Enhancement of xEV Motor Driving Force Control

Haibo Liu [1)] Kazuya Enokizono[2)] Masaru Kubota[3)] Sadahiro Akama[4)] Keisuke Kawai [5)]

DENSO CORPORATION, Kariya Aichi Japan
1) E-mail: haibo.liu.j3x@jp.denso.com
2) E-mail: kazuya.enokizono.j3n@jp.denso.com
3) E-mail: masaru.kubota.j2b@jp.denso.com
4) E-mail: sadahiro.akama.j6t@jp.denso.com
5) E-mail: keisuke.kawai.j5j@jp.denso.com

ABSTRACT:
To realize a carbon-neutral society, it is necessary to develop technologies to promote the electrification. The expansion of regenerative range of cooperative regenerative braking system is a promising technology. Here, two issues arisen in expansion of regenerative range (motor enhanced regenerative braking) are handled and solutions have been established, which are specified as (1) keeping regenerative braking until stop (2) continuation of regenerative braking in ABS actuation scenes. Regarding issue (1), a feed forward control is proposed which is independent from using wheel speed to ensure the vehicle stopping. At the same time, G omission and deterioration of drivability due to vibration caused by driveshaft twisting and gear-rattling is suppressed to an acceptable level. Dealing with issue (2), a PID control in piecewise function is suggested which is divided by differential of tire slip ratio. In small differential of slip ratio area, a linear PID control is deployed to control the motor force precisely to keep tire gripping. In the area where differential of slip ratio is large, an ID control is dominated to limit the change of slip ratio. The solutions of (1) and (2) were implemented in a rapid-prototype environment, and braking tests on dry road and μ-jump road (a sudden change in the road surface μ) had been conducted to confirm the validity of those control. It is proved that both the improvement of energy efficiency and keeping stopping distance equivalent to conventional ABS system are realized. In the future, further verification of the robustness will be carried out on xEVs.
KEY WORDS: motor, cooperative regenerative braking, motor enhanced regenerative braking, slip suppression control

1. Introduction

In automobile industry, electrification is rapidly accelerating toward the realization of a carbon-neutral society in 2050. [(1)] Taking advantage of the electrification, in comparison with conventional engine-powered vehicles which use friction brakes, electrified vehicles can save energy consumption by regenerative braking through converting vehicle's kinetic energy into electricity to batteries. [(2)]If regenerative braking can be achieved only by motors without using friction brakes, energy efficiency can be improved, but it is difficult to ensure all driving situations. As a result, a cooperative regenerative braking system with a combination of friction brakes and motor regenerative braking are mounted in electrified vehicles.

In this paper, focusing on expand the regenerative range with the motor, the effort was spent on developing motor control that takes advantage of features of precision and responsiveness to achieve both higher power efficiency and safer braking. The effectiveness and control validity were verified by vehicle and the result is reported as follow.

2. Issues to expand regenerative range

Cooperative regenerative braking by the coordination of friction brakes and motor regenerative braking is frequently used in hard deceleration and high speed scene, which requires the maximum torque and power of the motor. Since the amount of regenerative power of the motor is limited and can be derived by product of the torque and speed, motor’s available regenerative torque is restrained in the high speed range. Due to the restrain from current, motor torque is also limited in the low speed range. In those situation, the insufficient deceleration force is need to be compensated by friction brakes (Fig1).

Fig.2 shows the time chart of ABS (anti-lock braking system) operates in a cooperative regenerative braking system. When the road surface μ has a sudden shift (μ-jump), hydraulic ABS works to prevent tire from locking (Fig2) and stops regenerative braking.

In light of above, the following issues arise by motor enhanced regenerative braking.

- A motor that can generate large torque and power up to high vehicle speed is required, which have cost and layout trade off.
- If the regeneration torque is removed until the vehicle stops, the driveshaft will twist and the vehicle will swing back after stopping.
- The rattle caused by gear backlash vibrate the car body.
- Sudden changes in the road surface μ with high deceleration
- As the amount of regeneration increases, full battery situations will increase, and in the worst case, motor regeneration lapses and deceleration fails.

In this paper, two following issues are solved by utilizing the precise and highly responsive controllability of the motor. First solution is to suppress vehicle vibration caused by "twisting of the driveshaft" and "rattling of the gears" occurring just before a stop. Second solution is to achieve both the continuation of regenerative braking in ABS scene and vehicle stability from preventing tire from locking.

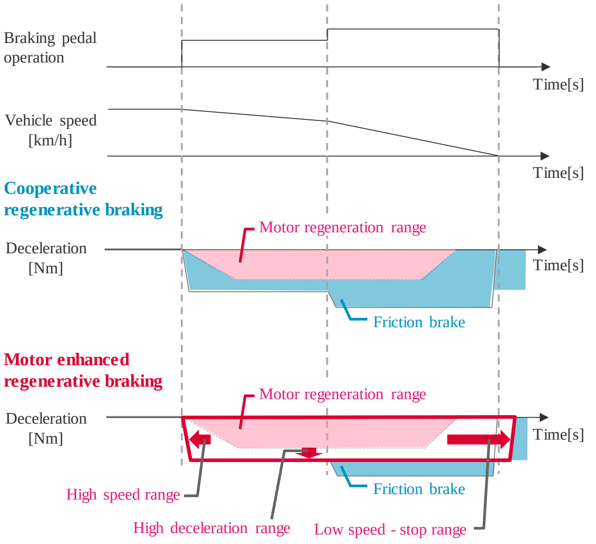

Fig1. Cooperative regenerative braking and regenerative enlargement range

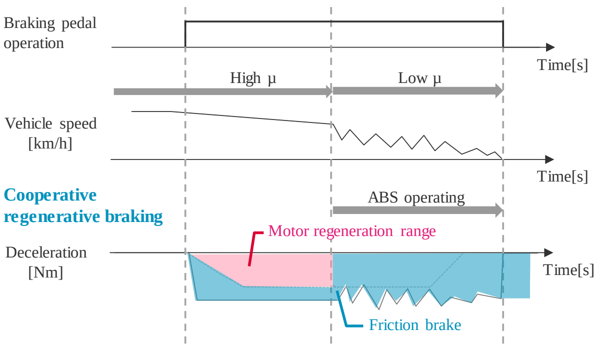

Fig2. The characteristics of conventional ABS.

3. Suppression of vehicle body swinging

3.1 Causes of vehicle body swinging

In conventional friction brake system, the tires can be controlled more directly as the braking rotor is mounted on wheel, whereas motor regenerative braking transmit torque to the tires via a less rigid driveshaft, so that the restoration of shaft twist after torque releasing causes the body swinging back (Fig3).

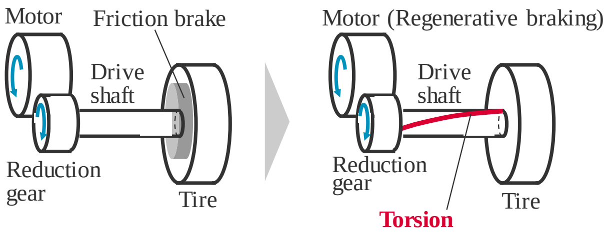

Fig3. Friction brake V.S. Regenerative braking

3.2 Approach to find the vehicle body swinging solution

In order to suppress vehicle body swinging back during stopping, it is necessary to suppress the torsion of the driveshaft caused by regeneration. There are two main methods can be considered. First method is by increasing the rigidity of the driveshaft and second method is by eliminating the twisting by motor control just before stopping. First method is unrealistic because it would increase the rigidity against torsion by increasing the diameter of the driveshaft. As a result, the vehicle weight and cost will increase offsetting the benefits. Second method can be realized by releasing elastic energy due to twist of the driveshaft just before stopping, but it requires the detection of extremely low vehicle speed just before stopping and precise motor torque control to release the energy as well as stopping and holding the vehicle at the same time. Obviously, the cost of the second one is less, so it is going to be well considered.

3.3 Detection of low vehicle speed near a stop

Conventional vehicles utilize wheel speed sensors to measure the vehicle speed. However, if the speed is extremely low like just before a stop, wheel speed sensors is not able to output pulse signal and the vehicle speed cannot be measured. On the other hand, using rotational speed of motor is a solution to estimate the vehicle speed, but the speed is difficult to be accurately calculated due to driveshaft torsion and torque fluctuations. Another alternative is to utilize the G-sensor, but since the inclination of sprung mass is superimposed on longitudinal G during deceleration, the vehicle speed estimation becomes complicated due to disturbance. Hereby, a stopping method that using vehicle motion equations instead of relying on vehicle speed estimation was devised.

Firstly, the development goal is going to be set. As shown in Fig4. The friction brake characteristic can be simply described as that friction force retains the dynamic friction force until a vehicle speed reaches low speed and decreases with a linear characteristic of a slope k to zero. This characteristic minimizes the reduction of vehicle deceleration until just before a stop, allowing occupants to stop without fear. Braking characteristics equivalent to conversional friction brake system were targeted for our control.

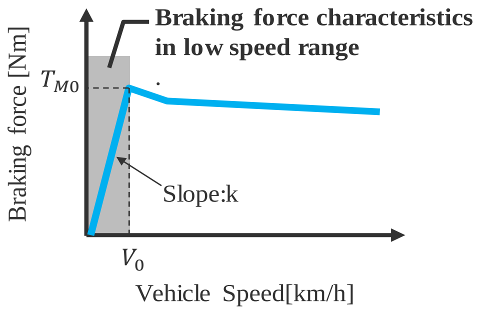

Fig4. Friction brake characteristics at very low vehicle speed

The motor regenerative torque T_M at a vehicle speed of 1 km/h or less is expressed by (1).

$$T_M = kV_x \tag{1}$$

Where k denotes the slope of motor regenerative torque to vehicle speed and V_x denotes vehicle body speed.

Here, the basic equation of vehicle motion as shown in Fig5 is utilized and expressed as (2).

$$F = \frac{T_M}{R} = m\frac{dV_x}{dt} \tag{2}$$

Where F is the driving force, R is the tire radius, and m is the vehicle weight.

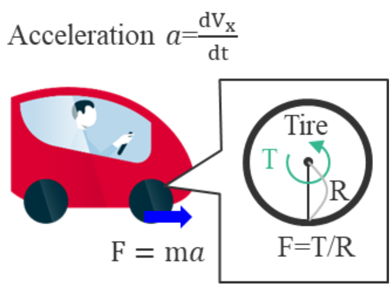

Fig5. Basic equations of vehicle motion

Substituting (2) into (1) yields (3).

$$V_x = V_0 e^{-\frac{t}{\tau}} \tag{3}$$

Substituting (3) into (1), motor torque can be expressed by (4).

$$T_M = kV_0^{-\frac{t}{\tau}} \tag{4}$$

Also, the time constant τ is as follows.

$$\tau = \frac{mR}{k} \tag{5}$$

Here, V_0 is the speed at which the vehicle speed is switched from using the wheel speed sensor, which is about 1 km/h.

When vehicle speed is less than 1km/h, by switching to motor torque control of (4), braking characteristics equivalent to friction brake can be realized without relying on the vehicle speed estimation. This allows the motor torque to be released from just before a stop, thereby the swinging back of the vehicle body is suppressed in thanks of less twist of driveshaft.

By adjusting the slope k of the motor regenerative torque to the vehicle speed, the time constant τ can be modified, but when the time constant τ is reduced, the vehicle vibration due to the rattle of the gears becomes apparent. Thus, the suppression of the vibration should be considered.

3.4 Suppression of vehicle vibration due to rattle of gears

As shown in Fig6, the regenerative force of the motor is transmitted to the tires via a gear in addition to the driveshaft. Due to the backlash in the gear, if the motor torque is steeply reduced, the vibration caused by the rattle of the gear will cause the shock of the tires via the driveshaft, which causes the vehicle vibration and deteriorates the ride quality.

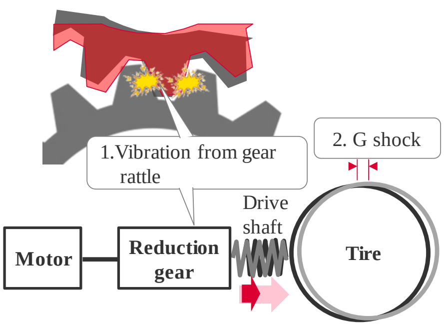

Fig6. Propagation of tires G variation due to motor torque variation

Therefore, it is necessary to adjust the time constant τ, which is restrained by G variation at which the occupants do not feel uncomfortable.

Thus, an equation is need to be derived for expressing the relation between vehicle vibration shock and torque releasing time constant τ to design a proper τ. The MG torque can be simplified into (6) by (4). T_{M0} is the braking force at vehicle speed V_0.

$$T_M = T_{M0} e^{-\frac{t}{\tau}} \tag{6}$$

If the driveshaft stiffness is K_{ds} and the twist angle of the driveshaft is θ, the equation of motion of the rotating body can be expressed as below (7).

$$T_M = K_{ds}\theta = T_{M0} e^{-\frac{t}{\tau}} \tag{7}$$

The maximum value of the motor speed ω_{max} can be derived by two-sided differentiation and sorting out equation (7).

Here, ω is the torsional speed of the driveshaft, and since the driveshaft rotates with the rotor of the motor, the motor speed can be approximated by ω.

$$K_{ds}\omega = -\frac{T_{M0}}{\tau} e^{-\frac{t}{\tau}} \tag{8}$$

$$\omega_{max} = -\frac{T_{M0}}{K_{ds}\tau} \tag{9}$$

Assuming that the rattle of gear backlash is a perfectly elastic collision, the energy conservation law of (10) is satisfied.

$$\frac{1}{2} I_M {\omega_{max}}^2 - \frac{1}{2}(I_M + I_{tire})\omega_2^2 = \frac{1}{2}\frac{T_1^2}{K_{ds}} \tag{10}$$

I_M, I_{tire} is the moment of inertia of the motor and tires. T_1 is the maximum impact torque transmitted from the driveshaft to tires at the time of impact. ω_2 is the speed after impact. In addition, the angular momentum conservation law is also satisfied, which leads to (11).

$$I_M \omega_{max} = (I_M + I_{tire})\omega_2 \tag{11}$$

The amount of G fluctuation of the vehicle body caused by backlash can be expressed by (12) using the impact torque T_1.

$$G_{shock} = \frac{T_1}{mgR} \tag{12}$$

Solving equations (9) - (12) in conjunction, the change in the vehicle vibration shock when the driveshaft torque is released with the time constant τ is given by (13).

$$G_{shock} = \frac{T_{M0}}{mgR}\sqrt{\frac{I_M I_{tire}}{K_{ds}(I_M + I_{tire})}} \cdot \frac{1}{\tau} \tag{13}$$

By using (13) to determine the design area of proper τ, a sensory test was carried out and is shown that if the G fluctuation caused by rattle is less than a threshold value, the occupants on board will not feel discomfort. From these requirement came from braking force decreasing and G shock, the proper design area for τ is shown in Fig7 in which ensures equivalent braking performance to friction brake and is of possibility to achieve both "Suppression of vehicle body swinging back" and "Suppression of vehicle body vibration by rattle of gears."

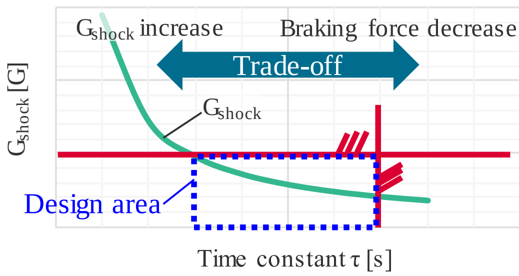

Fig7. Relationship between shock and time constant τ during untwisting

4. Continuation of regenerative braking during ABS

4.1 Tire lock suppression during regeneration expansion

If the vehicle is under suddenly deceleration situation on a low μ road surface, conventional hydraulic ABS operates to prevent tire from locking, which affect steerability. In above scene, the cooperative regenerative braking requests the host controller (EVCU) to stop regenerative braking after the tendency of locking is detected. Thereby, the decelerating task is handled to the hydraulic ABS and wheel speed is under controlled to maintain the steerability (Fig8). However, as shown in Fig9, if the regenerative braking deceleration (regenerative torque) by motor increases, tires will be locked before the arrival of command from braking controller due to communication delay. For that reason, the motor should be able to individually judge tire slip and keep braking as well as prevent tires from locking. Hereby, after motor detect slip of tires, a control to suppress tire locking through high controllability of motor's torque is suggested.

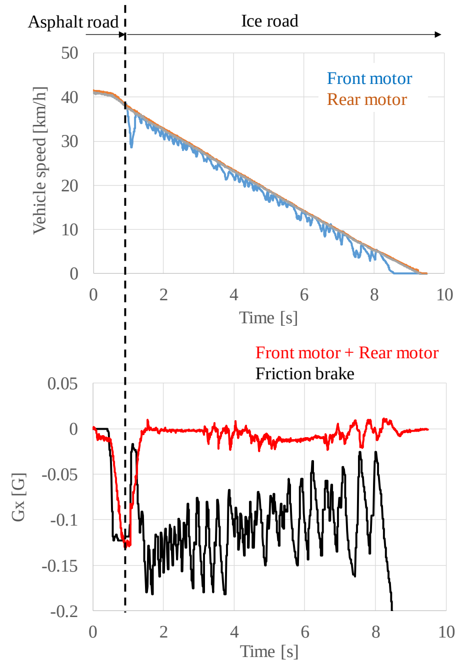

Fig8. µ jump during conventional cooperative regeneration

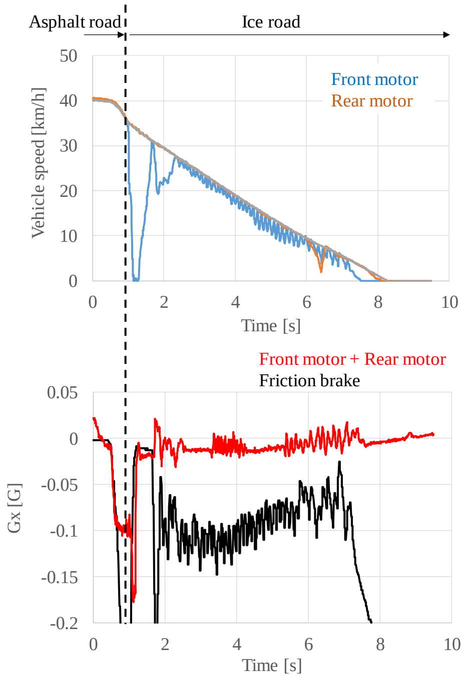

Fig9. µ jump when regenerative range is expanded

4.2 Slip suppression control utilizing motor

Fig10 shows the braking force characteristics of the tire. The horizontal axis is the slip ratio and the vertical axis is the tire braking force (coefficient of friction, µ), which is separated into the grip and slip areas divided by µ peak. In order to ensure braking performance (shortening the braking distance), it is preferable to maintain the braking force as close to the µ-peak as possible within the grip area.

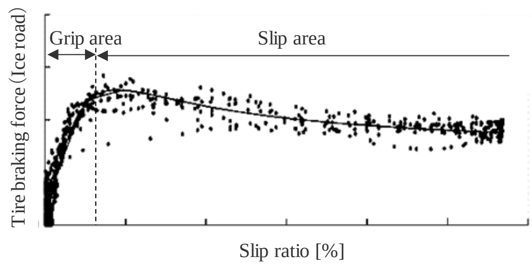

Fig10. Tire braking force characteristics

In order to control the slip ratio, controller that can follow the non-linear characteristics of the tire is required. It is desirable to utilize a normal linear controller (PID) rather than a complex controller. Tire has linear characteristics in the grip area. Therefore, we firstly considered a simple control method utilizing PID.

When the tire braking force is PID-controlled according to the slip ratio, deviation occurs due to overshoot. Due to the non-linear characteristics of the tire, this deviation will cause tire hunting between the slip area and grip area. As shown in Fig11, it is possible to suppress the deviation by reducing the proportional term P, but it takes long time to converge. This trade-off cannot be fulfilled by linear PID controller.

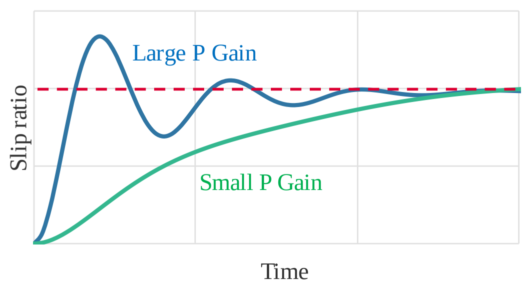

Fig11. Effect of P-term on control stability

In order to quickly suppress the deviation, the differential of slip ratio should be quickly restrained to zero. Therefore, a PID controller expressed by piecewise function which divided by differential of slip ratio is investigated whether it is possible to suppress the deviation while improving the responsiveness (Fig12). In the area where the differential of slip ratio is large, ID control increases responsiveness and quickly transitions the slip ratio from slip area to grip area. After that, the differential component becomes smaller. In small differential of slip-ratio area, proportional term P is added to improve convergence. As a result, both high response and suppression of deviation can be achieved (Fig13). The suggested equations of slip suppression control are shown in (14), (15), and (16).

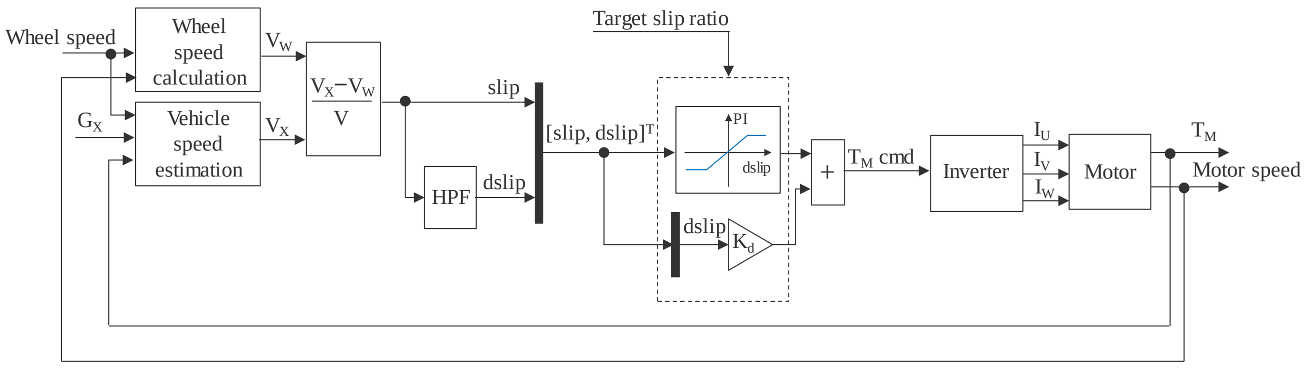

Fig12. Control block diagram

$$
\begin{aligned}
T &= T_D + T_{PI} \\
&= K_D dSlip + \int dTdt \\
&= K_D dSlip + (T_{PI\ old} + dT)
\end{aligned}
\tag{14}
$$

$$dT = f(Slip,\ dSlip) \tag{15}$$

Here、 $f(Slip,\ dSlip)$is shown as (16)

$$
f(Slip,\ dSlip) = \begin{cases} BSlip + A_0\ ; dSlip < -dSlip_1 \\ AdSlip + BSlip + C_0 ; dSlip < |dSlip_1| \\ BSlip + B_0 ; dSlip > dSlip_1 \end{cases} \tag{16}
$$

Where the meaning of parameters is shown as follow:

T_D: D control Torque, T_{PI}: PI control torque, K_D: D gain

Slip: tire slip ratio during brake, dSlip: differential of slip ratio

A_0: I gain lower bound factor in the area where dSlip is large.

B_0: I gain upper bound factor in the area where dSlip is large.

C_0: I gain parameter in the area where dSlip is small.

A: P gain, B: I gain

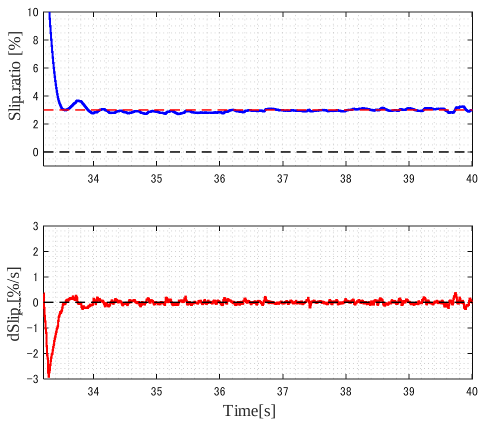

Fig13. Goal of the proposed PID control

4.3 Improvement of robustness considering tire wear

Since the nonlinear characteristics of tire change due to tire wear and tire replacement, the optimum slip ratio also changes. Therefore, a control to adjust the target slip ratio is added. A wheel speed hunting detection counter is provided as shown in Fig14. When the tire slip is detected after the slip suppression control has interfered and once the slip ratio convergent, the times of slip is countered up and the target slip ratio is lowered gently according to counter output value. As tires are under stably controlled, the counter stops increasing and the output target slip ratio is kept until next time starting the vehicle(Fig14).

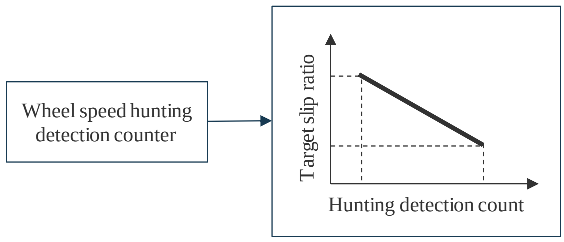

Fig14. Target slip ratio adjustment control

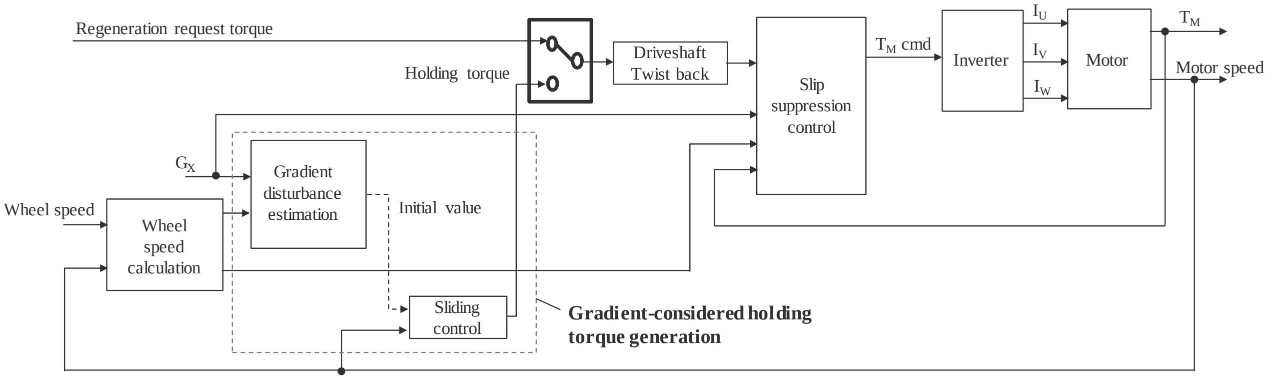

Fig15. Control block diagram

5. Experimental result

5.1 Test conditions and control block diagram

The developed control was evaluated on a hybrid electric vehicle RX 450hL under the conditions shown in Table 1. The control block diagram is shown in Fig15. In the control block diagram, a sliding control is implemented to estimate the road gradient and generate stop-holding torque when the vehicle is stopped on slope. In addition, a control to handover the braking task from regenerative braking to the friction brake is also considered when the motor regenerative torque is insufficient due to steep slope.

Table1. Experimental condition

	Suppression of vehicle body swinging test	Slip suppression test
Vehicle Speed	15km/h	40km/h
Road Surface	Asphalt road	Asphalt to Ice road (μjump)
Deceleration	-2.94m/s^2	-1.96 m/s^2

5.2 Suppression of vehicle body swinging test

τ and G_{shock} were matched before conducting the test, Based on the characteristics of the vehicle RX450hL, the appropriate values of τ are adjusted to 0.14s or less, so that G_{shock} is 0.05G or less. Fig16 shows the results when regenerative braking was performed only by the motor until stopping without implementing this control. An overshoot of braking G occurred at stopping which denotes the vehicle is swinging. On the other hand, Fig17 shows the result of implementing this control and it is possible to stopping the vehicle only with motor until stopping.

In addition, the effect of the improvement of the electricity consumption of the control is calculated by simulation (under WLTC mode). It was confirmed that the electricity consumption is improved by 0.7%.

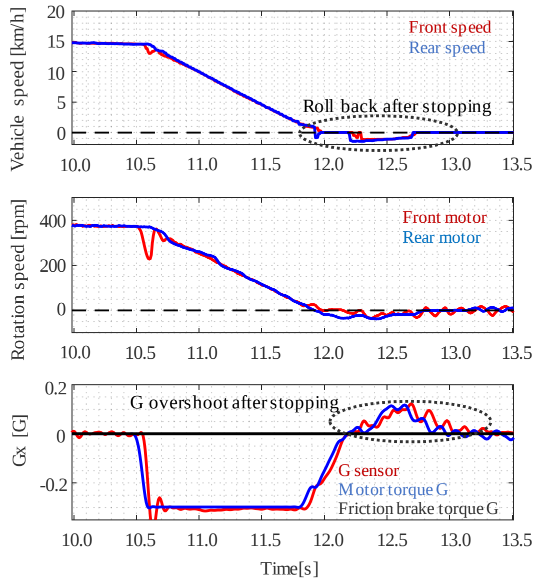

Fig16. Result without suppression swinging control

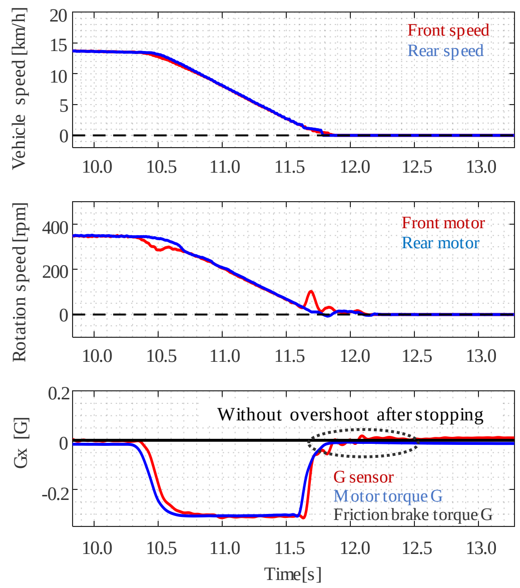

Fig17. Result with suppression swinging control

5.3 Slip suppression test

In order to confirm the slip suppression effect by the motor control, a braking test was conducted on the road whose surface suddenly changes from asphalt road (μ: 0.9 to 1.0) to ice road (μ: 0.1) during braking. Fig18 shows the result of conventional cooperative regenerative braking operation. Fig19 shows the result of proposed control. Table2 shows the deceleration and braking distance when the same test was performed three times. It was confirmed that braking performance is equal to or greater than that of conventional ABS.

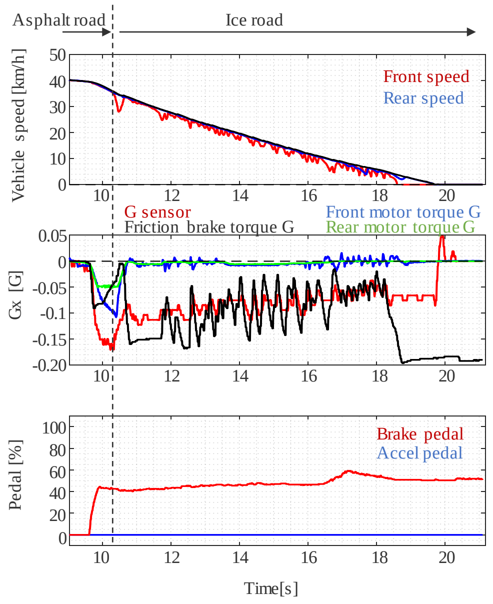

Fig18. μ jump behavior during cooperative regeneration (conventional ABS control operate)

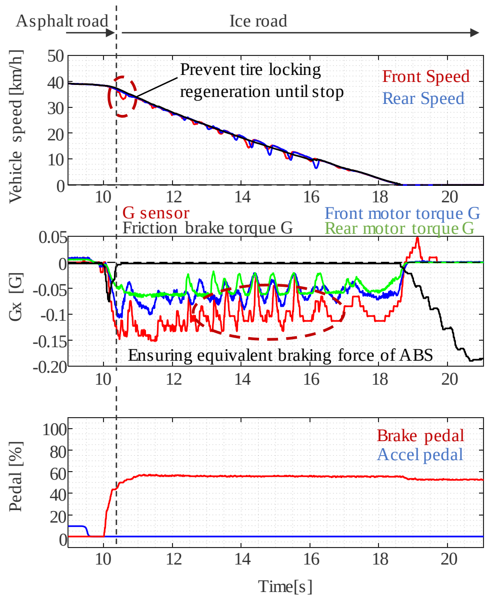

Fig19. μ jump behavior during regenerative range expansion (slip ratio control implementation)

Table2. Comparison of average deceleration and stopping distance after μ jump

Average deceleration $[m/s^2]$	Minute slip ratio control	Cooperative regeneration
1	1.21(37.82m)	1.15(40.71m)
2	1.37(37.12m)	1.19(40.57m)
3	1.36(37.81m)	1.07(39.74m)
Average	1.31(37.58m)	1.13(40.34m)

6. Conclusion

In this paper, motor control technologies are developed to enhance vehicle power efficiency by expanding the regenerative braking range until just before vehicle stops and in ABS operation scene. These technologies are realized by the cooperative regenerative braking system that takes advantage of the characteristics of electrification. Through actual vehicle tests, it has been confirmed to improve electricity consumption and provide stable braking performance.

In the future, a further research to verify the robustness of the control aiming for practical application will be carried out. The development of other new technologies which will improve the performance of electric vehicles to realize a carbon-neutral society will be continued as well.

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