Broadly-applicable Adaptive Damper System without Use of Dedicated Sensors

Ryoma KANDA* Yujiro NISHI* Masaki IZAWA**

ABSTRACT

An adaptive damper system that does not require the use of dedicated sensors has been developed. Employing the proportional relationship between front wheel speed and contact load, which change with input from the road surface, the system estimates the vehicle's vertical vibration state quantity using only the wheel speed, longitudinal acceleration, and yaw rate signals conventionally employed for systems such as Vehicle Stability Assist and airbag systems in Supplemental Restraint System. This does away with the necessity for the use of sensors for the measurement of the vertical acceleration of sprung mass and suspension stroke sensors. Using the estimated values, the system boosts ride comfort through the application of damping force control by adaptive dampers in order to control vertical vibration. In addition, the system also offers unsprung mass control and pitch and roll control in order to boost road-holding performance and handling and stability performance. The adaptive damper units feature a triple-tube configuration with externally mounted pressure-control solenoid valves, making it possible to set the damping force characteristic and the range of variation in damping force to match the weight, size, and concept of different vehicles. The developed system is capable of application to a wide range of vehicle categories.

1. Introduction

A variety of electronically controlled adaptive damper systems have been proposed in efforts to reconcile handling and stability and ride comfort. More recently, control systems incorporating real-time control logics that feed back vehicle state data have been proposed[(1), (2)]. Honda employs an electronically controlled adaptive damper system that it terms the Adaptive Damper System, or ADS, in mass-production vehicles. The ADS uses a control logic based on skyhook control in order to balance the isolation from road surface input obtained by minimizing the damping force of the dampers with vibration suppression performance in relation to sprung mass resonance. This control logic is widely known as a method of controlling vertical vibration of the vehicle body in order to increase ride comfort. It is necessary for the purposes of this control to know the vertical velocity of the sprung mass and the damper stroke velocity. To enable calculation of these parameters, the conventional ADS system configuration has featured three dedicated sensors to detect the vertical acceleration of the sprung mass and suspension stroke sensors positioned on each wheel, as shown in Fig. 1. Because this large number of sensors and the required wiring has increased the overall cost of the system, the application of ADS has been limited to luxury vehicles. In order to deliver the luxurious ride comfort realized by ADS to a greater number of users, it will be necessary to reduce the cost of the system and promote its application in a wider range of vehicle categories.

In order to realize this goal, methods of applying skyhook control without the use of sensors for the measurement of the vertical acceleration of sprung mass and suspension stroke sensors, the dedicated sensors employed in the ADS, have been proposed[(3), (4)]. These methods make active use of wheel speed signals[(5)-(7)], which fluctuate with road surface input and the vehicle's dynamic state during operation. Obtaining signals from wheel speed sensors already fitted as part of the Vehicle Stability Assist (VSA) system via a Controller Area Network (CAN), the methods estimate the vehicle's vertical vibration from changes in wheel speed and damp said vibration through skyhook control.

In order to apply ADS in mass-production vehicles, in addition to increasing ride comfort by damping the sprung mass, it would also be necessary to boost

* Automobile R&D Center

** Showa Corporation

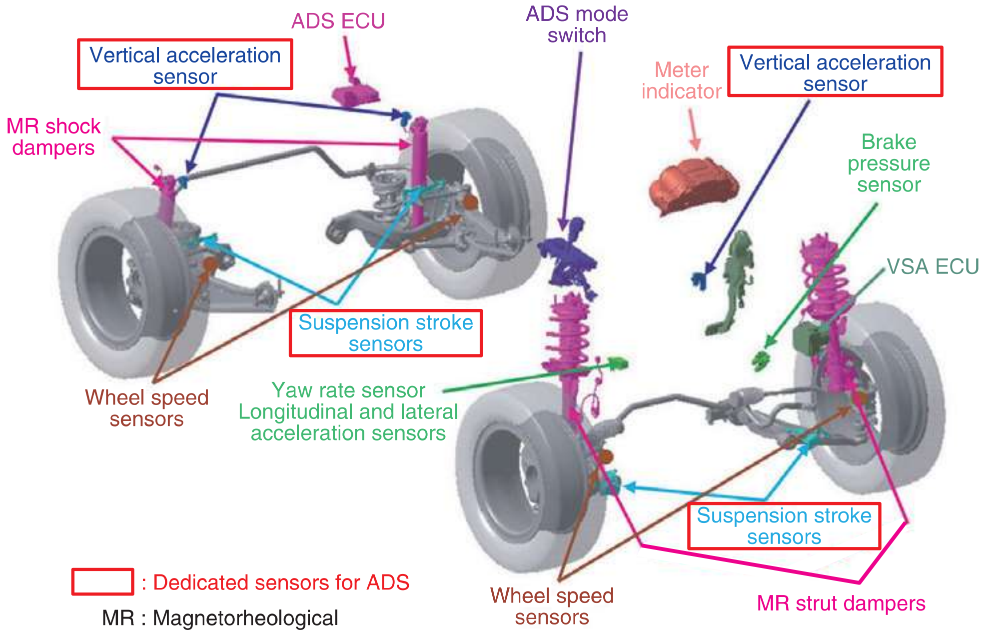

Fig. 1 Components of conventional ADS

road-holding performance and handling and stability performance by controlling posture changes in the pitch and roll directions while damping unsprung mass resonance. It would additionally be necessary to adjust the range of variation in damping force and the damping force characteristic of the adaptive damper units in accordance with the weight, size, and concept of a wide range of vehicle categories, from small cars to large vehicles such as SUV. The development process discussed in this paper created a system based on these considerations.

2. Development Aims

Looking toward the application of ADS in a wide range of mass-production vehicles, the following development aims were set in order to realize a system displaying a similar level of body damping performance to the conventional system while controlling system cost, and to enable adjustment of the damping force characteristic of the adaptive damper units in accordance with the weight, size, and concept of a range of vehicles.

(1) In addition to a skyhook control using only signals already employed by other systems, the realization of a control that suppresses unsprung mass resonance and changes in pitch and roll behavior in order to increase road-holding and handling and stability performance, while also maintaining ride comfort.

(2) Development of adaptive damper units in which the range of variation in damping force and the damping force characteristic can be adjusted for application in a wide range of vehicle categories.

3. Control Logic

3.1. Overview of Control Logic

This section will discuss the control logic employed in realizing the vehicle fitting and mass production of the system. ADS employs a control logic based on skyhook control in order to balance isolation from road surface input with the damping of sprung mass resonance. In addition, road-holding performance is increased by suppressing unsprung mass resonance, and handling and stability performance is increased by controlling changes in pitch and roll behavior resulting from acceleration and deceleration by the driver and steering input such as lane changes.

Figure 2 shows a block diagram of the control logic incorporated in an actual vehicle. As input signals to the control logic, vehicle speed at each wheel, longitudinal acceleration, lateral acceleration, yaw rate, and steering angle signals are supplied to the system by other systems [VSA, Supplemental Restraint System (SRS), and Electric Power Steering (EPS)] via a CAN.

The state quantity estimation part calculates the vertical velocity of the sprung mass and the damper stroke velocity necessary for the application of skyhook control based on changes in wheel speed[(3), (4)]. However, because in an actual driving environment changes in wheel speed during vehicle operation can be caused by factors including changes in vehicle speed and changes in the turning radius in addition to changes in the contact load, it is necessary to conduct pre-processing in order to remove these factors from the wheel speed signal in advance. In the wheel-speed-signal preprocessing block, processing is applied using the longitudinal acceleration signal and the yaw rate signal in order to remove changes in wheel speed due to acceleration, deceleration, and operation of the steering wheel from the front wheel speed signal. Following this, changes in the contact load are calculated from the preprocessed wheel speed signal and input to a quarter-car model for estimation (to be explained in the next section), making it possible to estimate the sprung mass velocity and damper stroke velocity for each wheel, which are essential parameters for the application of skyhook control.

The following four independent control blocks, each possessing a different function, are provided in the body control logic section; these control blocks use the estimated state quantities and other input signals in order to control changes in vehicle posture involving three degrees of freedom of the sprung mass of the vehicle body (vertical, pitch, and roll) and to control unsprung mass resonance.

(1) Skyhook control using estimated state quantity values controls sprung mass resonance while isolating the vehicle body from vibration, boosting ride comfort
(2) Pitch control using the longitudinal acceleration signal controls pitch behavior during acceleration and deceleration
(3) Roll control using the lateral acceleration signal and the steering angle signal controls roll behavior during operation of the steering wheel
(4) Unsprung mass control using the wheel speed signal controls unsprung mass resonance and boosts the road-holding performance of the tires

Pitch and roll control are determined on the basis of changes in the longitudinal and lateral acceleration signals, and unsprung mass control is determined on the basis of changes in the wheel speed signal. Output is restricted to a frequency range in which any effect on ride comfort will be minimized. The system's control output is set such that the level of control applied by each control block is adjusted in relation to vehicle behavior, and the maximum value for the output of each control block that will help ensure that there is no shortfall in damping force is selected.

3.2. Method of Estimation of Front Axle State Quantity

In order to apply skyhook control without the use of dedicated sensors, the system estimates the vertical velocity of the sprung mass and the damper stroke velocity, the essential parameters for control, from the wheel speed signal and other existing CAN signals. This section will discuss the method of estimation of the state quantity of vertical vibration using the proportional relationship between changes in wheel speed and changes in contact load.

In order to consider the effect of deflection of the tires due to fluctuation in the contact load as a result of road surface input, the relationship between vehicle speed and wheel speed in a vehicle traveling at a constant speed will be considered. With the vehicle speed as V and the tire radius as R, as shown in Fig. 3, and the standard tire radius during conversion from the angular velocity of the wheel to vehicle speed as R_0 (= constant), the vehicle speed calculated from the wheel speed sensor, V_0, is determined by Eq. (1)[(4), (7)].

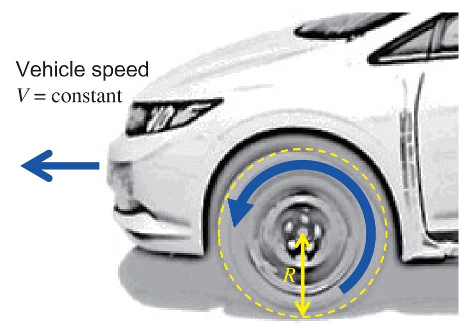

Fig. 3 Vehicle speed and wheel speed

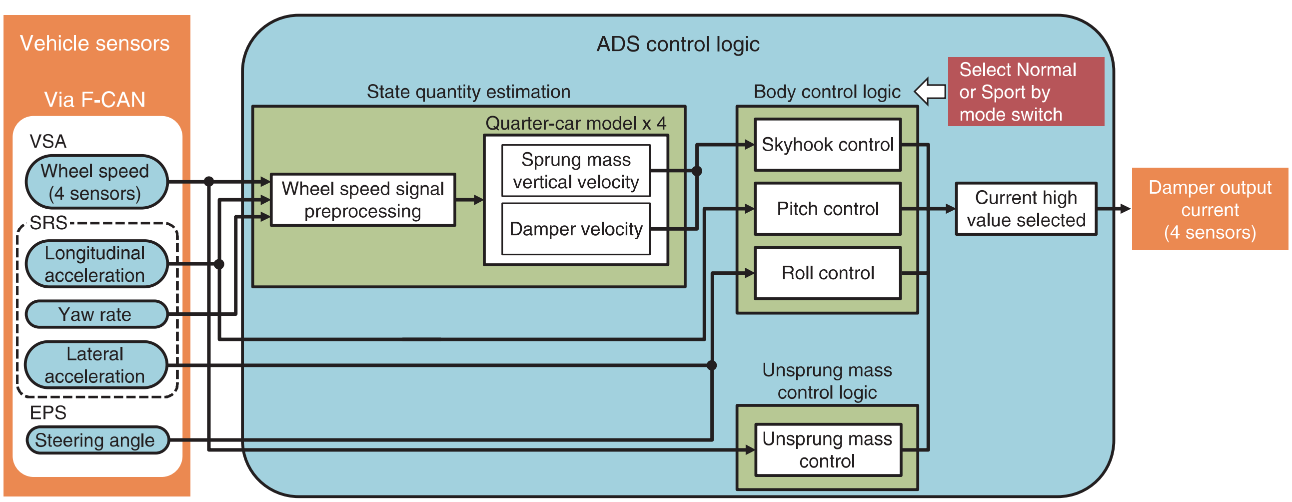

Fig. 2 Control logic block diagram

$$V_0 = \frac{R_0}{R} V \tag{1}$$

Considering the case in which, due to road surface input, R becomes ΔR lower than R_0 and V_0 becomes ΔV greater than V, Eq. (1) can be transformed into Eq. (2).

$$V + \Delta V = \frac{R_0}{R_0 - \Delta R} V \tag{2}$$

If, following Maclaurin expansion with consideration that $R_0 >> \Delta R$, the terms from the second term onwards are ignored as infinitesimal in order to simplify the expression of the relationship between ΔV and ΔR, Eq. (3) is generated.

$$\Delta V = \frac{\Delta R}{R_0} V \tag{3}$$

Expressing the change in contact load, ΔF, in terms of ΔR and the vertical spring constant of the tires, K_1, gives Eq. (4).

$$\Delta F = K_1 \Delta R \tag{4}$$

Here, the effect of the tire damping term is sufficiently low to be ignored. Substitution of the relationship shown in Eq. (3) into Eq. (4) gives Eq. (5).

$$\Delta F = \frac{R_0 K_1}{V} \Delta V \tag{5}$$

Because R_0, K_1, and V in Eq. (5) are constants, the change in the contact load, ΔF, is proportional to the change in wheel speed, ΔV.

Figure 4 shows the relationship between change in wheel speed and change in contact load for a front wheel, measured using a six-component force meter on the wheel of a vehicle proceeding straight ahead at 100 km/h on an undulating road. Signal processing was applied using a band pass filter to isolate the respective sprung mass resonance frequency bands in the wheel speed signal and the contact load signal. As Fig. 4 shows, change in the contact load displays a linear relationship with change in wheel speed in specific frequency bands, demonstrating that the relationship shown in Eq. (5) holds true under actual driving conditions[4].

Having demonstrated that change in the tire contact load can be determined from change in wheel speed, the determination of the vertical vibration state quantity, essential for the application of control, from change in the contact load will next be considered. Inputting the change in the contact load determined from the change in wheel speed, u_{1F}, to the unsprung mass, M_{1F}, in the three-degree-of-freedom single-wheel model shown in Fig. 5, makes it possible to calculate the vertical vibration state quantity, and to determine the sprung mass velocity, $\dot{x}_{2F}$, and the damper stroke velocity, $\dot{x}_{2F}$ - $\dot{x}_{1F}$, at the front wheel.

3.3. Method of Estimation of Rear Axle State Quantity

Figure 6 shows the relationship between change in wheel speed and change in contact load for a rear wheel during driving, with the same signal processing applied under the same conditions as for the front wheel in the preceding section. Because the relationship between the two parameters is not linear, as shown in Fig. 6, it would

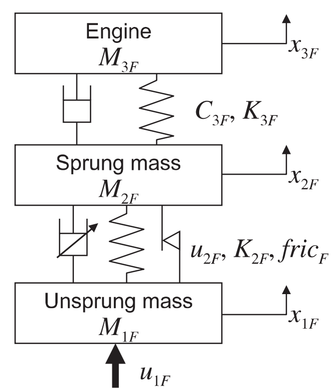

Fig. 5 Quarter-car model for estimation of state quantities for front axle

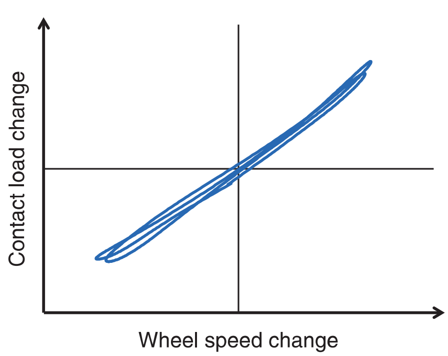

Fig. 4 Relationship between change in wheel speed and change in tire contact load for front wheel

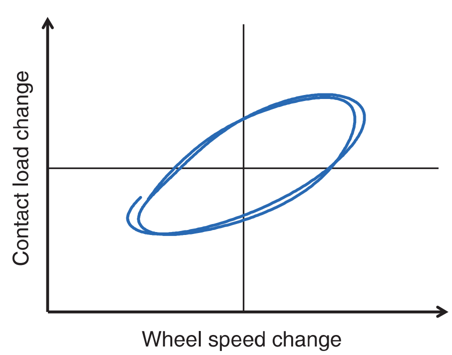

Fig. 6 Relationship between change in wheel speed and change in tire contact load for rear wheel

be challenging to estimate the vertical vibration state quantity for the rear wheels using the method discussed in the preceding section. This phenomenon is observed in suspension systems with a configuration that displaces the tire contact patch longitudinally with vertical displacement of the suspension. This configuration is designed to have an anti-lift effect that mechanically controls lift behavior in the rear wheels when the brakes are applied. For example, as the tire contact patch is displaced to the rear with displacement of the suspension towards the direction of compression with road surface input, the wheels move towards the rear of the vehicle body, and wheel speed is reduced. In the case of suspension systems employing an anti-lift geometry, change in wheel speed due to the longitudinal displacement of the wheels cannot be ignored, and because of this it is not possible to realize sufficient linearity between change in wheel speed and change in contact load[(4)].

As indicated above, it is challenging to estimate the contact load from wheel speed for the rear wheels, which often employ anti-lift geometry. Given this, a method that uses the front wheel speed and the estimated front wheel state quantity was proposed[(4)]. This method determines the vertical vibration state quantity for the rear wheels by employing the road surface displacement estimated at the position of the front wheels for the road surface displacement at the position of the rear wheels, with consideration of the time lag determined by the vehicle's speed and its wheelbase.

Figure 7 shows a block diagram of the principle used in estimation. The change in the contact load calculated from the change in wheel speed, u_{1F}, is input into the front wheel estimation model discussed above (Fig. 5), and the unsprung mass displacement, x_{1F}, is calculated when estimates of the state quantity for front wheel control are conducted. Replacing the tire vertical spring constant with K_{1F} and ΔR with x_{0F} - x_{1F} in Eq. (4), the contact load for the front wheels, u_{1F} (=ΔF), is determined by Eq. (6).

$$u_{1F} = K_{1F}\,(x_{0F} - x_{1F}) \tag{6}$$

If this equation is solved for road surface displacement at the front wheels, x_{0F}, road surface displacement at the contact patch for a front wheel can be calculated using Eq. (7).

$$x_{0F} = \frac{u_{1F}}{K_{1F}} + x_{1F} \tag{7}$$

If the wheelbase is termed WB, the time for the rear wheels to reach a point of the road surface passed by the front wheels, T_{delay}, can be expressed as Eq. (8).

$$T_{delay} = \frac{WB}{V} \tag{8}$$

Road surface displacement at the rear wheels estimated from the front wheels, x_{0R}, can therefore be expressed as Eq. (9).

$$x_{0R}(t) = x_{0F}(t - T_{delay}) \tag{9}$$

Inputting the estimated road surface displacement when the rear wheel passes, x_{0R}, to a two-degree-of-freedom rear-wheel estimation model makes it possible to calculate the vertical vibration state quantities for the rear wheels, and to determine the sprung mass velocity, $\dot{x}_{2R}$, and the damper stroke velocity, $\dot{x}_{2R}$ - $\dot{x}_{1R}$.

3.4. Estimation Results for Vehicle State Quantity

Because the estimated vehicle state quantities are intended for use in real-time control, the state quantities for the front wheels during actual driving are estimated using real-time processing. Figure 8 shows an example of the results. The graphs show the time series waveforms of the estimated values of the vertical vibration state quantities for a vehicle traveling straight ahead at a speed of 100 km/h on an undulating road. The upper graph shows the front sprung mass velocity, and the lower graph shows the front damper stroke velocity. Results for sprung mass velocity calculated

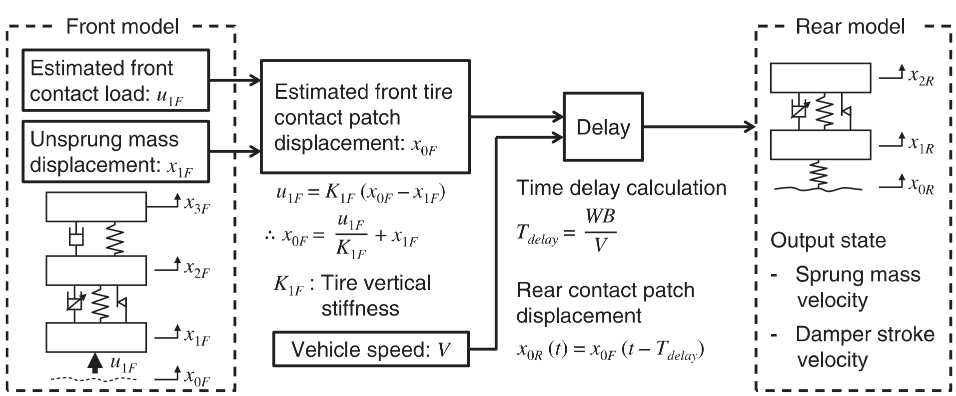

Fig. 7 Conceptual diagram of state quantity estimation for vertical vibration

from sprung mass acceleration sensor output and results for damper stroke velocity calculated from suspension stroke sensor output are superimposed on the estimated results. The results demonstrate that it is possible to estimate state quantities for the front wheel axle using the front wheel speed signal.

Similarly, Fig. 9 shows estimation results for the rear wheel axle. The graphs show that phase lag is reduced in results for the rear state quantities estimated from front wheel speed and estimated values in comparison to results estimated using the rear wheel speed signal.

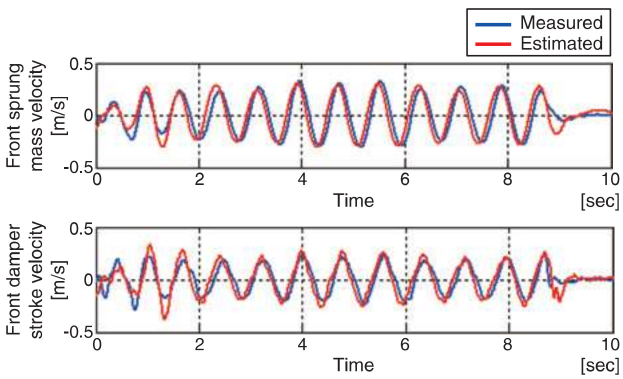

Fig. 8 Comparison of estimated waveforms of vertical vibration state quantity for front axle

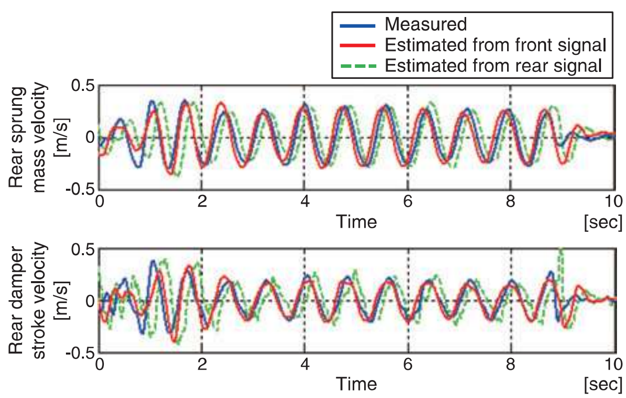

Fig. 9 Comparison of estimated waveforms of vertical vibration state quantity for rear axle

4. System Configuration

4.1. Sensors

Figure 10 shows the configuration of the developed system. Because the system does away with the three sensors for the measurement of the vertical acceleration of sprung mass and the suspension stroke sensors at each wheel employed in the conventional system, it has a simple configuration, with the only parts exclusively for ADS use being the control ECU and the adaptive damper units positioned at each wheel. This has done away with the weight and the cost of the sensors, sensor brackets, couplers, and wiring. Input signals are provided via the CAN. The wheel speed signal is obtained from the VSA system, the longitudinal and lateral acceleration signals and the yaw rate signal are obtained from the SRS airbag system, and the steering angle signal is obtained from the electric power steering (EPS) system.

4.2. Adaptive Damper Unit

Because the damping force changes with the weight, size, and concept of the vehicle, it was necessary to adjust the damping force characteristic of the adaptive damper units accordingly. In addition, a wide range of variation in damping force would also be necessary in order to make it possible for the effect of damping force control by the adaptive dampers to be displayed in a greater range of scenarios, and it was therefore desirable for damping force to be variable for both the tension and compression strokes of the dampers.

Based on these considerations, adaptive damper units employing replaceable, externally mounted pressure control solenoid valves were employed in the developed system. Because solenoid valves are able to adjust the pressure characteristic in relation to the oil flow rate, replacing the valves makes it possible to change the damping force characteristic of the adaptive dampers.

Figure 11 shows schematic diagrams of the configuration of the triple-tube adaptive dampers. The diagrams show schematic images of the respective oil flow during the tension and compression strokes. As the piston rod moves during the tension stroke, the piston valve closes and oil flows from the upper damper chamber to the solenoid valve via the inner tube. At this time, negative pressure in the lower damper chamber opens the bottom valve, and oil is supplied from the reservoir chamber. As the piston rod strokes during the compression stroke, the bottom valve closes, and increased pressure in the lower damper chamber opens the piston valve, and oil flows into the upper damper chamber. Following this, the flow becomes the same as that during the tension stroke, and oil is supplied to the solenoid valve. This means that oil flows to the solenoid valve from the same direction during both strokes, making it possible to control the damping force during both strokes using a single solenoid valve.

Figure 12 shows an example of a damping force characteristic. The graph shows the load characteristic in relation to piston velocity when currents that produce the minimum and maximum damping force are impressed in the coil. An example of the damping force characteristic of a conventional damper is included for comparison. Figure 12 shows that the current impressed in the coil is able to change the damping force characteristic during both tension and compression strokes. In addition, the adaptive damper damping force characteristic shown in the graph encompasses the damping force characteristic of the conventional damper.

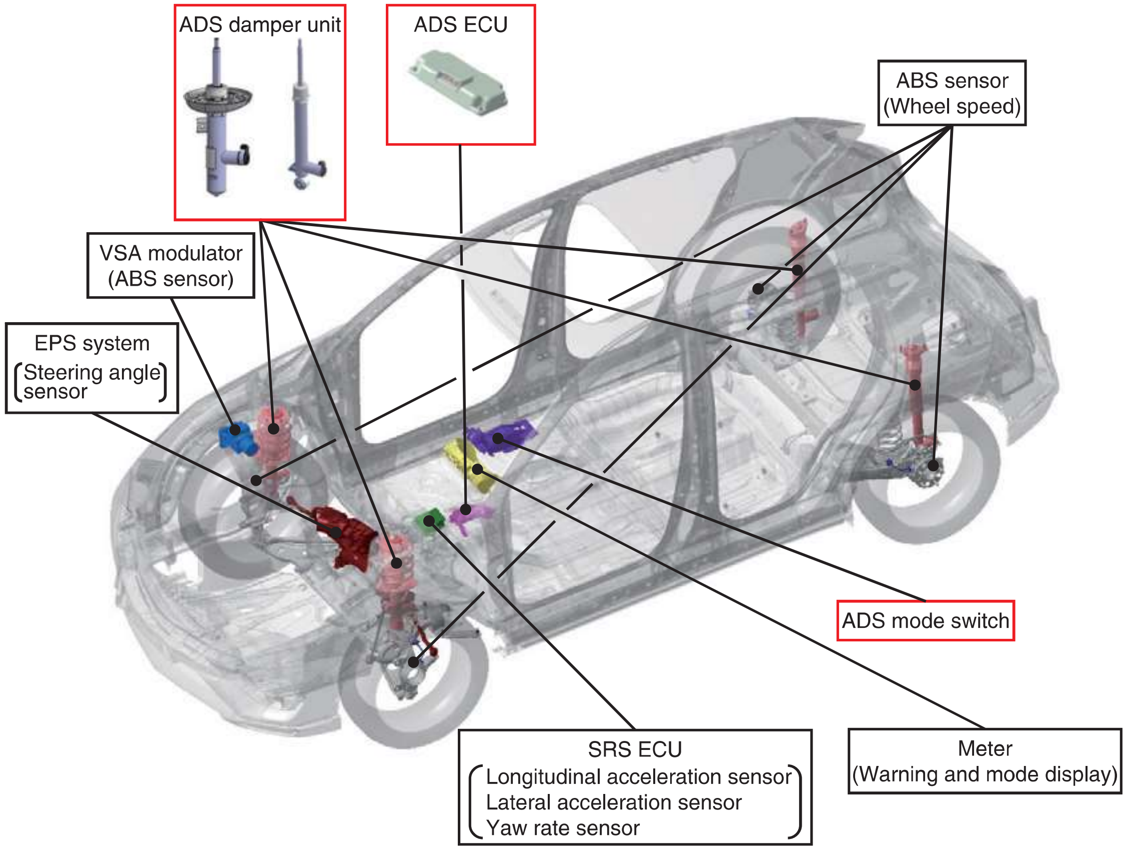

Fig. 10 Components of ADS without dedicated sensors

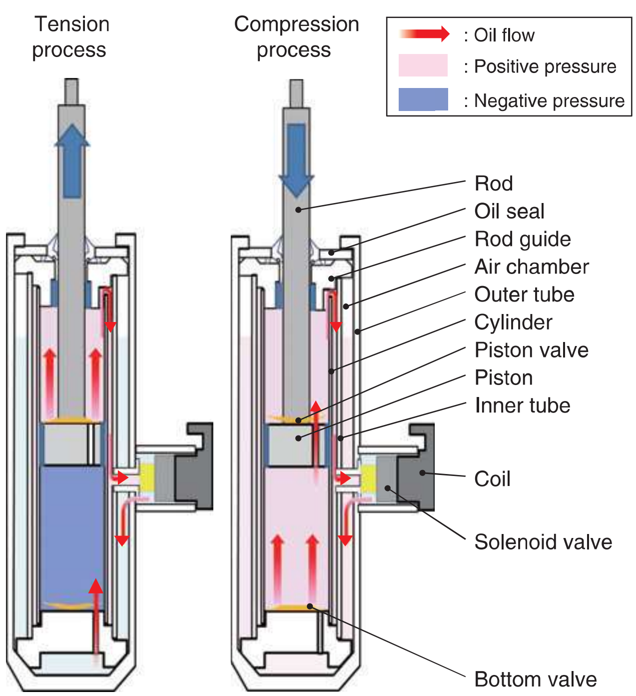

Fig. 11 Triple tube damper configuration

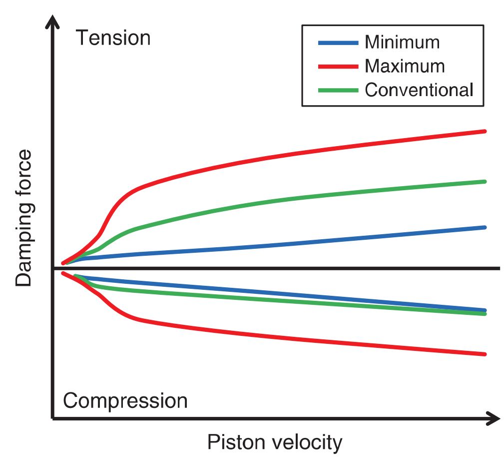

Fig. 12 Typical damper force characteristic

Figure 13 shows an example of the damping force characteristic of an adaptive damper unit adjusted through replacement of the solenoid valve. As the graph shows, it was possible to set the maximum damping force higher and the minimum damping force lower during both the tension and compression strokes.

The damping force characteristic of the adaptive damper unit can be changed by replacing the solenoid valve, enabling adjustment of the characteristic in accordance with the weight, size, and concept of vehicles in a wide range of categories.

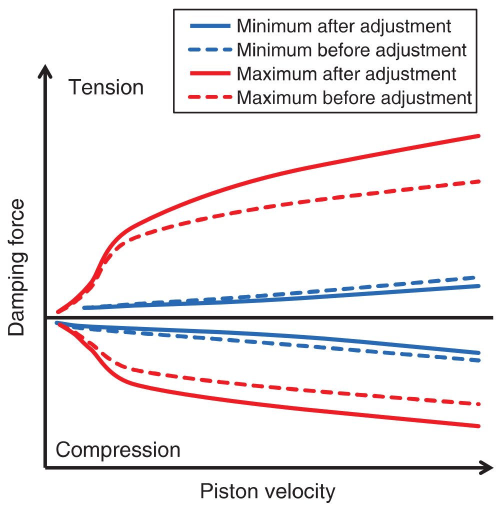

Fig. 13 Example of adjusted damper force characteristic

5. Actual Vehicle Trials

In order to verify the effectiveness of the developed system in an actual vehicle, acceleration sensors were fitted to the damper mounting points of a test vehicle and sprung mass acceleration signals were measured during actual driving. The developed system was fitted in the test vehicle, and the damping force of the adaptive dampers was controlled without the use of dedicated sensors. To make it possible to compare results for the developed system with conventional ADS using dedicated sensors, sensors for the measurement of the vertical acceleration of sprung mass and suspension stroke sensors for control purposes were also fitted in the vehicle, and a switch was installed to make it possible to switch between the developed system and the conventional system. Measurements focused on comparing vibration suppression performance in the 1 Hz to 2 Hz range of sprung mass resonance and isolation performance in the 3 Hz to 8 Hz range. The tests were conducted on a test course modeling an urban road with frequent input from the road surface in these frequency bands.

To begin, tests were conducted to determine the effect of conventional ADS using dedicated sensors in controlling vertical vibration of the vehicle body. Figure 14 shows an example of a comparison of the power spectral density (PSD) of the vertical acceleration of sprung mass at the rear wheels for control by conventional ADS, the use of minimum damping force without the application of control, and the damping force provided by a conventional damper. The results show that the application of control by the conventional ADS resulted in greater suppression of vibration close to the 1 Hz to 2 Hz range of sprung mass resonance than the use of minimum damping force. Control by the conventional ADS also reduced the level of vibration close to the 3 Hz to 8 Hz range for isolation to a lower level than the damping force provided by the conventional damper. These results indicate that skyhook control is able to control sprung mass vibration irrespective of the frequency range, and increases ride comfort against the use of conventional dampers. Unsprung mass control, which boosts road-holding performance, and pitch and roll control, which boost handling and stability, were also applied, and results showed no increase in the vibration level in the range for isolation or other effects that would result in a decline in ride comfort.

Next, PSD for the vertical acceleration of sprung mass was compared for conventional ADS and the developed system in order to compare the effectiveness of the systems in controlling vehicle vibration. Figure 15 shows the

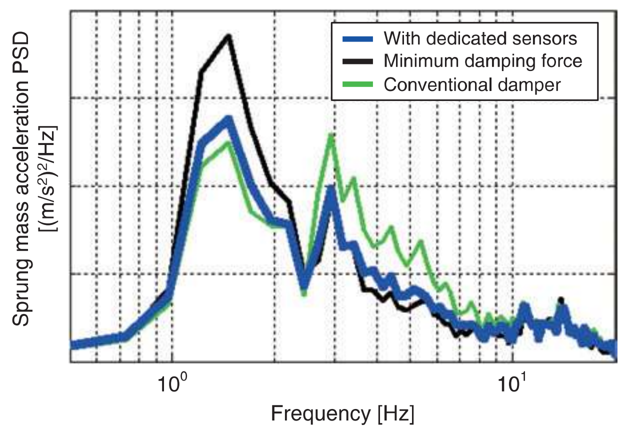

Fig. 14 Example of PSD of sprung mass acceleration for ADS using dedicated sensors

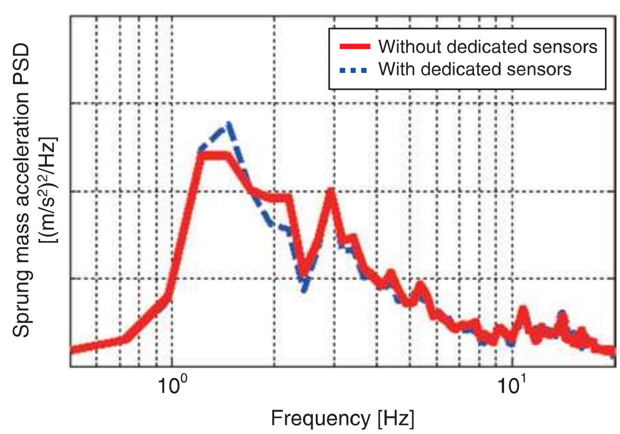

Fig. 15 Comparison of PSD of sprung mass acceleration for ADS using dedicated sensors and ADS using estimation

results of this comparison. The level of vertical vibration of sprung mass for the developed system is similar to that for conventional ADS in the range of sprung mass resonance and the range for isolation.

These results demonstrate that control using the developed system produces a similar vibration characteristic to control using the conventional system, for which sensors for the measurement of the vertical vibration of sprung mass and suspension stroke sensors are necessary, and that the developed system is able to realize the same effect in controlling vehicle vibration as the conventional system.

6. Conclusion

An ADS that reduces system costs while offering a wide range of damping force settings and applicability to diverse vehicle categories has been developed.

(1) The development of a control logic that estimates the state quantities of vertical vibration has made it possible to control vertical vehicle vibration using skyhook control without the need for dedicated sensors. In addition, the system also realizes unsprung mass control and pitch and roll control that maintain ride comfort.

(2) The system employs adaptive damper units with a triple-tube configuration that enables control of damping force during both the tension and contraction strokes using a single solenoid valve. The damping force characteristic can be adjusted for different vehicle types through replacement of the externally mounted solenoid valve.

The developed ADS has been employed in the Acura AVANCIER SUV, the UR-V, and the CDX compact SUV, and is scheduled for future application in other models.

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■ Author ■

Ryoma KANDA

Yujiro NISHI

Masaki IZAWA