

Optimization of Driving Force Distribution Control in All-Wheel Drive Based on Wheel Rotation Speed Difference Between Front and Rear

Shoichi OHYAGI*

Takayuki SEKI*

ABSTRACT

Electronically controlled all-wheel drive (AWD) systems that are built on a front-wheel drive base can control spinning of the front wheels by increasing the amount of driving force distributed to the rear wheels according to the speed difference generated between the front and rear wheels. On the other hand, the turning characteristics of the wheels change with the amount of driving force distributed. When the magnitude of this change is large, vehicle controllability may be degraded. Driving force distribution therefore needs to take into account the effects during turns that involve a greater wheel rotation speed difference between front and rear.

A spin suppression control system was therefore created that performs theoretical calculations of front and rear wheel rotation speed differences that occur due to turns and tire radius variations, and that takes those values as the target values for the system. The factors that cause front and rear wheel rotation speed differences are divided between front wheel spin and all others, and it was confirmed that this would enable convergence only on front and rear wheel rotation speed differences that are due to front wheel spin.

1. Introduction

The magnitude of the cornering force relative to the slip angle of the tire (cornering power) is known to vary according to the driving force on the tire in the longitudinal direction⁽¹⁾. AWD systems with electronic control and a clutch for driving force distribution can determine the amount of driving force distributed to sub-driving wheels by implementing electronic control of the clutch fastening force. Adjustment of the amount of driving force distributed in order to control cornering power and control the vehicle's turning state is a widely used practice⁽²⁾⁻⁽⁵⁾.

This kind of AWD system may also implement control to suppress the rotation speed difference⁽⁶⁾⁻⁽⁸⁾ by increasing the clutch fastening force according to the magnitude of the rotation speed difference between the front and rear wheels. This is control to reduce the slip ratio of the main driving wheel when the driving force applied to the main driving wheel exceeds the friction force limit of the tire, causing the wheel to spin, by using the sub-driving wheel to take up the surplus driving force. Such control can be expected to have the effects of mitigating the reduction in driving force due to spinning, the energy loss, and the load on the machine.

A rotation speed difference between front and rear wheels may also occur for reasons other than main driving wheel spin, such as differences in wheel direction or path. If spin suppression control does not take the influence of these factors into account, the system may estimate the amount of wheel spin to be greater, and a greater amount of driving force will be distributed than is needed to suppress spin. This distribution of an extra amount of driving force will change the turning characteristics, and the change may in some cases interfere with the turning state control by means of driving force distribution mentioned above. For this reason, spin suppression control needs to correct for turns. This used to be done by applying a correction value to the driving force according to the ratio of lateral acceleration⁽⁶⁾ or the speed difference between left and right wheels. In neither case, however, was correction applied on the basis of the calculated value for the rotation speed difference between front and rear wheels that occurs when turning.

This paper will discuss the causal factors in wheel rotation speed differences between front and rear wheels and will present a method of spin suppression control that does not interfere with turning state control by means of driving force distribution.

* Automobile R&D Center

2. Basic Theory of Spin Suppression

This paper will describe an AWD system of the kind shown in Fig. 1, in which the main driving wheel is a front wheel, the sub-driving wheel is a rear wheel, and there are differential gears that distribute the driving force equally between the left and right wheels, both front and rear, while also absorbing the rotation speed difference between left and right. When the rear wheels start turning before the front wheels in this kind of system, the torque transmission goes from the rear wheels to the front wheels. The rear wheels do not have their own source of driving force, so the torque transmission causes convergence of the rotation speed difference. Consequently, spin suppression will be considered only for the case of spin in the front wheels.

Spin suppression is implemented by increasing the amount of driving force distributed to the rear wheels in accordance with the amount of front wheel spin. The amount of spin can be determined by the ratio of the wheel rotation speed to the rotating directional component of the wheel center velocity vector. While the vehicle is turning, however, the wheel direction and the direction and magnitude of the wheel center velocity vector differ for each wheel. Therefore the amount of spin cannot be determined from the wheel rotation speed alone, and so, in order to determine the amount of front wheel spin from the rotation speed difference between front and rear wheels, the influence of the change in rotation speed that occurs while the vehicle is turning must be eliminated.

A rotation speed difference also occurs when the tire dynamic loaded radius differs between the front and rear wheels. This rotation speed difference is also a value that is unrelated to the amount of spin, and so, with the exception of cases of extreme rotation speed difference that result in excessive mechanical load, the influence of the tire dynamic loaded radius error must also be eliminated from the rotation speed difference between front and rear wheels.

This chapter will describe the conceptual approach

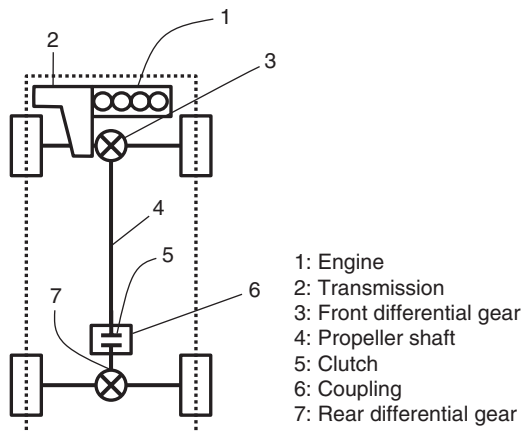


Fig. 1 Structure of AWD system

to correction for tire radius variation and the influence of vehicle turns on spin suppression control based on rotation speed difference between front and rear wheels.

V : Vehicle body speed

V_{ij} : Wheel center velocity

The character i indicates the front wheel (f) or the rear wheel (r).

The character j indicates the left wheel (l) or the right wheel (r).

R_{ij} : Tire dynamic loaded radius

ω_{ij} : Wheel rotation speed

s_{ij} : Longitudinal wheel slip ratio

α_x : Vehicle body longitudinal acceleration

δ : Steering angle

γ : Yaw rate

β : Vehicle body slip angle

l : Wheelbase length

l_f : Length of a perpendicular line from the vehicle center of gravity to the front wheel axis on a horizontal plane

l_r : Length of a perpendicular line from the vehicle center of gravity to the rear wheel axis on a horizontal plane

d_f : Front wheel tread

d_r : Rear wheel tread

h : Vehicle center of gravity height above ground

ω_f : Front wheel average rotation speed

ω_{in} : Input rotation speed of longitudinal driving force distribution clutch

ω_{out} : Output rotation speed of longitudinal driving force distribution clutch

A : Ratio of clutch input rotation speed to front wheel average rotation speed

B : Ratio of clutch output rotation speed to rear wheel average rotation speed

I_f : Inertia moment of front wheel rotating system

T_{att} : Driving torque generated by source of driving force (drive shaft equivalent)

T_r : Rear wheel driving torque

F_f : Tire reaction force in longitudinal direction on front wheel

g : Acceleration of gravity

2.1. Theoretical Value of the Rotation Speed Difference Between Front and Rear Wheels

This section will use the two-dimensional vehicle model shown in Fig. 2 for the sake of simplicity. The rotating directional components of the wheel center velocity for each wheel while the vehicle is turning are shown by Eqs. (1), (2), (3), and (4).

$$V_{fl} = V \cos(\delta - \beta) + \gamma(l_f \sin \delta + \frac{d_f}{2} \cos \delta) \quad (1)$$

$$V_{fr} = V \cos(\delta - \beta) + \gamma(l_f \sin \delta - \frac{d_f}{2} \cos \delta) \quad (2)$$

$$V_{rl} = V \cos \beta + \gamma \frac{d_r}{2} \quad (3)$$

$$V_{rr} = V \cos \beta - \gamma \frac{d_r}{2} \quad (4)$$

When driving force is being generated by the wheels, slipping occurs in a longitudinal direction. If the slip ratio under driving force is defined by Eq. (5), then the wheel rotation speed is shown by Eq. (6).

$$s_{ij} = \frac{V_{ij} - R_{ij} \omega_{ij}}{R_{ij} \omega_{ij}} \quad (5)$$

$$\omega_{ij} = \frac{V_{ij}}{(1 + s_{ij}) R_{ij}} \quad (6)$$

When converted into the rotation speeds (ω_{in} and ω_{out}) of input and output at the driving force distribution clutch, the rotation speeds of the left and right wheels are averaged out and shown by Eqs. (7) and (8), respectively.

$$\omega_{in} = A \frac{\omega_{rl} + \omega_{rr}}{2} \quad (7)$$

$$\omega_{out} = B \frac{\omega_{rl} + \omega_{rr}}{2} \quad (8)$$

The clutch slippage (hereafter represented as ΔN) is shown by Eq. (9).

$$\begin{aligned} \Delta N &= \omega_{in} - \omega_{out} \\ &= A \{ x_{fa} [V \cos (\delta - \beta) + \gamma l_f \sin \delta] + x_{fd} \gamma \frac{d_f}{2} \cos \delta \} \\ &\quad - B (x_{ra} V \cos \beta + x_{rd} \gamma \frac{d_r}{2}) \end{aligned} \quad (9)$$

$$x_{fa} = \frac{1}{2} \left[\frac{1}{(1 + s_{fl}) R_{fl}} + \frac{1}{(1 + s_{fr}) R_{fr}} \right] \quad (10)$$

$$x_{fd} = \frac{1}{2} \left[\frac{1}{(1 + s_{fl}) R_{fl}} - \frac{1}{(1 + s_{fr}) R_{fr}} \right] \quad (11)$$

$$x_{ra} = \frac{1}{2} \left[\frac{1}{(1 + s_{rl}) R_{rl}} + \frac{1}{(1 + s_{rr}) R_{rr}} \right] \quad (12)$$

$$x_{rd} = \frac{1}{2} \left[\frac{1}{(1 + s_{rl}) R_{rl}} - \frac{1}{(1 + s_{rr}) R_{rr}} \right] \quad (13)$$

2.2. Rotation Speed Difference Between Front and Rear Wheels that Occurs During Turns

If $s_{ij} = 0$ and $R_{fl} = R_{fr} = R_{rl} = R_{rr} = R$ in Eq. (9), the result will be Eq. (14).

$$\Delta N_{id} = \frac{A [V \cos (\delta - \beta) + \gamma l_f \sin \delta] - BV \cos \beta}{R} \quad (14)$$

Equation (14) does not include variation in the tire dynamic loaded radius, and it represents ΔN that does not involve slip in a longitudinal direction, so this value is a reference value.

2.3. Variation in Tire Dynamic Loaded Radius

When the tire dynamic loaded radius is different for each wheel, ΔN will always occur even if the vehicle is driving with every tire gripping. The following will describe the ΔN (ΔN_a) that is generated by tire dynamic loaded radius variation in a state of no slipping in a longitudinal direction.

Simplify Eq. (9) so that the slip ratio is zero for each wheel. If correction of the tire dynamic loaded radius variation is kept within the range of $\pm 5\%$, X_{fd} and X_{rd} are sufficiently small by comparison with X_{fa} and X_{ra} . Ignoring the terms X_{fd} and X_{rd} in Eq. (9), this can be shown by Eq. (15).

$$\Delta N_a = \frac{AX_{fa} [V \cos (\delta - \beta) + \gamma l_f \sin \delta] - BX_{ra} V \cos \beta}{R} \quad (15)$$

$$X_{fa} = \frac{1}{2} \left(\frac{R}{R_{fl}} + \frac{R}{R_{fr}} \right) \quad (16)$$

$$X_{ra} = \frac{1}{2} \left(\frac{R}{R_{rl}} + \frac{R}{R_{rr}} \right) \quad (17)$$

Equations (16) and (17) represent the average of the ratio of the reference tire radius to the radius of the wheels for the left and right front wheels and the left and right rear wheels, respectively. Here, consider the maximum value of ΔN_a .

$$\max (R_{fl}, R_{fr}, R_{rl}, R_{rr}) = R \quad (18)$$

In this case, it is clearly when $R_{rl} = R_{rr} = R$ that ΔN_a reaches its maximum (ΔN_{amx}).

$$R_{rl} = R_{rr} = R \quad (19)$$

In other words, this is under Eq. (20).

$$\Delta N_{amx} = \frac{AX_{fa} [V \cos (\delta - \beta) + \gamma l_f \sin \delta] - BV \cos \beta}{R} \quad (20)$$

Comparing Eq. (20) and Eq. (14), it is apparent that X_{fa} appears as a ratio of the coefficient A term (the rotation speed of the front wheel).

If the difference here between the actual ΔN (ΔN_{re}) and ΔN_{amx} is taken as ΔN (ΔN_{sl}), which is generated by the front and rear wheel slip ratio difference, then it follows that:

$$\Delta N_{re} = \Delta N_{amx} + \Delta N_{sl} \quad (21)$$

Considering the deviations of ΔN_{re} and ΔN_{id} (if there is no longitudinal slip and no variation in the tire dynamic loaded radius, then these will be zero), the result will be Eq. (22).

$$\begin{aligned} \Delta N_{re} - \Delta N_{id} &= (\Delta N_{amx} - \Delta N_{id}) + \Delta N_{sl} \\ &= \frac{A}{R} (X_{fa} - 1) [V \cos (\delta - \beta) + \gamma l_f \sin \delta] + \Delta N_{sl} \end{aligned} \quad (22)$$

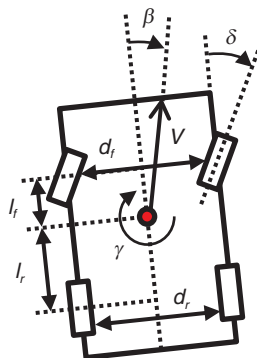


Fig. 2 Vehicle model

Transforming Eq. (22), the result will be Eq. (23).

$$\frac{R(\Delta N_{re} - \Delta N_{id})}{A[V \cos(\delta - \beta) + \gamma l_f \sin \delta]} + 1 = X_{fa} + X_{st} \quad (23)$$

$$X_{st} = \frac{R\Delta N_{st}}{A[V \cos(\delta - \beta) + \gamma l_f \sin \delta]} \quad (24)$$

If the time change in the dynamic loaded radius of each wheel is ignored in Eq. (16), then X_{fa} is a constant. In this case, the left side of Eq. (23) can be divided into the constant value X_{fa} and the variable term X_{st} .

3. AWD Driving Force Distribution Control that Includes Spin Suppression

This chapter will describe specific methods for implementing spin suppression control. Figure 3 shows a summary block diagram of this control. The amount of driving force distributed to the rear wheels is the sum determined by the amounts of driving force distributed by the base distribution controller and the rotation speed difference controller, which implements spin suppression. The amount of driving force distributed is determined predominantly by the base distribution controller, and the system is configured so that the rotation speed difference controller operates as a compensator when the driving force distribution to the rear wheels is insufficient, such as when the front wheels are tending to slip.

3.1. Base Distribution Controller

The maximum friction force of a tire is determined by the ground load and the coefficient of friction between the road surface and the tire. When the ratio of distribution of driving force between the front and rear wheels is decided by the ground load ratio, tire use can be well balanced, which makes this a useful method in terms of driving force performance and turn stability. When load shift due to longitudinal acceleration is taken into account, the ground load ratios u_f and u_r of the front wheels and rear wheels to the vehicle weight are expressed by Eqs. (25) and (26).

$$u_f = \frac{l_r}{l} - \frac{\alpha_x}{g} \cdot \frac{h}{l} \quad (25)$$

$$u_r = \frac{l_f}{l} + \frac{\alpha_x}{g} \cdot \frac{h}{l} \quad (26)$$

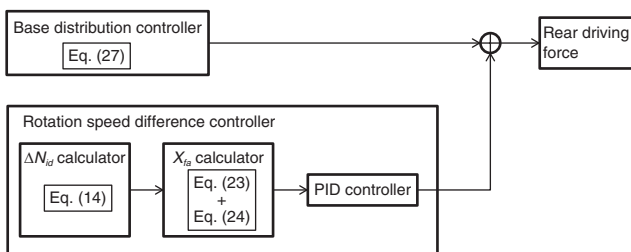


Fig. 3 Block diagram of control

The base distribution (T_b) is the value resulting from multiplication of the rear wheel ground load ratio by the estimated value of driving force generated by the source of driving force, or in other words, this is:

$$T_b = u_r T_{all} \quad (27)$$

3.2. Rotation Speed Difference Controller

The rotation speed difference control suppresses front wheel spin by increasing the distribution to the rear wheels when ΔN exceeds a certain target value. Providing an appropriate target value prevents spin suppression from disturbing the base distribution ratio in scenarios where it is unnecessary. By setting the target value to the maximum value envisioned when there is no spin, the system is configured to not do anything when ΔN_{re} is lower than the target value, and this configuration limits the amount of control implemented by the rotation speed difference control.

The control is configured with (1) a turn correction ΔN calculator that calculates ΔN when there is no longitudinal slip, (2) a ΔN variation correction unit that increases system robustness with regard to tire dynamic loaded radius variation, and (3) a ΔN servo-controller that determines the amount of increase in the distribution amount due to deviation in the actual ΔN and the derived reference ΔN .

3.2.1. Turn correction ΔN calculator

This unit calculates ω_{in} , ω_{out} and ΔN_{id} in cases where, given the driving state of the vehicle, no slip is envisioned to be occurring in a longitudinal direction. These figures are used for calculations in the ΔN variation correction unit, to be discussed later.

Note here that Eq. (14) includes the vehicle body slip angle. Cost-performance considerations, however, make it impractical to install an actual vehicle slip angle sensor to measure this quantity. For this system, therefore, the slip angle is estimated using a vehicle dynamics model⁽⁹⁾.

3.2.2. ΔN variation correction unit

This unit determines the correction factor X_{fa} for the ΔN that occurs due to the tire dynamic loaded radius variation, and calculates the reference ΔN . It suffices to choose a maximum value for the envisioned target value, and Eq. (20) can be used for the purpose. A steady-state value is derived by using a low-pass filter on the values calculated on the left side of Eq. (23).

The smaller the time constant in this low-pass filter, the shorter the time that the correction factor X_{fa} will remain a constant value. If it is too small, however, the calculated value also follows the time variation, and therefore diverges from the true correction factor X_{fa} .

There are some cases, however, in which robustness can be further heightened by causing this time variation to be followed to some extent. The left side of Eq. (23) may generate calculation errors relative to true values from sensor errors, errors in estimation from vehicle models, or

disturbances caused by uneven road surfaces. These can be considered values that vary over time. Since an increase in ΔN due to front wheel spin has a relatively fast response, it can be considered acceptable to follow the calculated value for time variation in a time constant that is sufficiently slower than an increase in ΔN that is assumed to be due to spin. However, continued spinning of the front wheels will end up generating a constant ΔN , so that the correction factor X_{fa} will need to be kept within the envisioned range of error.

The correction factor X_{fa} obtained is substituted in Eq. (20). This value is ΔN when there is no slip in a longitudinal direction, but when driven, the front-rear wheel slip ratio difference results in ΔN (ΔN_{sl}). This is handled by use of the allowable slip ratio difference ΔN (ΔN_{os}) as an offset. In other words, reference ΔN (ΔN_t) is expressed by Eq. (28).

$$\Delta N_t = \Delta N_{amx} + \Delta N_{os} \quad (28)$$

3.2.3. ΔN servo-controller

Proportional integral differential (PID) control is implemented on the basis of the deviations between ΔN_t obtained by the ΔN variation correction unit and ΔN_{re} . As described above, as a rule no control is implemented when ΔN_{re} is low relative to the target value, but the system is configured so that control will not become unstable.

PID was adopted for this ΔN servo-control in order to enhance convergence with regard to the steady-state deviation. In a state in which the deviation occurs continuously, however, the amount of control implemented will end up increasing without being limited by the integral term. When the vehicle is making turns with a short turning radius, constraints on the front and rear wheels may result in generation of braking force (the tight corner braking phenomenon).

In ΔN servo-control, therefore, the total rear wheel driving force distribution is as expressed in Eq. (29). Control is applied as a value that does not exceed the driving force generated by the source of driving force.

$$T_r \leq T_{all} \quad (29)$$

The following describes the principle of tight corner braking.

The amount of change in the front wheel rotation speed ω_f is expressed by Eq. (30).

$$I_f \frac{d\omega_f}{dt} = T_{all} - T_r - RF_f \quad (30)$$

Looking then at a steady state in which there is no change in the front wheel rotation speed, the left side of Eq. (30) becomes zero, and this can be transformed to Eq. (31).

$$T_r = T_{all} - RF_f \quad (31)$$

Braking force is generated when $F_f < 0$, but if $T_r \leq T_{all}$, then F_f will necessarily be positive. In other words, steady-state braking from constraints on the front and rear wheels can be prevented.

4. Confirmation of the Effectiveness of Spin Suppression Control

The controls described in chapter 3 were installed in a vehicle with electronically controlled AWD on a front-wheel drive base. The controls were then evaluated in terms of the validity of the reference rotation speed difference, the effectiveness of reference value correction, and the effectiveness of ΔN servo-control spin suppression.

4.1. Confirmation of the Validity of Reference Rotation Speed Difference

Figure 4 shows a comparison of the actual rotation speed difference ΔN_{re} when no driving force is applied (in other words, when the slip ratio of each wheel is virtually zero) with ΔN_{id} .

When the transient change component of ΔN_{re} is excluded, its waveform largely matches the ΔN_{id} waveform. This indicates that the calculated results for reference rotation speed difference are valid.

4.2. Confirmation of the Effectiveness of Reference Rotation Speed Difference Correction

By constantly monitoring the deviation between the reference rotation speed difference ΔN_t and the actual rotation speed difference ΔN_{re} , reference rotation speed difference correction promptly corrects for the ΔN that occurs due to such factors as tire dynamic loaded radius variation. Confirmation of the operation of reference value correction was carried out with regard to variations that occurred.

First, the value for front wheel rotation speed while the vehicle is turning in a constant circle was replaced by that value multiplied by a factor of X to artificially generate a tire dynamic loaded radius variation. It was then confirmed how ΔN_t would change with respect to ΔN_{re} , and how the correction factor X_{fa} would change. Figure 5 shows the results.

The point where ΔN_{re} increases instantaneously is the time when the front wheel rotation speed value is replaced. It is apparent that at the instant when ΔN_{re} changes, ΔN_t is

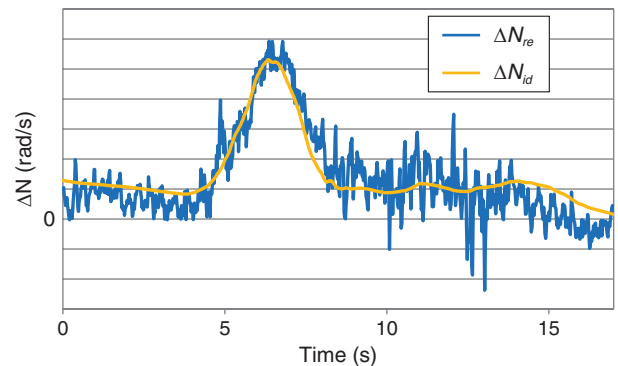


Fig. 4 Comparison of actual and calculated rotation speed difference

smaller, but the increase in X_{fa} causes ΔN_t to be corrected so that it again exceeds ΔN_{re} . As stated in section 3.2, rotation speed difference control is not implemented when ΔN_{re} is lower than ΔN_t . In other words, the tire dynamic loaded radius variation does not exert a continuous influence on the amount of driving force distributed, which is determined by the rotation speed difference control.

Next, the vehicle was driven with the front wheel air pressure lowered to create an actual front and rear wheel radius difference. Figure 6 shows the variations in ΔN_{re} , ΔN_t and X_{fa} that occurred when this was done. It is apparent that the existence of a radius difference between front and rear wheels means that there is less time when X_{fa} is one, and correction of ΔN_t is being implemented at all times. Moreover, the way that X_{fa} changes gently over time enables ΔN_t to follow the lower frequency component of ΔN_{re} .

4.3. Confirmation of the Effectiveness of Spin Suppression

Up to section 4.2, this paper has described methods for correcting the reference value without causing rotation speed difference control to operate when there is no spin. However, by correcting the reference value, correction factor X_{fa} reduces the steady-state deviation. This means that the amount of driving force distributed, which is determined according to the deviation, may be insufficient. Here the suppression effect of ΔN servo-control when there is front wheel spin was confirmed.

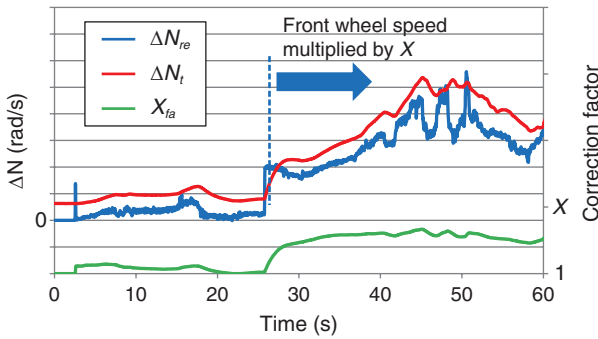


Fig. 5 Variation of correction factor for tire radius error

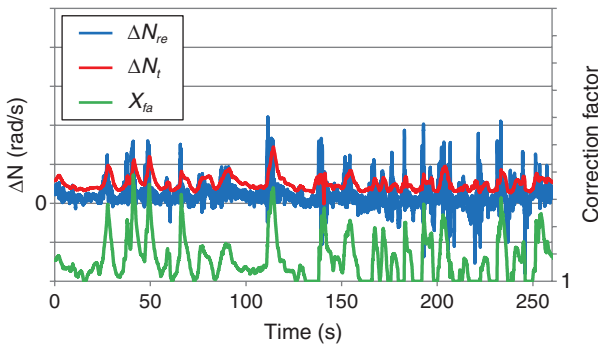


Fig. 6 Low front tire pressure driving data

Figure 7 shows the change component in rear wheel driving torque, which is determined by ΔN_{re} , ΔN_t and ΔN servo-control, when driving on a snowy road. ΔN_t does not follow the rapid change in ΔN_{re} when spin occurs, and ΔN servo-control therefore increases the rear wheel driving torque. It is apparent that this keeps ΔN_{re} in check. Moreover, the amount of ΔN servo-control implemented is largely zero, showing that the operation of spin suppression is kept in check when it is not needed.

Next to be considered is a scenario when front wheel slip can be expected, such as an uphill start from a complete stop on a snowy road. In such a case, the front wheels continue to slip for a relatively long time, causing a steady-state component in ΔN_{re} to occur. When correction is applied to this steady-state component, the amount of control implemented may be less than that needed to suppress spin.

As described in sub-section 3.2.2, therefore, a limiting value is applied within the range of error in which correction of X_{fa} is intended. Figure 8 shows the amount of ΔN_{re} , X_{fa} and ΔN servo-control implemented when starting from a complete stop on a snowy uphill road with the limit $1 \leq X_{fa} \leq X$.

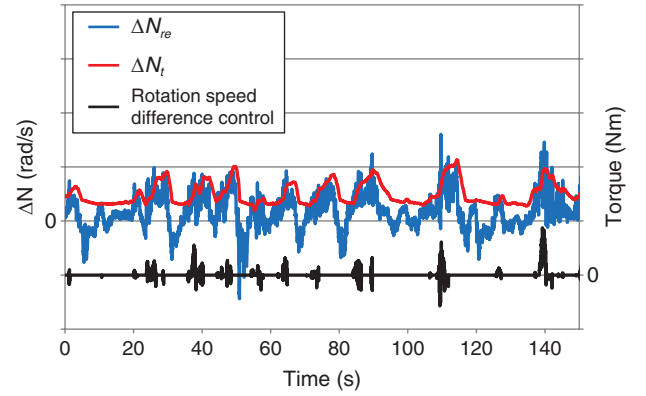


Fig. 7 Snowy road driving data

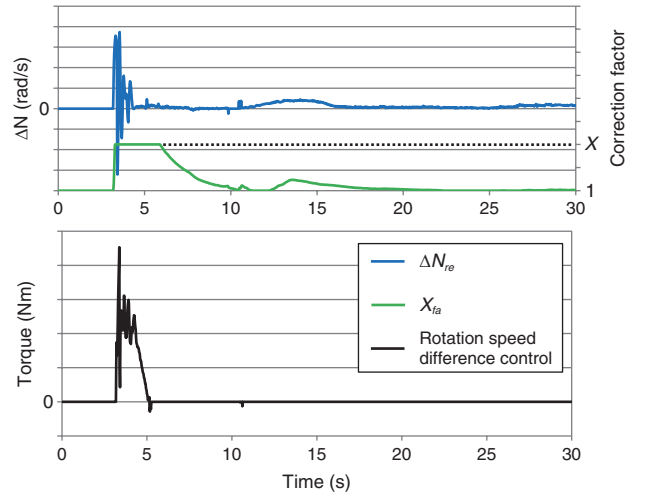


Fig. 8 Uphill snowy road driving data

It is apparent that X_{fa} rapidly increases with respect to ΔN_{re} , which is large when starting, and reaches the upper limit value X . This eliminates the decrease in deviation due to correction, so that PID is activated in ΔN servo-control, and the rotation speed difference converges.

5. Conclusion

In the implementation of spin suppression control that increases the amount of driving force distributed to the rear wheels in conjunction with the occurrence of spin in the front wheels, the rotation speed difference between front and rear wheels that occurs when the vehicle is turning was sought by geometrical means and applied to correct the target value. A correction factor determined from the steady-state deviation between the actual value and target value for rotation speed difference between the front and rear wheels was also used successfully to correct for rotation speed differences that occur due to such factors as variations in the tire dynamic loaded radius.

The use of these methods showed that spin suppression control would not operate even in vehicle states with a rotation speed difference between front and rear wheels that occurs when the front wheels are not spinning. This also showed that defining the range for the correction factor appropriately would allow limiting the reduction in the amount of spin suppression control implemented that occurs due to correction, and could rapidly produce convergence in the rotation speed difference between front and rear wheels.

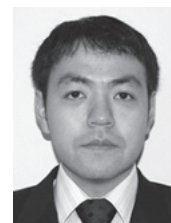
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■ Author ■



Shoichi OHYAGI



Takayuki SEKI