

Development of Electronically Controlled Braking System with High Performance and Applicability

Yusuke Kamiya¹, Yoshio Masuda¹, Masaaki Komazawa², Yasuji Mizutani², Takahiro Okano²

¹ADVICS CO., LTD., Chassis System Development Dept.1,

2-1, Showa-cho, Kariya, Aichi, 448-8688, Japan, yusuke_kamiya@nts.advics.co.jp

²TOYOTA MOTOR CORPORATION, Chassis Development Div.

1, Toyota-cho, Toyota, Aichi, 471-8572, Japan, masaaki_komazawa@mail.toyota.co.jp

Summary

This paper proposes an electronically controlled hydraulic braking system for advanced vehicles such as hybrid/pure electric and automatic vehicles. This braking system contains a high-pressure accumulator, brake-by-wire composition using a piston gap, and a hydraulic pressure controller combined with a mechanical valve and linear solenoid valves satisfying the requirements of various brake functions.

Keywords: braking, regenerative braking, electronic, HEV (hybrid electric vehicle), EV (electric vehicle)

1 Introduction

The research and development of pure electric vehicles (EV), hybrid electric vehicles (HEV), and fuel cell electric vehicles are conducted at several corporations, universities, and research institutes to obtain a solution for exhaust gas reduction and energy related problems. For the braking system of those vehicles, a regenerative-friction braking coordination technology, in which the brake force requested by the driver is controlled by blending the regenerative motor power and hydraulic pressure force, plays an important role in enhancing the fuel efficiency.

On the other hand, to improve the efficiency, safety, and convenience of operation, the behavior of a vehicle is controlled instead of the operation of the driver. For example, adaptive cruise control, intelligent parking assistance systems, and autonomous emergency braking systems are relevant. Furthermore, pure automatic vehicles have been researched and developed in various industries. As a part of these trends, sensing technologies that detect the conditions around vehicles have been attracting particular attention. However, a braking technology that controls the behavior of a vehicle after the detection of the conditions around the vehicle is still extremely important.

This paper reviews the requirements of such a braking system for these advanced vehicles and proposes a new electronically controlled hydraulic braking system and a key technology of the system to achieve high performance and applicability.

2 Requirements of Braking System for Advanced Vehicles

The following are the requirements for the braking system of the advanced vehicles introduced in the previous section.

2.1 Independent Power Source and Devices Controlling Arbitrary Pressure

For HEV/EV, it is necessary to have an energy source for the braking system independent of the engine manifold vacuum because the brake servo energy is not obtained when the engine stops in HEV and at all occasions in pure EV, which does not have an engine.

Now, regenerative-friction braking coordination that plays an important role in improving the fuel consumption of HEV/EV shall be described. For increasing the regenerative efficiency during deceleration, it is desirable to realize the brake force requested by the driver by combining a hydraulic brake and a regenerative brake as shown in Fig.1 [1]. In regenerative-friction braking coordination, the regenerative brake force changes continuously because of factors such as vehicle speed (regenerative motor revolution) and the charge of the battery. The total brake force combined with the friction brake force shall realize the deceleration requested by the user.

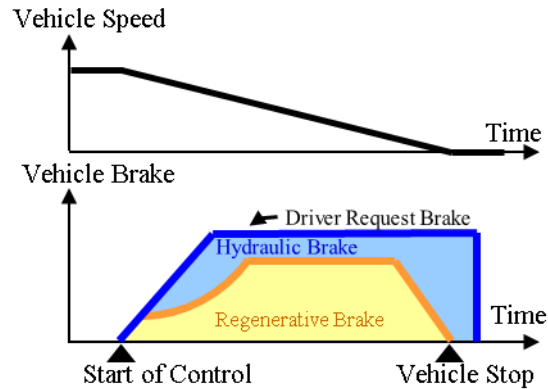


Figure 1: Behavior of the combination of the regenerative and hydraulic brakes.

On the other hand, traction control systems (TCS), electric stability control (ESC), and autonomous emergency braking functions require the generation of an optional brake force without the use of the brake pedal by the driver.

For that reason, a brake power source independent of the engine manifold vacuum servo and a device that can realize an arbitrary hydraulic pressure with high accuracy are needed. In this paper, the portion of the braking system that controls the hydraulic pressure is referred as the pressure application unit, and it consists of a brake power supply unit and a hydraulic pressure control unit.

2.2 Brake-by-wire Composition

In regenerative-friction braking coordination, the brake input portion that is operated by the driver and the brake output portion that provides the hydraulic pressure must be separated so that the hydraulic brake pressure is not generated directly by the brake pedal operation of the driver. In general, the composition that satisfies this requirement is referred as brake-by-wire composition.

In a brake-by-wire composition, the brake pedal senses the operation of the driver. The brake hydraulic pressure is generated electronically. When a failure of the braking system occurs, the brake-by-wire composition must be stopped and a backup mode in which the brake pressure is generated directly by the driver must be switched on.

From the point of view of brake pedal feeling, in the conventional braking system, a reaction force is generated by the brake caliper flexibility against the brake pedal operation. On the other hand, the brake-by-wire composition needs a device that can simulate the reaction force. For a good brake feeling, the brake pedal reaction force should be simulated without discomfort under any situation.

3 Proposal of Pressure Application Unit

This section describes a pressure application unit for the braking system described in the previous section.

3.1 Requirements for Various Brake Functions

Table 1 lists the requirements for each brake function of advanced vehicles.

Table 1: Outline of braking force control performance of pressure application unit required for each brake function.

		Required performance		
		Size of braking force	Rate of braking force change	Accuracy of braking force
Type of application	Regenerative-friction braking coordination	Small	Low	High
	Service brake	Small to Large	Low to Extremely High	High
	Autonomous emergency braking function	Large	High to Extremely High (according to the sensor type and compatible vehicle speed range of system)	Medium
	Traction control	Small	Medium	Medium
	Vehicle stability control	Large	High	Medium

3.1.1 Requirements for Regenerative-Friction Braking Coordination and Service Brake

Generally, the maximum deceleration realized by the regenerative brakes is in the range of 1–3 m/s². Furthermore, the fuel efficiency can be improved by increasing the maximum deceleration. However, increasing the maximum deceleration also increases the weight of the system and the vehicle. Therefore, the required vehicle braking deceleration is likely to remain in the range.

Regenerative brakes have three distinguishing features.

1. Regenerative brakes operate with high accuracy and fast response [2].
2. Unlike hydraulic brakes, which exert a braking force to stop a vehicle, regenerative brakes exert a braking force in the direction of travel as well as the opposite direction. Consequently, if the regenerative brakes are used continuously, the vehicle may start to travel in the opposite direction. Therefore, the action of the regenerative brakes must be halted before the vehicle stops.
3. Depending on the state of various systems, such as the battery charging state, it may not be possible to achieve the desired braking force using the regenerative brakes. However, it is possible to identify whether the regenerative brakes can achieve the desired braking force in advance.

An electronically controlled hydraulic unit must be capable of securing the required vehicle braking force while ensuring compatibility with these features of the regenerative brakes. Specifically, it is important to achieve accurate values of relatively small braking forces. However, from the standpoint of the speed of response, an extremely high performance is not a requirement because the operation of the regenerative brakes is identified and limited in advance.

If the function of the service brake is performed by the electronically controlled hydraulic unit during regenerative-friction braking coordination, the following points are also required.

4. A large braking force must be exerted to achieve the required vehicle deceleration.
5. Braking force must be changed rapidly in accordance with various brake pedal operations of the driver.
6. When a fault occurs in the electronically controlled hydraulic unit, the residual performance must be equivalent to that of the conventional system.

3.1.2 Requirements for Autonomous Emergency Braking Function

Autonomous emergency braking systems are control devices that use various sensors to detect vehicles, pedestrians, and objects around the vehicle and then generate a braking force regardless of the operation of the driver. These systems may cause driver discomfort if the braking force is generated too early. An electronically controlled hydraulic unit in an autonomous emergency braking system would be required to generate a vehicle braking force equivalent to 10 m/s² in as short time as possible. Because the timing of generation of the braking force that leads to driver discomfort is virtually unique to each driver, the applicable range of autonomous emergency braking functions can be enlarged by shortening the time in which the required braking force is achieved.

3.1.3 Requirements for TCS and ESC

TCS and ESC are applications that stabilize the vehicle motion by applying a braking force to each wheel individually in accordance with the behavior of the vehicle and the road surface. Excluding certain systems [3], this is usually achieved by transmitting the master cylinder pressure generated by pressure application actuators to each wheel appropriately by controlling the solenoid valves provided at each wheel.

TCS is achieved by a coordinated control with the engine and the motors as the power sources, mainly in accordance with the friction (μ) of the road surface, when the vehicle starts moving. The required master cylinder pressure is relatively low, and the rate of change of the master cylinder pressure is normal.

In contrast, ESC stabilizes the changes in the lateral behavior of the vehicle while driving using a coordinated control with the power sources described above. In this case, the required master cylinder pressure is high, and the rate of change is also high.

3.2 Proposal for Pressure Application Unit for Various Brake Functions

This subsection describes a system configuration that satisfies these requirements. Table 2 shows the relationship between the type of the pressure application unit used in the braking system and the hydraulic control performance.

Table 2: Relationship between pressure application unit and hydraulic control performance.

			Hydraulic control performance (the second row indicates the required level)		
Pressure application unit			Pressure increase performance	Pressure control accuracy	Residual performance if power supply fault occurs
Power supply	Hydraulic pressure control unit		High	High	Desirable performance
1	Auxiliary battery	Cylinder-driven electric motor	High	High	×*1
2		Pump and solenoid valve	Medium	High	×*1
3	High-pressure fluid	Linear solenoid valve	High	High	△*2
4	built up in accumulator	Regulator*3	Extremely High	Medium	○*4

Notes:

*1 If a backup power supply equivalent to the main power supply is present, the electric motor that acts as the pressure control unit will continue to function even if a fault occurs in the main power supply system enabling pressure control by the electronically controlled hydraulic unit.

*2 A relatively small backup power supply drives the linear solenoid valve enabling pressure control by the electronically controlled hydraulic unit even if a fault occurs in the main power supply system.

*3 Practically adopted only for the systems that drive the regulator using the brake pedal operation [4]

*4 Because the regulator can be driven using the brake pedal operation of the driver, the pressure control can be achieved by using the electronically controlled hydraulic unit even if a fault occurs in the main power supply system.

The first pressure application unit listed in Table 2 uses an auxiliary battery as the power supply and a cylinder-driven electric motor as the pressure control unit. This system has already been proposed [5] and adopted practically. The advantages of this system include high performance in increasing the pressure, accurate pressure control, and compliance with various applications. However, a major disadvantage of this system is that the pressure application unit itself stops operating if a fault occurs in the power supply system. Therefore, if the pressure application unit is used for the service brake functions as well, countermeasures such as redundant power supply systems should be considered.

The second pressure application unit listed in Table 2 uses an auxiliary battery as the power supply and a pressure control unit that combines a pump for increasing the pressure and solenoid valve. This system has also been adopted practically [1]. This system generally uses ESC actuators commonly adopted in conventional vehicles. Automakers are accustomed to using these actuators, thereby increasing the applicability of this system. However, the performance of this pressure application unit in increasing the pressure is comparatively poor than the other systems. For this reason, this pressure application device is rarely used when service brake functions are required.

The third pressure application unit in Table 2 uses the high-pressure fluid built up in the accumulator as the power supply and linear solenoid valves for the pressure control unit. Because this system has an extremely good performance in increasing the pressure, it is mainly adopted in strong hybrid vehicles for braking

The fourth pressure application unit in Table 2 uses the high-pressure fluid built up in the accumulator as the power supply. However, this system uses a regulator as the pressure control unit. This regulator mechanically controls the inflow, maintenance, and discharge of the high-pressure fluid in response to the brake pedal operation of the driver. This system has an extremely high performance in increasing the pressure, and it is mainly adopted in large vehicles [6]. In addition, although the pressure control accuracy is limited by the individual mechanical hysteresis of the regulator, the performance is sufficiently accurate to respond to the brake pedal operations of the driver. However, it has not yet been applied to applications requiring a highly accurate pressure control, such as regenerative-friction braking coordination.

In addition, this system can also be used for the hydraulic control for a certain period if a fault occurs in the main power supply system without requiring an auxiliary power supply.

An overview of the system is illustrated in Fig. 2. The system consists of a master cylinder with an input piston and an output piston, stroke simulator, power supply unit, hydraulic pressure control unit, anti-lock braking system (ABS), brake pedal stroke sensor, and an ECU. The hydraulic pressure control unit contains a regulator and solenoid valves.



In the master cylinder, an input piston linked with the pedal operation compresses the fluid in the first reaction force chamber connected to a stroke simulator, and a reaction to the foot on the pedal is generated. The first reaction force chamber is also connected to the second reaction force chamber placed at the rear of the output piston so that the fluid pressure of the reaction force chambers is maintained during the operation of the output piston by the fluid transfer. The two reaction force chambers are connected with a normally closed piston gap-holding valve. The second reaction force chamber is connected to a reservoir tank with a normally opened simulator release valve. If a malfunction of the system is detected by the ECU, to realize the deceleration with the brake pedal operating force, the piston gap-holding valve is closed and the simulator release valve is opened so that the output piston is operated by the input piston directly.

The power supply unit has an accumulator that stores the high-pressure fluid using a pump driven by an electric motor.

In the hydraulic pressure control unit, a linear solenoid valve for increasing the pressure and a regulator are connected to a high-pressure accumulator. The regulator controls the pressure of the servo chamber being driven by the pilot chamber pressure generated by the operation of the linear solenoid valves.

The master cylinder is operated by the pressure of a servo chamber located at its back face, and it supplies the fluid to four wheel cylinders via the ABS unit. The ECU calculates the required brake force based on the brake pedal operation detected by the pedal stroke sensor and distributes it to the regenerative brake force and friction brake force taking into account the status of the regenerative braking system and the vehicle. The master cylinder exerts a brake fluid pressure required for realizing the friction brake force by operating the pressure application unit (power supply unit and hydraulic pressure control unit).

As mentioned above, for reducing the fuel consumption, which is one of the most important features of HEV/EV, it is desirable to secure a regenerative brake as large as possible. To realize this, a piston gap is placed between the input piston, which is operated by the brake pedal operation of the driver, and the output piston, which is operated by the pressure application unit, so that the brake fluid pressure is not generated by the brake pedal operation of the driver. The required piston gap stroke is decided by the relationship between the pedal stroke and deceleration as well as by the recovered regenerative brake.

When the brake force requested by the driver is increased, the piston gap narrows down, and the output piston is operated by the pressure application unit to realize the friction brake force. When the output piston operates rapidly, a delay in the transfer of the fluid between the first reaction force chamber, which faces the input piston, and the second reaction force chamber, which is located at the back of the output piston, could occur. Then the pedal stroke fluctuates to compensate for the excess/lack of fluid in the first reaction force chamber, and the driver tends to feel discomfort at the brake pedal. With an increase in the pressure, the pedal moves forward because the fluid in the first reaction force chamber is insufficient, and with a decrease in the pressure, the pedal moves backward because the fluid in the first reaction force chamber is excessive. The delay in the transfer of fluid is generated by the orifice action of the piston gap-holding valve. The excess/lack of fluid was eliminated by using the device shown below.

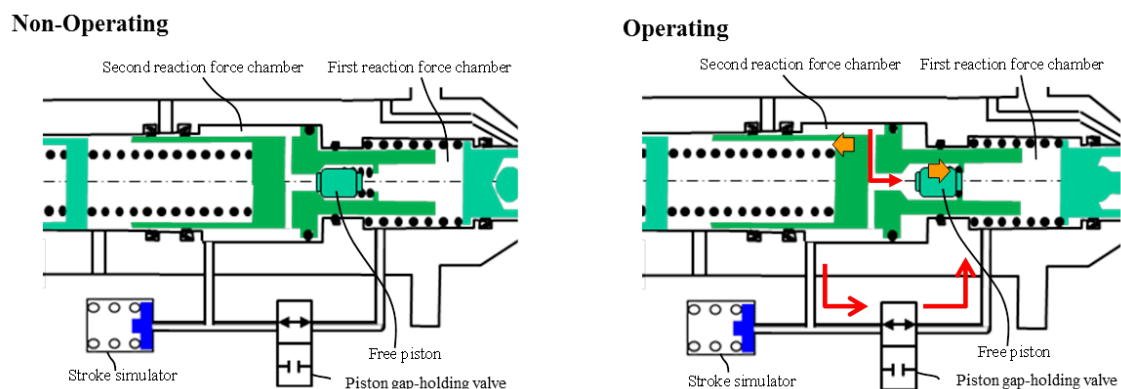


Figure 3: Reaction force pressure compensation mechanism

A reaction force pressure compensation mechanism with an interposed free piston is constructed by arranging a passage that penetrates the output piston and connects the first reaction force chamber with the second reaction force chamber. The excess/lack of fluid can be compensated indirectly by expanding or compressing the capacity of each chamber with the operation of the free piston.

5 Control Method of Pressure Application Unit

This section describes a control system for the pressure application unit that plays an important role in the realization of various brake functions.

5.1 Pressure Application Unit Control System Design

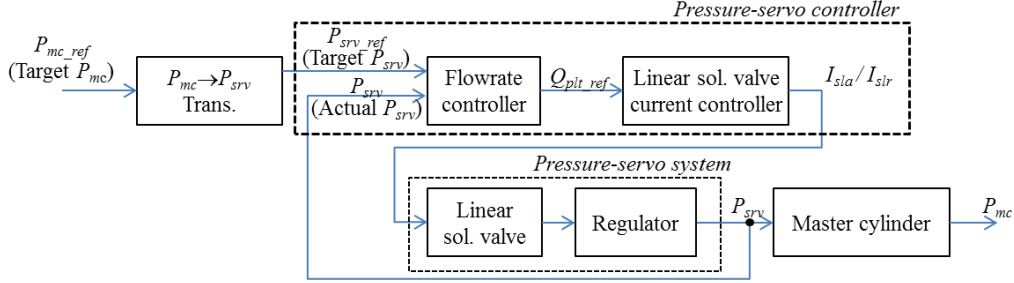


Figure 4: Block diagram of the pressure application unit control system.

Fig. 4 shows a block diagram of the pressure application unit control system. The pressure application unit consists of a pressure-servo controller (mainly comprising a microcomputer and integrated chips (ICs)) and a pressure-servo system (mainly comprising the linear solenoid valves and regulator). The pressure-servo system generates the servo pressure (P_{srv}) that is applied to the servo chamber contact with the back surface of the master cylinder. This generates a pressure (P_{mc}) in the master cylinder leading to the generation of a wheel cylinder pressure at each of the four wheels. Because the pressure application unit detects P_{srv} using the pressure sensors and controls P_{srv} using the linear solenoid valves and the regulator, the target servo pressure (P_{srv_ref}) is calculated considering the spring reaction force of the master piston at a predetermined master pressure (target master pressure: P_{mc_ref}) and the resistance of the sealing material.

The pressure-servo controller consists of a flow rate controller, which calculates the flow rate of the pilot chamber (Q_{plt_ref}) to be applied to the regulator based on P_{srv_ref} and P_{srv} , and a linear solenoid valve current controller, which generates the linear solenoid current for achieving Q_{plt_ref} .

The following section describes the operation of the pressure-servo system as outlined in Table 3.

Table 3: Outline of pressure-servo system operation.

State	Pressure-servo system operation	State	Pressure-servo system operation
No control		Release	
Apply		Hold	

The linear solenoid valve for increasing the pressure (SLA) is normally closed. One end of the valve is connected to the high-pressure accumulator (ACC) and the other to the pilot chamber inside the regulator valve. The SLA allows the high-pressure fluid from the accumulator into the pilot chamber in accordance with an opening angle adjusted by the electromagnetic force generated by the application of current (I_{sla}) to the valve. In other words, the SLA controls the flow rate from the accumulator based on the applied current to increase the pilot chamber pressure (P_{plt}).

The linear solenoid valve for decreasing the pressure (SLR) is normally open. One end is connected to the pilot chamber inside the regulator valve and the other to the reservoir tank (atmospheric pressure). The

SLR controls the discharge flow rate of the fluid from the pilot chamber to the reservoir tank in accordance with an opening angle adjusted by electromagnetic force generated by the application of current (I_{slr}) to a coil. In other words, the SLR controls the discharge flow rate based on the applied current to decrease the pilot chamber pressure.

The servo pressure control is performed by the regulator as follows. The regulator consists of a spool plug, pilot chamber, and servo chamber. The pilot chamber is connected to the input side of the spool plug, and the pilot chamber pressure is generated in accordance with the amount of fluid flowed in or discharged by the SLA and the SLR. In the process of increasing the pressure, the pilot chamber pressure is increased, thereby generating a force to push the spool plug toward the servo chamber. When the spool plug moves toward the servo chamber and the edge of the spool plug (for increasing the pressure) reaches the pressure increase port, the accumulator and servo chamber are linked. The high-pressure fluid then flows from the accumulator and increases the servo pressure. If the servo pressure is greater than the pilot pressure, a force is generated to move the spool plug back toward the pilot chamber breaking the link with the accumulator and stopping the increase in the servo pressure. In contrast, in the pressure reduction process, the pilot pressure is decreased, thereby pushing the spool plug back toward the pilot chamber. When the spool plug moves toward the pilot chamber and the edge of the spool plug (for reducing the pressure) reaches the pressure reduction port, the servo chamber and reservoir tank are linked. The fluid is then discharged into the reservoir tank, and the servo pressure decreases. If the servo pressure is smaller than the pilot pressure, a force is generated to move the spool plug toward the servo chamber breaking the link with the reservoir tank and stopping the reduction in the servo pressure. Finally, in the pressure maintenance process, the pilot pressure does not change. Therefore, the spool plug does not move, and the servo pressure does not change.

In this way, the spool plug is moved by controlling the pilot pressure enabling the generation of the desired servo pressure. In addition, because the regulator operates automatically to balance the input pressure (pilot pressure) and the output pressure (servo pressure), large output flow rates can be obtained with only small input flow rates allowing the use of the regulator as a flow rate amplifier.

The pressure-servo controller that functions to obtain the desired P_{srv} operates as follows. A particularly important element of the pressure-servo controller is the flow rate controller that calculates the target flow rate Q_{plt_ref} applied to the pilot chamber. The flow rate controller uses a control system with two degrees-of-freedom (2DOF) that is capable of tracking a target value and has excellent disturbance suppression characteristics to achieve a highly accurate pressure control required by the regenerative-friction braking coordination. Fig. 5 illustrates this 2DOF control system based on the flow rate controller.

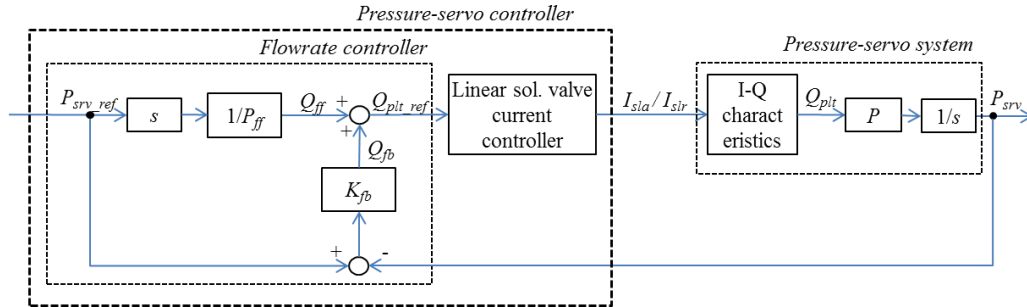


Figure 5: 2DOF control system including flow rate controller.

In this control system, K_{fb} is the feedback gain with respect to the servo pressure deviation. This is set by the PD control. P_{ff} is the model for plant P . Plant P indicates the regulator and, in macro terms, it functions to change the servo pressure in accordance with the pilot flow rate. Therefore, it can be modeled as

$$P = K_1 \times K_2 \quad (1)$$

K_1 is the servo pressure generation rate with respect to the unit of the pilot pressure. K_2 is the pressure generated per unit of fluid applied to the pilot chamber (pressure–fluid stiffness [MPa/cc]). In addition, the purpose of the differentiator in the feedforward loop is to convert the pressure (unit: [Mpa]) into flow rate (unit: [cc/sec.]). The target pilot flow rate, which is the output of the flow rate controller, is then converted into the target flow rate for SLA and SLR in the linear solenoid valve current controller after considering the flow rate–current characteristics of each linear solenoid valve.

5.2 Issues with the Developed Pressure Application Unit and Countermeasures

5.2.1 Compensation for Hysteresis of Spool Valve

This section considers the operation of the SLA when the pressure is increased. When the SLA stops operating, the spool plug moves from the servo pressure application port (δ_{app}) toward the force balance position. The force balance position (convergence position) is determined by the fluid leakage from the accumulator to the servo chamber and the fluid leakage from the servo chamber to the reservoir tank. Because the transfer of spool plug causes the contraction of the pilot chamber and an increase in the pilot pressure, the servo pressure also increases compared to the pressure achieved when the SLA is stopped. The same phenomenon occurs during pressure reduction, and the pilot and servo pressures decrease compared to the pressure achieved when the SLR is stopped.

These are not the desirable characteristics from the standpoint of tracking the target servo pressure. To enable the servo pressure to be controlled at the desired value, it is, therefore, necessary to estimate servo pressure fluctuation and to stop linear solenoid valve operation in advance. When the deviation of target servo pressure and actual servo pressure (ΔP_{srv}) is smaller than the threshold value, hysteresis compensation control is added to reduce the flow rate of linear solenoid valves. The servo pressure fluctuation after the valves operating is determined by amount of the pilot chamber contraction and expansion that is proportional to convergence position of spool plug. And the convergence position of spool plug is approximately proportional to the servo pressure. Therefore, the deviation thresholds to reduce the flow rate in advance are set as a variation in accordance with the servo pressure. Fig. 6 shows the relationship between the convergence position of the spool plug δ and the control threshold.

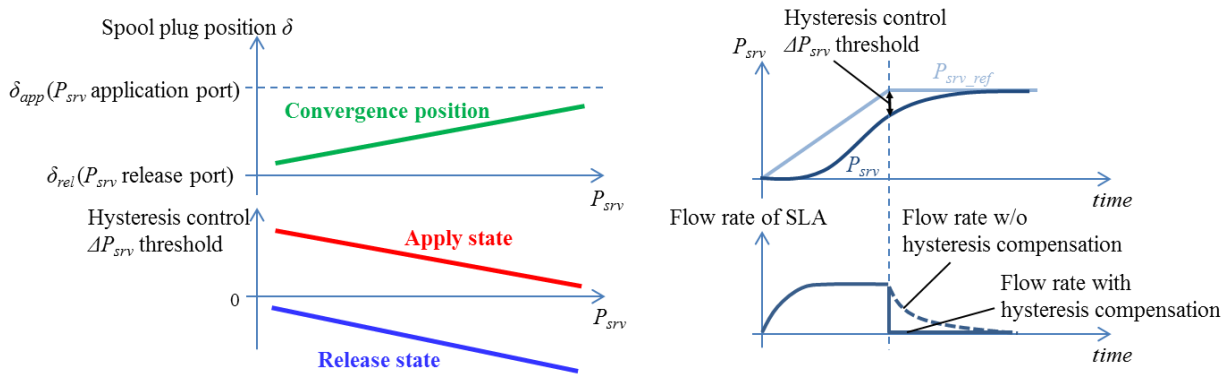


Figure 6: Relationship between the spool plug convergence position and the threshold of hysteresis compensation control.

5.2.2 Compensation for Dead Zone of Spool Valve Control

The spool valve control has a dead zone in which the pilot flow rate is not amplified. In other words, in the process of increasing the pressure, the servo pressure does not change from the start of the spool plug movement toward the servo chamber until the edge (for increasing the pressure) reaches the pressure increasing port. Likewise, in the pressure reduction process, the servo pressure does not change from the start of the spool plug movement toward the pilot chamber until the edge (for reducing the pressure) reaches the pressure reduction port. These non-linear characteristics cannot be ignored to achieve a highly accurate pressure control. Therefore, a control is added to quickly fill the dead zone in the spool valve control in the 2DOF control system described above. This is called the compensation control for the spool valve dead zone.

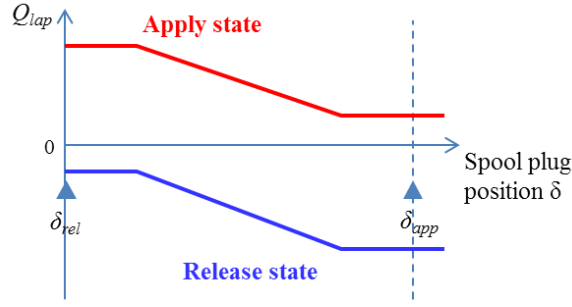


Figure 7: Flow rate at each spool plug position in the compensation control for spool valve dead zone.

Fig. 7 shows the flow rate Q_{lap} in the compensation control for the spool valve dead zone for each spool plug position δ . Although this compensation flow rate must fill the control dead zone in a short time, it is necessary to avoid an increase in the pressure due to an over-high flow rate or pressure reduction overshooting. Therefore, when the pressure is being increased, it is desirable to increase the flow rate close to the pressure reduction port and reduce the flow rate close to the pressure increment port. To achieve this, it is important to estimate the current spool plug position.

The previous sub-subsection described the servo pressure fluctuations that are caused by the pilot chamber contraction and expansion. The change in the pilot chamber volume is generated by the change in the spool plug position, so it is possible to estimate the spool plug position using a change in the servo pressure. The equation of balance of the force acting on the spool plug is described in (2).

$$\Delta P_{srv} \times A_{srv} + f_0 - f_1 = \Delta P_{plt} \times A_{plt} \quad (2)$$

In this equation, ΔP_{srv} is the change in the servo pressure after the linear solenoid stops operating, which is detected by a servo pressure sensor. Likewise, ΔP_{plt} is the change in the pilot pressure. A_{plt} is the cross-sectional area of the spool plug in the pilot chamber side, A_{srv} is the cross-sectional area of the spool plug in the servo chamber side, f_0 is the spring force of the spool plug, and f_1 is the sliding resistance force of the spool plug acting in the direction opposite to the movement of the spool plug. The change in the spool plug position $\Delta\delta$ is described in (3).

$$\Delta\delta = \frac{\Delta P_{plt}}{K_2 \times A_{plt}} \quad (3)$$

K_2 is the pilot chamber pressure–fluid stiffness coefficient. Using (2) and (3), $\Delta\delta$ is estimated by the change in the servo pressure at each calculated period, and then spool plug position δ is calculated by integrating $\Delta\delta$. The flow rate Q_{lap} is calculated from δ using the relationship expressed in Fig. 7.

Because the compensation control for the spool valve dead zone is a feedforward control, it is affected by the current–flow rate characteristics of the linear solenoid valves, variation in the length of the spool valve dead zone, and the like. Therefore, as a feedback element, the control ends once it is judged that the flow rate compensation corresponding to the spool valve dead zone has been achieved. This is defined as the point at which a certain change in the servo pressure has occurred with respect to the servo pressure at the start of the control.

Table 4 summarizes the performance requirements of this system and the method for achieving this performance based on the system configuration and control system design described in the previous and present sections. Fig. 8 shows the controller incorporating the compensation controls for the spool valve dead zone and hysteresis.

Table 4: Required performance and measures adopted by the developed system.

	Required performance		
	Size of braking force	Rate of braking force change	Accuracy of braking force
System configuration	High-pressure accumulator	Flow rate amplified by regulator	Pressure adjusted by linear solenoid valves
Pressure control method	-	2DOF control system	
		Compensation control for the spool valve dead zone	Compensation control for the spool valve hysteresis

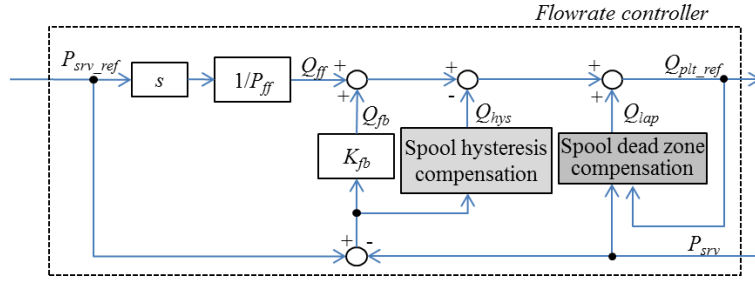


Figure 8: 2DOF flow rate controller with compensation controls for the spool valve dead zone and hysteresis.

5.3 Performance Experiment of the Proposed Pressure Control System

Fig. 9 shows the test results of the pressure control system described in this section.

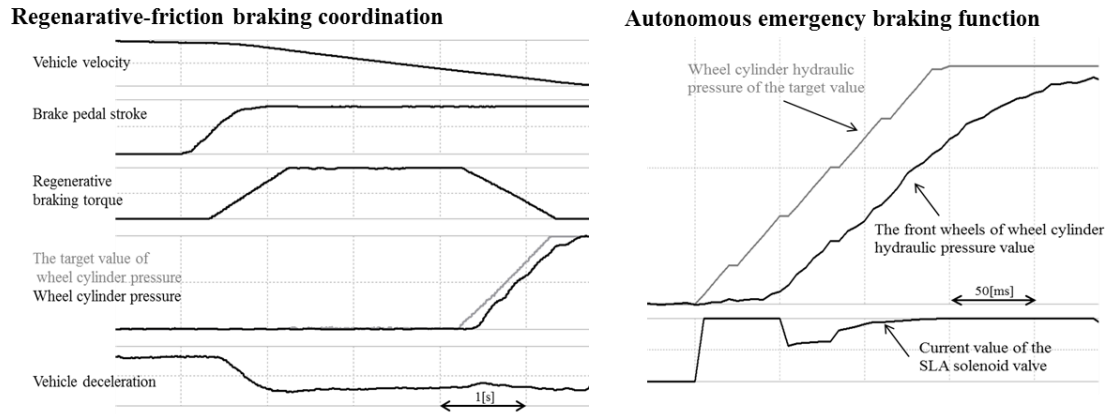


Figure 9: The pressure control results assuming the operation of the regenerative-friction braking coordination and autonomous emergency braking function.

In the results of pressure control tests performed by using the regenerative-friction braking coordination, the vehicle deceleration results at the time of replacement of the regenerative braking torque and the wheel cylinder pressure indicate that an excellent pressure control accuracy was achieved.

In the results, assuming the operation of an autonomous emergency braking function, the quick rise and objectives are reflected in the pressure control waveform. Thus, an application requiring a quick and large braking force can be realized.

6 Conclusion

This paper proposed an electronically controlled braking system, which contains a brake-by-wire composition and a pressure application unit. The brake-by-wire composition configured with the input piston and the output piston improves the regenerative efficiency, system reliability, and brake pedal feeling.

The pressure application unit combined with the linear solenoid valves and regulator achieves a hydraulic pressure control with high accuracy. In the control strategy, a 2DOF controller with spool hysteresis and a dead-zone compensator is proposed. Good performances of various brake functions are demonstrated with experiments.

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Authors



Yusuke Kamiya

He graduated in 2007 with a Master of informatics engineering from the University of Kyoto. He worked as an engineer for electronically controlled braking systems at Toyota Motor Corporation since 2007 and at ADVICS CO., LTD. since 2014.



Yoshio Masuda

He graduated in 2006 with a Bachelor of electrical and electronic engineering from the University of Mie. He worked as an engineer for electronically controlled braking systems at ADVICS CO., LTD. since 2006.



Masaaki Komazawa

He graduated in 1982 with a Master of mathematical engineering and instrumentation physics from the University of Tokyo. He worked as an engineer for electronically controlled braking systems at Toyota Motor Corporation since 1982.



Yasuji Mizutani

He graduated in 1991 with a Master of electrical engineering from Hokkaido University. He worked as an engineer for electronically controlled braking systems at Toyota Motor Corporation since 1991.



Takahiro Okano

He graduated in 2003 with a Master of electrical engineering from the University of Tokyo. He worked as an engineer for electronically controlled braking systems since 2003 at Toyota Motor Corporation and ADVICS CO., LTD.