

Torque Steer Simulation Technology for Performance Design in Model Based Development

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ABSTRACT

A differential gear model was devised that can be used to examine torque steer when the vehicle is launching from standstill in MBD performance design. The model was expanded to enable reproduction of the transient left-right torque bias ratio, which was previously treated as a constant, by adding an element that takes into account the left-right difference in rotational speed derived from uneven torsional stiffness of driveshaft. Conventional methods needed a complex model using a multibody dynamics model. The proposed method uses a simple essential model to express the torque steer mechanism. This enabled analysis of the contribution due to the difference in transient drive torque in MBD performance design. As a result, countermeasure methods could be quantitatively and efficiently examined in performance design in the initial stages of development. The proposed method was applied to development of 2016 model year mass-production vehicles, and its usefulness was confirmed.

1. Introduction

In addition to the basic performances of running, turning, and stopping, automobiles face increasing demands for safety, environmental, and other performances in order to achieve environmentally conscious motorization. However, the mechanisms and structures used to meet these demands are becoming increasingly complex, and shorter development periods and cost reduction are also demanded in automotive development. This is increasing the importance of model-based development (MBD) as a means for efficiently promoting development⁽¹⁾. Figure 1 shows a graphic representation of the mass-production vehicle development process as a V-model. The items for examination in the MBD area enclosed by the red border can be expanded by incorporating the elements of each automotive component into the simulation model⁽²⁾⁻⁽⁴⁾.

Vehicles that use a small displacement turbo engine (compact T/C) in the power unit or which are equipped with an electric motor, such as an EV, are becoming popular as means of enhancing environmental performance. Figure 2 shows an example of the torque performance of these power units. Compared to the conventional naturally aspirated engine, the compact T/C features a supercharging effect that enables production of high torque from a low rotational speed. In addition, the electric motor can produce even higher torque from lower rotational speeds compared to the

internal combustion engines.

In a front wheel drive vehicle, high drive torque when starting from a standstill may pull the steering wheel in a direction not intended by the driver, with the result that the vehicle attempts to turn. This phenomenon is called torque steer, and is generally an anxiety factor for drivers. Typical methods for reducing this torque steer include making the left and right drive shafts the same length, reducing the left-right difference in drive shaft torsional stiffness, reducing the left-right difference in the drive shaft and axle joint angles, shortening the king pin center offset, and so on^{(5), (6)}. In addition, geometry that shortens the king pin center offset

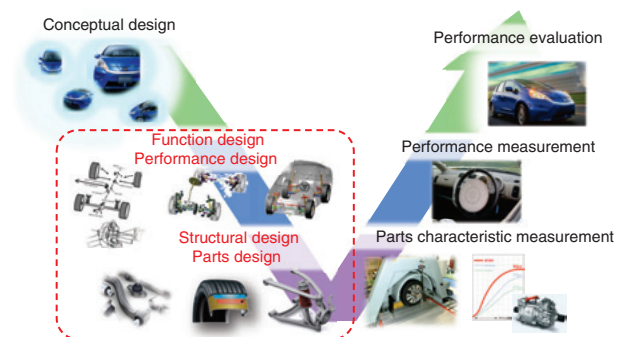


Fig. 1 Positioning of MBD in V-Model process

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is sometimes added to the suspension in vehicles equipped with high torque engines^{(7), (8)}.

In vehicle development, replacement with a compact T/C and mounting an electric motor can enhance both fuel economy and standing start performance, so these trends are continuing to increase. As a result, models with torque steer issues are expected to increase. This calls for development of simulation technology that enables stimulation of the influence of torque steer and formulate countermeasures in the initial stages of development in order to enhance development efficiency.

Torque steer characteristics are simulated by full vehicle model calculations using a multibody dynamics model such as shown in Fig. 3^{(9), (10)}. In analysis using a multibody dynamics model, high calculation accuracy can be obtained by creating detailed component models. However, it takes

much time and work to collect the information needed for modeling and to reflect it in the calculations. Furthermore, it is not easy to get all the information needed for modeling and to change the characteristics of the multibody dynamics model in the initial stages of development.

The left-right torque bias ratio of the differential gear can be treated simply as a constant. In this case, however, the difference in the drive torque distributed to the left and right wheels becomes larger than that in an actual vehicle. This makes it a challenge to simulate the influence of torque steer at the same accuracy as that of other performance design items in mass-production vehicle development.

Therefore, a differential gear modeling method that can be applied to performance design in the initial stages of development was studied by understanding the intrinsic mechanisms by which torque steer is caused.

2. Torque Steer Mechanism

Uneven torque around the king pin axis due to the uneven drive torque of the left and right wheels is transmitted to the steering wheel, resulting in torque steer. The following three mechanisms are the main factors causing uneven drive torque:

- (a) Left-right difference in drive shaft and axle joint angles
- (b) Left-right difference in drive shaft torsional stiffness
- (c) Left-right difference in torque distribution by the differential gear

The drive shafts are normally coupled with the axle via a universal joint so that they can track the motion of the suspension. If drive torque is transmitted in the condition with a joint in the angled condition, then torque component force is generated on that joint around the axis orthogonal to the axle. Figure 4 shows a schematic expression of the joint between the drive shaft and the axle.

If the drive torque distributed to the left and right drive shafts is T_i and the angle of the joint between the drive shaft and the axle is θ_{o_i} , then the torque M_{o_i} around the axis orthogonal to the axle is expressed by Eq. (1)⁽¹¹⁾ ($i = L$ or R).

$$M_{o_i} = T_i \cdot \sin \theta_{o_i} \quad (1)$$

In addition, if the king pin inclination angle is θ_{z_i} , then the torque $M_{KP_1_i}$ around the king pin axis is expressed by Eq. (2).

$$M_{KP_1_i} = M_{o_i} \cdot \cos \theta_{z_i} \quad (2)$$

Here, the joint angle and the king pin inclination angle vary according to the oscillation of the suspension and the power train. The left and right θ_{o_i} differ in many cases, and the left and right M_{o_i} also differ, so this causes torque steer.

The torque output from the power train is distributed to the left and right drive shafts via the differential gear. If there is a left-right difference in the drive shaft torsional stiffness at this time, then a transient left-right difference

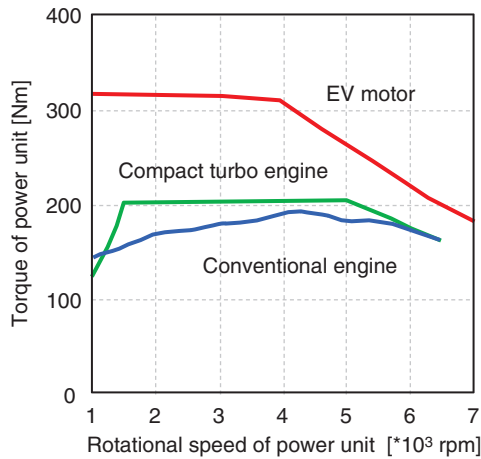


Fig. 2 Typical torque characteristics curves of different power units

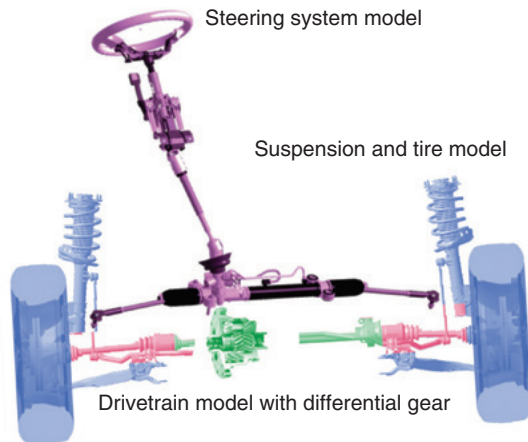


Fig. 3 Schematic image of component models for full vehicle torque steer simulation

in the rotational speed occurs. Ideally, the difference in the rotational speeds of the left and right driving wheels should be absorbed and equal drive torque should be transmitted to both tires. However, an actual differential gear has internal friction elements such as gear friction, so a left-right difference occurs in the torque distributed by the differential gear, within the range that the differential is limited by this friction. This torque difference becomes a left-right difference in the driving force, and a left-right difference also occurs in the torque around the king pin axes. This torque is called the torque around the king pin axis due to transient drive torque difference, and is a factor causing transient torque steer. Figure 5 shows a schematic expression of the drivetrain including the differential gear. If the tire radius is R and the king pin center offset is ζ_z , then the torque M_{KP_2-i} around the king pin axis due to transient drive torque difference can be expressed by Eq. (3).

$$M_{KP_2-i} = \frac{T_i \cdot \cos \theta_{o-i}}{R} \cdot \zeta_z \cdot \cos \theta_{z-i} \quad (3)$$

That is to say, when simulating torque steer, the simulation model should take into account the torque around the king pin axis due to the left-right difference in drive torque resulting from the differential gear and the left-right difference in drive shaft stiffness. The torque transmitted to the steering geometry can be calculated using the total force produced on the steering rack shaft (F_{Rack}), which is determined from the torque around the king pin axis and the link efficiency⁽²⁾. In order to reduce torque steer, it is necessary to reduce the maximum value of F_{Rack} .

In full vehicle model calculations using a multibody

dynamics model, the transient left-right torque bias ratio can be accurately reproduced by creating a detailed model of the differential gear geometry. However, even a multibody dynamics model is unable to reproduce the torque around the king pin axis due to transient drive torque difference if the left-right torque bias ratio is treated as a constant.

Therefore, a differential gear model was examined that can reproduce the left-right torque bias according to transient differences in drive shaft rotational speed, at an accuracy equivalent to that of a multibody dynamics model.

3. Differential Gear Model

3.1. Model Outline

If the drive torque distributed to the left and right drive shafts is respectively T_l and T_r , then the drive torque T and the left-right difference in drive torque ΔT are expressed by Eq. (4) and Eq. (5).

$$T = |T_l + T_r| \quad (4)$$

$$\Delta T = |T_l - T_r| \quad (5)$$

In addition, ΔT is expressed by Eq. (6) and Eq. (7) using the torque bias ratio (TBR), which is the ratio of the torque of the low rotational speed side to the torque of the high rotational speed side of the differential gear.

$$TBR = \left(\frac{T}{2} + \frac{\Delta T}{2} \right) / \left(\frac{T}{2} - \frac{\Delta T}{2} \right) \quad (6)$$

$$\Delta T = T \frac{(TBR-1)}{(TBR+1)} \quad (7)$$

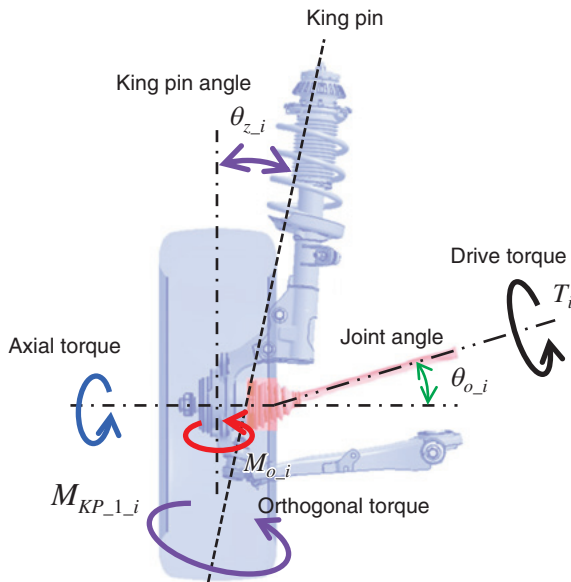


Fig. 4 Schematic expression of drive torque components around wheel hub

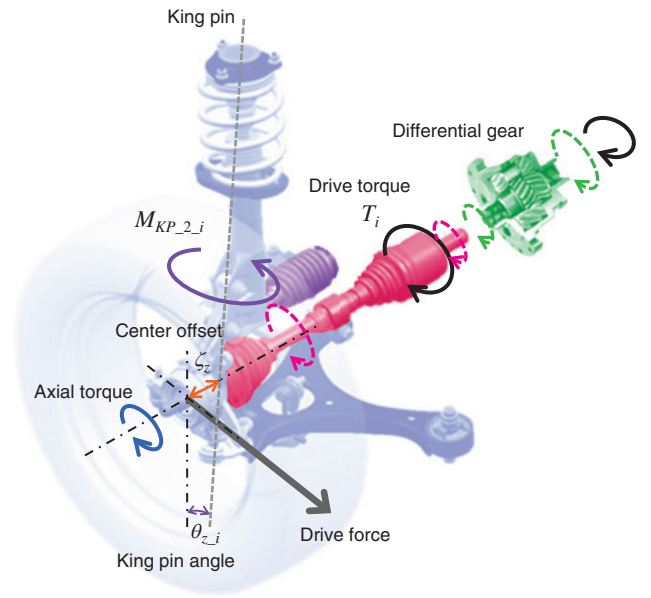


Fig. 5 Schematic expression of transient torque around king pin due to drive shaft stiffness

In a differential gear model without internal friction, the drive torques T_l and T_r , distributed to the left and right sides, are the same, and TBR is 1. In an actual differential gear, however, the differential is limited by internal friction, so a left-right difference occurs in the drive shaft rotational speeds. As a result, T_l and T_r are not the same, and TBR is a value greater than 1.

Meanwhile, the drive torque T_i , distributed to the drive shafts, produces a torsional angle θ_i in accordance with the drive shaft torsional stiffness k_i as shown in Eq. (8).

$$\theta_i = \frac{T_i}{k_i} \quad (8)$$

Here, if the left and right torsional stiffness differ, then a difference also occurs in the torsional angle, and a left-right difference occurs in the drive shaft rotational speed. From Eq. (8), this transient difference in rotational speed is the time series change in the torsional angle difference $d\Delta\theta/dt$, as expressed by Eq. (9).

$$\frac{d\Delta\theta}{dt} \approx \frac{1}{2} \frac{dT}{dt} \left(\frac{1}{k_l} - \frac{1}{k_r} \right) \quad (9)$$

Equation (9) shows that the transient difference in rotational speed $d\Delta\theta/dt$, depends on the time series change in the drive torque dT/dt . That is to say, the left-right difference in drive shaft torsional stiffness becomes a difference in rotational speed $d\Delta\theta/dt$, which causes fluctuations in the difference in drive torque ΔT . As a result, the previously described TBR also fluctuates, making a coupled issue that is a challenge to solve analytically.

The proposed differential gear model was modeled on the assumption that TBR changes in accordance with the difference in rotational speed when the left-right difference in drive shaft rotational speed is smaller than the boundary rotational speed difference N as shown in Fig. 6. In addition, the model that treats TBR as a constant is also shown for comparison.

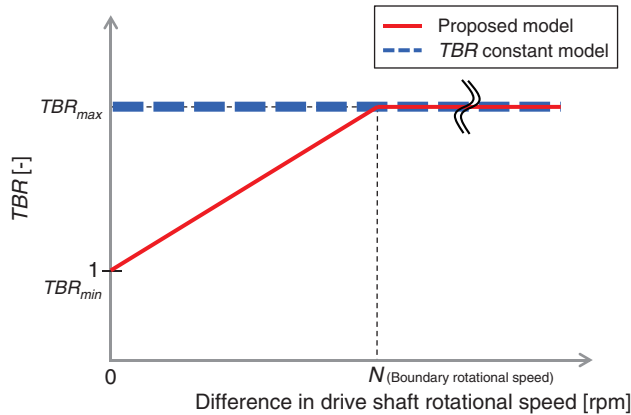


Fig. 6 Proposed TBR characteristic of differential model

Table 1 Calculation parameters and conditions

Weight: 1400 kg, Wheel base: 2.6 m, Center offset: 50 mm

Items	Note
Proposed method	TBR is variable depending on differences in rotational speed ($1 \leq TBR \leq 2.25$)
TBR constant	TBR is constant at design specification value ($TBR = 2.25$)

In contrast to the constant TBR case, the TBR of the proposed model varies according to the left-right difference in drive shaft rotational speed. TBR is the lower limit value at a rotational speed difference of zero, and increases linearly together with the rotational speed difference until it reaches the upper limit value at the boundary rotational speed difference N , after which it remains constant.

Given the above assumption, TBR is expressed by Eq. (10) as a function of $d\Delta\theta/dt$, and the left-right drive torque difference ΔT is expressed by Eq. (11).

$$TBR \left(\left| \frac{d\Delta\theta}{dt} \right| \right) = \begin{cases} \frac{(TBR_{max}-1)}{N} \cdot \left| \frac{1}{2} \frac{dT}{dt} \left(\frac{1}{k_l} - \frac{1}{k_r} \right) \right| + 1 & \left(\left| \frac{d\Delta\theta}{dt} \right| < 1 \right) \\ TBR_{max} & \left(\left| \frac{d\Delta\theta}{dt} \right| \geq N \right) \end{cases} \quad (10)$$

$$\Delta T = T \cdot \left[TBR \left(\left| \frac{d\Delta\theta}{dt} \right| \right) - 1 \right] / \left[TBR \left(\left| \frac{d\Delta\theta}{dt} \right| \right) + 1 \right] \quad (11)$$

Equation (11) shows that ΔT increases together with T , and Eq. (10) shows that TBR increases together with dT/dt . Therefore, the maximum left-right difference in drive torque is produced when T^*dT/dt is the maximum value. The torque $M_{KP,2}$ around the king pin axis due to transient drive torque difference can be obtained from Eq. (3) by using the left-right difference in drive torque ΔT . With regards to the drive torque distributed to the left and right wheels at this time, the side with high drive shaft torsional stiffness becomes the high torque side.

3.2. Verification of Effects

Torque steer simulation results obtained by a model that embeds the proposed model in an MBD performance design tool and the conventional model that treats TBR as a constant were compared with the calculation results obtained by a full vehicle model. Table 1 shows the calculation conditions.

Figure 7 shows the time series response of the drive torque used in the calculation T^*dT/dt , and F_{Rack} obtained using the proposed model. The maximum F_{Rack} is produced after around 1.4 s when T^*dT/dt is the maximum value.

Figure 8 shows a graph with the relationship between the difference in rotational speed and TBR at the maximum F_{Rack} in the MBD performance design tool and full vehicle model calculation results, superposed on the characteristic

assumed in Fig. 6. This shows that the results obtained by the MBD performance design tool using the proposed model are nearly the same as the results obtained by the full vehicle model.

Figure 9 compares the F_{Rack} produced by the drive torque in each model. Here, the contributions of the drive torque difference and the drive torque transmission of the axle joint to F_{Rack} are indicated by different colors for each mechanism. This shows that the calculation results using the proposed model match the calculation results of the full vehicle model, which indicates that torque steer can also be quantitatively calculated in the MBD performance design tool.

In contrast, when TBR was treated as a constant,

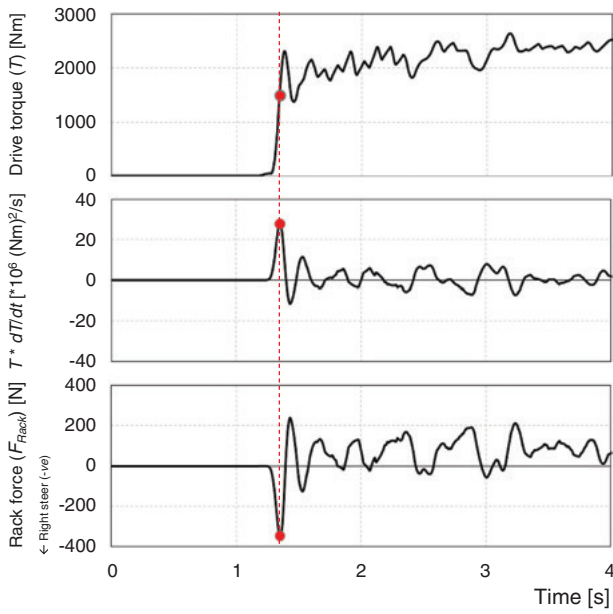


Fig. 7 Input torque (upper), proposed index (middle), and F_{Rack} (bottom) from proposed simulation

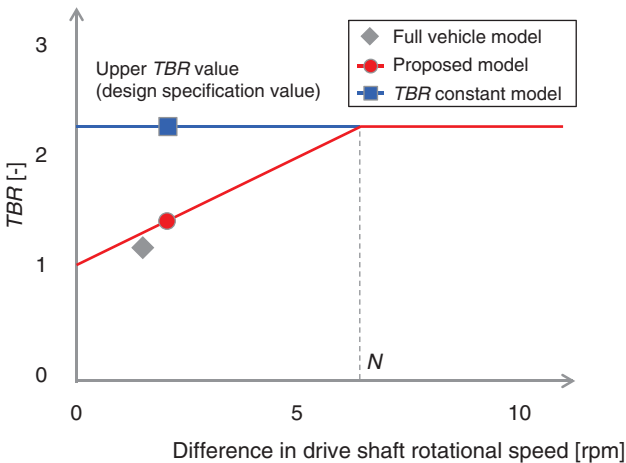


Fig. 8 TBR characteristics used in calculation

the contribution due to the drive torque difference was calculated to be larger, so F_{Rack} was larger than the detailed calculation result of the full vehicle model and the simulation accuracy also decreased. This means that TBR is assumed to be larger than the actual value in Fig. 8.

In addition, the effects of changes in the drive shaft torsional stiffness, the axle joint angle, and the king pin center offset were verified using the proposed model. Figure 10 compares the contributions to F_{Rack} with the

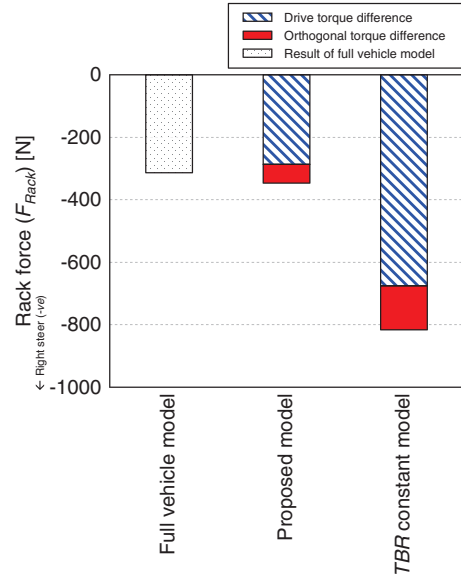


Fig. 9 Difference in F_{Rack} between proposed and TBR constant model compared with result of full vehicle model

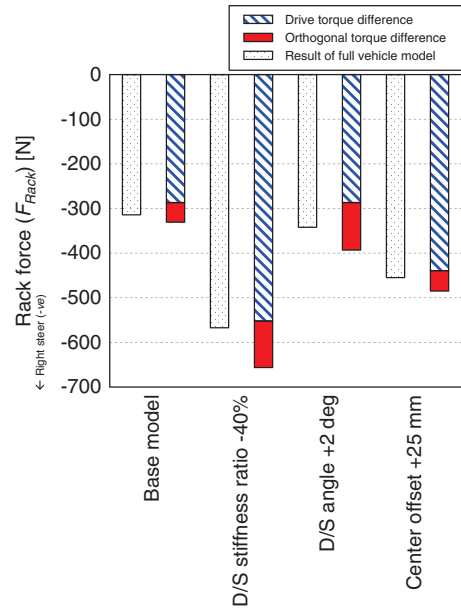


Fig. 10 F_{Rack} factor analysis results of proposed model compared with results of full vehicle model

Table 2 Calculation parameters for model verification

Weight: 1400 kg, Wheel base: 2.6 m, Center offset: 50 mm

Items	Note
D/S stiffness ratio -40% (Drivetrain effect)	Decrease D/S stiffness ratio by 40%
D/S angle +2 deg (Drivetrain effect)	Increase D/S joint angle by 2 deg
Center offset +25 mm (Chassis effect)	Increase center offset of front suspension geometry by 25 mm

D/S: Drive shaft

calculation results of the full vehicle model. Table 2 shows the calculation conditions. In each case, F_{Rack} closely matched the calculation results of the full vehicle model.

These results confirmed that the proposed model can accurately reproduce transient changes in TBR at an accuracy equivalent to that of a more detailed multibody dynamics model. In addition, it was found that a torque steer simulation model that incorporates the proposed model can accurately reproduce the left-right drive torque bias. This demonstrated that the proposed model can be used to accurately and efficiently examine various countermeasure methods based on drivetrain and chassis components, such as drive shaft and suspension characteristics, from the initial stages of development.

4. Case of Application

This chapter describes a case of applying a differential gear model constructed using the developed technology to MBD performance design in development of a 2016 model year mass-production vehicle platform. This development planned to change the power unit to a compact T/C in order to enhance fuel efficiency, and there were concerns that the increased drive torque would cause torque steer. Therefore, the developed technology was used to quantitatively simulate the influence of torque steer and formulate countermeasures.

Figure 11 shows the sensitivity of the drivetrain and chassis countermeasure methods to F_{Rack} , compared to the initial specifications for this case. The countermeasure methods assumed changes to the drive shaft stiffness, drive shaft joint angle, and king pin center offset. Each item was varied by up to $\pm 10\%$ of the design value, and the sensitivity was calculated. The horizontal axis expresses the contribution to F_{Rack} due to transient drive torque difference, and the vertical axis expresses the contribution to F_{Rack} due to the drive torque transmission of the axle joint.

Contribution analysis shows that increasing the drive shaft torsional stiffness by approximately 6% is an effective countermeasure method for the development vehicle to

Table 3 Countermeasure list

Weight: 1530 kg, Wheel base: 2.7 m, Center offset: 78 mm

Items	Note
D/S stiffness	Increase D/S stiffness by +6%

achieve the target of a F_{Rack} equivalent to that before the change to a compact T/C.

The developed technology can be used to examine specifications while quantitatively understanding how each countermeasure method contributes to each torque steer mechanism. This makes it an effective method for performing streamlined performance design.

Table 3 shows the calculation condition for the specification with torque steer countermeasures applied to a vehicle equipped with a compact T/C.

Figure 12 shows the F_{Rack} calculation results for the development vehicle before and after changing to a compact T/C and for the proposed specification. This shows that increasing the drive shaft stiffness by 6% as a countermeasure achieves the same F_{Rack} as that before changing to a compact T/C.

These results show that application of a simulation model constructed using the developed technology enables examination of countermeasure methods using drivetrain and chassis components based on quantitative simulation of torque steer in the initial stages of actual platform development, thus demonstrating the usefulness of this technology.

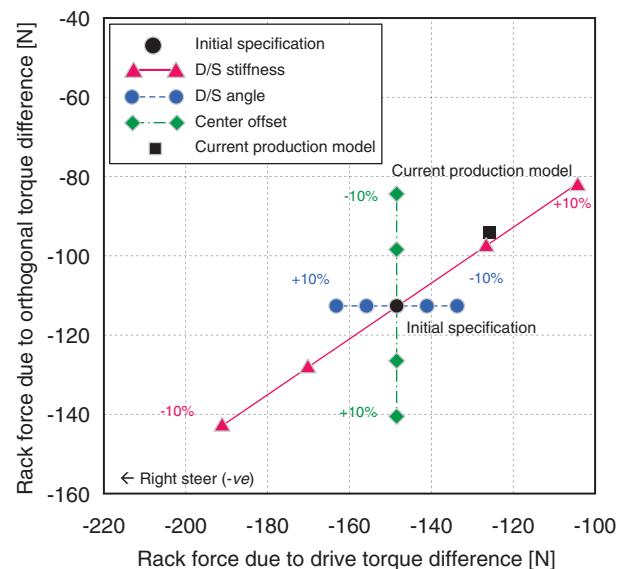


Fig. 11 Sensitivity of countermeasure items

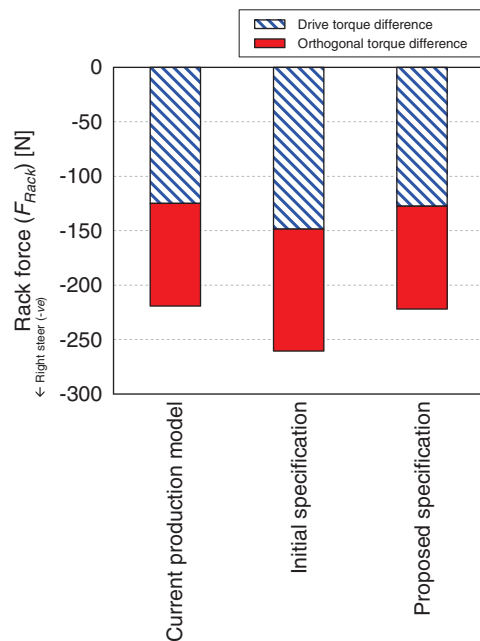


Fig. 12 Difference in F_{Rack} between initial specification and proposed specification

5. Conclusion

A simulation model was proposed that can reproduce the transient left-right torque bias ratio at an accuracy equivalent to that of a more detailed multibody dynamics model.

- (1) The mechanisms that cause the left-right uneven drive torque that is a factor resulting in torque steer were modeled by division into three left-right difference elements.
- (2) A differential gear model was devised that treats *TBR* as variable by the rotational speed difference. This enabled calculation at the same accuracy as that of a detailed model.
- (3) Application of the developed technology to an MBD performance design tool enabled quantitative and efficient examination of countermeasure methods during performance design in the initial stages of development.

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