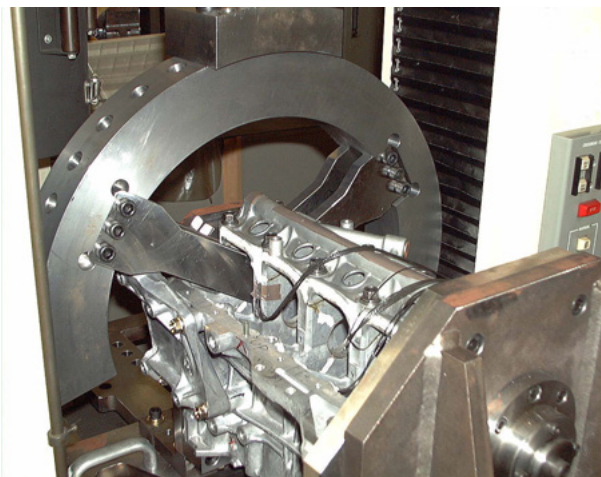


Fig. 2 Calibration jig

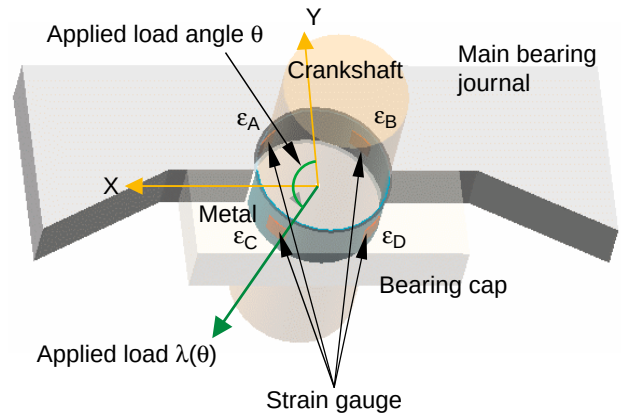


Fig. 3 Attached Strain Gauges and Applied Load

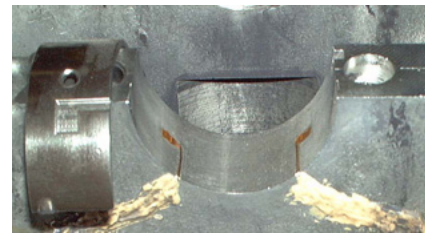


Fig. 4 Strain gauge attachment

The load applied was from 0 to 20,000 N. Measurements were taken every 500 N, in 10° increments from 0° to 360° . Special care was taken with machining the main bearing metals as the shape had a large effect on the oil film thickness. FEM analysis was applied to investigate the machined shape so that changes to the roundness of the internal surface of the main bearing journal could be minimized. Fig. 5 shows the difference between the circumscribing circle and inscribing circle concentric with the reference circle that was determined by the least-squares method applied to the measured values. Table 2 and Fig. 6 give the FEM analysis results of the machined shape. A high concentration of stress was apparent around the initial shape of the machined area of the bearing metals. Thus, these were not adopted for testing. However, by correcting the shape of the machined area, the roundness of these parts was roughly equivalent to that of the mass-production parts and a relatively uniform stress could be obtained. Therefore, the parts with the modified machined area could be used in the tests.

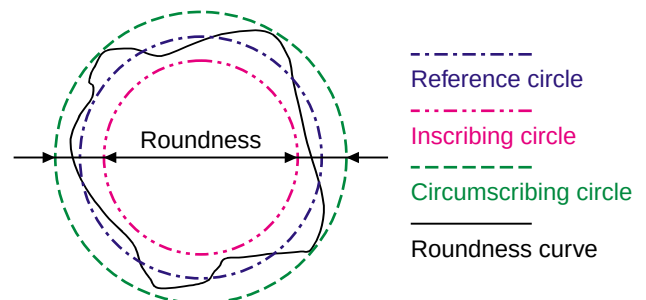


Fig. 5 Roundness measuring method

Table 2 Roundness measurement result

	Roundness (μm)
Mass-produced metal	10.25
Successful notch	9.05
Failed notch	20.01

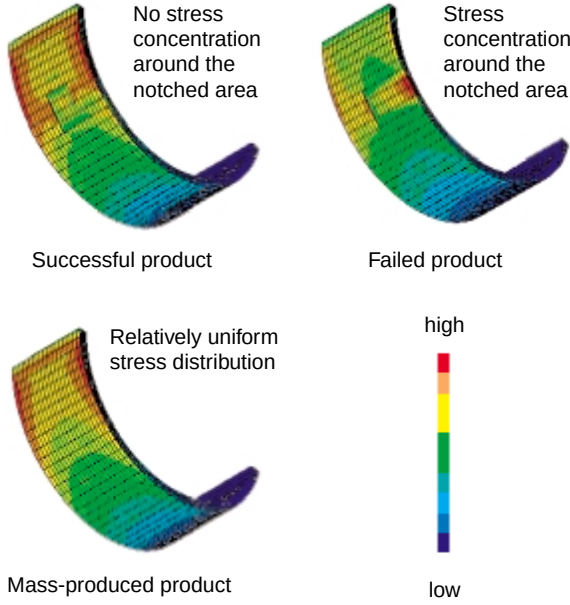


Fig. 6 FEM analysis of bearing metal (Stress distribution)

2.5 Calculating the Calibration Data

2.5.1 Calibration Data Measurements

The measurement results for this calibration test are shown in Fig. 7. The strain output for load exhibits good linear characteristics.

Thus, we decided to use the calibration sensitivity vector determined by the least-square method as the calibration data. Considering the possibility of non-linearity in the high-load

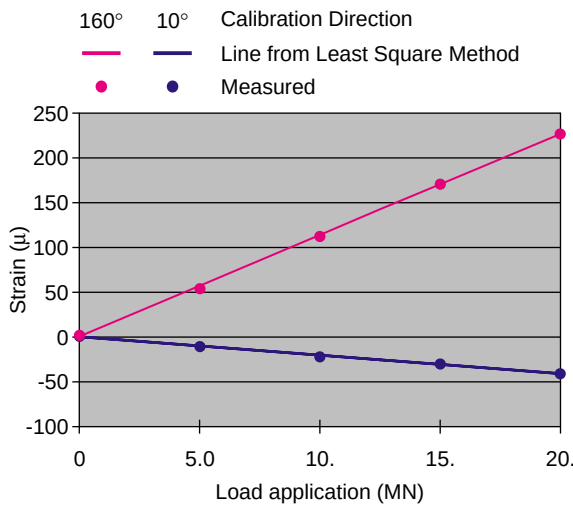


Fig. 7 Calibration test result (for 10° and 160° load direction)

range, calibration data were determined with a load from 0 to 20,000 N. The load direction is defined according to the coordinate system shown in Fig. 3. Calibration was carried out every 10° of crankshaft rotation. However, since a resolution of 1° was required for calculating the excitation forces, it was converted by Lagrange interpolation to calibration data for 1° increments.

2.5.2 Calibration Curve Calculations

The calibration curves shown in Fig. 8 were constructed by interpolating and plotting the calibration sensitivity vector that had been obtained from the test results for an applied calibration load direction from 0° to 360°.

Generally, when there is a perpendicular force applied to a structure like a main bearing journal, we can expect the strain distribution to be like that shown in Fig. 9.

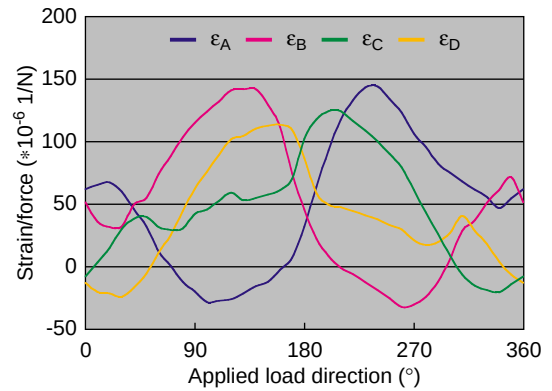


Fig. 8 Calibration curves

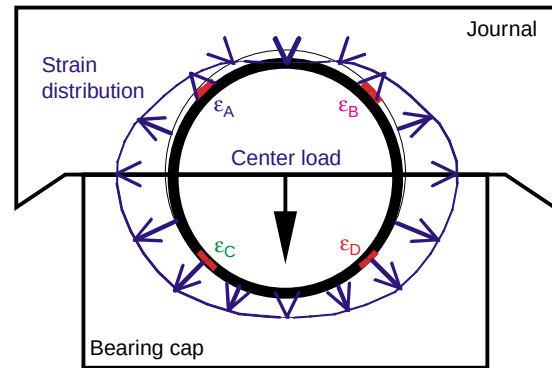


Fig. 9 Strain distribution for a vertical load

This is nearly symmetrical to the right and left of the applied load, but with the journal block side having slightly higher rigidity and lower stress, and the bearing cap side having slightly lower rigidity and higher stress. Also, the side opposite the direction of applied load can be considered to be subjected to compressive stress. The maximum stress would exist at a position $\pm 90^\circ$ from the load, but since the joint face is at this position, the point of maximum stress is shifted slightly. When the direction of the applied load is slightly moved to the right or the left of a perfectly vertical applied load, the stress distribution is also slightly altered.

The degree of this change depends on the rigidity of each area.

When inspecting the calibration curves with the above points in mind, there is nearly a left-right symmetry in the range from 0° to 180°, while the journal block and bearing cap rigidity differences can also be seen. Thus the results can be judged as qualitatively appropriate.

2.6 Measurement Methods

2.6.1 Measurement Conditions

Engine excitation force measurements were performed with the engine load absorbed by a dynamometer. The conditions for the tests are shown in Table 3.

Table 3 Measuring conditions

Engine speed range	1600 to 6000 rpm
Dynamometer torque	Wide open throttle 40 Nm While motoring
Lubricant temperature	85 °C to 95 °C
Coolant temperature	82 °C

2.6.2 Data Sampling Method

Multi-point simultaneous sampling was given the highest priority when obtaining data. The data was processed with the results used in this method shown in Fig. 10.

During engine operation, the pulse (at top dead center)

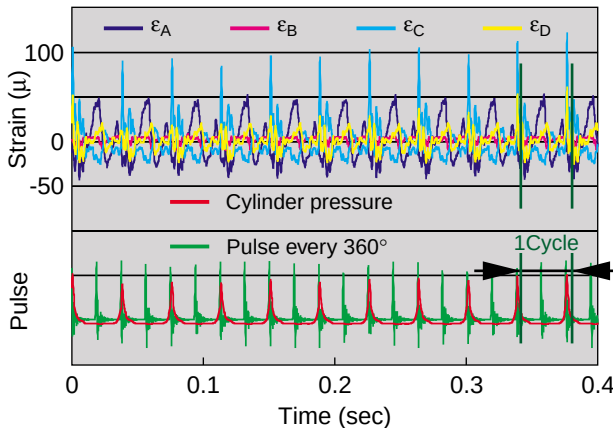


Fig. 10 Pulse examples under running conditions (wide open throttle 4000rpm)

and the cylinder pressure were measured by a revolution pulse sensor. Using the 360° pulse immediately prior to the peak cylinder pressure as start of a cycle, and the next 360° pulse immediately prior to the peak cylinder pressure as the end of the cycle, the crankshaft has rotated by an absolute angle of 720°. The sampled data are expressed as the average of 5 or more cycles.

2.7 Excitation Force Calculations

When the load direction is changed based on considerations of the tendencies for right-left symmetry and the vertical asymmetry of the strain distribution for the vertical load shown in Fig. 9, and when the output changes of each strain gauge are considered, the direction of the excitation forces can be determined if strain data is obtained from at least four locations.

The actual measurement data vectors from the four places on each main bearing journal are ϵ_A , ϵ_B , ϵ_C , and ϵ_D . These are determined, in the same way, from the strain calibration tests. As calibration load direction θ_k , magnitude $\lambda_k(\theta_k)$, and the strain ϵ_x ($x = A \sim D$) of the said locations can all be determined from the calibration tests, the sensitivity calibration value vectors, $K_A(\theta)$, $K_B(\theta)$, $K_C(\theta)$, and $K_D(\theta)$, are determined in the following way.

$$K_x(\theta_k) = \frac{\epsilon_x}{\lambda_k(\theta_k)} \quad \text{Eq. (1)}$$

The main bearing journal strain ϵ_x can be determined in the following way, using sensitivity calibration value vectors ($x = A \sim D$).

$$\epsilon_x = \lambda(\theta) \cdot K_x(\theta) \quad \text{Eq. (2)}$$

The main bearing journal strain relative error function $E(\theta)$ is obtained from the above.

$$E(\theta) = \sum_{x=A}^D \{\epsilon_x - \lambda(\theta) \cdot K_x(\theta)\}^2 \quad \text{Eq. (3)}$$

According to this measuring method, the $\lambda(\theta)$ and θ that minimize the relative error function $E(\theta)$ are determined and used as the excitation force and direction.

In Eq.(3), assuming that the q is fixed, the $l(\theta)$ resulting in a minimum $E(\theta)$ can be expressed in Eq.(4), below.

$$\lambda(\theta) = \frac{\sum_{x=A}^D \epsilon_x \cdot K_x(\theta)}{\sum_{x=A}^D K_x^2(\theta)} \quad \text{Eq. (4)}$$

In Eq.(4), the $\lambda(\theta)$ was calculated for θ throughout the range from 0° to 359° for ϵ_A to ϵ_D to determine the θ_0 where $E(\theta)$ is at a minimum as well as the corresponding $\lambda(\theta_0)$. The magnitude of the excitation force $\lambda(\theta_0)$ and the excitation force direction θ_0 were obtained from the above calculations.

3. Measurement Results

The excitation forces determined by this method are shown in Figs. 11-13.

Fig. 11 gives the excitation force of each main bearing journal for the crankshaft angle. When the No. 1 cylinder ignition is near 0°, the No.1 and No.2 main bearing journal excitation forces are at their peaks. This indicates that the greatest excitation force for each bearing journal changes in accordance with the firing sequence (1-3-4-2).

Fig. 12 is a Lissajous figure for the direction of the generated engine excitation force. The excitation force is at an angle approximately 15° from the vertical. Considering that this was mainly caused by the downward load from the

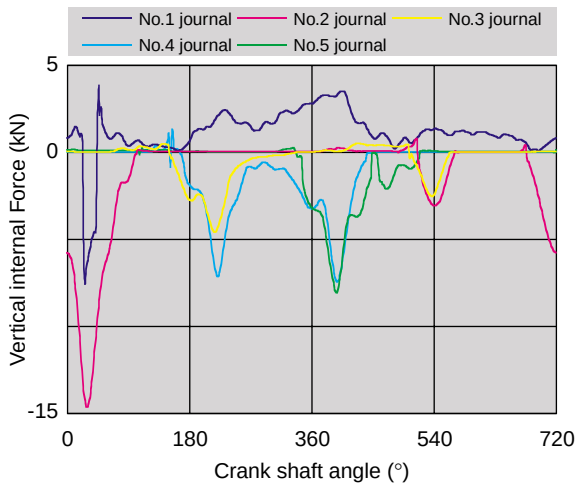


Fig. 11 Internal force at each main bearing journal (wide open throttle 1600rpm)

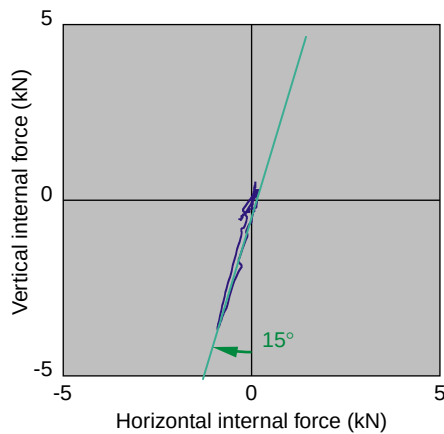


Fig. 12 Lissajous figure for internal forces

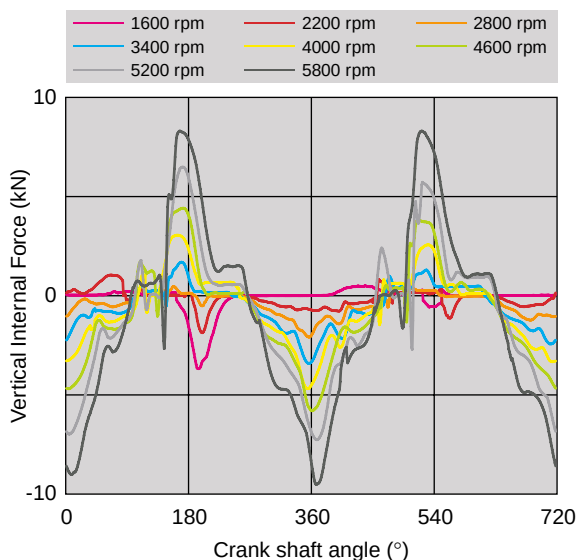


Fig. 13 Internal force variation (40Nm torque, no.3 journal)

crankshaft balance rate and that the ignition angle was set at 15° before TDC, the results seem appropriate.

Fig. 13 shows the changes in engine excitation forces that accompany increases in rpm. In the comparatively low-speed range, the effect of ignition pressure, which decreases, is large. However, in the comparatively high-speed range, we can see the significant effects from the inertia force that brings about a clearly defined peak every 180° .

The foregoing results are both non-contradictory and qualitatively appropriate. Since these results are also sufficiently linear, it is believed that accurate measurements can be made.

4. Summary

A method for directly measuring the magnitude and direction of the constantly changing excitation forces was developed. The direction and absolute value of the excitation forces were determined from the strain measurements taken during operation by multiple strain gauges attached to the internal surface of each main bearing journal. They were determined when the relative error of the strain estimated from the calibration curve was at a minimum. Measurements of the excitation forces were taken for a Civic 4-cylinder engine in the speed range of 1,600 rpm to 6,000 rpm. Consistently non-contradictory and appropriate results were obtained. This measurement method for determining excitation forces can be carried out without using large-scale measuring equipment, unlike the conventional indirect methods, making it relatively simple to measure the excitation forces.

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Prediction Technique for Gear Noise Transfer in Engine Mounting Systems

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ABSTRACT

Prediction methods using finite element method (FEM) analysis were investigated to construct technology for predicting the aggravation of gear noise and other high frequency vibrations due to engine mount transfer in the initial development stages. Engine mounts are normally comprised of rubber clasped between metal brackets on the power plant and chassis sides. Coupled vibration including these metal parts is a problem for high frequency vibration transfer. This study showed the mount vibration transfer characteristics are a composite of the response characteristics of each metal part, and that simple evaluation by the response level can be used to predict the aggravation of noise in the cabin. In addition, modeling the chassis frame and case to which the metal fittings are joined, and modeling the clamping rigidity of the joints using spring elements, made it possible to accurately calculate the frequency response (error of resonance frequency: approximately 5%).

1. Introduction

Reducing transmission gear noise is an important issue from the viewpoint of improving automobile product appeal. Routes for gear noise into the front seat are divided into structural-borne noise, such as through the engine mounts and cables, and airborne noise radiated from the transmission case. Currently, gear noise transferred from the engine mounts is a large contributing factor. Many development man-hours are spent to coping with this problem. Research thus far has shown that there is little attenuation of vibration in engine mounting systems due to resonance of the metal fittings on the power plant and chassis sides which encase the mount rubber. This results in vibration which is transferred at certain frequencies, thus aggravating gear noise in the front seat. However, it was difficult to quantitatively predict the amount of aggravation in advance. Accurate prediction techniques are desired due to recent trends toward shorter development periods and reduced numbers of prototypes. Therefore, this study investigated the following items.

- (1) Methods of evaluating the appropriateness of the transfer level by calculating the gear vibration transferred through the engine mounts.
- (2) Model creation techniques for simple and accurate calculations.

2. Investigation of Calculation Evaluation Methods

2.1 Analysis of Gear Vibration Transfer

Gear meshing vibration is transmitted through the transmission case. Part of this vibration is transmitted to the chassis through the brackets fixing the transmission, the rubber and the metal fittings fixing the rubber. Therefore, the problem is thought to be the total transfer characteristics of the transmission case, mounting system and chassis frame. Fig. 1 shows a comparison of the condition where gear vibration is transmitted at the peak vibration while cruising

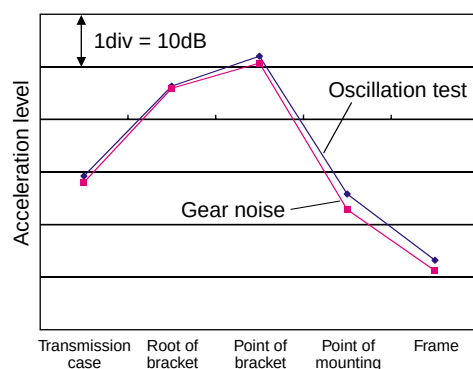


Fig. 1 Mode shape of mounting system

and the vibration mode where the tip of the transmission side bracket is excited by hammering excitation. In addition, Fig. 2 shows an example mounting system layout.

The graph shows the size of the gear vibration component while cruising at each point and the response at that same

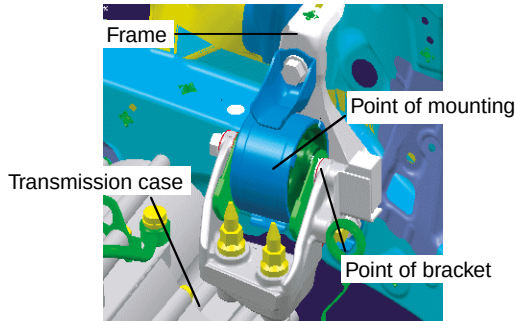


Fig. 2 Layout of a mounting system

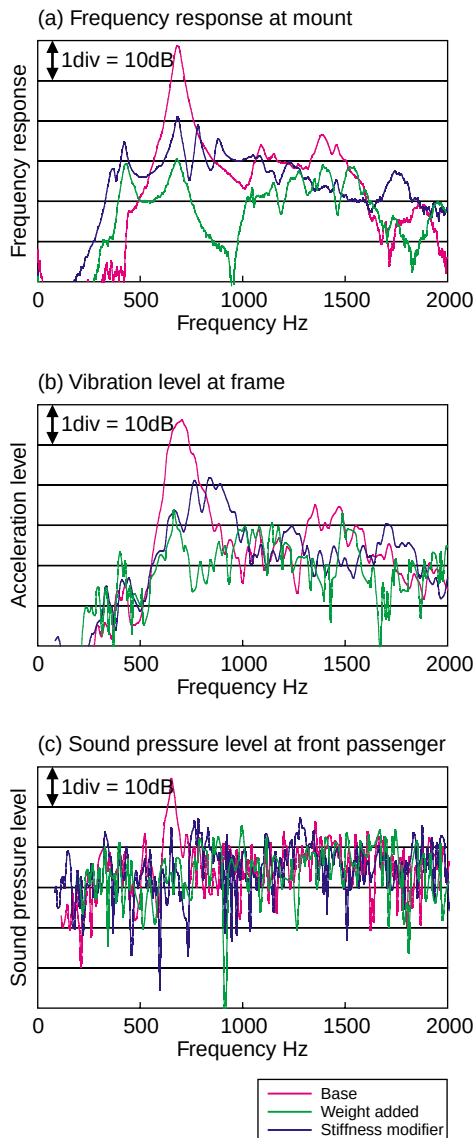


Fig. 3 Sound and vibration level reduction effect of improvement of frequency response

point when the tip of the power plant side is excited. These results show that hammering-induced coupled vibration and gear vibration transfer while cruising have almost the same mode. In addition, Fig. 3 shows the hammering-induced frequency response (chassis side tip response to transmission side tip excitation), the gear vibration level on the chassis side while cruising, and the gear noise level in the front seat.

These graphs compare three different types of mounting system specifications: the base specifications without countermeasures, specifications with weight added to change the resonance frequency, and specifications with the bracket stiffness changed. The metal bracket and mount fittings resonate with the base specifications resulting in sharp peaks in each of noise, vibration and frequency response measurements. However, the resonance frequency is shifted for each of the countermeasure specifications. The frequency response is improved and the sound and vibration peaks decrease. This indicates that there is a correlation between the frequency response characteristics of the mounting system and gear noise, and that gear vibration transfer can be evaluated by this response level. Therefore, frequency response characteristics (chassis side tip response to transmission side tip excitation) were calculated through simulations.

2.2 Simplification of the Model

As stated in 2.1. above, there is a correlation between the mount vibration while cruising and the gear noise in the front seat. The problem with this gear vibration is thought to be the vibration mode of the total transfer route. However, modeling the entire system is impractical due to system resources and other limitations, thus it was decided to calculate the chassis and transmission sides separately with the rubber as the boundary. First, the coupled condition of the metal bracket and mount fittings was analyzed through hammering excitation tests. Fig. 4 shows a comparison of the discrete responses for the transmission (a) and chassis (b) sides, the overall response characteristics due to the mounting tip response to bracket tip excitation (d), and the composite of the discrete responses (c).

This comparison shows that the total response characteristics due to transmission bracket tip excitation (d) exhibit features of the discrete metal bracket and mount fitting responses (a) and (b), and closely match the composite (combined) characteristics (c) of the two discrete responses.

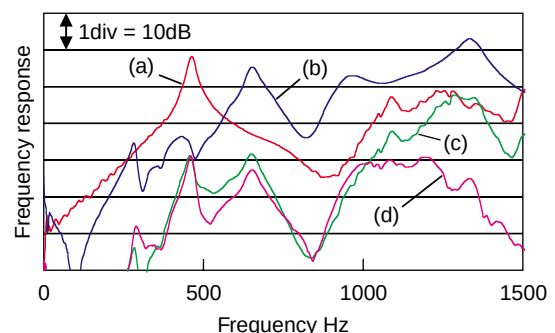


Fig. 4 Analysis of frequency response

In other words, the attenuation of the rubber has comparatively flat characteristics with respect to frequency, and the rubber stiffness is also sufficiently low compared to the metal fittings, so the metal fitting modes on the transmission and chassis sides are independent. Therefore, it can be assumed that only the load is transmitted between the metal fittings, which means that simple prediction of the total frequency response characteristics is possible by calculating and combining the frequency response characteristics (point inertance) of the discrete metal bracket and mount fittings.

2.3 Composite Inertance Calculation Results

Fig. 5 shows the results (hereafter “composite inertance”) obtained by calculating and combining the discrete bracket and metal fitting inertance for the three specifications described in Fig. 3 of section 2.1. Like sound, vibration and hammering-induced frequency response, the peak level relationship for composite inertance also showed the largest peak for the base specifications, with lower peaks for each of the countermeasure specifications. In addition, it was confirmed that gear noise also decreased when the calculated composite inertance was lower.

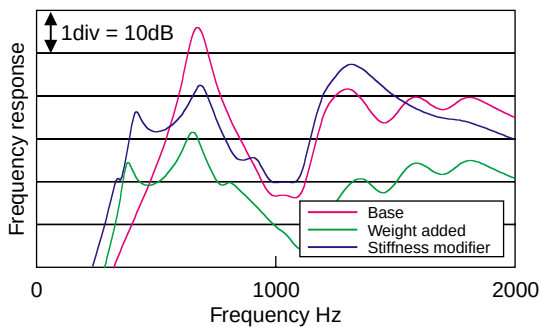


Fig. 5 Total frequency response function

2.4 Judgment Criteria for Composite Inertance Level

As described above, aggravation of gear noise can be predicted by the composite inertance level. Therefore, in actual development target levels can be established for each characteristic by considering the chassis acoustic characteristics, excitation source size and required performance based on the composite inertance level of the mounting systems for mass production models developed thus far.

3. Bracket Frequency Response Analysis

3.1 Difference in Results due to Boundary Conditions

When performing eigenvalue analysis and response analysis, calculations can be performed with sufficient accuracy as shown in Fig. 7 in the discrete free-state (unclamped and with no added objects) shown in Fig. 6.

However, in the mounted condition (joined to the case and chassis frame), the results vary widely according to the clamping conditions and other factors. Fig. 8 shows a



Fig. 6 Bracket model

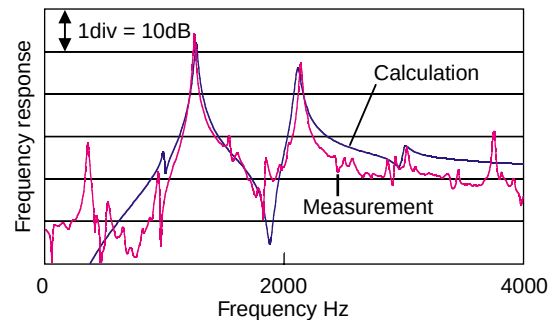


Fig. 7 Frequency response of bracket

Solver : CATIA FEM
Auto mesh : CATIA FEM
Mesh size : 5 mm ~ 10 mm
Damping ratio : 0.05

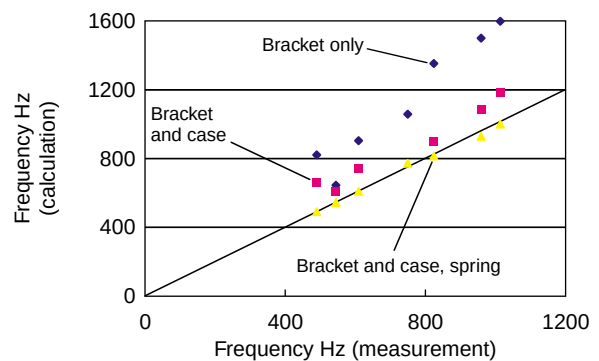
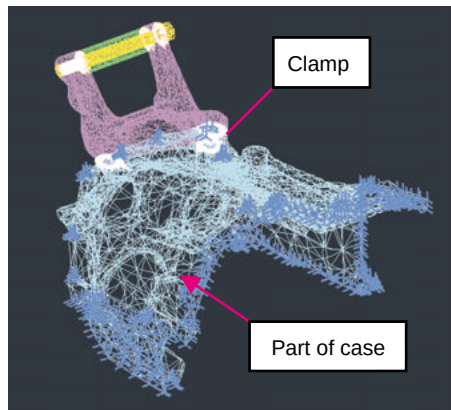


Fig. 8 Resonance frequency

comparison of the calculation results and the actual measured resonance frequency due to hammering excitation. This graph shows the values under three types of clamping conditions for each of the discrete bracket (mount clamping), bracket and case, and case and bracket coupled by spring element models.

These results show that the model consisting of only the metal bracket fitting does not match with the measured values, so the case and chassis frame must be modeled as shown in Fig. 9, or the joint must be modeled in order to obtain accurate results (error: 5% or less).

(1) Case and bracket



(2) Frame and mounting

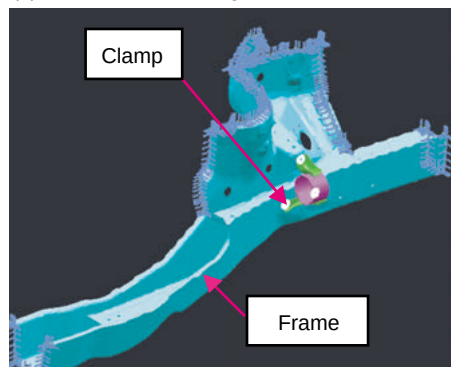


Fig. 9 FE model

that a thick plate and test piece were joined with contact elements and downward pressure was applied over the approximate area of the washer diameter. Fig. 11 shows the thickness of the clamped plate and the contact diameter. As shown in this figure, the plate deforms into a bowl shape and the contact area becomes smaller as the plate thickness decreases. Accordingly, these calculations showed that for a steel plate approximately 2 mm thick, even if the drawing shows a wide contact area, the range coupled by the spring elements is approximately the diameter of the clamping washer.

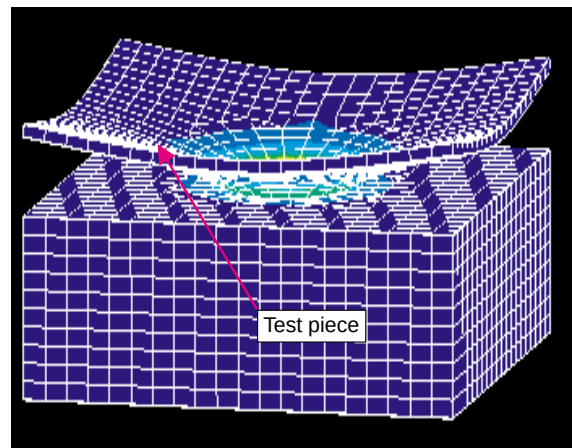


Fig. 10 FE model of contact analysis

3.2 Bolt Clamping Conditions

3.2.1 Spring Coupling Concept and Spring Constants

The mount and bracket fittings are bolted to the chassis frame and the transmission case. These types of joints are generally said to have lower stiffness than normal structures.⁽¹⁾

This study created a model assuming that springs equivalent to the bolt stiffness and clamping portion stiffness exist on the contact surface. The static stiffness k of the bolt clamping portion was assumed to be $k = k_t + k_c$, where k_t is the bolt spring constant and k_c is the clamped portion spring constant. Using an approximation formula⁽²⁾, k_t and k_c were calculated as approximately 10^5 to 10^6 N/mm (determined by the material characteristics and shapes of the bolt and clamping portion).

3.2.2 Differences due to Spring Coupling Range and Plate Thickness

When the contact surface of the bolt clamping portion is modeled using spring elements, the coupling range is also important. The actual contact surface area and the contact surface on the drawing may differ according to the clamping portion shape. Therefore, contact analysis was performed using finite element method (FEM) analysis using simple shapes, and the clamped portion thickness and clamped contact area were investigated. Fig. 10 shows the model used for the calculations.

The calculation analyzed the contact condition assuming

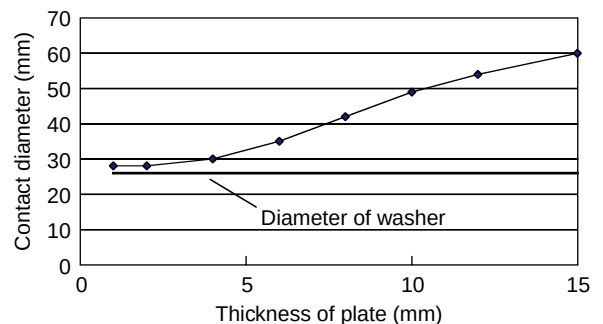


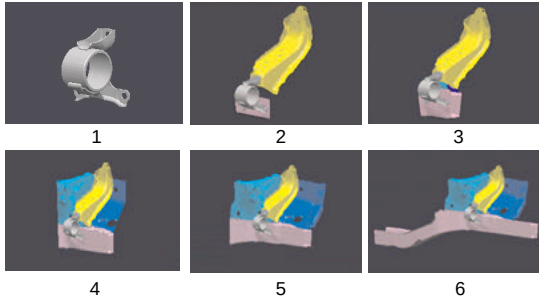
Fig. 11 Diameter of contact surface

3.3 Extent of Modeling

Fig. 12 shows the extent of modeling and a comparison of the calculated resonance frequency and the actual measurements for each model.

This figure shows that the calculation for the discrete metal fitting (model 1) produces a resonance frequency that is too high as mentioned previously, but that if even a small part of the chassis frame is modeled (model 2), the results approach the actual measurements. Thus, this study performed calculations for models 4 and higher in Fig. 12, which is thought to be a sufficient extent of modeling.

(1) Extent of FE model



(2) Comparison of resonance frequency

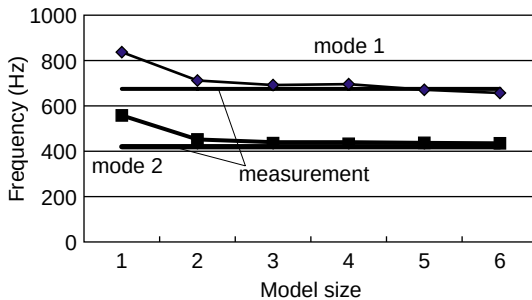
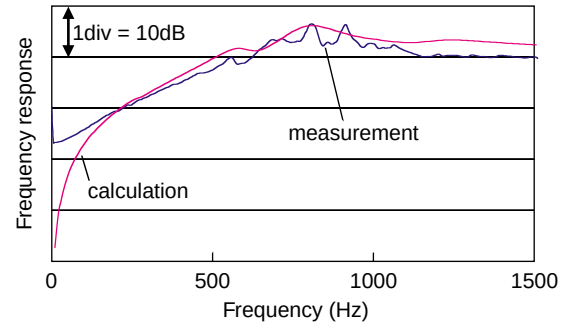


Fig. 12 Relationship of resonance and model size

(1) Example 1



(1) Example 2

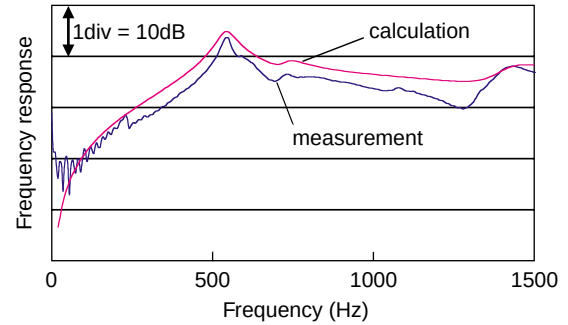


Fig. 13 Comparison between calculation and measurement for frequency response

3.4 Response Analysis Results

Fig. 13 shows an example comparison of the analysis results under the above conditions and actual measurements due to hammering. In addition, Fig. 14 shows an example of eigenvalue analysis results.

As shown in these figures, the calculated point inertance of the metal bracket and mount fittings matches closely with the hammering excitation test results using actual vehicles. Thus, a composite inertance such as shown in Fig. 5 can be accurately calculated using either of these results. In addition, it was confirmed that analysis can be performed in approximately one day by creating a mesh and calculating directly from the CATIA 3D model which is the product drawing. Therefore, the mount transfer characteristics can now be accurately predicted and optimized at an early timing.

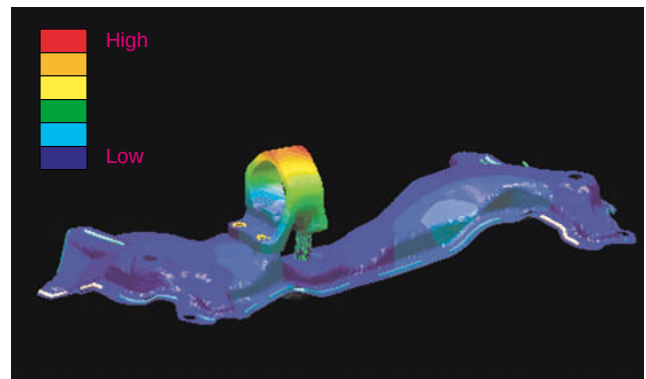


Fig. 14 Displacement of mounting at resonance frequency

4. Conclusion

This study yielded the following results.

- (1) Gear vibration transfer by the engine mounts can be simply evaluated by the composite value level of the response analysis results for the discrete metal bracket and mount fittings in the engine mounting system.
- (2) The necessary know-how was obtained to enable comparatively simple yet sufficiently accurate calculations (error: 5% or less) by creating models to the necessary extent and spring elements for the bolt clamping portions.
- (3) The traditional problem of gear noise peaks at certain speeds is mostly due to structural-borne noise transferred through the engine mounts and other elements as

described in this study. The transfer characteristics of each part can be predicted and development efficiency can be vastly improved in the future by using the prediction technique proposed in this study from the initial development stages.

CAE is progressing and we are approaching an environment where virtually anyone can perform FEM analysis. However, despite this it is surprisingly difficult to apply FEM analysis to real noise and vibration problems at actual development sites and achieve satisfactory results. The contents of this study may not be high level in terms of vibration science and calculation technology, but they are

reported as an example of an application which can contribute to reducing man-hours at development sites amidst the need to shorten development periods. It is the authors' hope that these results will prove a useful reference in improving development efficiency using simulations in the future.

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Development of a Night Vision System (Vehicle-Pedestrian Collision Avoidance System)

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ABSTRACT

A night vision system has been developed to help reduce vehicle-pedestrian accidents occurring at night. High temperature objects presumed to be pedestrians are detected by processing the images from infrared stereo cameras mounted on the front of the vehicle, and the possibility of a collision is judged by calculating the position and relative moving vector of the pedestrian. In addition, voice guidance is provided and a highlighted infrared image of the pedestrian is displayed at the bottom of the front windshield using a head-up display (HUD) installed in the instrument panel.

As a result of confirming detection performance on a test course, it was found that the possibility of collisions with pedestrians on the road and crossing the road could be judged.

Further, as a result of confirming evasive action on the part of the driver with respect to the provided information, this system was found to be a useful means of reducing vehicle collisions with pedestrians. Further development is needed to bring this kind of system to the point of practical application.

1. Introduction

Approximately 10,000 traffic accident fatalities occur in Japan annually. Some 30% of these are a result of collisions with pedestrians, the vulnerable parties in traffic.⁽¹⁾

We are presently working on a practical application of a pedestrian injury-mitigation system in order to reduce the possible consequences resulting from vehicle-pedestrian accidents. This paper reports on our examination of systems for providing drivers with advance notice about the presence of pedestrians.

In order to examine the system functions and specifications, we first conducted a detailed investigation of accidents involving pedestrian fatalities. Fig. 1 shows the results of our investigation.

These results showed that:

- (1) As shown in graphs A), B), and C), the majority of accidents involving collisions with pedestrians on the road or crossing the road take place when vehicles are traveling straight on straight roads at night.
- (2) As shown in graph D), the majority of these accidents involve collisions with vehicles traveling at speeds of 40-80 km/h.

As a result, we developed a night vision system that would help detect pedestrians on the road or crossing the road that are difficult to see from vehicles driving at night

as shown in Fig. 2, and would notify the driver by providing voice guidance and displaying infrared camera images.

We also verified the usefulness of this system by means of tests to detect pedestrians crossing the road and obstacle avoidance tests.

2. System concept

2.1 System functions and configuration

The following target requirements were established for the purpose of setting system functions:

- (1) The information provided will be able to call the driver's attention to obstacles and will allow judgment of forward conditions.
- (2) The system will be capable of detecting objects ranging in size from a pedestrian's head to a full-body (20 x 20 cm to 2 x 2 m) while taking into consideration targets hidden by oncoming vehicles and other such factors on ordinary roads.
- (3) The system will provide a time margin (information provision time) of 3.5 seconds or more⁽²⁾ after information provision and before collision, and will provide the detection range needed for this.

We decided to satisfy requirement (1) by using voice guidance and a head-up display (HUD) of images of nighttime forward conditions from an infrared camera.

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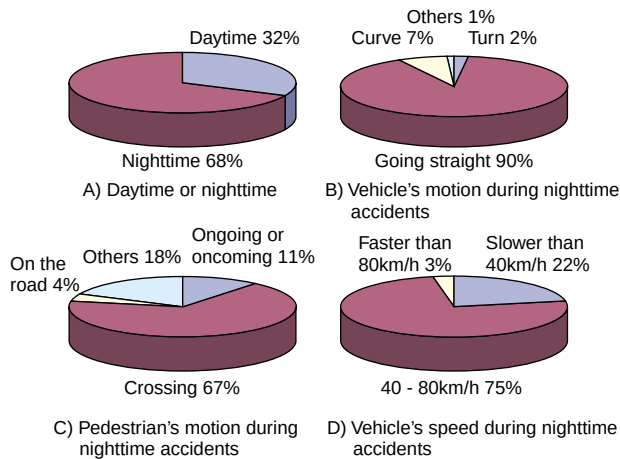


Fig. 1 Accident investigation of pedestrian fatalities

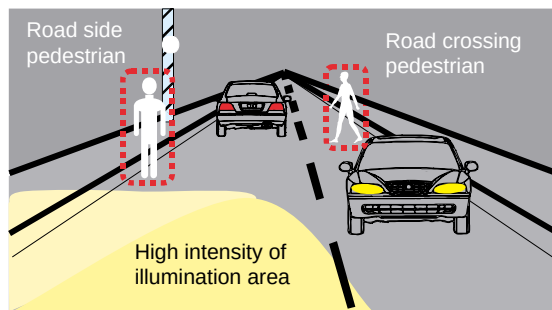


Fig. 2 Example of pedestrians who the driver should notice

We decided that detection capabilities satisfying requirements (2) and (3) could be obtained by using infrared stereo cameras, calculating the distance to high temperature objects that can be perceived as pedestrians, and thus judging the possibility of a collision. Fig. 3 shows the layout of the system on an actual vehicle, while Fig. 4 shows information provision in operation.

2.2 Operating conditions for information provision

The target area for information provision by this system is the region that is difficult to see with headlights and so is subject to a visual enhancement system. This was defined as the region extending outward from the high intensity of illumination area of the headlights (which extends up to 30 m forward).

Based on result (2) from the investigation of pedestrian accidents, it was decided that the information provision system would operate at cruising speeds of 40 km/h and above. The objective was to satisfy the requirement of an information provision time of 3.5 seconds or more for driving speeds up to 80 km/h. This is the 30-80 m detection area shown in Fig. 5.

2.3 Detection and judgment conditions

The two detection areas and the judgment conditions shown in Fig. 5 were established in order to assist in the detection of pedestrians subject to possible collision.

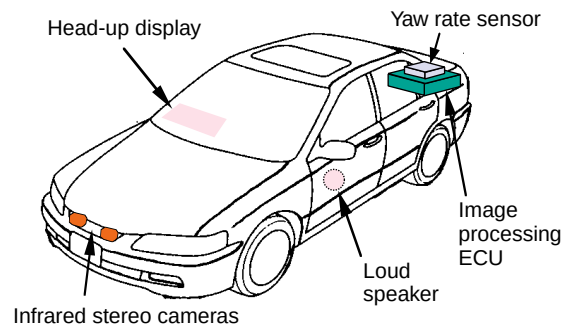


Fig. 3 System component locations

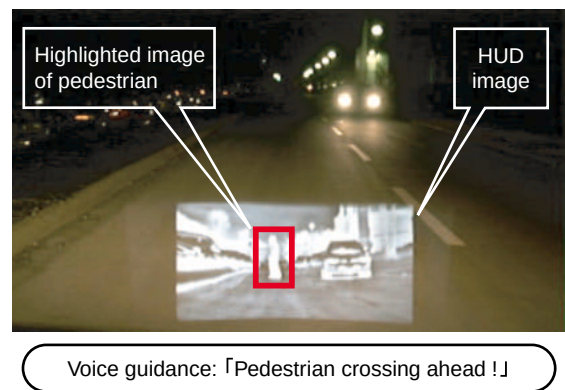


Fig. 4 Driver's view of the HUD image and voice guidance

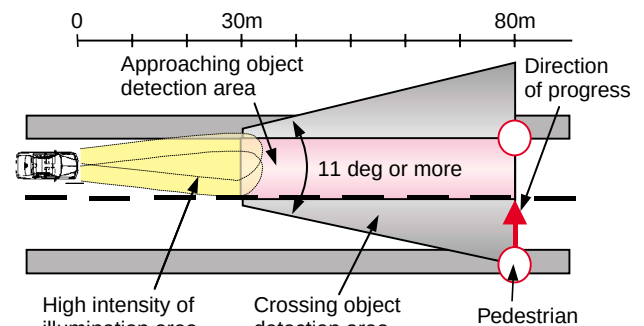


Fig. 5 Pedestrian detection area

2.3.1 Detection area and judgment of road-side pedestrians

The system detects pedestrians walking along the road at a rate of 5 km/h or less within an area the body width of the vehicle with an additional margin of 1 m to either side.

When the subject vehicle is turning, the system uses the yaw rate and vehicle speed to estimate the predicted course of the vehicle, and changes the shape of the detection area accordingly.

2.3.2 Detection area and judgment of crossing pedestrians

The condition for a collision between the subject vehicle and an object crossing the road is that, when seen from the subject vehicle, the object azimuth θ (in degrees) is constant.⁽³⁾ The azimuth θ is determined by the speed of

one's own vehicle V_j and the crossing speed V_h of the object according to the following equation:

$$\theta = \tan^{-1} \frac{V_h}{V_j} \quad (1)$$

Assuming that the pedestrian is crossing at the rate of 1 m/s and that the minimum operating speed for carrying out information provision is 40 km/h, then the detection area for crossing objects will be the total imaging range of the infrared cameras across an angle of 11 degrees or more.

Objects with a greater crossing speed will only appear for a short time as images in the infrared camera's angular field of view. Although they can be detected, the time for information provision will be reduced.

3. Detection processing algorithm

3.1 Definition of coordinate system

This system uses two coordinate systems, the 3-dimensional coordinate system based on the vehicle body as shown in Fig. 6, and the image coordinate system.

The origin O of the 3-dimensional coordinate system is the lens center of the camera on the right. Right-hand rotation of the axis is treated as positive. The image coordinate system is the plane in the 3-dimensional coordinate system where $Z=f$ (where f is the camera focal length), and the origin o is the point of intersection with the Z -axis.

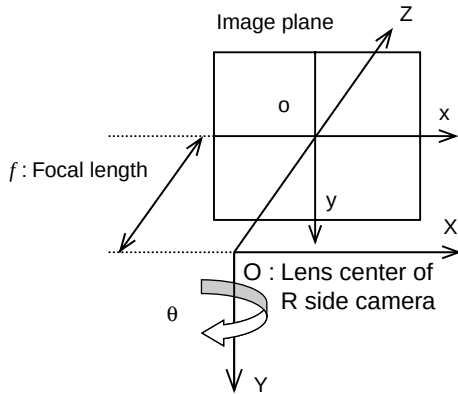


Fig. 6 Definition of coordinate system

3.2 Pedestrian isolation and tracking over time

Measurement of the difference between atmospheric temperature and pedestrian skin surface temperature in all seasons showed that a temperature difference of about 5°C occurs at atmosphere temperatures up to about 30°C. The measurement results are shown in Fig. 7.

Consequently, high temperature objects such as pedestrians appear in the infrared image with a higher brightness than background objects.

We confirmed the possibility of rendering pedestrians in images by driving on ordinary roads in every season, collecting image data, and performing two-level renditions of it. Fig. 8 shows an infrared image taken in the summer and the results of the two-level rendition.

The system on an actual vehicle generates a histogram of the brightness of the right-hand image from the infrared

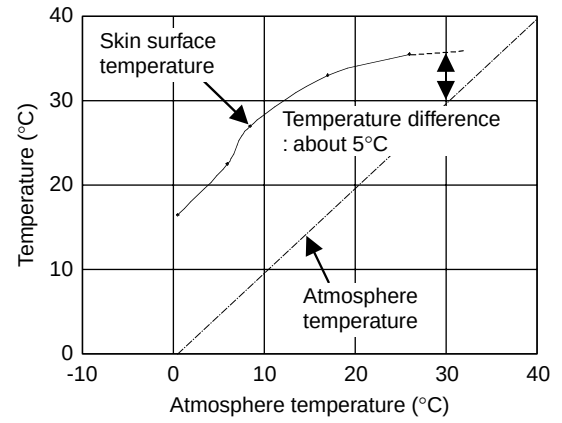


Fig. 7 Temperature differences in skin surface and atmosphere during all seasons



Fig. 8 Infrared image and result of two-level rendition under summer time conditions

stereo cameras and sets the intermediate value in the frequency distribution as the threshold value. The pedestrian in the right-hand image is elicited in this way.

The equality of the object in the right-hand image is judged at each point in time according to such conditions as the position of its center of gravity and characteristics of its geometry to track the object over time. Fig. 9 shows the tracking status of the object in the right-hand image over time.

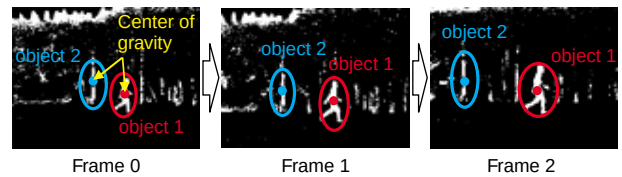


Fig. 9 Object tracking

3.3 Calculation of the object's 3-dimensional position coordinates

As shown in Fig. 10, the left and right images from the infrared stereo cameras display a difference of lateral position (disparity) of dn (in pixels) according to the distance to the object.⁽⁴⁾ The area around the object in the right-hand image is extracted and used to perform a correlation calculation with the left-hand image. In this way, the disparity dn is calculated. Next, equation (2) is used with the thus calculated disparity to derive the distance to the object Z_i (in meters).

The center of gravity position $q=(x_i, y_i)^T$ in the image

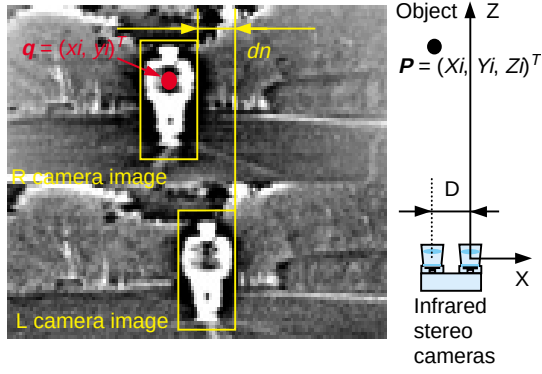


Fig. 10 Disparity of infrared stereo cameras

and the calculated value Z_i (in meters) are then used in equations (3) and (4) to calculate the 3-dimensional position of the object $P=(X_i, Y_i, Z_i)^T$.

D is the distance (in meters) between cameras, while f and p are the focal length (in meters) and pixel pitch (in m/pixel) of the cameras.

$$Z_i = f \times \frac{D}{dn \times p} \quad (2)$$

$$X_i = \frac{x_i \times Z_i \times p}{f} \quad (3)$$

$$Y_i = \frac{y_i \times Z_i \times p}{f} \quad (4)$$

3.4 Calculation of the object's relative moving vector

The calculated position of the object along the X-axis can be changed significantly by slight changes in the turning angle of the vehicle while it is moving. Therefore, it is difficult for the system when operating from a traveling vehicle to identify the difference in the 3-dimensional position of a crossing pedestrian along the X-axis.

Consequently, as shown in Fig. 11, the system uses the vehicle speed and yaw rate sensors to calculate the distance traveled (T) and turning angle (q) of the subject vehicle. Equations (5) and (6) are then used to compensate for the difference.

However, $P'[t]=(Xa[t], Ya[t], Za[t])^T$ and $P[t]=(Xc[t], Yc[t], Zc[t])^T$ are the 3-dimensional positions A and C, respectively, of the object as calculated at time t .

$$\begin{bmatrix} Xc[t] \\ Yc[t] \\ Zc[t] \end{bmatrix} = \begin{bmatrix} \cos \theta & 0 & -\sin \theta \\ 0 & 1 & 0 \\ \sin \theta & 0 & \cos \theta \end{bmatrix} \begin{bmatrix} Xa[t] \\ Ya[t] \\ Za[t] \end{bmatrix} + \begin{bmatrix} Tx \\ 0 \\ 0 \end{bmatrix} \quad (5)$$

$$Tx = \frac{360 \times T}{2 \times \pi \times \theta} \times (1 - \cos \theta) \quad (6)$$

Fig. 12 shows the group of object position $P[t]=(Xc[t], Yc[t], Zc[t])^T$ during the period from time $t=0$ (the past) to $t=n$ (the present) with compensation for the turning angle of the subject vehicle using equations (5) and (6).

The straight-line directional vector $L=(Lx, Ly, Lz)^T$ is obtained by performing principal component analysis⁽⁵⁾ on the center of gravity position coordinate $Q=(\bar{X}, \bar{Y}, \bar{Z})^T$ and the matrix of cross-correlation coefficient for the group of the object position ($P[t]=, t=0, 1, \dots, n$) shown in equation

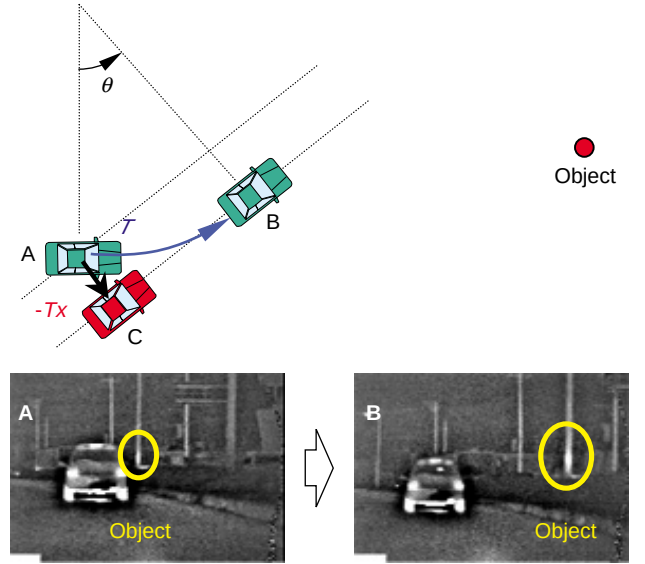


Fig. 11 Compensation for the object's 3D position

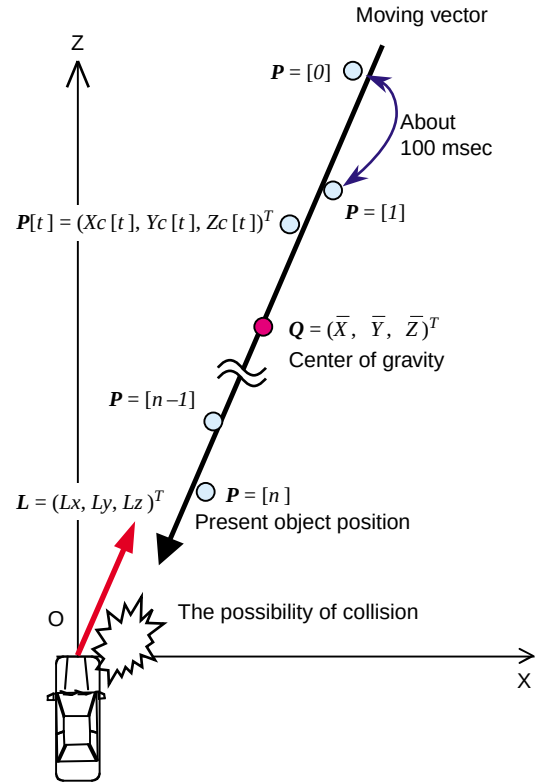


Fig. 12 Predicted 3D moving vector

(7). In this way, the relative moving vector for the object is determined as the straight line shown in equation (8).

The relative velocity (V_s) is calculated from the changes occurring over time in the object position.

$$\begin{bmatrix} \bar{X} \\ \bar{Y} \\ \bar{Z} \end{bmatrix} = \frac{1}{n+1} \begin{bmatrix} \sum_{i=0}^n Xc[i] \\ \sum_{i=0}^n Yc[i] \\ \sum_{i=0}^n Zc[i] \end{bmatrix} \quad (7)$$

$$\frac{X - \bar{X}}{L_x} = \frac{Y - \bar{Y}}{L_y} = \frac{Z - \bar{Z}}{L_z} \quad (8)$$

3.5 Processing to judge the possibility of a collision

The two detection judgments described in section 2.3 regarding detection judgment conditions are carried out with respect to an object that is up to 2 m in height above the road plane and that satisfies the conditions in equation (9) for the calculated relative velocity (V_s) and the driving velocity V calculated from the speed sensor.

$$|V - V_s| \leq \frac{V}{2} \quad (9)$$

3.6 Processing flow

In order to realize real time processing, the system separates processes (1) and (2). These are carried out on independent operation cycles, with process (1) occurring at an NTSC frame rate of 33 msec, and process (2) occurring every 100 msec.

Process (1): Using only the right-hand image from the infrared stereo cameras, this process performs two-level rendition, tracking, and image display of the object.

Process (2): This process calculates the 3-dimensional position of the object from the stereo images, estimates the relative vector, and judges the possibility of a collision.

The pedestrian detection processing flow is shown in Fig. 13.

4. Hardware

4.1 Infrared stereo cameras

The infrared stereo camera system is made up of two 320 x 240-pixel uncooled infrared cameras that have a horizontal field of view of 12 degrees and a vertical field of view of 9 degrees. The distance between the cameras is 360 mm.

The characteristics of the infrared stereo cameras must meet the following four conditions in order to measure the distance to the object correctly:

- (1) The two cameras must have synchronous detection.
- (2) The two cameras must have equivalent image brightness characteristics.
- (3) The two cameras must be in alignment.
- (4) The optical systems of the cameras must not have any distortion.

Synchronous detection was realized by fabrication of prototype synchronous detecting circuits. The cameras in the present prototype system demonstrated sufficiently similar image brightness characteristics.

Corrections for the alignment of the cameras and any distortion in their optical systems were made in the software.

Fig. 14 shows the layout of the prototype infrared stereo cameras when installed on an actual vehicle.⁽⁶⁾

4.2 Yaw rate sensor

The system uses an optical fiber gyroscope with a dynamic range of ± 50 deg/s and a minimum detection resolution of about 0.03 deg/s to calculate the turning angle of the vehicle in order to estimate the relative moving vector and change the shape of the detected area.

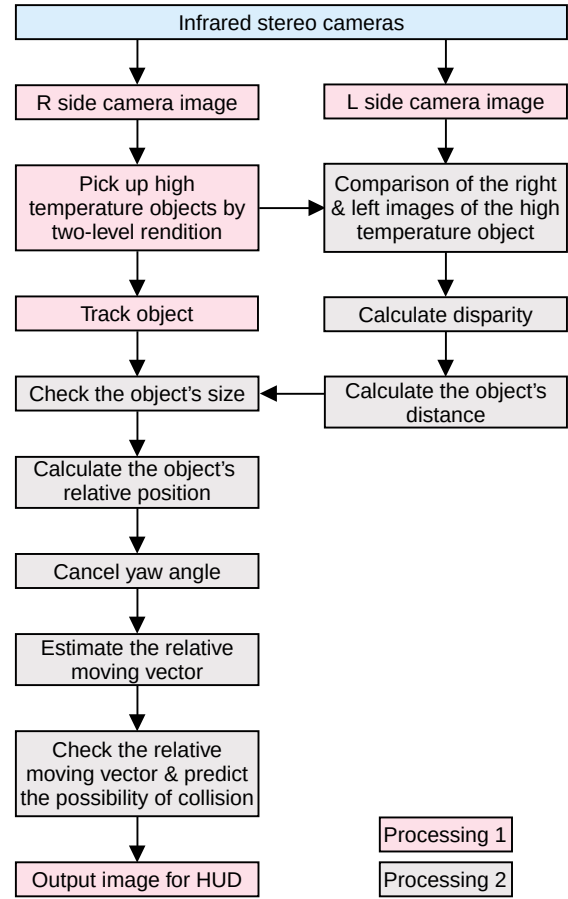


Fig. 13 Processing flow of pedestrian detection

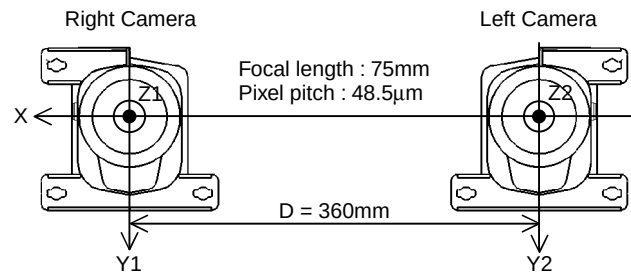
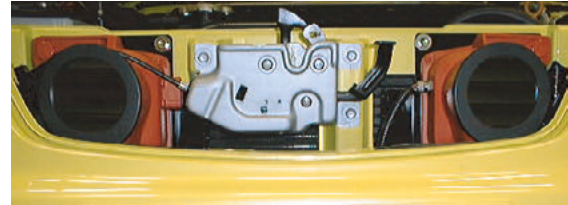


Fig. 14 Infrared stereo cameras Installed in the car

4.3 Image processing ECU

The left and right images from the infrared stereo cameras are output as NTSC signals. These are merged in a 640 x 480-pixel image split horizontally into two (left and right camera images each in a 640 x 240-pixel format) from which 8-bit gray-scale images are sampled at a video rate (30 frames/second). (See the infrared stereo cameras screen image in Fig. 10.)

Speed, yaw rate, and other such information about the vehicle are read in a 33 msec cycle, along with the images. The configuration of the image processing ECU is shown in Fig. 15.

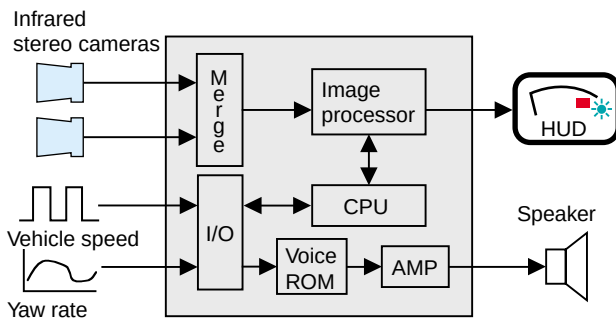


Fig. 15 Image processing ECU

4.4 Head-up display (HUD)

The system displays an 11- x 4-degree (horizontal x vertical) field of view image in order to be capable of displaying the detection range for pedestrians crossing the road and a full-body view of a pedestrian 30 m ahead.

The HUD visible to the driver has a 10- x 3.5-degree (horizontal x vertical) display. This size was selected because it is sufficient to allow a natural-feeling perception of a pedestrian's direction and distance when a pedestrian ahead of the vehicle is displayed as described above.

The display is located in the lower front of the driver, so that the angle when looking down is 5 degrees. The focus point of the image is approximately 2 m forward, near the

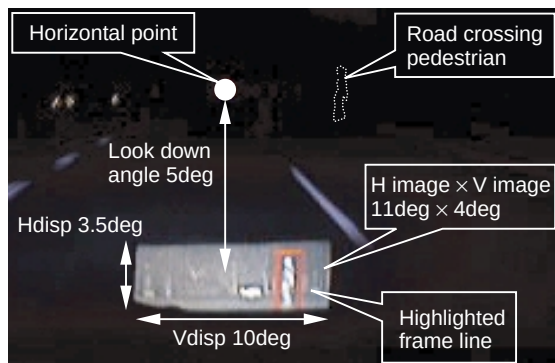


Fig. 16 Projected images of HUD



Fig. 17 Appearance of the HUD

front edge of the hood.

The brightness of the visual image is 0-50 cd, and the color of the detected pedestrian's highlighted image is an easily-recognizable red. (See Fig. 16.) An exterior view of the prototype HUD is shown in Fig. 17.⁽⁶⁾

5. Test results

5.1 Object detection results

Two verification tests were carried out to determine whether the prototype system satisfied the detection performance parameters as they had been set.

5.1.1 Relative velocity calculation test

Fig. 18 shows the results of relative velocity calculations conducted when driving at 60 km/h toward a stationary object. (The horizontal axis shows the distance to the object, and the vertical axis shows the relative velocity error.)

Moving at a speed of 60 km/h, the forward distance to the object at 3.5 seconds before collision is approximately 60 m. Objects to be detected should have a relative velocity within 60 ± 30 km/h, as determined by equation (9).

The relative velocity calculation for a stationary object at a distance of 60 m has an accuracy within ± 25 km/h, as shown in Fig. 18. Therefore, a pedestrian walking at a velocity of no more than ± 5 km/h will have an estimated relative velocity calculation accuracy within ± 30 km/h. This will satisfy the detection condition of a relative velocity within 60 ± 30 km/h.

Similar tests were also conducted at cruising speeds of 40-80 km/h, and these confirmed that the system could detect a pedestrian 3.5 seconds before collision.

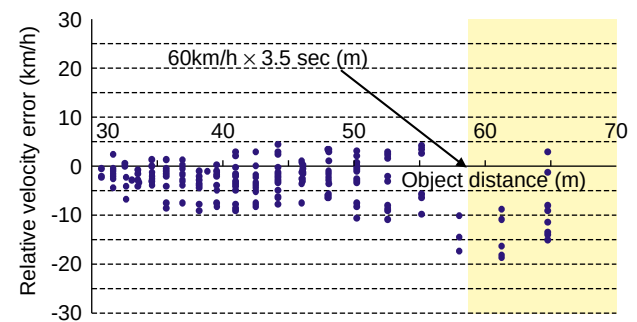


Fig. 18 Relative velocity error of the object distance

5.1.2 Relative moving vector calculation test

Fig. 19 presents the results from calculation of the relative moving vectors of a pedestrian crossing the road at approximately 1 m/s and a stationary object (a streetlight) approximately 60 m ahead of a vehicle traveling at 50 km/h.

From calculation of the relative moving vectors, the crossing velocity of the pedestrian was 1.02 m/s, and the velocity of the streetlight was 0.08 m/s. This shows that the lateral moving vectors are being calculated with an accuracy within ± 0.1 m/s.

The calculation of the relative moving vectors at a point 3.5 seconds before collision and approximately 50 m away yielded the result of 40 km/h. This satisfies the conditions of equation (9), and shows that the system is capable of detection and judgment of pedestrians crossing the road.

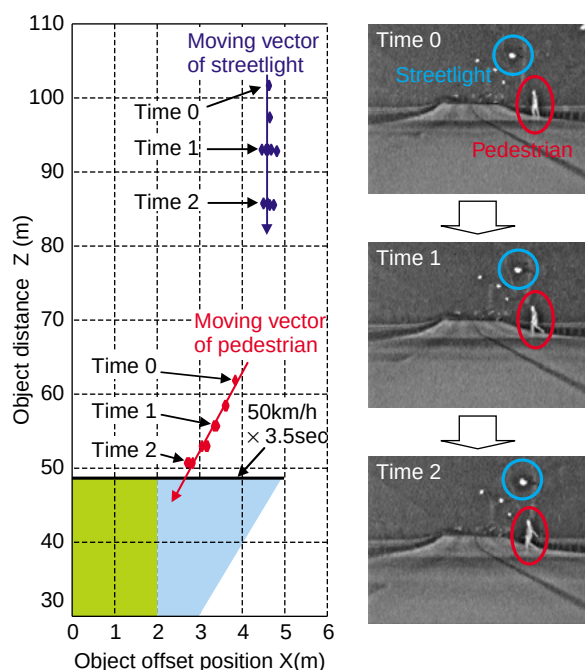


Fig. 19 Detected moving vector and infrared images

5.2 Evaluation of the information provision method

Nighttime driving conditions that make obstacles difficult to see were reproduced on the test course, and multiple techniques for providing information about obstacles were implemented. Comparisons were then done of the timing of the start of the collision avoidance action for each technique.

- (1) Comparison of the information provision methods
Voice guidance only and voice plus highlighted infrared image
- (2) Comparison of the image display positions
LCD on the instrument panel and the HUD (See Fig. 20.)

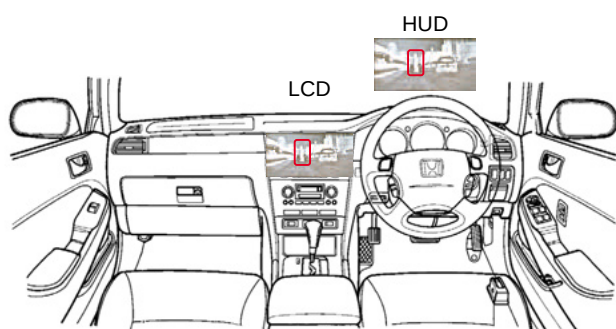


Fig. 20 HUD and LCD display position

5.2.1 Test methods

Fog lamps were used as the only illumination to limit the nighttime range of vision to 20-30 m, and the test subjects were instructed to cruise at a constant speed of 40 km/h. Information was then provided on the obstacles. Obstacles were placed randomly at two locations, one directly in front of the vehicle and one on the centerline, in order to avoid a simple reaction when the information was provided.

Information provision was implemented at a distance of 40 m (3.6 seconds before collision). The test conditions are shown in Fig. 21.

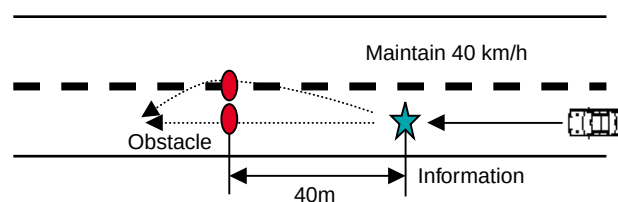


Fig. 21 Interface test condition

5.2.2 Test results

Collision avoidance actions varied with the test subjects. Therefore, the faster of steering avoidance and braking avoidance was selected as the timing for the start of collision avoidance action (response time). Fig. 22 shows the comparison results for the different information provision methods, and Fig. 23 shows the comparison results for the different display positions.

The results from the two tests show that collision avoidance could be initiated about 1 second in advance with voice plus HUD. With voice only and with LCD display, however, the timing for the start of collision avoidance was 1 second or more in many cases. This indicates that voice plus HUD providing a highlighted infrared image is more effective with respect to collision avoidance by the driver.

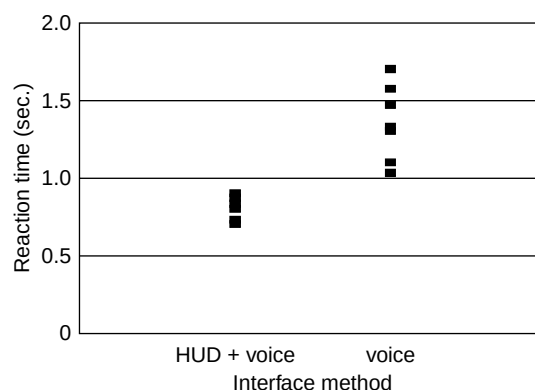


Fig. 22 Comparison of voice and IR image with voice

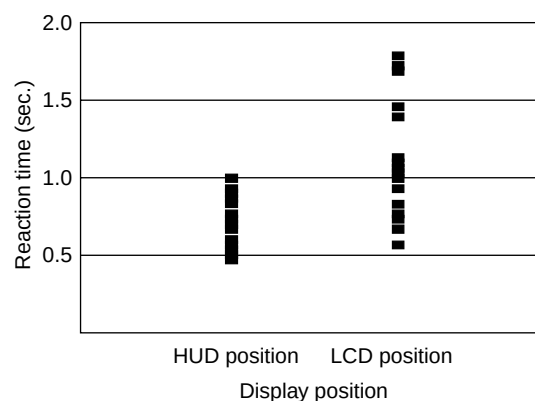


Fig. 23 Comparison of HUD and LCD positions

6. Conclusion

Based on an analysis of nighttime vehicle-pedestrian accidents, we created a system that uses infrared stereo cameras, an image processor, voice guidance and HUD to notify drivers who are driving at night of the presence of pedestrians on the road and crossing the road. This project yielded the following two results:

- (1) We created an algorithm to judge the possibility of collision with high temperature obstacles that can be perceived as pedestrians by means of infrared stereo cameras and image processing. We determined that this enables judgment of the possibility of a collision 3.5 seconds in advance at driving speeds of 40-80 km/h.
- (2) We created an interface that assists in the detection of pedestrians and initiates the possibility of collision avoidance within 1 second from information provision by using voice guidance and a HUD to display a highlighted infrared image that allows comparison of the virtual image to the real roadway scene observed by the operator.

Further development is needed to bring this kind of system to the point of practical application.

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Rear-end Collision Velocity Reduction System

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ABSTRACT

Through the use of traffic accident statistics and questionnaire surveys, some causation factors for rear-end collisions were obtained. Next, accident analysis was performed through simulations based on these factors. The results indicated that many rear-end collisions are caused by momentary inattention during daily driving. Therefore, a collision avoidance assist system was developed that can be useful in these situations. In developing this system, a warning timing that does not interfere with the driver was set so as to reduce the potential for a less effective warning, and a system structure was adopted where delays in recognition and judgment are compensated for by automatic braking. The automatic brake activation method was investigated through calculations and actual vehicle tests. The activation timing was set so as not to interfere with normal driver operations, and the deceleration rate was set to assist the driver's avoidance operations. As a result, the basis for a system to help reduce rear-end collisions was developed. Further development is needed to bring this kind of system to the point of practical application.

1. Introduction

Various systems have been proposed by different companies for the Advanced Safety Vehicle (ASV) project promoted by the Ministry of Transport (since 2001 known as the Ministry of Land, Infrastructure and Transport). These proposals include the use of radar and other technologies for sensing the external environment to determine conditions in the surroundings and make judgments accordingly to help reduce traffic accidents from occurring or to reduce accident severity. The ASV project is now entering its second phase.

Systems using radar, cameras, and other such external environment sensing technologies to predict collisions with preceding vehicles and apply automatic braking to help reduce collisions from occurring have been proposed since Phase 1 of the ASV project. However, due to technical and conceptual difficulties such systems have not reached the point of practical application.

Instead of total automatic collision avoidance, the present system concept is a collision avoidance assistance system centered on the driver and limited to applications to rear-end collisions. This paper reports on the development of this system.

2. General accident analysis

First we analyzed 1997 traffic accident data. The analysis of traffic accidents involving death or injury in 1997 shows that rear-end collisions make up approximately 30% of the

total accidents (Fig. 1).⁽¹⁾⁽²⁾ Then the characteristics of rear-end collisions were analyzed.

Fig. 2 presents the frequency of rear-end collisions in relation to the speed at which the driver recognizes the danger. This indicates that more rear-end accidents occur at low and medium cruising speeds. Fig. 3 shows the human factors of the persons principally concerned who caused the accident. This data indicates that recognition failure caused by inattention to the conditions ahead is often the cause of

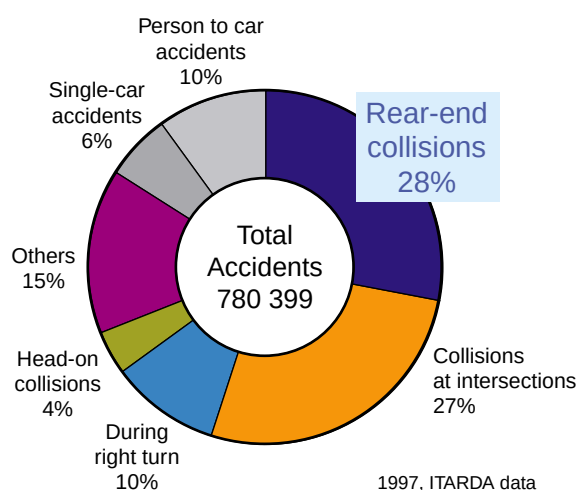


Fig. 1 Composition of traffic accidents per year (1997)

* Tochigi R&D Center

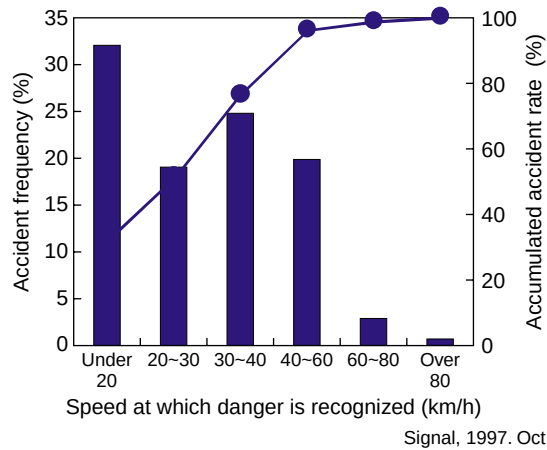


Fig. 2 Frequency and accumulated rate of rear-end collisions in relation to the speed at which danger is recognized

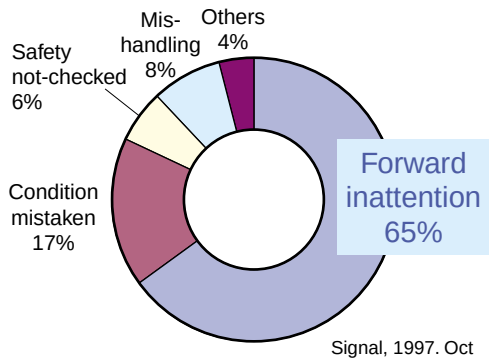


Fig. 3 Factors in rear-end collisions

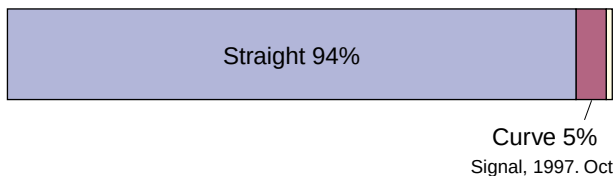


Fig. 4 Road situation during rear-end collisions

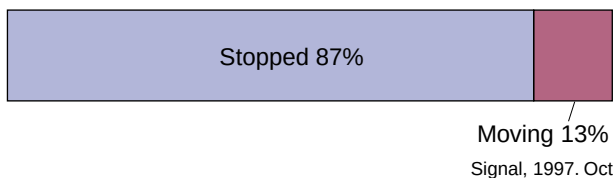


Fig. 5 State of vehicle ahead during rear-end collision (according to statistics)

rear-end collisions. Fig. 4 shows the road type at rear-end collision locations, which indicates that approximately 90% of these accidents occur on straight roads while the number occurring on curves is extremely small. Fig. 5 indicates the state of the preceding vehicles involved in these rear-end collisions. Approximately 90% of rear-end collisions are with stationary vehicles.

The above accident analysis suggests that a common scenario for the occurrence of rear-end collisions is delayed recognition of a preceding vehicle due to inattention to the conditions ahead while driving at low to medium speeds on a straight road.

3. Analysis of accident conditions

3.1 Questionnaire survey

We conducted a questionnaire survey to investigate the conditions that were involved when actual accidents took place.

The survey was provided to 57 people within Honda who had actually caused rear-end collisions and people who had nearly such accidents.

We began by inquiring about the state of the vehicles hit, and learned that collisions most commonly occurred while the vehicle ahead was still moving (Fig. 6). This contradicts the previous results shown in Fig. 5. Further analysis of the accident conditions indicated, as can be seen in Fig. 7, that approximately half of the cases cited “delay in recognition of deceleration” and “greater deceleration than

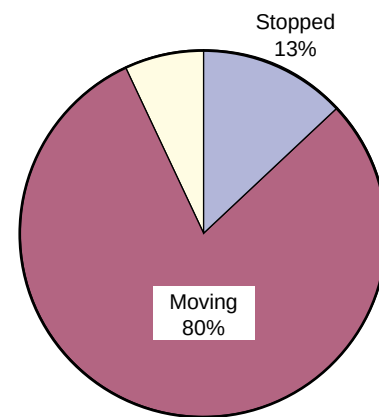


Fig. 6 State of vehicle ahead during rear-end collision (according to questionnaire)

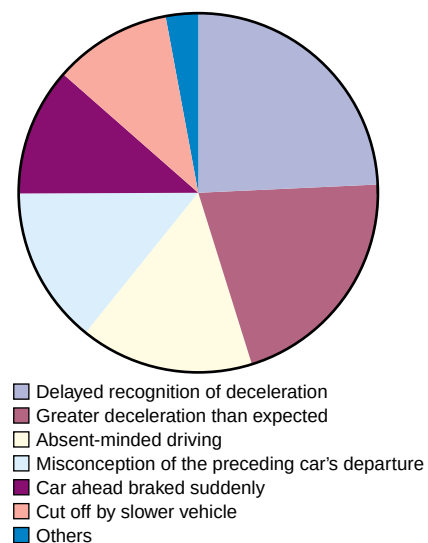


Fig. 7 Factors involved in rear-end collisions (according to questionnaire)

expected” as factors in causing or nearly causing a collision. In other words, the cause was not that the driver had failed to watch the road ahead for too long a time. Instead, that the driver knew there was a vehicle ahead but neglected to pay sufficient attention, and thus caused or nearly caused a rear-end collision.

3.2 Simulation analysis

An analysis was then carried out to determine the degree of inattention that resulted in these accidents, taking into account the general accident analysis and the questionnaire survey results. That is, it could be considered understandable if a rear-end collision followed some degree of prolonged inattention due to, for example, drowsiness. The large number of accidents, however, made it unlikely that the majority of drivers were experiencing such prolonged inattention, nor did the questionnaire survey uncover anything to support such a factor.

Consequently, we used simulation to investigate the possibility that the accidents could have been caused by short-term or momentary inattention.

Under the simulation scenario, we hypothesized that drivers were momentarily inattentive as they followed the preceding vehicle, which decelerated during this very moment of inattention. This possible scenario was determined based on the questionnaire, according to which the majority of accidents occurred under these circumstances.

The scenario conditions specified a vehicle headway time in relation to the preceding vehicle (equal to the distance between the vehicles divided by the subject car speed) of 1.5 seconds, with the accident occurring when the vehicle ahead decelerates (deceleration of 0.3 G) in the instant when the subject car driver is inattentive. After a set period of 1, 1.5, and 2 seconds of inattention, the driver pays attention forward, recognizes what is happening, makes a judgment, and acts to decelerate. The driver’s response time (the time from the start of attention to the beginning of deceleration) at this point is set at 1.3 seconds.⁽³⁾⁽⁴⁾ At the time an action to decelerate is taken, the deceleration is 0.8 G. The simulation was carried out with three different levels of driver inattention.

Fig. 8 presents the simulation results. The horizontal axis is the speed of the subject car, while the vertical axis is

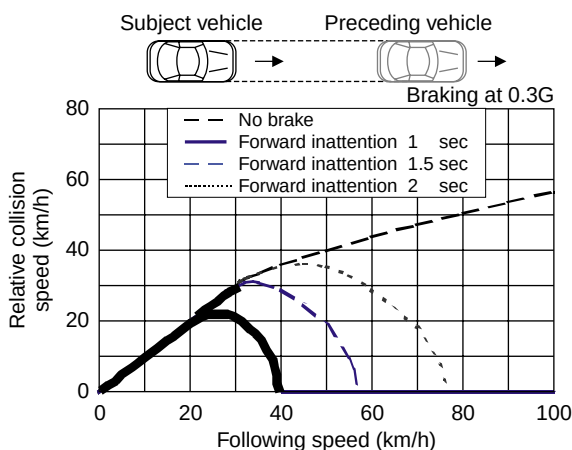


Fig. 8 Results of simulation (while following)

the relative speed at the time of collision. As Fig. 8 indicates, the period of inattention makes a difference. A rear-end collision takes place when driving at low to medium speeds, and a collision does not take place when driving at higher speeds. This result is in line with the results shown in Fig. 2, which shows that the majority of rear-end accidents occur at low to medium speeds. The bold line in Fig. 8 represents the situation when the vehicle ahead decelerates to a complete stop and the subject vehicle collides with it from the rear. In the statistics, an accident that occurs in this way with a travelling vehicle that has come to a stop is included in “Stopped” in Fig. 5. The simulation results suggest that the main causal factor in rear-end collisions is not necessarily a prolonged period of inattention alone. Instead, rear-end collisions can also occur as a result of momentary inattention when driving at low to medium speeds.

Another simulation was then carried out with the scenario of a rear-end collision with a stationary vehicle. This simulation was conducted using a parameter defined as the number of seconds in advance that the driver recognized the stationary vehicle. Fig. 9 shows the results when the driver takes action to decelerate in the specified response time (the 1.3 seconds explained above) after recognizing the stationary vehicle. Examination of these results shows that a rear-end collision will not take place at low to medium cruising speeds when recognition takes place 2 - 2.5 seconds before collision. A rear-end collision will only take place if the driver has been inattentive for a considerable time regarding a vehicle that has already come to a stop, and only recognizes the situation approximately 2 seconds before collision. It seems unlikely that there are many drivers of this kind.

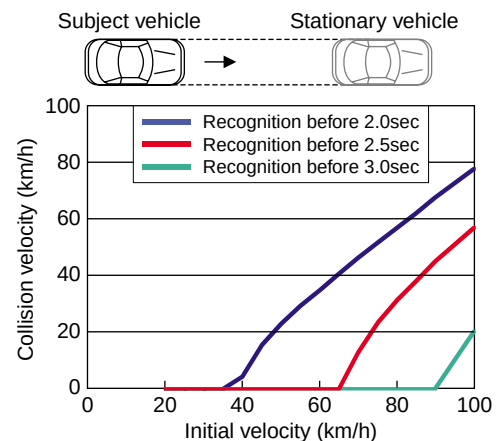


Fig. 9 Results of simulation (with stationary vehicle)

3.3 Period of inattention while driving

An eye camera was then used for measurements to determine the extent of inattention to the conditions ahead during ordinary driving (Fig. 10). This measured the time spent checking for safety, operating the audio system, and performing other actions ordinarily taken during driving that involve looking away from the front. The horizontal axis in the graph shows the forward inattention duration, and the

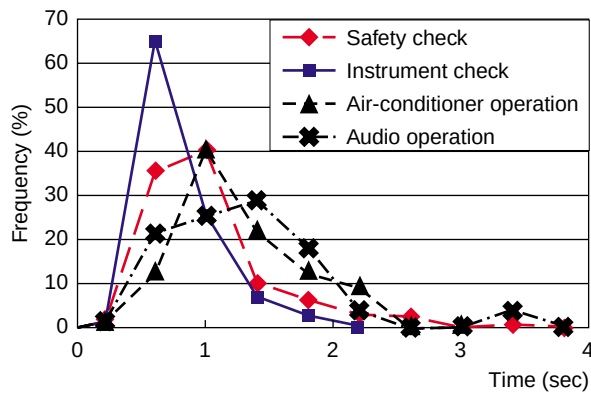


Fig. 10 Forward inattention duration

vertical axis shows frequency of occurrence. As the results in Fig. 10 show, ordinary actions such as operating the audio system or checking instruments can result in momentary inattention of 1 - 2 seconds. It is possible that if the preceding vehicle were to decelerate during the seconds of forward inattention caused by ordinary actions, that a rear-end collision could result. This can be considered the cause of many rear-end collisions that occur at low to medium cruising speeds. Based on the above analysis results, the objective of this research project was set as the development of a basis for a collision avoidance assist system that would be useful in the avoidance of rear-end collisions caused by the inattentiveness of the driver occurring even during normal driving. The system was developed as a collision avoidance assist system rather than an automatic avoidance system in order not to contribute to unsafe driving or invite driver overconfidence.

4. System concepts

4.1 Concepts of a warning and automatic brake system

A warning system was first considered to provide the driver with recognition assistance by means of voice messages or other warnings during times of possible collisions. In this case, it is necessary for the warning to function with timing that allows leeway for the driver to hear the warning, make a judgment, and carry out a maneuver.

Consequently, driver response time to warnings was analyzed using a driving simulator. The test was set up so that the driver would look off to the side while driving behind a vehicle moving ahead. During that time, the forward vehicle would decelerate, and a warning would sound 2 seconds before the predicted collision. Fig. 11 shows the response time that elapsed between hearing the warning sound and initiating braking. From this a response time of about 0.6 - 1.4 second is needed. In the case of a warning system, therefore, it is necessary to sound the warning with a timing that allows for this response time as well as the time required by drivers to decelerate their vehicles.

When extra time is allowed in order to adapt to a larger number of drivers, however, the alarm may sound frequently in normal situations. There is the risk that drivers would then become accustomed to the warning, so its intended

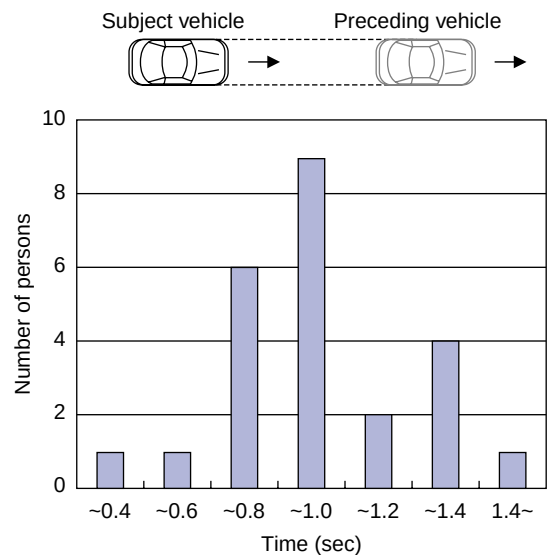


Fig. 11 Driver's response time (by driving simulator)

effect could end up being diminished. On the other hand, when the warning timing is reduced, there is less of a time margin for judgment and action after hearing the warning. Therefore, there is a possibility that the warning will not be fully effective.

This system, therefore, also utilizes automatic braking to further help assist the driver in avoiding a collision. The timing for the activation of the warning was set to try to minimize interference with the driver during normal driving operation so as not to minimize the effect of the warning. The system was also configured with automatic brakes to help compensate for delays in judgment and action by the driver.

4.2 Study of the control timing

The distance required for collision avoidance by braking alone is ordinarily greater than that required for collision avoidance by steering. The attempt to avoid a collision with certainty by means of braking alone, therefore, may in some cases interfere with the driver's evasive steering maneuvers.⁽⁵⁾

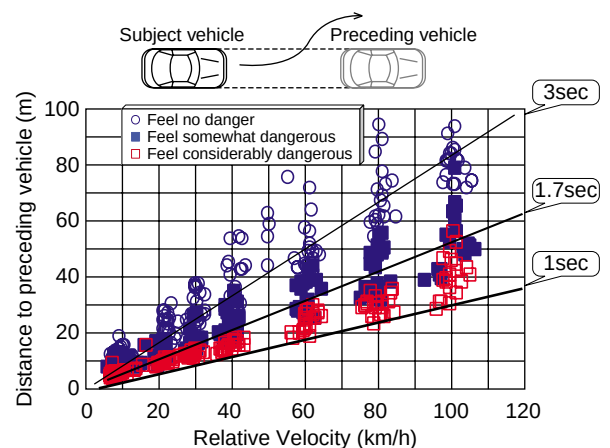


Fig. 12 Avoidance operation timing test

We then studied the timing for system operation by measuring the actual timing of evasive steering actions taken by drivers. Fig. 12 shows the results for measurement of distance from the preceding vehicle at the time evasive steering action is initiated with respect to the car's relative speed of approach to the preceding vehicle. The horizontal axis shows the relative speed and the vertical axis shows the distance at the time avoidance operations are initiated. The symbols in the graph indicate the points at which the driver felt the need for avoidance operations were feel no danger, feel somewhat dangerous, and feel considerably dangerous (at the limit for such operations). The lines in the graph show the margin of time before collision (equaling the distance between vehicles divided by their relative speed).

The above findings led to consideration of the two points below with regard to the timing for activation of a warning in this system. The first was that frequent warnings should not make the driver so accustomed to warnings that the warning becomes ineffective. The second was that drivers should not be encouraged to rely excessively on the warnings to the point that they feel inattention would be permissible until a warning is sounded. Taking this perspective, we decided to set the timing for warnings so that they would not occur frequently during ordinary driving, but would be generated approximately 1.7 seconds before the predicted collision when involving avoidance operations the driver considers somewhat dangerous. The timing for the automatic braking (termed emergency braking) was set to avoid interference with driver operations as much as possible and when involving avoidance operations the driver considers considerably dangerous. The timing was set at approximately 1 second before a predicted collision, which would activate the brakes rarely in ordinary driving.

4.3 Automatic brake setting concept

The timing for the activation of the warning is set at the point where ordinary avoidance operations would be initiated, which is the point where danger of collision is increasing, so the warning must be recognizable by the driver. It was decided, therefore, that the warning should not consist of sound alone, but should also include mild automatic braking to warn the driver through a physical

channel. This was termed warning braking. Various studies and experiments regarding the intensity of the warning braking (its deceleration) were conducted repeatedly, and the appropriate value was determined to be about 0.1–0.2 G. At this setting, the brakes would enhance the warning effect in conjunction with the sound while also compensating for any delay in deceleration operations by the driver. In addition, this setting should not interfere with the driver's ability to perform avoidance steering operations.

The next point to consider was the amount of deceleration by the automatic brakes that operate approximately 1 second before a predicted collision (termed emergency braking). One approach was to apply the maximum amount of automatic braking, since the situation was a predicted collision. However, experiments verified that with the inclusion of deceleration from the use of warning braking, the combined use of deceleration plus evasive steering operations would make it more possible to avoid collision. When setting the amount of emergency braking, therefore, we took into consideration how the activation of emergency braking would alter driver attitude and other factors, and how it would thereby affect the driver's evasive maneuvering. The system is also set up to consider the braking force applied by drivers when they recognize a dangerous situation in addition to the automatic braking force.

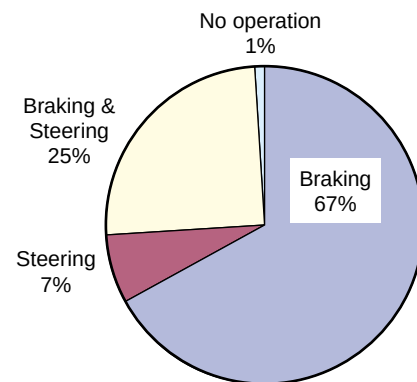


Fig. 13 Avoidance operation during a rear-end collision

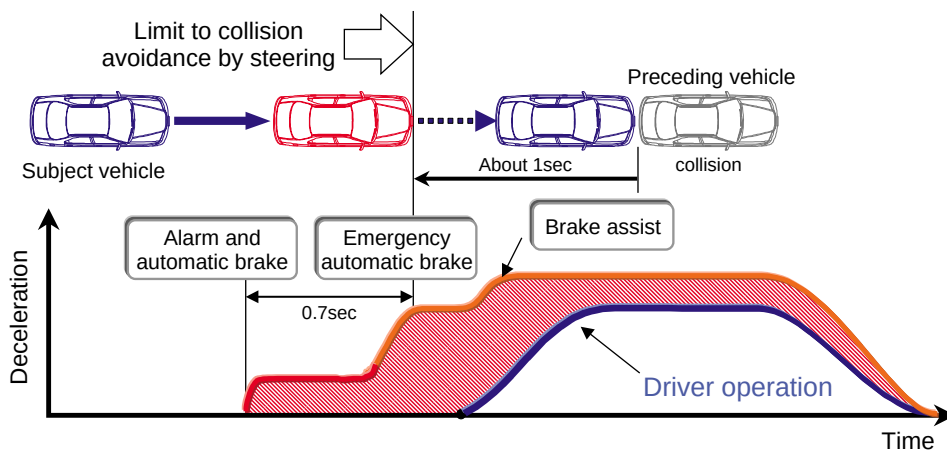


Fig. 14 System operation

The questionnaire results show that when drivers caused or nearly caused a rear-end collision, approximately 70% of them applied the brakes (Fig. 13). The combination of driver operations and automatic braking was, therefore, considered to bring about the appropriate amount of deceleration. Deceleration by automatic braking also increases the margin of time before a collision, thus providing the driver a greater margin of time for judgment and action. This could be expected to have the effect of making the application of the brakes by the driver more likely even for those drivers whose slower response times for making judgments and taking action made them previously less able to apply their brakes. Fig. 14 presents the sequence of system operation modes.

5. System overview

5.1 System configuration

The system configuration is shown in Fig. 15. Millimeter wave radar, which excels at environmentally-resistant sensing, was used for locating vehicles to the front. Table 1 presents the major specifications. The range is about 100 m, which provides detection capabilities approximately 3 seconds in advance at a relative speed of 100 km/h. The detection angle extends 16° to left and right to provide detection of preceding vehicles on straight and gently curving roads.

The system is equipped with a wheel speed sensor to measure the subject vehicle speed and a yaw rate sensor to measure the amount it turns. Using these quantitative vehicle condition values, the system extrapolates the course of the subject vehicle as shown in Fig. 16 and estimates the amount of the subject vehicle's lateral travel at the location of the obstacle. In addition, the system is equipped with a warning

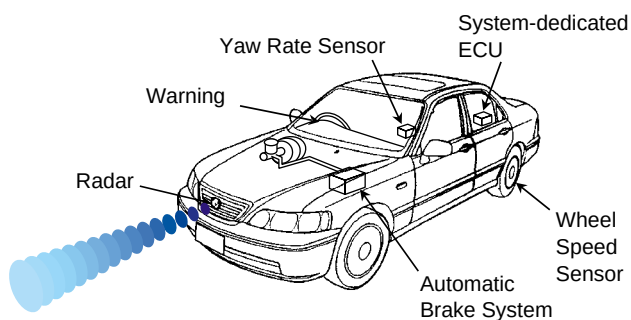


Fig. 15 System configuration

Table 1 Major specifications of the radar sensor

Detection Area	Horizontal Vertical	280 mrad 66 mrad
Maximum Range		100 m
Range Resolution		0.1 m
Range Accuracy		±1.0 m
Sampling Time		0.1 sec

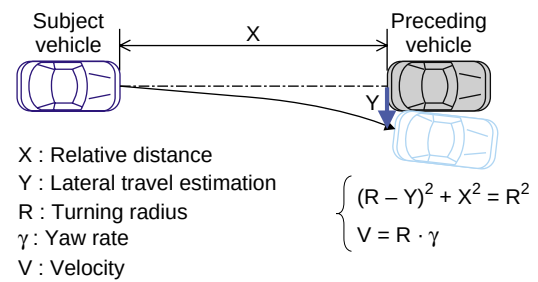


Fig. 16 Prediction of the vehicle locus

unit that warns the driver when the system goes into operation. A hydraulic actuator is also installed to activate the automatic brakes.

5.2 Control logic

Fig. 17 presents the basic flow of system control.

The system recognizes preceding vehicles in the front by radar, and then determines the relative distance to the subject vehicle ($X1$), relative speed, and angle of direction. It also estimates the subject vehicle's course using the speed and yaw rate measurements, and calculates the estimated amount of lateral travel. Then it uses the location of the obstacle and the estimated amount of lateral travel of the subject vehicle to determine the amount of overlap between the subject vehicle and the obstacle, after which it determines the amount of lateral travel (Y) required by the subject vehicle so that it can be steered away to avoid the obstacle. The system then calculates the relative distance ($X2$) required to carry out that lateral travel. The calculated distance and the relative distance to the obstacle are used then to judge

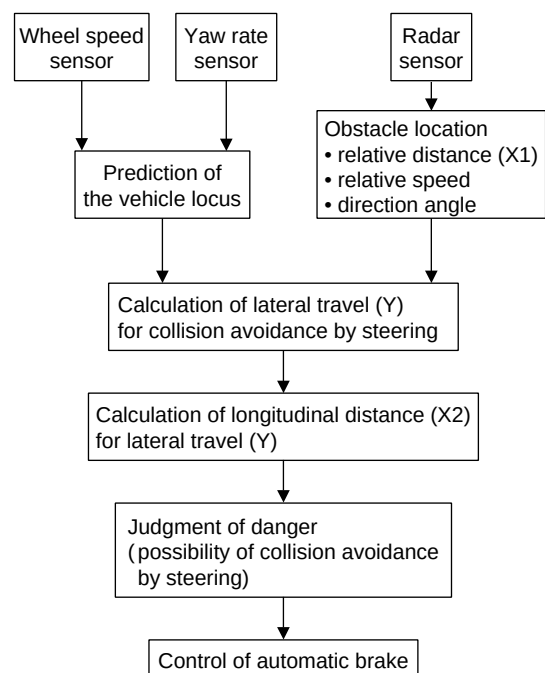


Fig. 17 Block diagram of basic control

the danger and carry out automatic braking control and warning. When $X1$ becomes shorter than $X2$ ($X1 < X2$), automatic braking is activated.

With the physical relationship between the subject vehicle and the preceding vehicle determined in this way, the lateral amount of avoidance achieved by steering will differ between situations when the subject vehicle is located immediately behind the preceding vehicle (full overlap) and when it is located in an offset location. Therefore, the system makes its judgment by replacing this difference in the lateral amount of avoidance by steering with the danger value.

6. System effectiveness

6.1 Simulation results

Simulation was used to demonstrate the system's usefulness in operation. A comparison with system operation using only the warning (by sound) was also carried out in order to determine the additional effectiveness of automatic braking. The response time for driver brake operation after hearing the warning was set at 1 second, which is the average value for the response times with the driving simulator shown in Fig. 11. The emergency braking activation deceleration when the system goes into operation was set at 0.6 G.

The simulation used a scenario in which the preceding vehicle decelerates with the subject vehicle behind it. The simulation conditions specified a vehicle headway time of 1.5 seconds, when following the preceding vehicle, and a 0.3 G deceleration by the preceding vehicle. The deceleration of the preceding vehicle generates a relative speed between it and the subject vehicle; as the relative distance between the vehicles decreases the danger of collision increases. At a point in this range, the system initiates operation. The driver is considered to operate the brakes 1 second after hearing the first warning sound, and the deceleration at the time of driver operation is 0.8 G. The results are shown in Fig. 18. The horizontal axis is the speed of the subject vehicle when following, and the vertical axis is the relative speed at the time of collision. Figure 18 indicates that rear-

end collisions can be reduced when driving at low to medium speeds, and that the collision speed is lowered at higher speeds, when the system has automatic braking. These results and the results in Fig. 2 suggest that it is possible to reduce rear-end collisions caused by momentary inattention.

When the simulation allowed only for the warning (by sound) without any automatic braking, the collision speed is lowered by a smaller amount than when the system has automatic braking, even when assuming the same driver response time (1 second) for applying the brakes after hearing the warning (by sound). Use of only the warning system, therefore, requires earlier warning timing, confirming the usefulness of the addition of the automatic braking system.

6.2 Actual vehicle test results

The next step was to use an actual vehicle to measure the collision speed reduction effect with respect to a stationary object. The test method involved placing an obstacle representing a vehicle directly in front of the subject vehicle, which drives toward it. The system initiates operation when the danger of collision increases. Here, too, the driver carries out braking operations approximately 1 second after the warning. Other system's parameters are the same as the simulation.

The test results are shown in Fig. 19. The horizontal axis is the relative speed (subject vehicle speed), and the vertical axis is the speed at the time of collision. These results indicate that rear-end collisions can be reduced at low to medium driving speeds, where the relative speed is low, and collision speed is reduced at higher speeds, where the relative speed is high.

In both the simulations and the actual vehicle tests, the data indicates that the majority of rear-end accidents occur at low to medium speeds, as shown in Fig. 2. Therefore, this system can be considered to have met the developmental objective of this research project, which was the development of a basis for a collision avoidance assist system that would be useful in the avoidance of rear-end collisions caused by the inattentiveness of the driver.

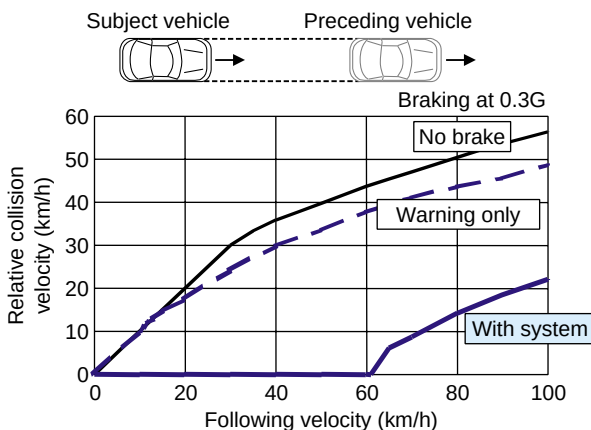


Fig. 18 System effect (according to simulation)

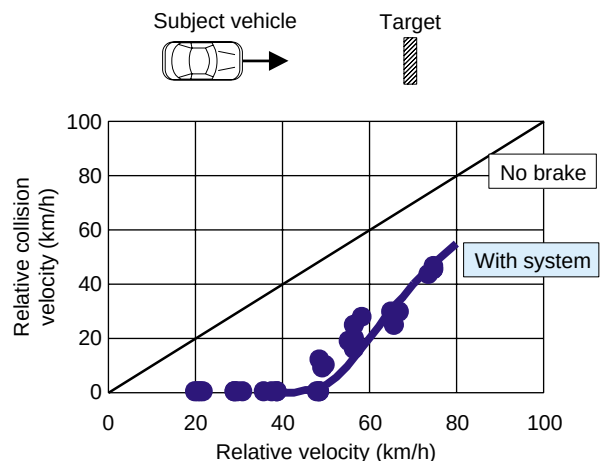


Fig. 19 System effect (according to test results)

7. Conclusion

We carried out development of a system that uses millimeter wave radar technology to sense the external environment in conjunction with a warning and automatic braking system. This made it possible to compensate for delays in the driver's recognition, judgment, and action, making the system useful in reducing rear-end collisions.

In developing this system,

- (1) we found that rear-end collisions can occur during ordinary driving as a result of momentary inattention,
- (2) a system was developed to reduce interference with driver operations and support avoidance operations, and
- (3) it was determined that this system is useful in reducing the number of rear-end collisions that take place at low to medium speeds.

However, other issues such as mistaken operation and operational failure also must be analyzed. It will also be necessary to carry out further verification of system reliability under a variety of environmental conditions. In addition, we must take into account changes in awareness on the part of the system users. This present system is a collision avoidance assist system that is intended to do no more than compensate for driver delays in deceleration operations. It is not an automatic avoidance system. It should not have the contrary effect of encouraging unsafe driving by leading drivers to rely entirely on the system. Furthermore, the system cannot prevent rear-end collisions in all situations. Therefore, before bringing this kind of system to the point of practical application we need to address these issues.

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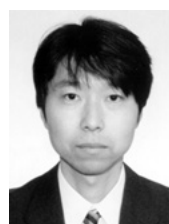
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Development of 7003 Aluminum Extrusions for Motorcycle Frame Applications

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ABSTRACT

The 7003-T4 alloy as an acceptable material for production of a motorcycle frame was developed. In the development of the material, a designed experiment was planned by which nine 7003 alloy variants were cast and homogenized. The cast billets were extruded into hollow frame tubes on a production extrusion line and the natural aging response of the material was evaluated. The tendency for recrystallized grains at the surface of the extruded tubes was measured and confirmed to be at an acceptable level for painting. The reversion and recovery of mechanical properties during paint curing was evaluated and curing limits were established for mass production. From the results, parametric equations were derived to describe the relationships between mechanical properties, composition and natural aging time to insure design compliance. This development has resulted in a frame which is 35% lighter than the tubular steel construction applied to previous models.

1. Introduction

In order to reduce vehicle weight and meet the demands of the customer, a new dual spar aluminum frame design has been developed for the 2001 Honda Goldwing shown in Figure 1. A suitable U.S. aluminum alloy, one recognized by the U.S. aluminum industry and capable of performing according to the design requirements, had to be developed for extrusion into hollow tubes for the frame body.

A 7003 Al-Zn-Mg alloy was developed for extrusion into frame tubes for the new Goldwing. Customized alloys based on the 7003 chemistry are currently used to produce frame tubes found on Honda sport class and dirt motorcycles built in Japan. The development of these alloys resulted in restricted chemical compositions to optimize mechanical properties and limit the recrystallized layer depth. The latter control being especially important for those frames having an anodized surface finish. In order to establish a cost merit for the extrusion of aluminum tubes in the U.S., restrictions on the alloy composition had to be avoided. It was decided to apply a standard 7003 alloy without restrictions on the

chemical composition. The standard 7003 alloy is compared to the Honda custom alloy in Table 1. Several designed experiments were conducted as a means to define the aging relationships for the standard alloy. Based on the results of the experiments, the material was found to meet the design requirements, and paint curing guidelines were successfully



Fig. 1 2001 Honda Goldwing with aluminum frame body

Table 1 Standard alloy vs. Honda custom alloy compositions

Alloy	Si	Fe	Cu	Mn	Mg	Cr	Zn	Zr	Ti	Al
Standard 7003	0.30 max.	0.35 max.	0.20 max.	0.30 max.	0.5-1.0	0.20 max.	5.0-6.5	0.05-0.25	0.20 max.	Bal.
Custom alloy	0.05-0.20	0.15-0.30	0.05-0.15	0.20-0.30	0.50-0.80	0.05 max.	5.5-6.1	0.15-0.25	0.01-0.06	Bal.

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established. This paper describes the procedures used to develop the 7003-T4 alloy as an acceptable material for production of a lightweight, high strength motorcycle frame. The authors are hopeful that this data is a useful benchmark tool for designers and engineers considering extruded aluminum tubing for similar applications.

2. Description of 2001 Honda Goldwing Frame

2.1 Design Requirements

The frame is a welded assembly consisting of extruded tubes and aluminum die castings. The dual spars or main tubes which extend from the head pipe to the pivot plates on either side, are multi-void extrusions having a 3mm outer wall thickness. These tubes serve as the primary load bearing structures in the welded assembly and are the focus of this material development. The down tubes are dual-void extrusions that extend from the head pipe to the front engine bracket locations. The tubes comprising the rear frame are round hollow extrusions which support the seat and saddle bags.

The lightest frame is achieved by minimizing the wall thickness of each extruded tube. Wall thicknesses are based on the mechanical strength of the aluminum and in the case of the new frame, a custom alloy served as the basis for calculating the thickness targets. The standard 7003 alloy had to perform as well as the custom alloy during tensile tests to prevent thickness increases leading to a heavier frame. The development of the 7003 material, resulted in a frame body having an average weight of 21kg. Although the 20kg target was slightly exceeded, a 35% decrease in frame weight compared to the welded tubular steel construction found on the previous Goldwing was realized. The frame is shown in Figure 2.

Exposing a bit of the frame body provides a mechanical personality but requires that the exposed area have a high quality surface appearance. Several surface finish options were considered at the onset of development. The original

idea was to apply a bright anodized surface for a shiny metallic appearance. The anodized option requires special control over the depth of the recrystallized layer which generates at the die interface during the extrusion process. Since grain growth following recrystallization is unrestricted at the surface, the surface grains become large and the surface appears coarsened⁽¹⁾. This coarsening condition is called "orange peel" and an example is shown in Figure 3. Orange peel becomes more noticeable when the grains are strained during bending. Anodizing also enhances this effect, so a powder paint system was chosen as an alternative to prevent strict control over the recrystallized layer depth and increased extrusion costs. The paint would sufficiently cover the recrystallized surface and provide good corrosion resistance. Control over the depth of the recrystallized layer was still required, but to a lesser degree. The frame material design requirements are given in Table 2.

2.2 Material Selection

Initially, several alloys were considered for the application. The selection criteria are shown in Table 3.

6061 is commonly applied to automotive structural applications where strength, corrosion resistance and weldability are required. Although 6061 is easily arc welded, the room temperature hardness recovery in the heat affected zone is not good and adequate strengths could not be achieved.

7075 was also briefly considered. The high Zn and Mg content in this material promote significant strengthening during natural aging. The Cu content (typically near 1.6%) is a disadvantage since increasing Cu widens the melting range and reduces the solidus temperature⁽²⁾. The effect is

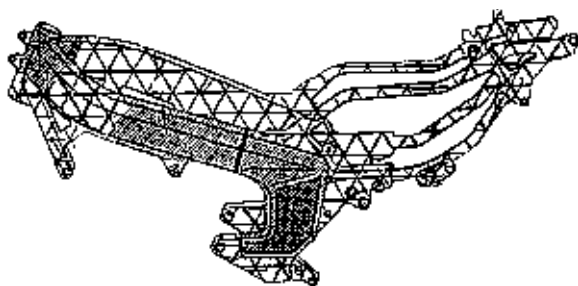


Fig. 2 Frame body construction

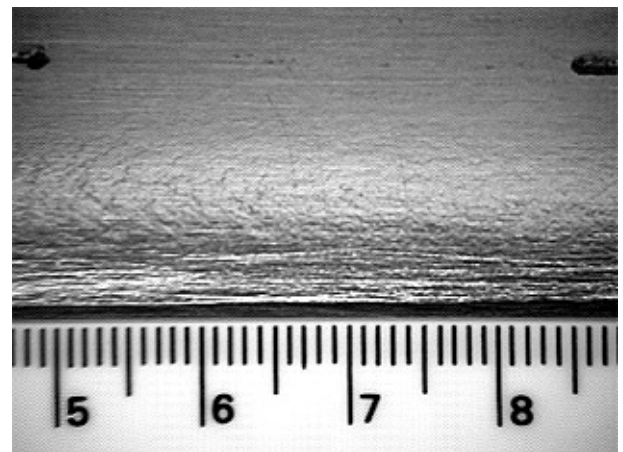


Fig. 3 Orange peel effect

Table 2 Frame material design requirements

Criteria	7 day aging	30 day aging	90 day aging
Tensile strength (MPa)	235 min.	275 min.	294 min.
Yield strength (MPa)	118 min.	176 min.	186 min.
Elongation (%)	13 min.	13 min.	13 min.
Recrystallized layer depth (mm)	25 max.		

Table 3 Aluminum alloy selection criteria

Criteria	Honda Custom Grade	6061-T6	*7003-T4	7075-T6
Tensile strength (MPa)	235	310	324	570
Yield strength (MPa)	118	275	187	505
Elongation (%)	13	12	19	11
Youngs modulus (GPa)	71	69	71	72
Extrudability (hollows)	Good	Excellent	Good	Poor
Formability	Fair	Fair	Fair	Poor
Weldability	Good	Excellent	Good	Poor
Hardness recovery after welding	Good	Poor	Good	–
Availability	Poor	Excellent	Fair	Excellent
**Material costs	\$5.00/kg	\$3.88/kg	\$4.77/kg	No data

*Typical properties after standard aging 7 days at RT

**Includes casting and conversion. Shipping costs are included for the Honda Custom Grade.

to increase the potential for cracking during arc welding. Extruding 7075 into shapes having thin wall sections is also difficult and distortion following press quenching is eminent. Based on these producibility issues, 7075 was eliminated as a candidate for this frame tube application.

The 7003 alloy was selected as the best option for production of the new frame tubes. The Zn and Mg content provide sufficient response to natural aging although the allowable ranges are wider than those of the Honda custom alloy. The low Cu content (0.2% maximum) narrows the melting range and increases the solidus temperature which improves resistance to weld cracking when a high Mg filler metal is applied⁽²⁾. The custom material was developed with anodized frames in mind and Cr promotes yellowing of the anodic oxide surface layer, therefore Cr is controlled to a low level (0.05% maximum) to improve the appearance of the anodized frames. The new Goldwing frames will be painted, so the yellowing effect from Cr is not relevant and higher levels may be tolerated. The standard 7003 was found to meet the material requirements and was chosen for production of the frame tubes.

2.3 Manufacturing Sequence

The manufacturing sequence required to create the frame body is shown in Figure 4. First, round ingots, 178mm in diameter and 1524mm long are continuously cast and homogenized. The heads and tails are cropped to eliminate extreme variations in composition and microstructures. The ingots are heated to 510 °C and direct extruded into multi, dual void, or hollow tubes. As the tubes exit the extrusion dies, they are quenched and the natural aging cycle begins. The extruded tubes are next cut to length and formed by bending into the appropriate shapes. Minor machining and drilling is performed and the tubes are prepared for welding.

The structure takes shape when the cast head pipe, pivot plates and engine brackets are joined to the frame tubes by manual arc welding. The frames pass through a paint booth where a polyester urethane powder is applied. Finally, the powder is cured at 210 °C for 48 minutes and the frames are ready to be assembled into motorcycles.

3. Experimental

3.1 Basis

The aging of rapidly quenched Al-Zn-Mg alloys from room temperature to relatively low aging temperatures is accompanied by the generation of solute rich zones known as GP (Guinier-Preston) zones⁽¹⁾. Strengthening is the result of interference between moving dislocations and these spherical shaped zones. With increasing aging time, GP zones increase in size, and the strength of the alloy increases⁽¹⁾.

Extended aging at temperatures above room temperature transforms the GP zones in alloys with relatively high Zn-Mg ratios into the equilibrium precipitate known as $MgZn_2$ or M phase⁽¹⁾. In the case of the 7003 alloy where Zn is in excess of Mg, the sequence is in the order GP zones, coherent precipitate, and M phase ($MgZn_2$)⁽¹⁾. Heating and cooling of the extruded tubes during production of the frame body changes the mechanical state of the alloy according to this aging sequence. In the case of the painting process, the stable tubes in the T4 condition, are exposed to moderately high temperatures during the curing step. This exposure causes the sequence to reverse which decreases the hardness, and therefore the strength, of the tubes. This phenomenon, known as reversion, is caused by Zn and Mg diffusion back into the solvent aluminum and subsequent shrinking of the GP zones.

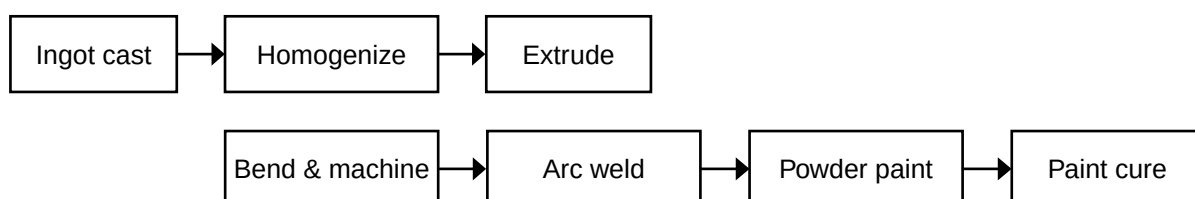


Fig. 4 Manufacturing process

Table 4 3x5 design experiment: Chemical variants

Chemical variants		Magnesium (%)		
		0.5	0.75	1
Zinc (%)	5.00	A1	A2	A3
	5.75	A4	A5	A6
	6.50	A7	A8	A9

3.2 Experimental Procedures

3.2.1 Mechanical Properties vs. Natural Aging

Relationships between mechanical properties, composition and natural aging time were empirically developed.

The increase in strength during natural aging is a result of the growth of GP zones rich in Mg and Zn solute atoms, therefore these elements were chosen as factors and adjusted to produce nine variants covering the entire 7003 chemical range in the standard alloy. The chemical variants are shown in Table 4. Time and temperature were chosen as the aging factors. These levels were set at 7, 30, and 90 days. With production locations in mind, the aging temperature levels were set at -10 °C, 25 °C and 50 °C to reflect possible natural aging temperature scenarios.

Tensile coupons were machined from the sample tubes representing each of the nine chemical variants described in Table 4. Coupons were made from tubes representing the front, middle and rear of each cast ingot in order to capture inherent process variation. The aging start time ($t=0$) was re-established by solution heat treating the coupons in a small electric furnace at 435 °C for 1 hour and quenching in air to represent press quenching. All tensile coupons were produced according to ASTM B557. The duplicate flat coupons had a gage length of 50mm and the testing was conducted according to ASTM E8.

3.2.2 Recrystallization Tendency

Temperatures at the material/die interface are high enough during press quenching to cause simultaneous recovery, recrystallization, and grain growth. Surface quality of the extruded tubes is dependent on the depth of the recrystallized layer and the size of the recrystallized grains at the surface. Coarse grains, which become strained during bending produce an undesirable "orange peel" effect. Controlling recrystallization during extrusion is crucial for anodized frames, since anodizing amplifies the orange peel effect. The selection of powder paint reduces the necessary control over the recrystallized layer depth, however it does not eliminate orange peel as a factor which reduces appearance quality. In order to estimate the recrystallization tendency in the 7003 alloy, a section from each of the alloy variants was taken normal to the extrusion direction. The sections were polished, etched to reveal the recrystallized layer, and observed under a light microscope.

3.2.3 Reversion and Recovery Evaluation

The paint curing cycle was established for mass production by determining the effect of curing temperature and time on the reversion and recovery of mechanical properties.

The painting process consists of pretreatment, powder paint application, and paint curing stages. The gas fired

Table 5 Reversion and recovery experiment

Process variable	Normal production		
	Trial 1	Trial 2	Trial 3
Line speed (m/min.)	1.22	1.83	1.83
Pretreatment (°C)	177	177	177
Zone 1 (°C)	199	199	210
Zone 2 (°C)	199	204	210
Total cure (mins.)	60	48	48

curing oven contains two zones, each having independent temperature control. Frames are placed on hangers and transferred through the curing oven via an overhead conveyor. Curing temperature and time are partially determined by the paint selection since each paint is delivered with a recommended cycle developed to provide the best adhesion and appearance for a given substrate. In the case of the Goldwing frames, a polyester urethane resin material was selected. The best adhesion and appearance is achieved by curing between 182 °C and 216 °C for at least 15 minutes. The curing time is controlled by the speed of the overhead conveyor carrying the painted frames into the curing oven. Minimizing curing time results in maximum line speed and production output but requires a temperature setting near the high side of the supplier's recommended range.

Three trials were conducted in order to establish the optimum curing cycle under normal production conditions. The variables for each trial are given in Table 5. In each trial, the pretreatment temperature was fixed at 177 °C and painting was done according to mass production guidelines. Upon completion of each trial, the test frames were inspected for paint adhesion and appearance. Reversion and recovery of mechanical properties was quantified by measuring the change in hardness in a section cut from the main tube of each test frame over a period of 50 days.

4. Experimental Results

4.1 Mechanical Properties vs. Natural Aging

4.1.1 Composition Effects

The 7 days aging condition was chosen to illustrate the relationship between mechanical properties and composition. One week is the typical time between press quenching and frame welding during production, so it is convenient to express the strength of the formed tubes after natural aging for 7 days. The results of tensile strength, yield strength, and elongation averages are shown in Figures 5, 6, and 7. The data shows that tensile and yield strength targets, set by design, can be achieved across the entire chemical range when natural aging occurs at room temperature or above. The aging sequence is slowed significantly when freezing temperatures are encountered so that the 7-day strength targets are only achieved when the Zn and Mg percents are at the high end of the specification range. Acceptable elongation is also achieved for all chemistries. A premise for using the standard 7003 alloy was that custom chemical control be avoided. Based on the results, customizing the alloy composition is not required.

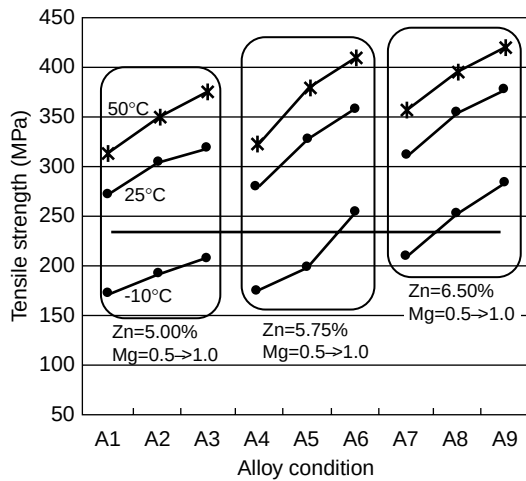


Fig.5 Copmosition impact on tensile strength at -10 °C, 25 °C, 50 °C

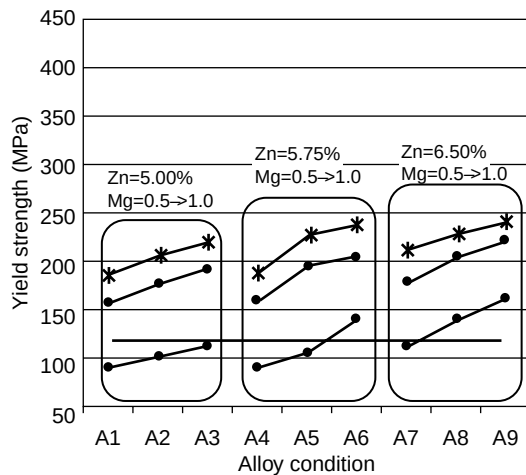


Fig. 6 Composition impact on yield strength at -10 °C, 25 °C, 50 °C

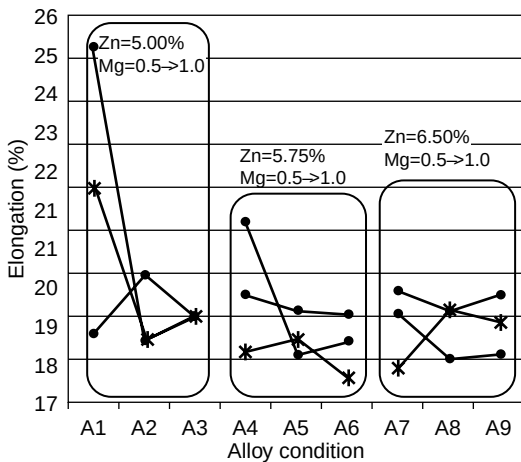


Fig. 7 Composition impact on elongation at -10 °C, 25 °C, 50 °C

4.1.2 Natural Aging Effects

Figures 8, 9, and 10 illustrate the natural aging response at -10 °C, 25 °C, and 50 °C for 7, 30, and 90 days. The data shows that the tensile and yield strength targets can be achieved after a 7-day natural aging cycle except for the case where aging occurs at -10 °C. The mechanical strengths increase as natural aging proceeds and the increase in strength is proportional to aging temperature. The natural aging rate is inversely proportional to aging time due to the slowed growth of GP zones. Under the conditions of this aging study, the tensile strength increases at a rate 0.8MPa/day at -10 °C, 0.7MPa/day at 25 °C, and 0.5MPa/day at 50 °C. The yield strength increases at a rate of 0.6MPa/day at -10 °C, 0.4MPa/day at 25 °C, and 0.3MPa/day at 50 °C.

Although these general trends are known and were expected, the data allows quantification based empirical relationships. Regression analysis was performed to generate trend lines and equations for each condition. A statistical

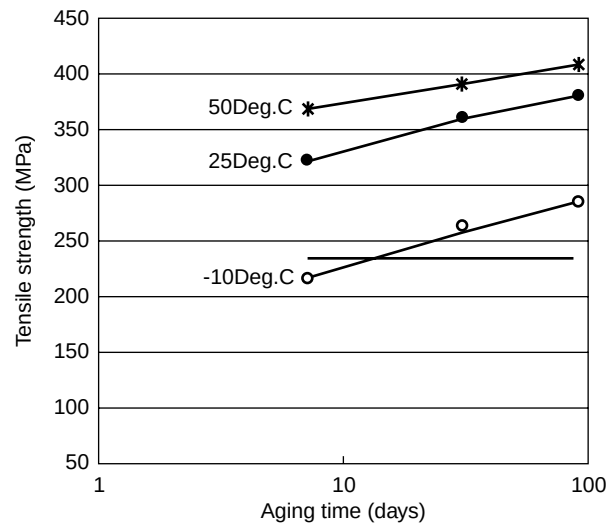


Fig. 8 Aging impact on tensile strength

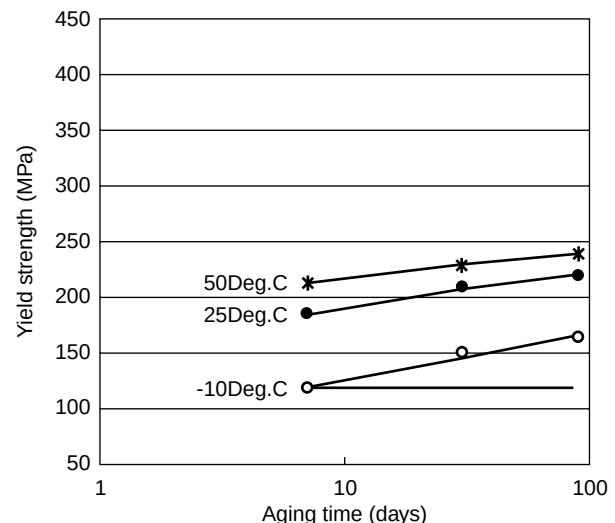


Fig. 9 Aging impact on yield strength

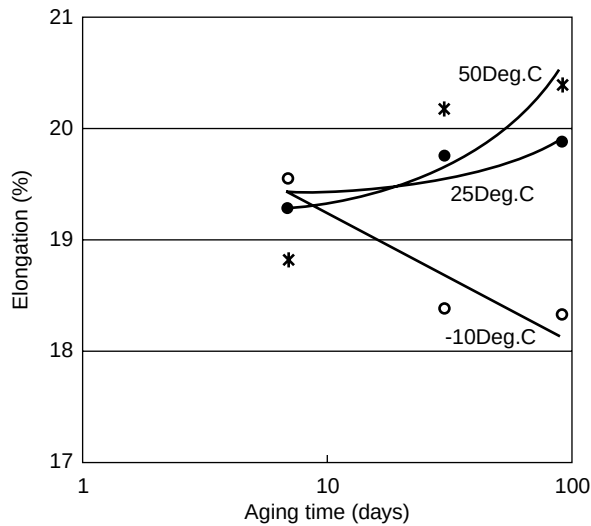


Fig. 10 Aging impact on elongation

check on the accuracy of the strength vs. natural aging equations was made by calculating the R^2 value. The equations and R^2 values are given in Tables 6 and 7. Test error prevented an accurate trend analysis on the elongation data.

The mechanical properties given by the standard 7003 alloy are sufficient after naturally aging for at least 7 days, except when exposed to freezing temperatures. To prevent exposure to freezing temperatures likely when transporting the extruded lengths during the winter, extrusion and bending was established at the same plant location. This decision prevents unexpected changes in the mechanical response to tube bending and improves control over the dimensions of the finished tubes for easier alignment during welding. Special chemical control is not required to guarantee compliance with the design criteria and the standard 7003 was confirmed to be acceptable for frame production.

4.2 Recrystallization Tendency

The normalized results from each worst case depth measurement are shown in Figure 11 and a photomicrograph showing a typical recrystallized layer is shown in Figure

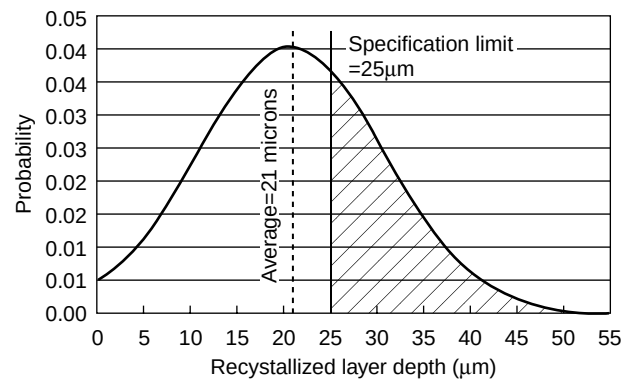


Fig. 11 Recrystallization tendency in 7003-T4

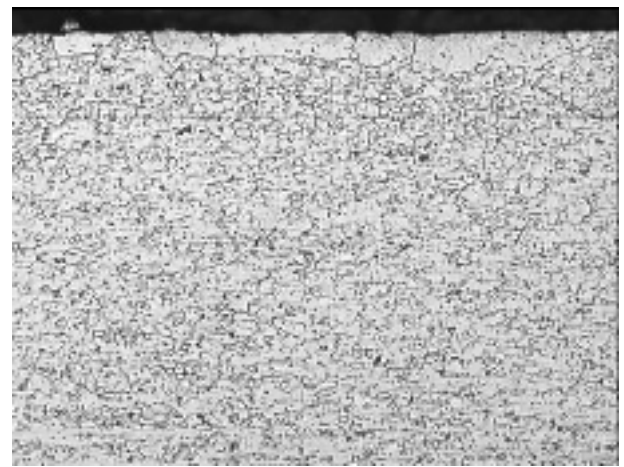


Fig. 12 Typical recrystallized layer in 7003-T4, 200x HF

12. It is seen that the average recrystallized layer depth is less than the specification limit given in Table 2, however there is the potential to exceed this limit. The actual appearance of mass produced frames was acceptable to the visual criteria, so no further action was needed to reduce the potential for recrystallization.

Table 6 Equations, strength vs. chemistry at 7 days

Property	Natural Aging Temperature	Equation
Tensile Strength (MPa)	25°C	$58.31 + (117.75 * \%Mg) + (30.39 * \%Zn)$
Yield Strength (MPa)		$26.61 + (79.19 * \%Mg) + (19.22 * \%Zn)$

Table 7 Equations and R^2 values, strength vs. natural aging

Property	Natural aging temperature	Equation	R^2 Value
Tensile strength (MPa)	-10°C	$26.86 \ln(\text{days}) + 167.76$	0.98
	25°C	$25.99 \ln(\text{days}) + 270.05$	0.99
	50°C	$14.94 \ln(\text{days}) + 341.69$	1.00
Yield strength (MPa)	-10°C	$19.08 \ln(\text{days}) + 82.50$	0.98
	25°C	$14.89 \ln(\text{days}) + 158.19$	1.00
	50°C	$9.87 \ln(\text{days}) + 197.53$	0.99

4.3 Reversion and Recovery During Paint Curing

Hardness of the frame material was considered fully recovered when a hardness of 75 HRF was achieved. This value represents the as-welded hardness of the extruded tubes prior to paint curing. Tensile tests were conducted on samples having a maximum hardness of 75 HRF to guarantee acceptable strength and elongation at this target hardness. Based on the correlation, the tube material was considered acceptable when recovery to at least 75 HRF was obtained.

Reversion was found to occur in the 7003 tube material during each of the three paint curing trials. The A356.0 casting material, heat treated to the T6 condition, was not affected by exposure to curing temperatures. This is primarily due to the stability of the Mg_2Si precipitate formed during heat treating. Reversion was found to occur in the welds joining the tubes to the castings, however the high Mg content in the A5356 filler metal promotes rapid recovery and the hardness returned to the as-welded condition within 14 days. These results are illustrated in Figure 13.

Results from the trials conducted under normal paint curing conditions are given in Figure 14. The line speed settings for Trial 1 and Trial 2 were slow enough to allow a significant decrease in material hardness. In each of these two cases, the recovery time was nearly 15 days. Increasing the line speed at both the pretreatment and curing locations during Trial 3 acts to minimize reversion, despite higher curing temperatures. The frame material exposed to the conditions of Trial 3 required only 12 days to recover to the target hardness. This was the minimum recovery time recorded for the three Trials, therefore the mass production paint curing conditions were set to equal those applied to Trial 3.

5. Conclusions

- (1) The standard 7003-T4 alloy, produced in the U.S., has been developed for production of extruded frame tubes for the luxury touring motorcycle.
- (2) The standard material is shown to meet strength and surface recrystallization requirements for the lightweight, painted frame design.
- (3) Empirical relationships have been established to define mechanical properties as a function of alloying and natural aging.
- (4) A paint curing process has been defined to guarantee the mechanical properties in the final painted product.

6. References

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7. Acknowledgements

The authors are grateful for the efforts of Dave Childree (Kaiser CFT) and Honda of America Motorcycle Paint Department for their support during this material development. Special gratitude is given to all team members associated with this local development.

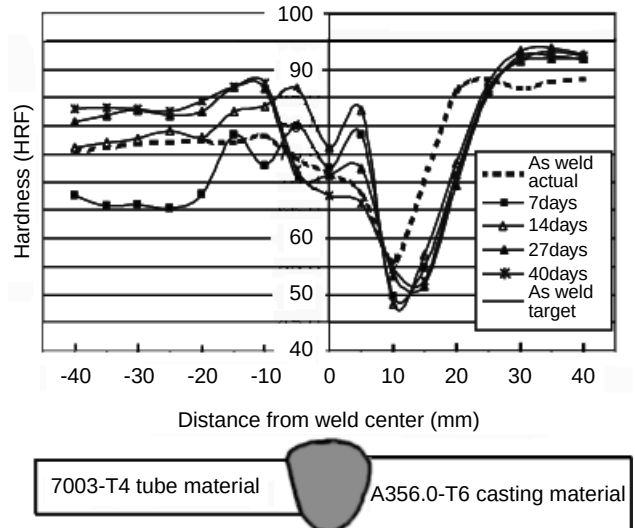


Fig. 13 Reversion and recovery in the weldment

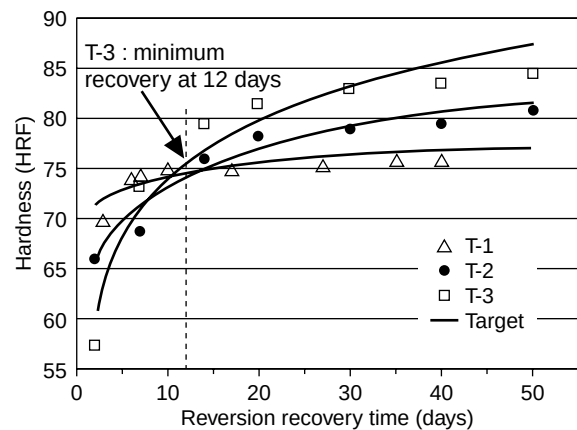


Fig. 14 Reversion recovery: Normal cure trial results T1-T3

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Development of an Alloy for High Pressure Die Cast Pistons

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Yukio SUGIMOTO* Takanori SATO*

ABSTRACT

High pressure die casting material was developed for the use of pistons in various mass-produced, small, general-purpose engines. This new material enables a significant reduction in casting tact time, allows a continuous line through the casting and machining processes, which is impossible to achieve by the permanent mold gravity die casting method (GDC), and contributes to production cost reduction. This material provides a material hardness of HRB65, which is equivalent to the material used in conventional methods, due to the high speed filling of the molten metal into the dies and the quick solidification of it. The material also accomplishes the high temperature durability requirements after undergoing a short T5 process. For the stabilization of the material properties in mass production, various alloy prototypes were produced and evaluated on strength, residual stress, seizure, microstructure, and actual engine durability in order to determine the optimal alloy composition.

1. Introduction

Small, general-purpose engines are now used in many different applications. In Japan, 6 to 7 million units are produced annually, and accompanying this growth, the manufacturers are exhaustively studying and implementing measures for reducing product costs. Honda released GC engines⁽¹⁾ and Mini 4-Stroke engine series⁽²⁾ in 1997 and we have continued to supply high-performance, low-priced engines. In respect to specific engine components, we have developed a single cam resin gear, an integrated cylinder head and block, and a FCD crankshaft among other components in an effort to reduce costs while maintaining function.

The use of aluminum alloy for piston materials is already common practice for reducing the weight. The production of general-purpose engine pistons at Honda is primarily done by permanent mold gravity die-casting (GDC). Recently, however, the small-displacement engines (22-50 cc) manufactured by Honda and other firms⁽²⁾ have begun to use pistons made by high-pressure die-casting (DC). Cost reductions from the shape of general-purpose engine pistons can already be said to have about reached their limit. On the other hand, when considering cost reductions from the manufacturing process used by Honda for multi-type, large-volume production (4 million units/year), the greatest opportunity lies within the use of a continuous production line that incorporates casting, heat treatment, machining and other operations of the process.

The most critical technical issue to be solved for realizing a continuous production line involving casting, heat treatment and machining is the time required for the heat treatment process; this process takes much longer than other processes, thereby inhibiting the application of efficient, continuous production throughout the processes. The current requirements for heat treatment time are:

- 5 to 6 hours for the T7 heat treatment of GDC pistons, and
- 4 to 5 hours for the T5 heat treatment of DC pistons.

There is no possibility of realizing a continuous production line unless this time is shortened to about 30 minutes at most.

This paper describes the course of development of the piston material that combines the advantage of short process time inherent to the DC production method with rapid solidification. The results of this material's evaluation are also presented.

2. Objectives

The major objective of changing the casting process from the conventional GDC method to the DC method was to create a continuous production line for the manufacturing process by synchronizing the casting and heat treatment processes with the machining process as this would result in a considerable improvement in production efficiency. A further objective of applying the DC method was to achieve the following by taking advantage of the rapid solidification and material characteristics inherent to this method:

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and material characteristics inherent to this method:

- Omission of the solution heat treatment
- Significant reduction in the casting tact time
- Minimization of the amount of machining by improving the dimensional accuracy of the as-cast product

The development of a material suitable for the new rapid heat treatment had the following aims:

- Providing mechanical characteristics such as high-temperature tensile strength and fatigue strength
- Providing physical characteristics such as high dimensional stability and thermal conductivity

3. Experimental method

3.1 Test Materials

The GDC casting material, AC8H, with 0.8wt.% of iron added to protect the mold was used for the tests to identify differences in the casting methods. This alloy was named DC8H, and its chemical composition along with that of AC8H are shown in Table 1. Alloys DC8Z/#1-#10 were used to examine the alloy composition. These were prepared for use in testing in the form of alloys exclusively for DC casting by adding various amounts of Si, Cu and Mg to the HD2 base metal, HDS. Table 2 shows the target compositions of the alloys examined, together with the melting point (mp) measured just before casting. Melting point was measured with a recorder specially designed for (AE Meter), which allowed the measurement of liquidus temperature through solidus temperature of each Al alloy sample as shown in Fig. 1-a, and the cartridge shell cup for AE meter shown in Fig. 1-b.

Each test material for a piston with a diameter of 68 mm was melted in a graphite crucible and cast from molten metal maintained at a temperature of 700°C with a 350-ton cold chamber die-casting machine. Various test pieces were then

cut from each cast piston. The shapes of the test pieces used in the tensile strength test and fatigue strength test are shown in Figs. 2-a and 2-b, respectively.

3.2 Mechanical and physical properties

The tensile strength test was conducted with an Instron universal testing machine at a strain rate of 0.3%/min up to 0.2% proof strength, and at a cross-head rate of 3 mm/min after reaching 0.2% proof strength. The test temperatures

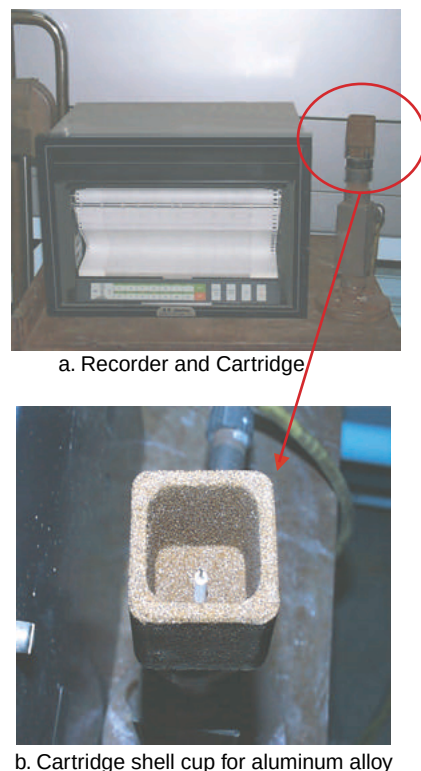


Fig. 1 Recording thermometer for molten metal

Table 1 Chemical compositions

Composition	Cu %	Si %	Mg %	Zn %	Fe %	Mn %	Ni %	mp °C
AC8H	3.0	11.2	0.9	0.08	0.35	0.09	0.03	569
DC8H	3.0	11.3	0.9	0.07	0.82	0.08	0.03	569

Table 2 Chemical compositions for alloy designing

Composition	Cu %	Si %	Mg %	Zn %	Fe %	Mn %	Ni %	mp °C
AC8H	2.5 ~ 3.5	10.5 ~ 11.5	0.8 ~ 1.4	< 0.10	< 0.40	< 0.10	< 0.05	569
DC8Z/#1	5.0	14.0	2.0	< 2.5	< 1.1	< 0.5	< 0.5	550
DC8Z/#2	4.0	14.0	2.0	< 2.5	< 1.1	< 0.5	< 0.5	549
DC8Z/#3	3.3	14.0	2.0	< 2.5	< 1.1	< 0.5	< 0.5	552
DC8Z/#4	3.3	14.0	1.5	< 2.5	< 1.1	< 0.5	< 0.5	553
DC8Z/#5	3.3	13.0	1.5	< 2.5	< 1.1	< 0.5	< 0.5	555
DC8Z/#6	3.3	11.5	1.5	< 2.5	< 1.1	< 0.5	< 0.5	549
DC8Z/#7	3.3	13.0	1.0	< 2.5	< 1.1	< 0.5	< 0.5	553
DC8Z/#8	3.3	13.0	2.0	< 2.5	< 1.1	< 0.5	< 0.5	547
DC8Z/#9	3.3	14.0	1.2	< 2.5	< 1.1	< 0.5	< 0.5	558
DC8Z/#10	3.5	13.0	1.4	< 2.5	< 1.1	< 0.5	< 0.5	554

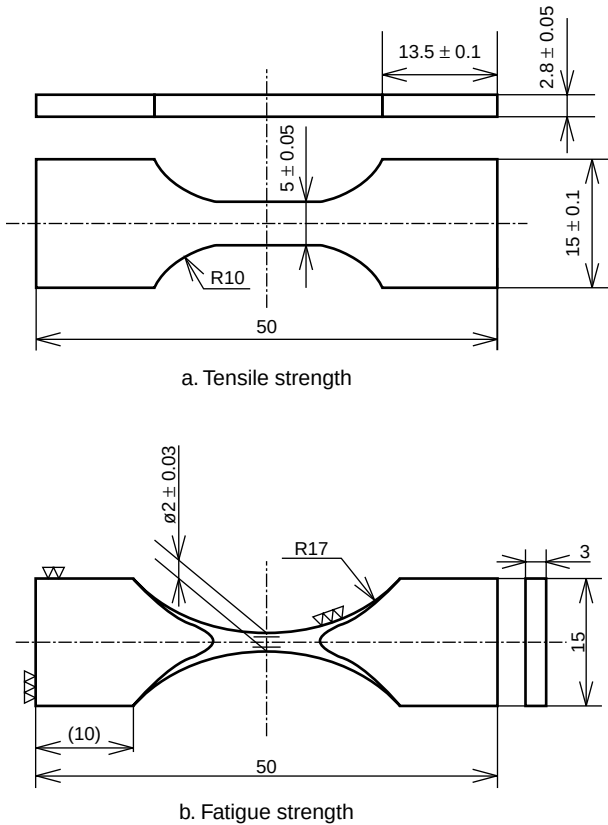


Fig. 2 Tensile and fatigue strength test piece under high temperature

were 200°C and 300°C, and the tensile strength test was performed after soaking the test piece at each temperature for 15 minutes. The fatigue strength test was conducted with an electro-hydraulic servo type fatigue strength tester with a capacity of ± 1 tf. Test conditions applied were a sine waveform for the applied load, a stress ratio of -1, a test frequency of 5 Hz to 17 Hz, and 1.00×10^7 test cycles. Testing was conducted under these conditions with test piece temperatures of 200°C and 300°C.

The hardness test and measurement were based on the Rockwell hardness testing method (JIS Z2245). The thermal expansion coefficient was measured over a range of 20-400°C by the TMA method, thermal conductivity was measured according to the laser flash method, and density was measured according to the Archimedes method.

4. Results

4.1 Effects of the different casting methods and heat treatments on the dimensional stability of the piston

In order to assess the necessity of heat treatment due to the change from the GDC method to the DC method, the effects of the thermal load on the dimensions and residual stress of the AC8H-T7 (GDC) and DC8H-F (DC) piston materials are shown in Table 1. In addition, the effects of changes in heat treatment conditions on the dimensional stability and residual stress of the DC8H material were also investigated.

4.1.1 Dimensional stability and internal residual stress of the DC material without any heat treatment

The change in piston diameter, which is used as an indicator of dimensional stability, was determined from the dimensional difference before and after evenly heating in an oven to a temperature that simulated the thermal load during actual engine operation. A machined piston to which no heat treatment had been applied was measured, before being placed in the uniform heating oven at 230°C for 10 hours. The piston was then removed from the oven and allowed to cool down to room temperature, after which the dimensions were measured again.

The residual stress was measured after drilling a hole of 30 mm in diameter in the center of the piston crown to reduce its rigidity and attaching strain gauges at the locations shown in Fig. 3. After leaving the strain gauges attached to the piston for at least 10 hours, they were connected to a data logger and zero-balanced. The piston was cut at the centerline between the strain gauges (section x-x in Fig. 3) to release the residual stress. The strain was measured after cutting, and then converted to the internal residual stress. Two mass-produced AC8H-T7 and two prototype DC8H pistons were tested to evaluate the dimensional changes and internal residual stress, and strict precautions were taken to avoid measurement error, such as loading the pistons into the heating oven simultaneously.

The change in diameter in the thrust direction (major diameter) and the piston pin direction (minor diameter) before and after thermal load tests for each of the AC8H-T7 and DC8H-F pistons are shown in Figs. 4-a and 4-b. The results of measuring the internal residual stress are shown in Fig. 5. Compared to the AC8H-T7 piston, which increased in diameter only about 20 μm , the DC8H-F piston increase by nearly three times that amount (about 60 μm). In addition, since the measured stress values during parting were positive, the residual stress was determined to be in the centripetal direction thus compressing the piston. In comparative terms, the DC8H-F piston exhibited values for the residual stress about one to six times greater than those for the AC8H-T7 piston.

On the basis of these experimental results, DC pistons without heat treatment were found to be difficult to achieve dimensional stability and internal residual stress equivalent to those of GDC pistons.

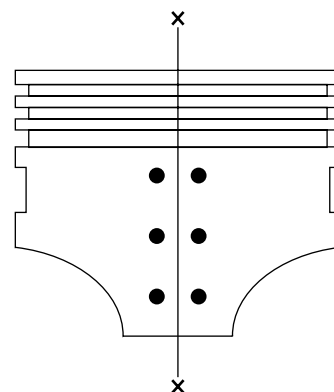


Fig. 3 Measurement points for residual stress

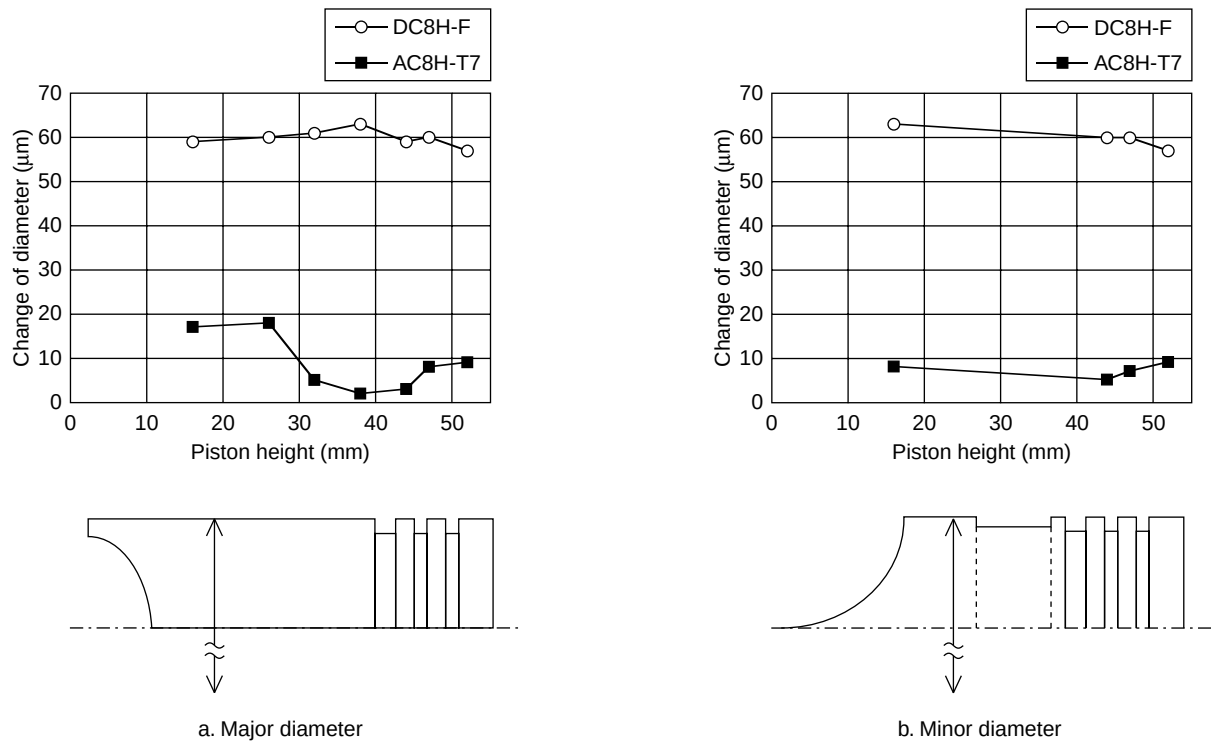
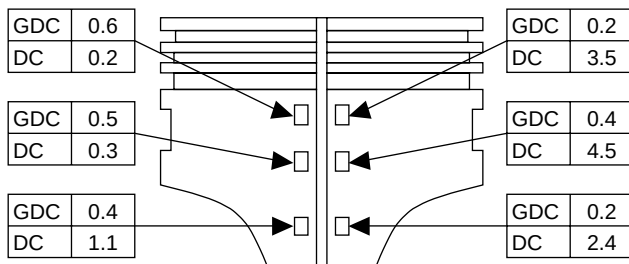


Fig. 4 Changes in major and minor diameters at each piston height after heating (230°C-10h)

Fig. 5 Residual stress after heating (kgf/cm²)

4.1.2 Heat treatment conditions and dimensional stability of the DC8H material and piston

The measured changes in diameter and internal residual stress of the pistons made of the DC8H material and heat treated at temperatures of 190°C, 225°C, 250°C and 275°C for 10, 20 and 30 minutes are shown in Figs. 6-a and 6-b. The thermal load test conditions were as that described in section 4.1.1; that is, the parts were heated for 10 hours at 230°C in a uniform heating oven. It was determined that as the temperature of heat treatment increased, the time needed for the internal stress to be released decreased. The heat treatment conditions required to obtain a change in diameter and internal residual stress equivalent to those of the AC8H-T7 piston described in the previous section were heating at a temperature of at least 225°C for 10 minutes or more. The

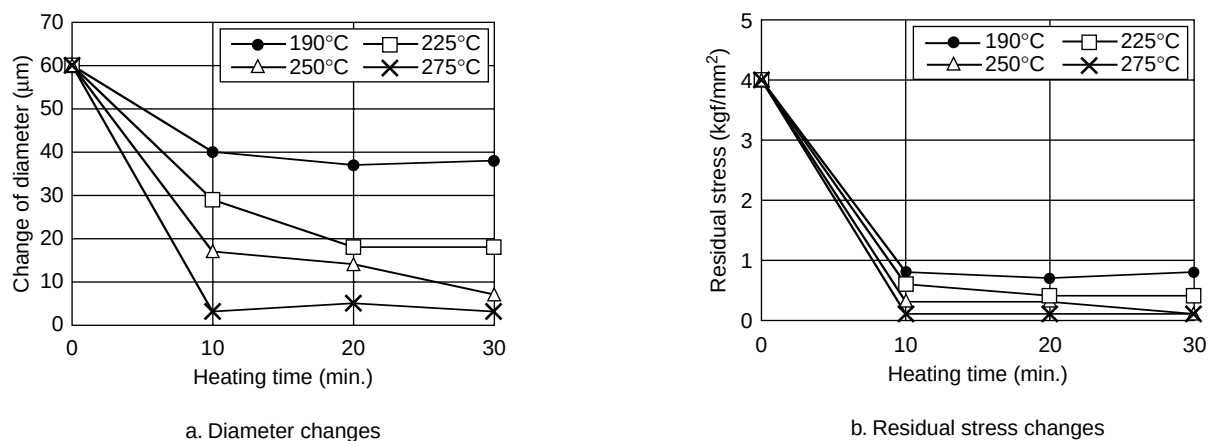


Fig. 6 Influence of heat treatment condition, temperature and time, on major diameter and residual stress

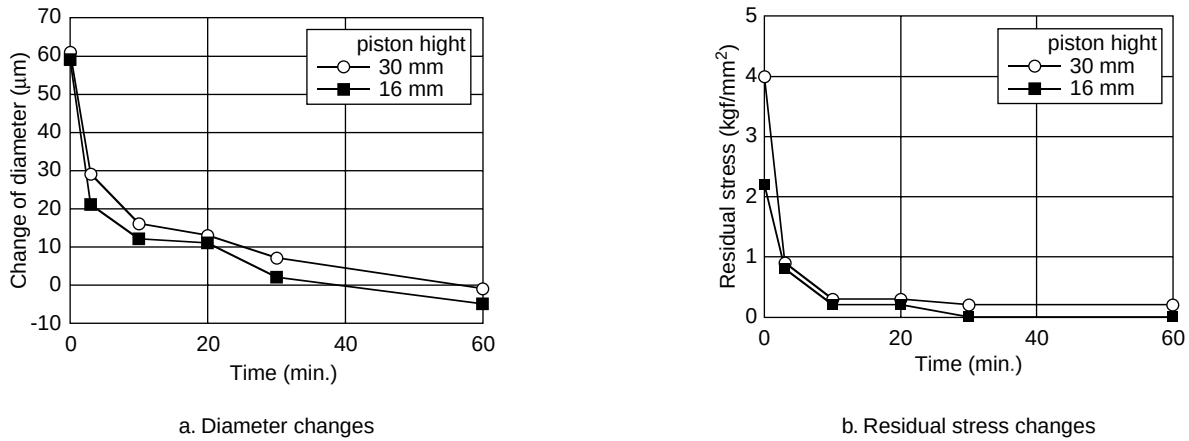


Fig. 7 Influence of heat treatment time on diameter and residual stress changes at 250°C

from an increased heat treatment duration at 250°C are shown in Figs. 7-a and 7-b. The graphs show that piston diameter and internal residual stress changes comparable to those of the AC8H-T7 piston could be achieved with a heat treatment duration of about 2 minutes.

Since the AC8H piston is subjected to the T7 heat treatment, the internal residual stress during casting is considered to be negligible. Thus, the cause of the internal residual stress and dimensional changes that occurred is presumed to be strain resulting from machining. In addition, the cause of the large dimensional change exhibited by the DC8H piston is believed to be the release of both the internal residual stress resulting from being subjected to high-pressure solidification during casting and the strain resulting from machining.

4.2 DC piston alloy composition

The results of an investigation of the dimensional stability associated with the DC method have been described in the previous section. In this section, experimental results are presented of the basic characteristics required of materials in consideration of the control range for the alloy composition when using DC cast alloy materials DC8Z/#1-#10 shown in Table 2. The casting (molten metal) temperature was the regular setting standard of 130-150°C above the melting point, based on the measured melting points of these alloys, and casting was carried out at 700°C.

4.2.1 Material hardness after casting

The molten metal temperature from the start of casting up to 100 shots, and the temperature of the piston crown immediately after removing it from the mold (referred to as the work temperature) are shown in Fig. 8 for each alloy. The molten metal temperature could be reliably controlled to $700 \pm 10^\circ\text{C}$ throughout the complete casting test, while the work temperature could be reliably controlled to $380 \pm 20^\circ\text{C}$, after increasing during the first 20 casting shots while the mold warmed up.

The work temperature and hardness (HRB) of the as-cast black skin during the first 10 casting shots for the 10 types of alloy are shown in Fig. 9. The hardness of the as-cast black skin material was measured after cooling with

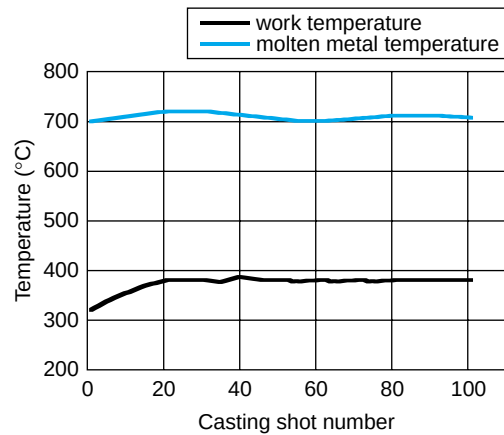


Fig. 8 Changes of molten metal and piston-top temperature after casting with the increase of casting shot number

water following the DC process. The first shot, having a low work temperature, was the hardest. As the work temperature subsequently stabilized, the hardness of the as-cast black skin material also became stabilized. Alloy #6 exhibited considerable variations in initial hardness, while alloy #9 exhibited a decrease in initial hardness as the work temperature increased, thus indicating that these alloys would not be suitable for practical use. An evaluation of the relationship between the added amounts of Mg, Si and Cu and the initial material hardness for the 10 types of alloy showed that the target hardness of HRB 65 or higher could be achieved with minimum contents of 1.4% Mg, 13% Si and 3.3% Cu.

4.2.2 Heat degradation characteristics

The effects of temperature and soaking time on the hardness of test pieces DC8Z/#5-T5 and AC8H-T7 after their heat treatment (250°C, 30 minutes, T5) are shown in Figs. 10-a and 10-b. In this experiment, the test pieces were reheated after the heat treatment and then allowed to cool to room temperature, before the hardness was measured.

The decrease in hardness at each soak temperature

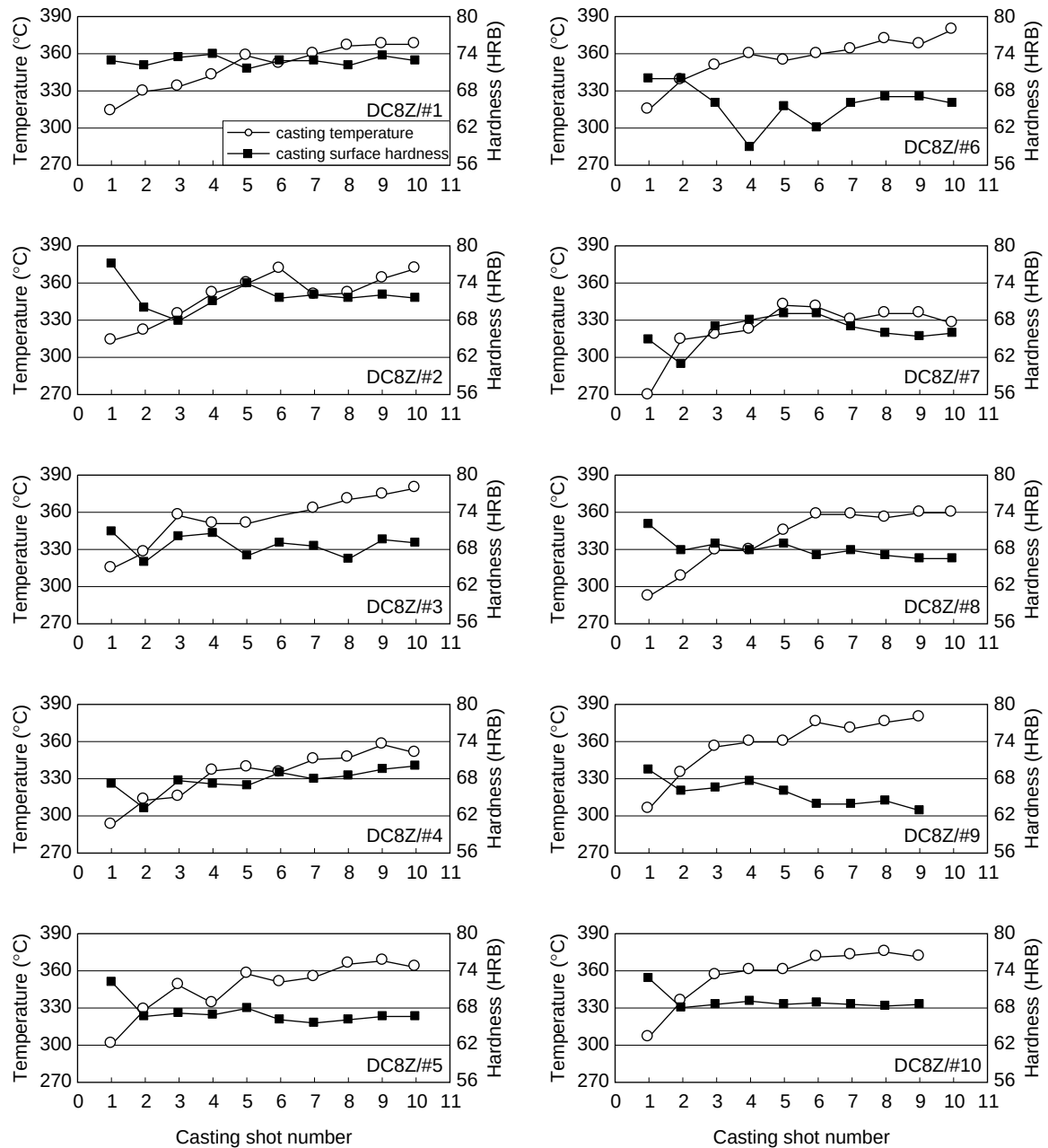


Fig. 9 Stability of alloy depending on composition of base material

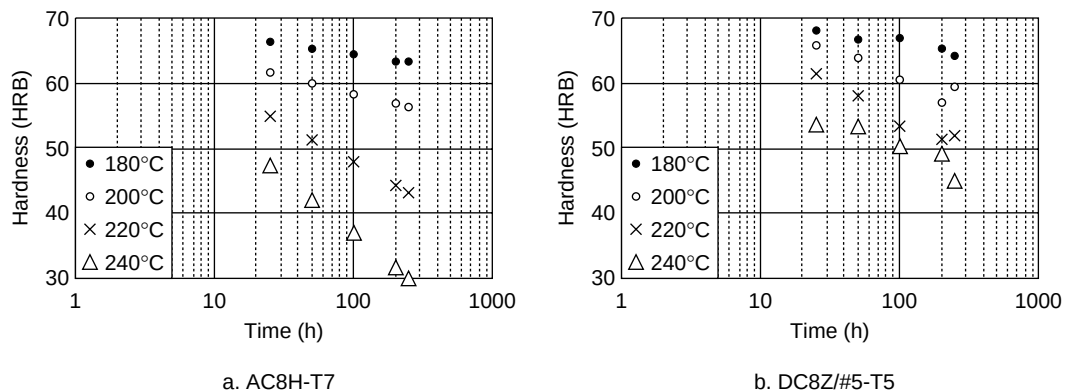


Fig. 10 Changes of hardness with after heating at various temperatures

The decrease in hardness at each soak temperature exhibited by the DC8Z/#5-T5 material was smaller compared to the AC8H-T7 material. The results reflect the study of Fukai *et al.* with AC8A and B materials.⁽³⁾ The difference in softening is thought to be due to the following: 1) the increasing fineness of the crystal grain size due to the rapid solidification characteristics of the DC process; and 2) the fine undissolved intermetallic compounds resulted from not performing a solution treatment were maintained in an evenly dispersed state, and these compounds act to resist movement of the dislocation line. Thereby the decrease in hardness after heating was reduced.

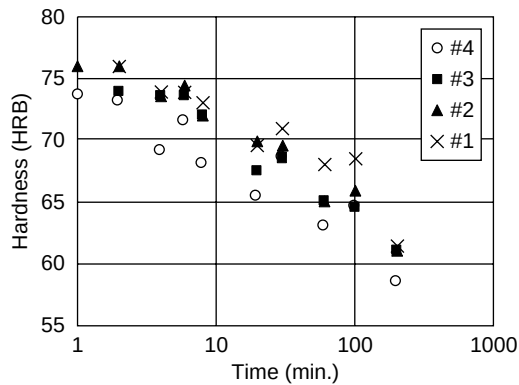
4.2.3 Selection of the optimum alloy composition and heat treatment conditions

Since the change in piston diameter was stabilized by heat treating at 225°C for at least 10 minutes as described in section 4.1.2, the change in hardness due to age hardening of each alloy when using the temporary T5 heat treatment conditions of 250°C for 30 minutes was evaluated as shown in Figs. 11-a and 11-b. If the target initial hardness is set to be a minimum of 65 (HRB), the average value for the GDC piston (AC8H-T7), alloys #7 and #10 are not suitable. In addition, since the test piece for alloy #1 occasionally exhibited brittle failure during machining, it was also excluded for the poor machinability.

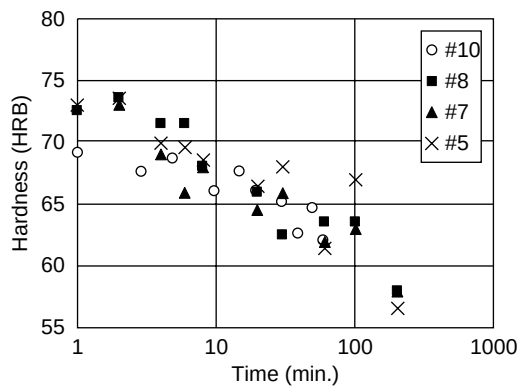
After evaluating the 10 types of alloy (DC8Z) according to various criteria, DC8Z/#2, #3, #4, #5 and #8 were determined to have desirable mechanical and physical properties as a piston alloy. A new alloy based on the range

of these compositions was designated as HAS14C4. The upper, middle, and lower limits of this alloy composition are shown in Table 3. Table 4 shows the mechanical and physical properties of HAS14C4 following a T5 heat treatment (250°C for 30 minutes) in comparison with those of AC8H-T7. In comparison with AC8H-T7, the thermal conductivity is about 25% lower, while the other physical properties are similar. However, the mechanical properties such as the hardness, tensile strength and fatigue strength have all improved. The thermal conductivity of HAS14C4 is equivalent to 0.256, the value for a high-performance engine piston material (A351), and the difference from AC8H-T7 is considered to be within the practically allowable range.

Photographs of the microstructures of AC8H-T7 and the upper and lower limit compositions of HAS14C4 are shown in Figs. 12-a, -b, and -c. The structure after solidification of the AC8H-T7 material contains white crystal grains of soft α -Al over a broad range, and large acerous crystals of eutectic Si is visible. On the other hand, the structure after solidification of the HAS14C4 material exhibits the distribution of extremely fine α -Al, Mg_2Si , eutectic Si and so forth. In addition, primary crystal of Si appearing in a planar form in the microstructure is distributed in larger numbers and more uniformly than in the AC8H-T7 material. This microstructure is believed to be responsible for the mechanical properties being equal to or better than those of the AC8H-T7 material.



a. DC8Z/#1 to #4 alloys



b. DC8Z/#5, 7, 8, 10 alloys

Fig. 11 Relationship between hardness and heat treatment time at 250°C

Table 3 Chemical composition of HAS14C4

	Cu %	Si %	Mg %	Zn %	Fe %	Mn %	Ni %	Al
Upper	4.0	14.0	2.0	< 2.5	< 1.1	< 0.5	< 0.5	Bal.
Middle	3.8	13.5	1.7	< 2.5	< 1.1	< 0.5	< 0.5	Bal.
Lower	3.5	13.0	1.4	< 2.5	< 1.1	< 0.5	< 0.5	Bal.

Table 4 Physical & mechanical property of HAS14C4

Physical property	Test condition	AC8H-T7	HAS14C4 250°C × 20 min.
Thermal expansion coefficient ($\times 10^{-6}$) ($^{\circ}\text{C}^{-1}$)	Room temp. ~ 100°C Room temp. ~ 200°C	19.2 ~ 20.8	19.4 ~ 20.3
Thermal conductivity ($\times 10^{-6}$) (W/m·°C)	Room temp.	135 ~ 142	100 ~ 105*
Modulus of longitudinal elasticity (MPa)	Room temp.	735 ~ 775	745
Density	—	2.68 ~ 2.74	2.68 ~ 2.79
Hardness (HRB)	—	64 ~ 68	66 ~ 68
Tensile strength (MPa)	200°C 300°C	2.12 ~ 2.60 0.74	2.30 ~ 2.80 1.29 ~ 1.42
0.2% proof strength (MPa)	200°C 300°C	1.98 ~ 2.05 0.57	1.89 ~ 2.40 1.00 ~ 1.19
Fatigue strength 10^7 cycle (MPa)	200°C 300°C	0.74 ~ 0.78 0.33	0.83 ~ 0.88 0.41

* Since the thermal conductivity of A351 is 107 which is almost equivalent to that of HAS14C4, this result is considered to be acceptable.

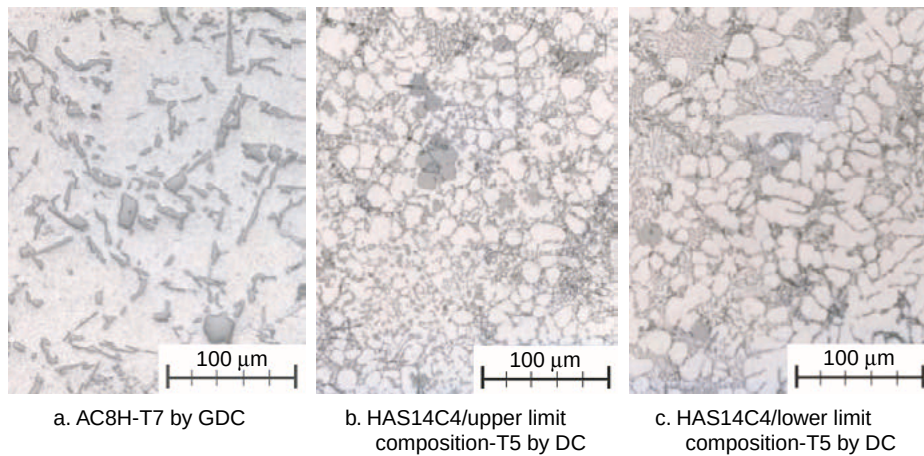


Fig. 12 Comparison of microstructures with AC8H-T7, HAS14C4/upper limit and HAS14C4/lower limit composition

4.3 Compatibility test

4.3.1 High-temperature piston fatigue strength test

A diagram of the high-temperature piston fatigue tester is shown in Fig. 13. The results of tests on an HAS14C4-T5 piston made by the DC process and on an AC8H-T7 piston made by the GDC process are shown in Fig. 14. Piston failure in the high-temperature fatigue strength test began with the formation of cracks from the pin hole which propagated toward the top of the piston pin boss in both pistons. Fig. 14 shows the change in safety factor with the number of fatigue cycles based on the maximum combustion pressure, and demonstrates that HAS14C4-T5 was superior

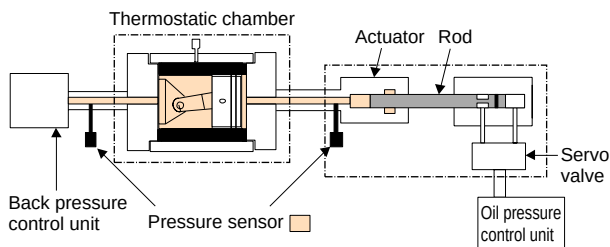


Fig. 13 Diagram of high-temperature piston fatigue tester

to AC8H-T7. This result is consistent with the results for heat degradation characteristics described in section 4.2.3.

4.3.2 Seizure limit test

(1) Piston test

The experiment on seizure limit, using a test piece cut from the piston, was performed with the chip-on-disk friction tester shown in Fig. 15. After attaching the test piece (6 x

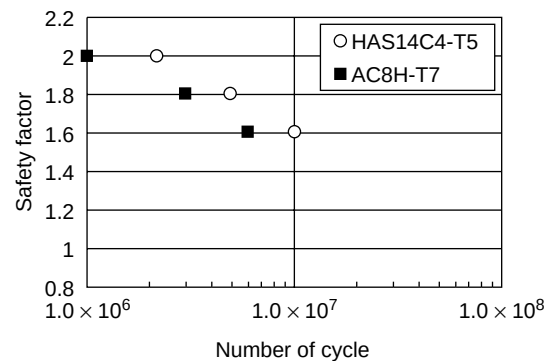


Fig. 14 Fatigue strength indicated by safety factor. Safety factor 1 equals to maximum cylinder pressure.

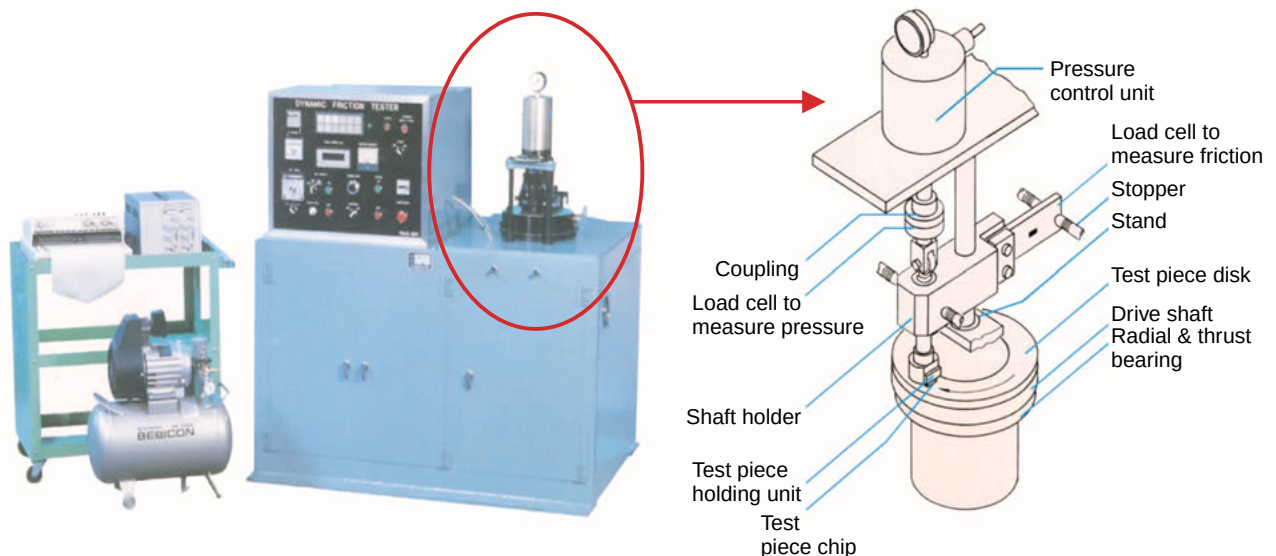


Fig. 15 Diagram and photograph of friction tester

17 mm) to the holding unit on the tester, a load was applied by the holding unit that pressed the test piece against a rotating disk made of FC250, a material equivalent to that of the cylinder liner. The rotating disk was rotated at a constant peripheral speed of 16 m/sec against the static test piece. After providing a break-in period of 3 minutes with a selected load, during which oil was dripped on to the test piece at the rate of 240 cc/min, the supply of oil was discontinued. The time till seizure was then measured. If seizure did not occur within 20 minutes, the test was terminated after 1×10^7 cycles in the same manner as a typical fatigue test, and the load at that time was determined to be the “seizure limit.”

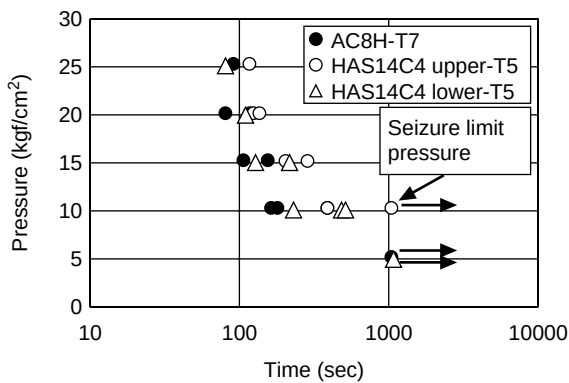


Fig. 16 Result of seizure test by friction tester

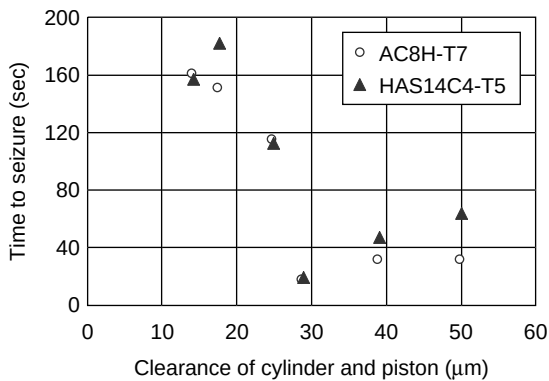


Fig. 17 Comparison of time to seizure between AC8H-T7 and HAS14C4-T5 under the dry seizure engine test

A comparison of the seizure limit for materials with upper and lower limit alloy HAS14C4-T5 compositions and AC8H-T7 is shown in Fig. 16. The newly developed material demonstrated a seizure limit superior to that of AC8H-T7. The comparison of the solidified metal microstructures described above indicates that soft α -Al crystals, which are responsible for seizure, constitute only a small proportion of the surface area of the HAS14C4-T5 material. Moreover, the microstructure of HAS14C4-T5 contains a uniform dispersion of intermetallic compounds and primary crystal of Si that contribute to the reduction of frictional resistance. These factors can be considered to be responsible for the higher seizure limit of this new material.

(2) Engine test

4-stroke general-purpose engines are often started without being filled with oil or supplied with engine oil because 2-stroke engines are more commonly used today as general-purpose engines. The resistance to seizure was evaluated by a dry seizure test assuming the first case, and by a wet seizure test assuming the second case. The engine for the dry seizure evaluation test was prepared by degreasing the cylinder bore during assembly and applying a single drop of oil. Testing was conducted by starting the engine without engine oil, immediately running the engine at 3600 rpm in the no-load state and measuring the time until seizure occurred. The test was conducted with six different values for clearance between the piston and cylinder.

The results of the dry seizure evaluation test for the AC8H-T7 and HAS14C4-T5 pistons are shown in Fig. 17. The HAS14C4-T5 piston tended to show higher seizure resistance than the AC8H-T7 piston. Photographs of the piston surface following the seizure test are shown in Fig. 18. The seizure marks on the AC8H-T7 piston were wider than those of the HAS14C4-T5 piston, and the marks from melting tended to be larger.

The engine used in the wet seizure evaluation test was assembled after applying the usual amount of oil to the cylinder bore and filling oil to the lower limit of the oil level gauge. After initially running the engine for 10 minutes at the rated load, the oil was gradually drained from the engine, and the engine was run until it stopped because of seizure. The amount of oil remaining in the engine was measured, and the appearance of the piston was observed after the engine had seized. The remaining oil in the engine at seizure was 70 cc for the AC8H-T7 piston and 58 cc for the HAS14C4-T5 piston. Photographs of the external appearance of these pistons are shown in Fig. 19. A comparison of the effects of seizure on the piston surface

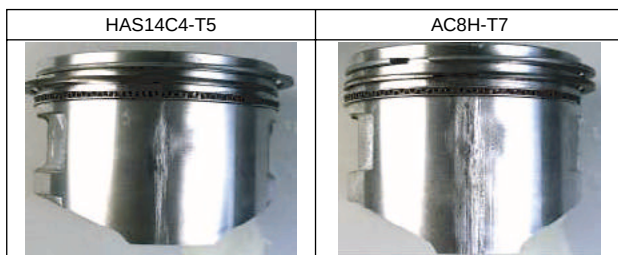


Fig. 18 Pistons after the dry seizure engine test

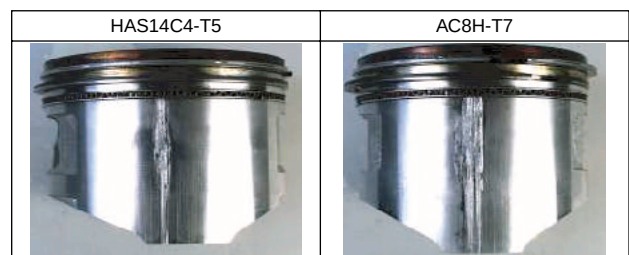


Fig. 19 Pistons after the wet seizure engine test

yielded results similar to those of the dry seizure evaluation, and the seizure resistance of the HAS14C4-T5 piston was again confirmed to be superior to that of the AC8H-T7 piston.

5. Conclusion

We have developed a new piston alloy by the DC process that offers a considerably shorter casting and heat treatment time than the conventional process to allow the use of a continuous production line. This piston alloy has mechanical properties that are equal to or better than conventional alloys used with the GDC process, under heat treatment conditions of 250°C for 30 minutes, and in seizure resistance tests it demonstrated the performance is also equal to or better than that of existing alloys.

Continuous production lines for the manufacture of general-purpose pistons are likely to be installed in the near future to enhance productivity considerably. There are great expectations for the results of research into new materials with enhanced properties and their application to manufacturing lines to improve productivity.

Acknowledgments

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Development of a Chromium-free Pretreatment System for Aluminum Bodies

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ABSTRACT

A chromium-free surface treatment method was developed for aluminum bodies. By adding fluorine ions and iron complex compounds to the current zinc phosphate pretreatment bath, an improved zinc phosphate film is formed on aluminum. Also, increasing the manganese in the pretreatment bath improves the alkaline resistance properties of the zinc phosphate film. As a result of these changes, the zinc phosphate film has improved secondary adhesion after water conditioning and improved filiform corrosion resistance without using a chromium rinse. Therefore, chromium can be eliminated from the paint process of aluminum bodies. This, in turn, reduces environmental burdens previously posed.

1. Introduction

It is necessary to consider the post-painting corrosion resistance when using aluminum in automotive bodies just as is true with the use of steel. Thus, research is being conducted on various painting techniques. Painting methods for aluminum automotive bodies and their associated pretreatment processes developed to date are shown in Fig. 1. In Japan, chromating was briefly used as the pretreatment method. However, currently the main method applied for aluminum/steel mixed metal pretreatment is a zinc phosphate treatment, which enables the painting of hybrid bodies. With this treatment, a free fluorine ion (F^-) is added to the zinc phosphate solution used for steel bodies, allowing a zinc phosphate film to be formed on aluminum.

However, it is known that the addition of fluorine ions lowers the adhesion of the paint film. This is believed to be caused by fluorine compounds adhering to the surface of the zinc phosphate film and impeding the adhesion of the paint film. One method for improving adhesion is to apply a chromium rinse after phosphating to remove the fluorine compounds.

It is also difficult to form a zinc phosphate film on a sanded area of an aluminum body (an area where a body stamping deformation or welding irregularity has been corrected by grinding the surface with sandpapers, etc.). Even with the addition of fluorine ions, a comparison between such areas and normal areas, where no abrasion has been applied, shows them to be inferior in post-painting corrosion resistance. To solve this problem, a chromium corrosion-resistant pigment is added to the cathodic electrocoat (E-coat) to assist in corrosion resistance.

However, a surface treatment that utilizes chromium has

a negative impact on the environment and entails costly wastewater treatment, so the use of a chromium rinse is gradually declining. It has also been decided in Europe that the amount of hexavalent chromium used in corrosion-resistant coatings will be restricted in the future.

Therefore, we began the development of a surface treatment for aluminum that does not require the use of chromium. We report in this paper the development of a method for adding fluorine ions and an iron complex compound to the zinc phosphate solution as reaction accelerators. This method enables the elimination of materials containing chromium from the painting process.

2. Current Surface Treatment Methods

Corrosion resistance in automotive bodies must take into account various difficulties in paint adhesion such as paint film blisters, delamination and peeling, and resistance to such types of post-painting corrosion as scab and filiform corrosion. It is known that the corrosion resistance of aluminum is generally superior to that of steel in the untreated state. However, when a zinc phosphate solution for steel is applied to aluminum, an adequate film cannot be formed and the resulting corrosion resistance is inadequate.

When treating aluminum, therefore, fluorine ions have conventionally been added to the zinc phosphate treatment solution as an etching agent to accelerate the formation of the zinc phosphate film. Fluorine ions are added as fluorosilicic acid (H_2SiF_6) and sodium fluoride ($NaHF_2$), and controlling their concentration regulates the degree of film formation.

When aluminum ions (Al^{3+}) are eluted in the treatment solution and accumulate beyond a fixed amount, they inhibit

* Tochigi R&D Center

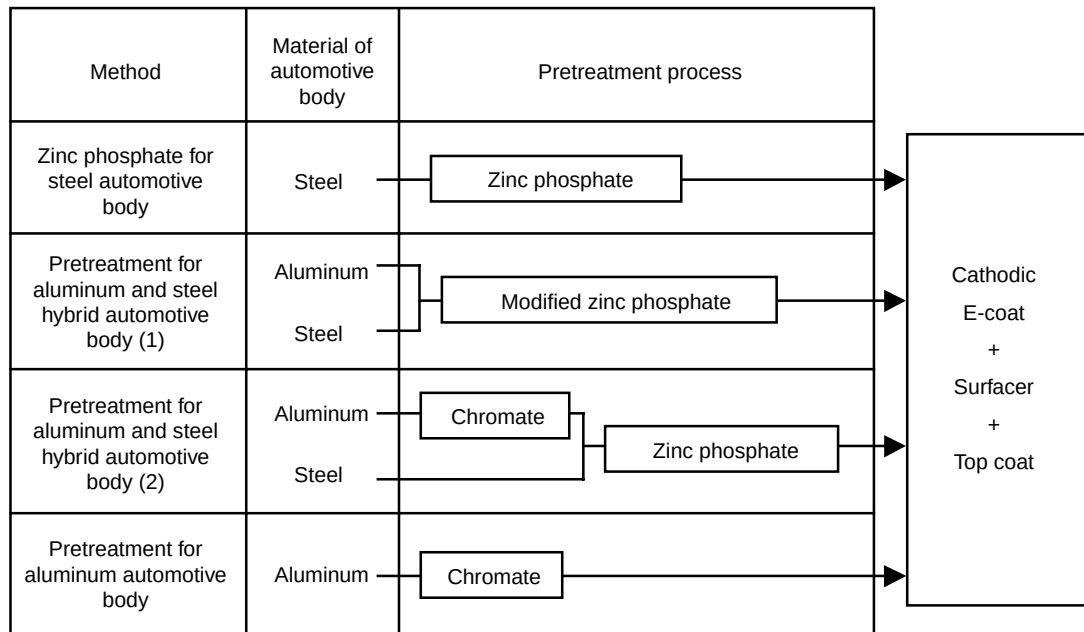


Fig. 1 Pretreatment methods for aluminum and steel bodies

the precipitation reaction of the zinc phosphate film. Thus, they must be removed.⁽¹⁾ The zinc phosphate reaction on aluminum and removal of the aluminum ions from the solution can be described by the following reactions:⁽²⁾

- (1) Etching the aluminum surface

$$2\text{Al} + 6\text{HF} \rightarrow 2\text{Al}^{3+} + 6\text{F}^- + 3\text{H}_2 \uparrow$$
- (2) Precipitation of a zinc phosphate film resulting from an increase in pH value of the aluminum surface

$$3\text{Zn}(\text{H}_2\text{PO}_4)_2 \rightarrow \text{Zn}_3(\text{PO}_4)_2 \cdot 4\text{H}_2\text{O} (\text{Hopeite}) + 4\text{H}_3\text{PO}_4$$
- (3) Removal of aluminum ions from the solution

$$\text{Al}^{3+} + 3\text{Na}^+ + 6\text{F}^- \rightarrow \text{Na}_3\text{AlF}_6 (\text{Cryolite}) \downarrow$$

In this way, the addition of fluorine ions to the current zinc phosphate treatment for aluminum and steel hybrid bodies improves the zinc phosphate properties on the aluminum and allows for a continuous treatment.

However, it is known that an excessive concentration of fluorine ions reduces the degree of adhesion of the paint film. This is thought to be caused by fluorine compounds

adhering to the surface of the aluminum blocking the adhesion of the paint film. To improve adhesion, it is necessary to carry out a chromium rinse in the post-treatment process to remove the fluorine compounds. Fig. 2 compares the results of a water-resistance test with and without a chromium rinse after the treatment. The results demonstrate that the chromium rinse improved the water resistance.

Even with the addition of fluorine ions, however, it is difficult to achieve sufficient film coverage over the scratches on a sanded aluminum surface, with the result that the filiform corrosion resistance is lowered on the sanded surface. It is known that the addition of a chromium corrosion-resistant pigment to the E-coat improves the filiform corrosion resistance. Fig. 3 shows the effect of this addition.

As already explained, it has been necessary to use materials containing chromium to maintain good paint adhesion and filiform corrosion resistance with the current zinc phosphate treatment for aluminum and steel hybrid automotive bodies.

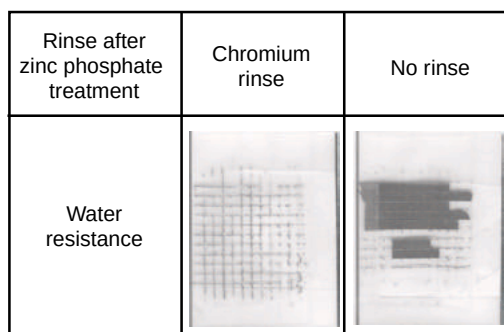


Fig. 2 Paint adhesion on sanded area after water resistance test

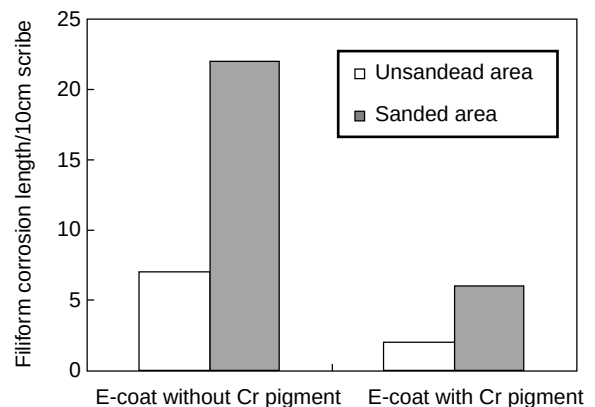


Fig. 3 Effect of corrosion-resistant pigment in E-coat

3. Improvement in Zinc Phosphate Treatment Properties with the Addition of an Iron Complex Compound

To ensure the post-painting corrosion resistance of aluminum without using a chromium rinse or chromium corrosion-resistant pigment in the E-coat, we concluded that it would be necessary to form a fine and uniform zinc phosphate film and improve its coverage (the percentage of the surface covered) by increasing the reactivity of the treatment solution.

The effectiveness of iron ions (Fe^{3+}) as an etching accelerator for aluminum in an acidic solution is well known.^{(3),(4)} It is also known that the addition of iron ions to the zinc phosphate solution forms a fine and uniform zinc phosphate film on a steel surface. However, iron ions in the zinc phosphate solution used on automotive bodies (pH 3 –

3.5, NO_2^- ions contained) form an iron phosphate sludge, making it impossible to stably maintain a fixed amount of iron ions over a long period of time.

We therefore attempted to reproduce the effect of iron ions as just described on the formation of a zinc phosphate film on aluminum, by maintaining the ions in the zinc phosphate solution as a complex. We used ammonium iron citrate ($[\text{Fe}\{\text{C}_6\text{H}_4\text{O}_7(\text{FeO})(\text{NH}_4)_2\}_2(\text{OH})_2\}_3\text{C}_6\text{H}_5\text{O}_7 \cdot x\text{H}_2\text{O}$) as the iron complex compound. Figs. 4 and 5 respectively show the phosphating time plotted against the change in coating weight and the precipitation of crystals in the film. The material used in both cases was a 6000 series aluminum alloy. The results show that the addition of the complex compound increased the precipitation of zinc phosphate film and improved the film coverage compared to the current treatment.

Fig. 6 shows the electrical potential during the etching of a 6000 series aluminum alloy. As the intention here was to observe the etching behavior during the zinc phosphate

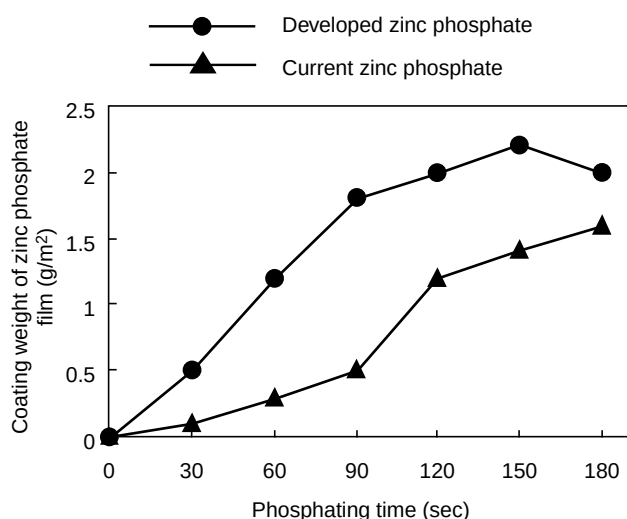


Fig. 4 Phosphating time vs. coating weight of the zinc phosphate film

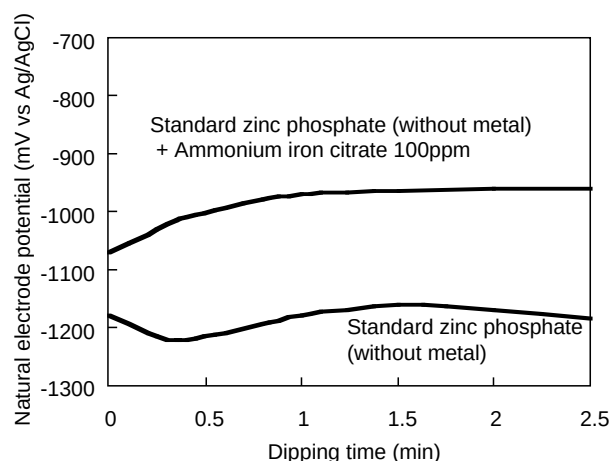


Fig. 6 Change of electrical potential during the etching reaction of aluminum

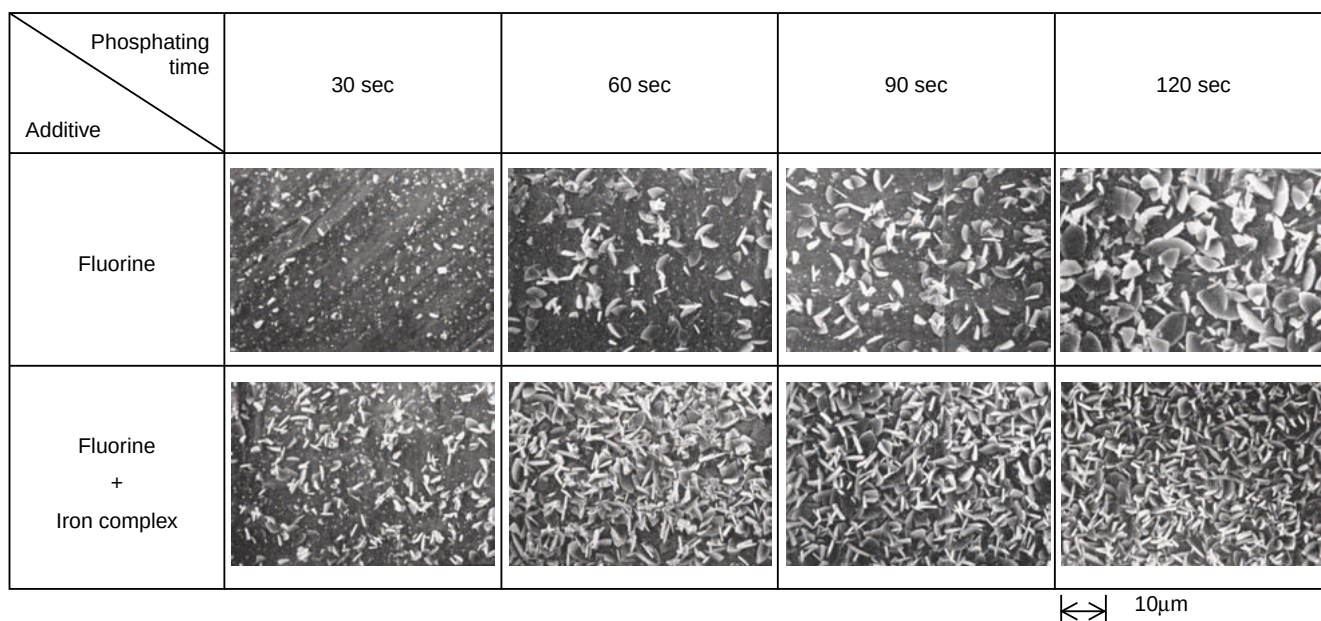


Fig. 5 SEM images of zinc phosphate films

reaction, we extracted the zinc (Zn), nickel (Ni) and manganese (Mn) (components of the film formation) from the treatment solution. With the addition of the complex compound, the electrical potential shifted to noble during the etching process. Based on this, we hypothesize that the aluminum etching reaction is accelerated due to the acceleration of the cathode reaction.

4. Improvement of Water Resistance

It is necessary to grant sufficient attention to paint film adhesion, because when it declines, the appearance of the body is compromised due to peeling, blisters, etc., eventually leading to the development of corrosion under the film.

It is thought that there are various controlling factors in paint film adhesion performance. These factors include the paint materials, the method of painting, the chemical conversion coating method and the metal materials. Various experimental methods have been devised to evaluate paint film adhesion. In this section, we will discuss the relationship between the zinc phosphate treatment and water resistance.

Various improvements have so far been made to the zinc phosphate solution for steel, with the aim of ensuring water resistance. For example, when galvanized steel was first used for vehicle bodies, Mn was added to the zinc phosphate solution. This improved the quality of the film by replacing part of it with Mn, and it limited condensation during warm-water dipping. As a result, the water resistance was improved.⁽⁵⁾

The zinc phosphate film on aluminum is also thought to be improved by Mn, but the water resistance declines on areas where it is difficult to form a zinc phosphate film such as sanded areas. It is also known, as already described, that if excess fluorine ions are added to improve the film coverage, fluorine compounds are precipitated on to the aluminum surface, causing a decline in the water resistance.

In contrast, a treatment with zinc phosphate to which iron complex compounds have been added demonstrates an improvement in the water resistance, as shown in Table 1. The figures shown as the surface F/Zn ratio represent the

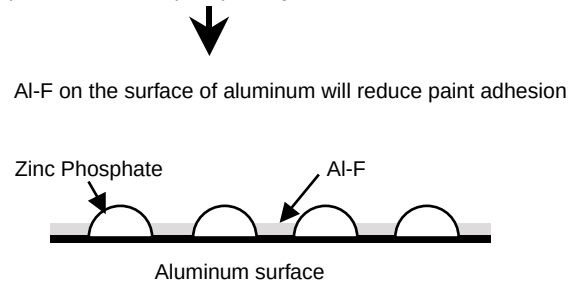
ratio of the atomic densities of fluorine (F) and zinc (Zn) on the aluminum surface after the zinc phosphate treatment. These figures were calculated from X-ray photoelectron spectroscopy (XPS) measurements.

A high water resistance resulted when the drain time (the time from phosphating to the water rinse) was less than 5 sec (i.e. immediate water rinse after phosphating) and when the phosphating time was 10 min. The resulting F/Zn ratio was zero in this case, so it is clear that fluorine compounds had not been formed.

These results enable the mechanism for the improvement in water resistance to be deduced. As Fig. 7 shows, when an area of the aluminum surface remains incompletely covered by the film, fluorine ions in the solution react with the incompletely covered area until the zinc phosphate solution is removed by the water rinse, and compounds of aluminum and fluorine (Al-F compounds) are precipitated. It is thought that these Al-F compounds inhibit adhesion. As Figs. 4 and 5 show, the phosphate film coverage was improved with a treatment by zinc phosphate to which an iron complex compound has been added, because Al-F compounds would not be readily produced in this case. When the water rinse was conducted immediately after phosphating, no Al-F compounds were formed because the fluorine ions had been removed. The coverage on the aluminum surface of the zinc phosphate film was sufficient with a treatment time of 10 minutes, so Al-F compounds had not been formed.

We can further point to the surface F/Zn ratio as a good indicator for judging water resistance.

(1) Drain time after phosphating



(2) Al-F doesn't appear on the surface of aluminum covered with fine zinc phosphate film

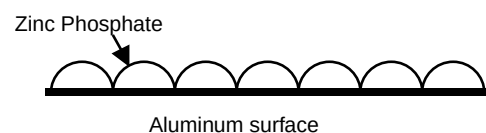


Fig. 7 Mechanism for the decrease in the paint adhesion

Table 1 Water resistance and F/Zn ratio

	Water resistance (Percentage of remained paint)	F/Zn ratio
Zinc Phosphate for aluminum and steel (STD condition)	0~50%	5.1
New zinc phosphate (STD condition)	100%	0.1~1.3
New zinc phosphate (Drain time < 5 sec)	100%	0
New zinc phosphate (Phosphating 10min)	100%	0

* STD condition : Phosphating time 2min, Drain time 30sec

5. Improvement in Filiform Corrosion Resistance

Filiform corrosion is one form of atmospheric corrosion that occurs with painted metals, etc., taking the particular form of corrosion growing from a damaged area or from the edge of the paint film in a thread-like form across the

surface. Filiform corrosion is particularly common with aluminum, and sufficient attention to the phenomenon is necessary in the body painting process. External factors that have been reported to influence the appearance and growth of filiform corrosion include relative humidity, paint materials, and pigments.⁽⁶⁾ As mentioned earlier, one method for improving the filiform corrosion resistance of aluminum when using the current zinc phosphate treatment is the addition of a chromium corrosion-resistant pigment to the E-coat.

A model of filiform corrosion on aluminum has been presented⁽⁶⁾ (see Fig. 8) in which the tip of the filiform corrosion (the anodic section) is in an acidic environment, while the area around the tip (the cathodic section) is in an alkaline environment.

Fig. 9 shows a comparison of the filiform corrosion resistance between the current zinc phosphate treatment for hybrid aluminum and steel bodies and the same treatment with an iron complex compound added. It is clear that the filiform corrosion resistance was improved by the newly developed treatment.

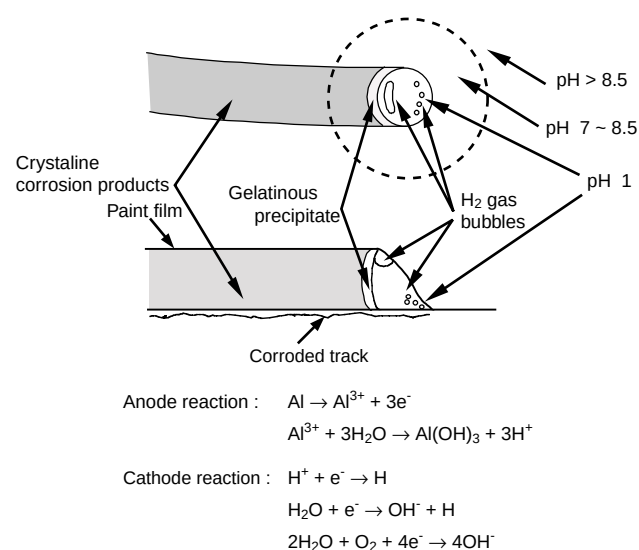
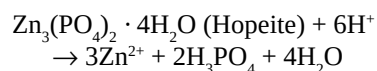


Fig. 8 Mechanism of filiform corrosion

We believe this improvement is due to the increased coverage of the zinc phosphate film causing the anodic section to be neutralized by the film according to the following formula, leading to suppression of the anodic reaction:



The filiform corrosion resistance was also improved by increasing the concentration of Mn in the zinc phosphate solution. We believe that increasing the amount of Mn in the film by increasing the concentration of Mn improved

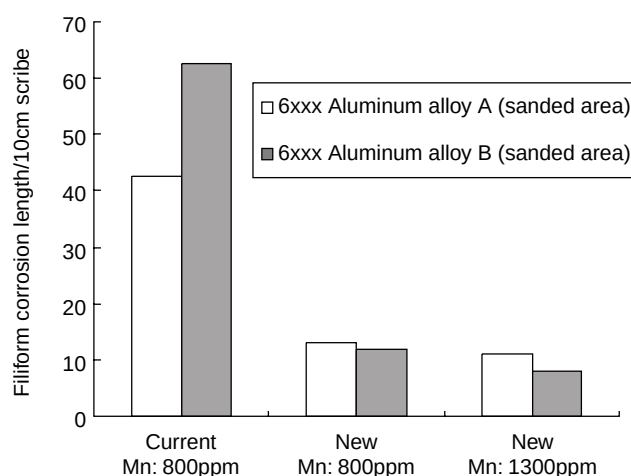


Fig. 9 Evaluation of filiform corrosion for each treatment

Table 2 Relationship between the concentration of Mn ions and alkaline resistance properties

Concentration of Mn (ppm)	Dissolved weight in alkaline (g/m ²)
800	0.2036
1300	0.0643

Current zinc phosphate treatment system

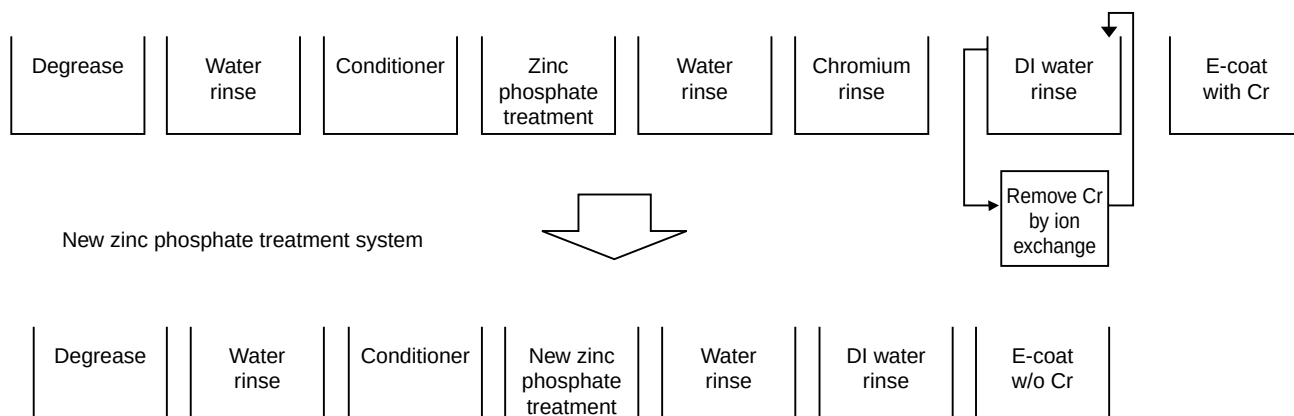


Fig. 10 New zinc phosphate treatment process

the alkaline-resistant property of the film, thus suppressing the filiform corrosion.

Table 2 shows the relationship between the concentration of Mn and the alkaline-resistant property of the zinc phosphate film.

The results of this study indicate that it is possible to obtain sufficient water and filiform corrosion resistance from body paint without the use of chromium. The removal of chromium from the painting process will contribute to a reduction in environmentally harmful substances. Fig. 10 shows an outline of the surface treatment systems.

6. Conclusion

In addition to the currently used fluorine ions, an iron complex compound was introduced as an additive to the zinc phosphate treatment for hybrid aluminum and steel bodies to increase reactivity. We also conducted studies on the concentration of Mn that would be required to improve the film.

- (1) The addition of the iron complex compound accelerated the zinc phosphate precipitation reaction, thus improving the coverage of the phosphate film on aluminum. This enabled the zinc phosphate film to be precipitated on to sanded areas which has been difficult with the current treatment.
- (2) Thus, it was possible to suppress the precipitation of Al-F compounds, which negatively affect paint film adhesion, and to ensure sufficient water resistance without a post-treatment chromium rinse. The filiform corrosion resistance was also improved in comparison with the current treatment.
- (3) We obtained a zinc phosphate film with superior alkaline-resistant properties by increasing the concentration of Mn in the treatment solution.
- (4) These results make it clear that it is possible to ensure sufficient filiform corrosion resistance without the addition of a chromium corrosion-resistant pigment to the E-coat.

Acknowledgments

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Acidic Bath by the AC Impedance Method, Hyomen Gijutsu, 43, No.3, p.205-210 (1992)

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Toshiya NISHINO



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Technical Papers Submitted to Out-of-company: January, 2000 to June, 2000

(English Papers only)

Date	Conference	Title	Author	Organization
Jul. 1, 2000	JSAE/JSAE Review Vol.21, No.3	Analysis of Sealing Behavoir of Fleid Sealant in Steel Plate made Engine Oil Pan	Kazuo Sakurahara Masayasu Sakata Takumi Ishigaki	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Development of a 100% Recyclable Instrument Panel	Sukekuni Itoh Keiichi Honjo	Tochigi R&D Center Tochigi R&D Center
		Analysis of Piston Top Rings Half-Stick Phenomena Using Optical Technology	Yoshiaki Akimoto Takeshi Maruyama	Tochigi R&D Center Tochigi R&D Center
		High-Resolution range sensor system with stereo camera	Chiaki Aoyama Kenichi Mineta Yasushi Okada	Wako Research Center Wako Research Center Wako Research Center
Jul. 7, 2000	On evolutionary optimization with approximate fitness functions (Las Vegas, USA)	Genetic and Evolutionary Computation Conference	Yaochu Jin Markus Olhofer Bernhard Sendhoff	Honda R&D Europe Honda R&D Europe Honda R&D Europe
Jul. 16, 2000	Congress of Evolutionary Computation 2000 (San Diego, USA)	Evolutionary optimised ontogenetic neural networks with incremental problem complexity during development	Bernhard Sendhoff	Honda R&D Europe
	CNS2000 Computational Neuroscience Meeting (Brugge, Belgium)	Fast Dynamic Organization without Short-Term Synaptic Plasticity: A New View of Hebb ' s Dynamical Assemblies	Julian Eggert Berthold Baeuml J.L.van Hemmen	Honda R&D Europe DLR Wessling Technische Universitat Muenchen
		Cortical Synfire Activity : Configuration Space and Survival Probability	Marc-Oliver Gewaltig Markus Diesmann Ad Aertsen	Honda R&D Europe Albert-Ludwigs-Universitat Freiburg Albert-Ludwigs-Universitat Freiburg
		Two Separate Processing Streams in a cortical- Type Architecture	Tobias Rodemann Edgar Koerner	Honda R&D Europe Honda R&D Europe

Date	Conference	Title	Author	Organization
Jul. 23, 2000	SCI2000/ISAS2000 Proceedings (Orland, USA)	Developing Brain-like Computer with Memory-Based Architecture	Hitoshi Yamada Yohane Takeuchi Takashi Tominaga Tadahide Kato Noriyuki Aibe Gen Matsumoto Michinori Ichikawa Hideaki Ono	Brain-Operative Device Lab., Brainway Group, Brain Science Institute, RIKEN Brain-Operative Device Lab., Brainway Group, Brain Science Institute, RIKEN Brain-Operative Device Lab., Brainway Group, Brain Science Institute, RIKEN Brain-Operative Device Lab., Brainway Group, Brain Science Institute, RIKEN Brain-Operative Device Lab., Brainway Group, Brain Science Institute, RIKEN Brain-Operative Device Lab., Brainway Group, Brain Science Institute, RIKEN Brain-Operative Device Lab., Brainway Group, Brain Science Institute, RIKEN Wako Research Center
Jul. 28-29, 2000	Vibro Acoustic/International Auto SEA Users Conference (San Diego, USA)	Analysis of Vehicle Road-Noise Using Statistical Analysis	Kazuhiro Misaji Toshinobu Saito Yuji Kuruhara Tsuyoshi Yamashita	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
Aug. 20, 2000	Tohoku University/The Third International Symposium on Shape Memory Alloys and Related Technology (Sendai)	High-Speed Respose of SMA Actuators	Jinhao Qiu Junji Tani Yuta Urushiyama	Tohoku University Tohoku University Tochigi R&D Center
Aug. 22-24, 2000	AVEC/Advanced Vehicle Control Conference 2000 (Detroit, USA)	Navigation System of Autonomous Vehicle Control System Installed in ICV S	Kazuya Tamura Yoshimi Furukawa	Tochigi R&D Center Tochigi R&D Center
Sep. 6, 2000	8th AIAA/USAF/NASA/ISSMO Symposium on Multidisciplinary Analysis and Optimization (California, USA)	Multi-Objective Optimization for the Transonic Compressor Stator Blade	Yoshihiro Yamaguchi Toshiyuki Arima	Wako Research Center Wako Research Center
	AVL LIST GMBH/AVL Congress September 7th-8th, 2000 (GRAZ, Austria)	Optimization of Interior Sound for Sports Cars	Yasuo Ishii Siegmond Thomann Franz.K.Brandl Tsuyoshi Yamashita	Tochigi R&D Center AVL List AVL List Tochigi R&D Center
Sep. 11, 2000	Intertech Consulting Conference & Studies/Hybrid Vehicles 2000 Intertech's International Conference (Ontario, Canada)	Development of an Energy Storage System for HEVs	Naoki Kinoshita	Tochigi R&D Center

Date	Conference	Title	Author	Organization
Sep. 11-12, 2000	Institute for Motorcycle Safety e.V./Safety Environment Future (Munich, Germany)	Research of an Emission Reduction System for Small Motorcycles	Shunji Akamatsu Shuichi Ishida Akira Hamauzu	Asaka R&D Center Asaka R&D Center Asaka R&D Center
		Development of a Tyre Tube with Puncture Prevention Function for Large Motorcycles	Toshio Yamagiwa Kengo Nakayama Akiko Tanaka	Asaka R&D Center Asaka R&D Center Asaka R&D Center
	SAE/SAE TOPTEC Variable Valve Actuation TOPTEC: The State-of-the-Art/ (Detroit, USA)	HONDA Variable Valve Train Systems	Chikara Tanaka	Tochigi R&D Center
		Directly Actuated Variable Valve Control System, "HYPER-VTEC"	Takaaki Tsukui	Asaka R&D Center
Sep. 12, 2000	European Biophysics Conference (Munich, Germany)	Activity-Gated Attention in the Visual System	Julian Eggert	Honda R&D Europe
Sep. 20-23, 2000	IRCOBI/International Research Council on the Biomechanics of Impact 2000 (Montpellier, FRANCE)	Pedestrian Head Impact Conditions Depending on The Vehicle front shape And Its Construction - Full Model Simulation	Yutaka Okamoto Tomiji Sugimoto Koji Enomoto Junichi Kikuchi	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center PSG Co.,Ltd.
		Development of a New Biofidelic Leg for Use with a Pedestrian Dummy	Marlon Artis Jason Mcdonald Richard White Tsai-Jeon Huang Tariq Shams Nangarajan Rangarajan Akihiko Akiyama Masayoshi Okamoto Ryouzo Yoshizawa Hirotoishi Ishikawa	GESAC Inc, GESAC Inc, GESAC Inc, GESAC Inc, GESAC Inc, GESAC Inc, Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Japan Automobile Research Institute
Sep. 25-29, 2000	ISATA/33rd ISATA Conference (Dublin, Ireland)	Thermal Generation and Entropy Control for Advanced Batteries	Noboru Sato Naoki Kinoshita	Tochigi R&D Center Tochigi R&D Center
2000.9.26	IUTAM/SMART Structures and Structronic Syetems 2000 (Magdeburg, Germany)	Buckling of Curved Column and Twinning Deformation Effect	Yuta Urushiyama David Lewinnek Junji Tani Jinhao Qiu	Tochigi R&D Center Tochigi R&D Center Tohoku University Tohoku University
Oct. 1-3, 2000	IEEE/IEEE Intelligent Transportation System Council 2000 (Dearborn, USA)	Workload of Using a Driver Assistance System	Jun Tanaka Shinnosuke Ishida Satoshi Kondo Hiroyuki Kawagoe	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Status and Future Direction of Intelligent Drive Assist Technology	Yoshimi Furukawa	Tochigi R&D Center

Date	Conference	Title	Author	Organization
Oct. 2, 2000	Int. Symposium on Metal Hydrogen Systems - Fundamentals and Applications (Noosa, Australia)	Thermodynamic Properties of catalytically enhanced sodium aluminum hydride	Ikuya Yamashita Ragaiy A. Zidan Satoshi Takara Craig M. Jensen	Wako Research Center University of Hawaii University of Hawaii University of Hawaii
Oct. 4-5, 2000.10.4	IEEE/IEEE Intelligent Vehicles Symposium 2000 (Dearborn, USA)	Autonomous vehicle control system of ICVS City Pal: Electrical Tow-bar Function	Kazuya Tamura Yoshimi Furukawa	Tochigi R&D Center Tochigi R&D Center
		Sensor and Data Fusion Design and Evaluation with a Virtual Environment Simulator	Keith Redmill Kazuya Tamura	OSU Tochigi R&D Center
		Introduction of Honda ASV-2	Akio Takahashi Nobuyoshi Asanuma	Tochigi R&D Center Tochigi R&D Center
		Fail Output Algorithm of Vision Sensing	Morimichi Nishigaki Masakazu Saka Hiromitsu Yuhara Tomoyoshi Aoki Makoto Kawai	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
Oct. 5-6, 2000	vka, ika, RWTH, VDI/9th Aachen Colloquium/(Aachen, Germany)	A modeling approach for suspension friction consist of several sets of frictions and springs in series	Jun Nakahara Masaaki Minakawa Michael Gipser Juergen Wimmer	Tochigi R&D Center Honda R&D Europe Esslingen University of Applied Sciences, Automotive Engineering Dept. Cosin Software GmbH, Germany
Oct. 15-18, 2000	EVAA, Electric Vehicle Association of the Americas/The 17th International Electric Vehicle Symposium & Exposition (Montreal, CANADA)	Development of Generation II Propulsion Motor for EV, FCV	Kazuaki Takizawa Hiroyuki Sato Tomoyuki Ito	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Development of a High Performance Nickel-Metal Hybride Battery for Smaller Electric Vehicles	Kazuhiisa Okamoto Kazuhiko Yagi Takashi Ishikura Noboru Sato	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
Oct. 17, 2000	SAE/Convergence 2000 (Dearborn, USA)	Development of the Motor Assist System for a Hybrid Car-INSIGHT	Shinobu Ochiai Kenji Nakano	Tochigi R&D Center Tochigi R&D Center
Oct. 17-19, 2000	The Japan Gas Association/7th International Conference and Exhibition on Natural Gas Vehicles (Yokohama)	The success of Civic GX in emerging NGV market	Toshiyuki Suga Tadashi Muraishi Akira Murakami Shigeo Hidai	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Development of Valve Seat Material for CNG Fueled Engine	Katsuaki Sato Teruo Takahashi	Tochigi R&D Center Nippon Piston Ring Co.,Ltd.

Date	Conference	Title	Author	Organization
Oct. 23-25, 2000	Executive Conference Management/The Seventh International Conference TPOs IN Automotive 2000 (Novi, MI, USA)	Application of Co-Injection Molding Technology for Bumper Recycling	Atsushi Takeuchi	Tochigi R&D Center
Oct. 25, 2000	MIT Consortium/MIT Consortium Nagoya Meeting (Nagoya)	Development of a Power Assist Motor for Hybrid Car-INSIGHT	Keiichi Yamamoto	Tochigi R&D Center
Oct. 27-28, 2000	μ - Club/XX.International μ -Symposium (Bad Neuenahr, Germany)	Research on Tribological Characteristics Between Al-MMC Brake Disk and Friction Material	Katsuhiro Shibata Kenichi Kawasaki	Tochigi R&D Center Tochigi R&D Center
Oct. 31, 2000	JAST/Synopses of the International Tribology Conference 2000 (Nagasaki)	On Characterizing CVT Fluids by Means of the Mean Coefficient of Friction for CVT Using Metal V-belts	Shigeru Kanehara Toru Fujii Osamu Fujimura Ryuhei Kataoka	Tochigi R&D Center Doshisha University Doshisha University Doshisha University
	JAST/Proceedings of the International Tribology Conference 2000	On Characterizing CVT Fluids by Means of the Mean Coefficient of Friction for CVT Using Metal V-belts	Shigeru Kanehara Toru Fujii Osamu Fujimura Ryuhei Kataoka	Tochigi R&D Center Doshisha University Doshisha University Doshisha University
Nov. 1, 2000	JSAE/JSAE Review Vol.21, No.4	Introduction to the Optically Accessible Engine for Investigation of In-Cylinder Phenomena	Takashi Sasaki Kazuo Yoshida Yasuhiro Urata Kiyoshi Ishii	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Development of Aluminum Body for the Most Fuel Efficient Vehicle	Masaaki Saito Shuichiro Iwatsuki Kunihiro Yasunaga Kimitaka Ando	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Stereo Vision Sensor	Hiromitsu Yuhara Akira Terauchi Masakazu Saka Masayuki Habaguchi Makoto Kawai	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Honda R&D Americas Tochigi R&D Center
	Japan Society of Waste Management Experts/ The Journal of Materials Cycles and Waste Management Vol.2, No.2	Evaluation of the Effect of Automotive Bumper Recycling by Life-Cycle Inventory Analysis	Minoru Makuta Yuichi Moriguchi Yasoi Yasuda Shigeho Sueno	Tochigi R&D Center National Institute for Environmental Studies University of Tsukuba University of Tsukuba

Date	Conference	Title	Author	Organization
Nov. 4, 2000	SOCIETY FOR NEUROSCIENCE 2000 ABSTRACT FORM (New Orleans USA)	STUDIES ON EXPERIENCE DEPENDENT LEARNING FOR RELATIONSHIP BETWEEN SIGNS AND LOCATIONS WITH A ROBOT	Hideaki Ono Hitoshi Yamada Yohane Takeuchi Takashi Tominaga Tadahide Kato Gen Matsumoto Michinori Ichikawa	Wako Research Center Brain-Operative Device Lab., Brainway Group, Brain Science Institute, RIKEN Brain-Operative Device Lab., Brainway Group, Brain Science Institute, RIKEN Brain-Operative Device Lab., Brainway Group, Brain Science Institute, RIKEN Brain-Operative Device Lab., Brainway Group, Brain Science Institute, RIKEN Brain-Operative Device Lab., Brainway Group, Brain Science Institute, RIKEN
Nov. 6-8, 2000	STAPP Advisory Committee/44th STAPP Car Crash Conference STAPP Car Crash Journal, Vol.44 (Atlanta, USA)	Development and Validation of the Finite Element Model for the Human Lower Limb of Pedestrians	Yuukou Takahashi Yuji Kikuchi Atsuhiko Kounosu Hirotoshi Ishikawa	Tochigi R&D Center Tochigi R&D Center Japan Automobile Research Institute Japan Automobile Research Institute
Nov. 6-9, 2000	VERTIS/7th World Congress on Intelligent Transport Systems (Turin, Italy)	Intelligent Technologies of ASV (The Research and Development of ASV-2 in Honda)	Nobuyoshi Asanuma Makoto Kawai Akio Takahashi Ryoichi Shigenari Kazuo Ochi	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Asaka R&D Center Asaka R&D Center
		Vision Sensing in the City Traffic	Hiromitsu Yuhara Masakazu Saka Morimichi Nishigaki Tomoyoshi Aoki	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
		Effects of Intelligent Drive Assist Control System on Driver Control Performance from the View Point of Active Safety	Yoshimi Furukawa Masato Abe	Tochigi R&D Center Kanagawa Institute of Technology
		Evaluation of Driver Assistance System	Shinnosuke Ishida Jun Tanaka Satoshi Kondo Hiroyuki Kawagoe	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center
Nov. 15-17, 2000	Mechanical Dynamics Italy s.r.l./15th ADAMS European User Conference (Rome, Italy)	Simulation of metal pushing type v-belt for CVT	Ken Kanokogi Hiroaki Hoshino Hideaki Yoshida Hiroyuki Akagi	Tochigi R&D Center Mechanical Dynamics Japan K.K. Tochigi R&D Center Tochigi R&D Center
Nov. 16-17, 2000	ESI/PAM Users' Conference in Asia2000, PUCA MM (Tokyo)	Development of a Finite Element Model for the Analysis of Lower Limb Injuries in Pedestrians	Yuji Kikuchi Yuukou Takahashi Atsuhiko Kounosu Hirotoshi Ishikawa	Tochigi R&D Center Tochigi R&D Center Japan Automobile Research Institute Japan Automobile Research Institute

Date	Conference	Title	Author	Organization
Nov. 17, 2000	ICONIP-2000 7th International Conference on Neural Information Processing	Evolution of Adaptive Nonlinear Models	Martin Kreutz Anja M. Busse Bernhard Sendhoff	University Dortmund Ruhr-University Bochum Honda R&D Europe
		Evolutionary Optimization for Problem Classes with Lamarckian Inheritance	Michael Husken Bernhard Sendhoff Edgar Komer	Ruhr-University Bochum Honda R&D Europe Honda R&D Europe
		On the Intergration of Biological Constraints into the Evolution of Artificial Neural Systems	Bernhard Sendhoff Ursula Komer	Honda R&D Europe Honda R&D Europe
Dec. 4, 2000	MOVIC2000 Committees/MOTION and VIBRATION CONTROL 2000 (Sydney, Australia)	Development of Active Control System for Low Frequency Road Noise Inside Automobiles Combined with Audio System	Hisashi Sano Akira Takahashi Toshio Inoue Tsuyoshi Yamashita Kenichi Terai Yoshio Nakamura	Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Tochigi R&D Center Matsushita Electric Industrial Co.,Ltd. Matsushita Electronic Components Co., Ltd.
	Numecal International/NUMECA WORLDWIDE USER MEETING (Brussels, Belgium)	Stage Analysis of Turbine	Naoki Kuno Kouji Murata Toyotaka Sonoda	Wako Research Center Wako Research Center Wako Research Center
Dec. 7-8, 2000	Technische Akademie Esslingen/Technical Academy Symposium (Ostfildern, Germany)	Honda's Current and Future Engine Technologies. As a Solution to the Global Environmental Problems	Tomohiko Kawanabe Kaoru Horie	Tochigi R&D Center Tochigi R&D Center

Editor's Notes

As is only appropriate at the start of the 21st century, this edition of the *Technical Review* begins with technology related to “ASIMO.” Papers on robotic technology have been suggested as candidates for inclusion in the *Review* numerous times in the past, but somehow it never occurred. However, the contributions we assembled for this issue convinced us that the time was right.

The *Technical Review* came into being in the latter half of the 20th century, and as the quality and quantity of the papers has grown, it moves into the 21st century with a continued broadening of its missions. One of these missions has been to produce English translations of all the technical papers and introductions of new technologies from 2000 forward. Providing access to these papers from the homepage of the Technology and Information Department has allowed them to be used as reference materials by a greater number of researchers at Honda’s domestic and overseas research centers, a system which is showing clear results.

In addition, objective third-party evaluations of the quality are increasingly high. One indication is that the articles appearing in the *Review* have a high republication rate in the *JSAE Review* published by the Society of Automotive Engineers of Japan, Inc. Our papers are also becoming known as they are increasingly quoted as references by external organizations. In the same way, opportunities to make presentations based on articles and papers from the *Honda Technical Review* at international conferences and domestic academic meetings are increasing.

We have been publishing the *Technical Review* twice a year since 1999. If we can say that this is an established publication based on this increase in paper quantity, we can also say that the positive evaluations from outside the company are proof of improvements in quality. The *Review* is a forum for presenting our engineer’s research, the results of this research and the subsequent contributions Honda makes to society. The expectations of the journal continue to grow as one of Honda R&D’s missions. We hope that you will continue to actively utilize the *Review* as a corporate and personal resource.

(Noboru Sato: Tochigi R&D Center)

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