

Technique for Simulating Vehicle Handling in Very Low Speed Range for Model Based Development

Takayuki TOYOSHIMA*

Yoshitomo AZEKATSU*

Yoshiaki KOBAYASHI*

Naofumi IKEDA*

Hiroki TAKANO*

Isao OBATA*

ABSTRACT

A conventional performance design simulation model, used in the development of mass production vehicles, was taken as the base to which a steering system model was newly added on to create a technique capable of simulating vehicle handling at very low speeds by performing a bulk calculation of those indices together during the initial stages of development. As a result, there is no longer a need to create separate simulation models for each index, and the efficiency of performance design has been enhanced. Formerly, the performance index of steering return torque could not readily be examined by means of simulation during the initial stages of development, but this can now be estimated seamlessly from high speeds to very low speeds by changing the vehicle speed condition. This technique is effective for performance design during the initial stages of development, including for investigation of specifications to achieve a balance between vehicle handling at the very low speed range and handling stability at the high speed range, for analysis of factors in torque around the king pin axis, and other such investigation. It has been applied to the development of chassis platforms for compact vehicles that were adopted, starting with the 2014 model year FIT.

1. Introduction

In recent years, model based development (MBD) has become an effective means of raising the efficiency of automobile development, which is growing increasingly complex^{(1), (2)}. Figure 1 shows the development process for mass-production vehicles following the V-model. MBD is a development method that uses simulation to advance the examination of specifications at the stages of function design, performance design, and so on, in the area at the bottom left of Fig. 1 shown enclosed by a red border.

In MBD for mass-production vehicles, it is important to calculate a variety of characteristic values within a limited time and to create an environment that allows information to be shared promptly⁽³⁾. This approach is also called on to respond to requests for investigation of performance design items that formerly were not readily subject to calculation or measurement.

One dynamic performance design item in the high speed range that has had to be created along the way is the model for prediction of lap time⁽⁴⁾ and another is the model that can compute predicted crosswind stability⁽⁵⁾, which is an item of performance robustness with regard to external disturbance. These and other such items were part of the measures taken

to expand the MBD domain. Recent years have also brought increasing request to investigate vehicle handling in the very low speed range that supports automatic parking functions and autonomous driving functions (hereafter very low speed range vehicle handling). Figure 2 shows the vehicle speed ranges covered in vehicle dynamics simulations created to date for the initial stages of development. Conventional

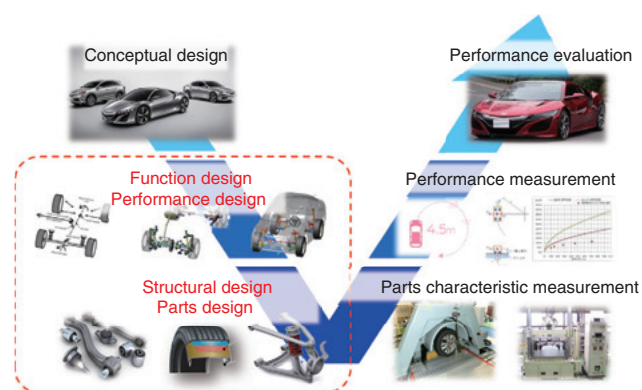


Fig. 1 Positioning of MBD in V-Model

* Automobile R&D Center

simulation covered vehicle speeds ranging from tens of kilometers per hour to hundreds of kilometers per hour, but did not support study of the very low speed range. The reason for this is that conventional simulation for use in MBD linearized the characteristics of vehicles in a driving state and approximated them with a laterally symmetrical model in order to make calculation more efficient.

In scenarios that have a vehicle driven with its steering wheel turned sharply to one side, however, there is an issue with conventional models because they could not increase the accuracy of their calculation due to the influence of the non-linear characteristics and the lateral asymmetry of the steering system. Consequently, vehicle dynamics characteristics that take into account non-linearity and asymmetry when driving in the very low speed range were investigated using full vehicle simulation. Figure 3 shows a conceptual image of a full vehicle simulation model. This approach has to model systems at the level of individual

components, and such a model could not readily be created with the limited information available during the initial stages of development. This was not suited as a predictive model for use in MBD of mass-production vehicles in the initial stages of development.

There have been instances of research in predictive models that are limited to very low speed ranges⁽⁶⁾, but there have not been many in simulation models for use in MBD. Therefore a new MBD simulation model was created that can perform bulk calculation of dynamic performance across a wide range of vehicle speeds from very low speed to high speed using only the information available during initial development.

2. Aims

With vehicle speeds of 10 km/h or less defined as the very low speed range, the following were the objectives set for a simulation model that would enable investigation of dynamic performance from very low speed to high speed under the same environment.

- (1) It is a simulation model capable of prediction using only information available in the initial stages of development.
- (2) It is a model capable of simultaneously predicting characteristic values for vehicle handling in a very low speed range and characteristic values for handling stability when driving at high speed.
- (3) It is a model capable of outputting prediction results with accuracy and efficiency that are usable in the development of mass-production vehicles.

3. Vehicle Dynamics Characteristics at the Very Low Speed Range

Research on vehicle handling at the very low speed range has been conducted from a variety of angles⁽⁷⁾⁻⁽¹⁰⁾. Vehicle dynamics characteristics at the very low speed range are closely related to vehicle handling in such scenarios as left and right turns at intersections, U-turns, and turning one direction and then another when parking. Indices of vehicle handling at the very low speed range can be classified under vehicle maneuverability and torque characteristics generated in the steering wheel. A prime example of the former type of index is minimum turning radius, and of the latter type of index is steering return torque. These two indices were taken as the objects of evaluation for the present research.

3.1. Minimum Turning Radius

Minimum turning radius is defined as the radius of the trajectory drawn by the tire ground contact center when the vehicle is moving forward slowly with the steering wheel turned all the way in one direction^{(11), (12)}. Figure 4 shows an overview. The four wheels all describe a different radius on the turn trajectory, but for this paper, the radius B of

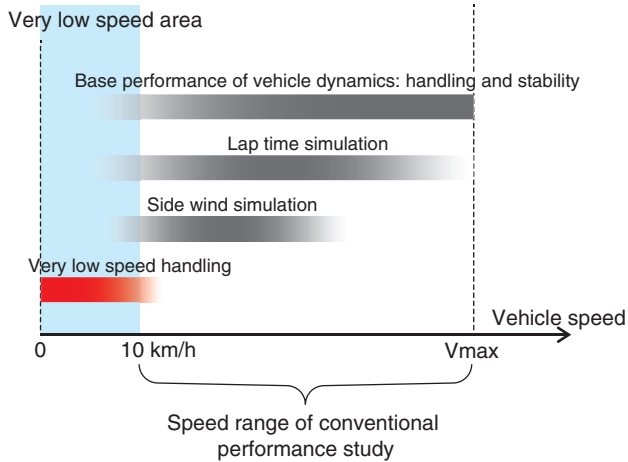


Fig. 2 Domain of performance design

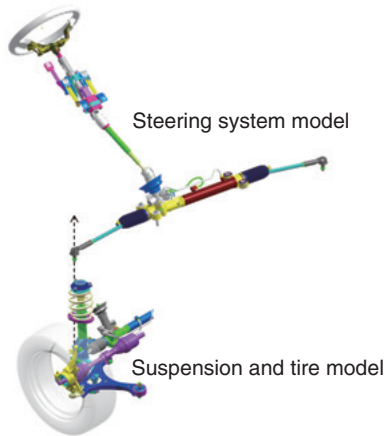


Fig. 3 Image of full vehicle simulation model

the trajectory drawn by the tire ground contact center on the outside end of the front axis is taken as the minimum turning radius. When a vehicle that is making a constant turn with a fixed steering angle gradually reduces its speed, ordinarily the turning radius also grows smaller, as shown in Fig. 5. The turning radius becomes smallest when the vehicle speed approaches zero while turning with the sharpest steering angle, and in this paper, that is taken as the minimum turning radius R_{min} .

3.2. Steering Return Torque

When the steering wheel is turned, the torque that works to return the wheel to the straight ahead state is called the steering return torque (M_{Steer}). The solid line in Fig. 6 represents M_{Steer} with respect to the steering wheel angle (A_{Steer}). Depending upon whether the steering wheel is turned farther toward the turn or turned back toward the center, the friction (M^*) is added to M_{Steer} in the opposite direction, respectively. Consequently, M_{Steer} with respect to A_{Steer} forms the Lissajous figure of an S-curve with a width shown by the dotted lines on the graph.

Figure 7 shows a simplified image of a typical

steering system. When M^* elements are excluded, M_{Steer} is proportional to the input of the resultant force (F_{Rack}) from the left and right of the rack, while A_{Steer} is proportional to the rack stroke (S_{Rack}), and their relationship is described by Eq. (1).

$$M_{Steer} / A_{Steer} = F_{Rack} / S_{Rack} \quad (1)$$

According to Eq. (1), the A_{Steer} - M_{Steer} characteristic and S_{Rack} - F_{Rack} characteristic will form similar figures. For the present simulation, therefore, the relationship between A_{Steer} and M_{Steer} is replaced with the relationship between S_{Rack} and F_{Rack} , based on this principle, and the aim is made to calculate the S_{Rack} - F_{Rack} characteristic.

With regard to steering return characteristics, it intuitively makes sense for F_{Rack} to increase uniformly with respect to S_{Rack} . Even when assist control from a motor or other such source is taken into consideration, it is more advantageous for it to change uniformly than to have it change by reversing its polarity at unspecified points⁽¹³⁾⁻⁽¹⁵⁾. In terms of ease in returning the steering wheel to the straight forward state, it is also desirable for the steering

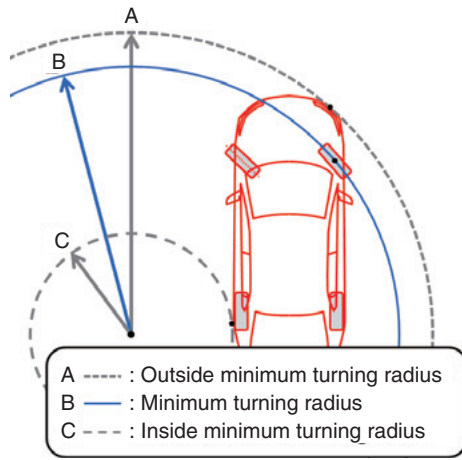


Fig. 4 Scheme of minimum turning radiuses

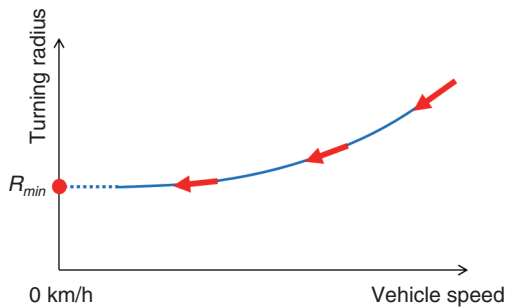


Fig. 5 Turning radius with respect to vehicle speed

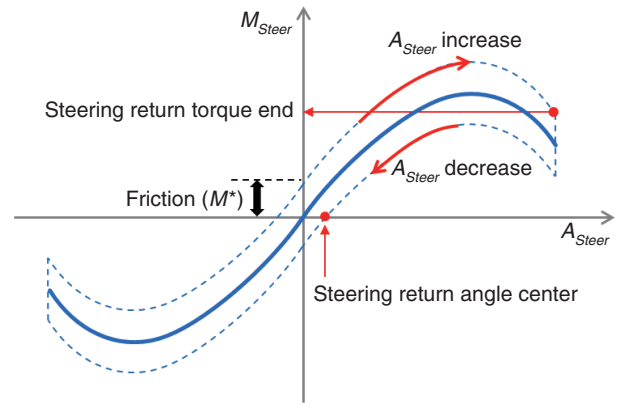


Fig. 6 Steering return torque (M_{Steer})

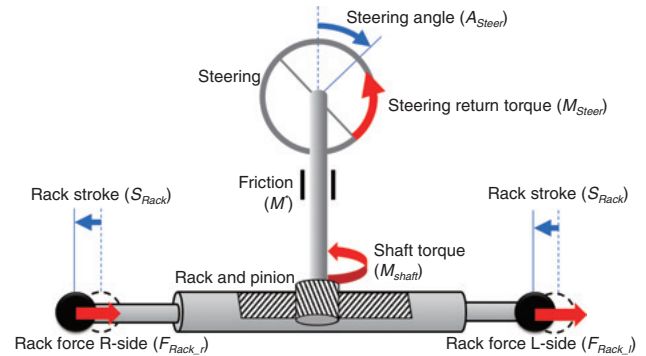


Fig. 7 Steering system

return angle center to be smaller. In other words, it has also been observed that F_{Rack} increases uniformly with respect to S_{Rack} , and at the point where $F_{Rack} = 0$, it is desirable for S_{Rack} to be as small as possible.

4. Means of Accomplishment

4.1. Overview of Predictive Model

Figure 8 shows the elements added to the conventional simulation model. In order to obtain R_{min} with sufficient accuracy, the basic elements of the suspension and steering system that make up the Ackermann geometry were modeled and incorporated. Incorporating the concept of Ackermann geometry produces a difference in the changing left and right wheel angle (δ) and the changing tire slip angle (θ). Taking the difference in the left and right wheel angle (δ) change and tire slip angle (θ) change also into consideration, the relational expressions for the tire lateral force (F_y) and drag (F_d) are determined for each wheel. The steered wheels F_y are converted into the torque around the king pin axis (M_{KIP}) on the left and right, respectively. M_{KIP} is transmitted separately to left and right through interposed basic elements making up the Ackermann geometry and become F_{Rack} . The ratio of F_{Rack} to M_{KIP} is referred to here as link efficiency.

When link efficiency can be defined, then F_{Rack} can be derived from M_{KIP} and S_{Rack} can also be derived from δ . In this way, the R_{min} and S_{Rack} - F_{Rack} characteristic can now be predicted simultaneously with the prediction of performance indices in other vehicle speed ranges.

Given the above, the basic concepts added to the conventional model to form the means of accomplishment can be classified under the following three headings.

- (1) Prediction of F_y for each of the four wheels taking the Ackermann ratio into consideration
- (2) Prediction of M_{KIP} for left and right independently
- (3) Prediction of S_{Rack} and F_{Rack} that takes link efficiency into consideration

4.2. Force Exerted on Contact Patches of the Four Wheels

When the wheel angle becomes larger while turning in the very low speed range, F_y with differing directions is generated in the left and right wheels. That is accompanied by the generation of F_d and in order to maintain the vehicle at a constant speed, a drive force commensurate with F_d will also be needed. These forces cannot be ignored when driving in the very low speed range. The forces generated in the tire contact patch were therefore classified under the following three headings and incorporated into the model.

- (A) Lateral force generated in the direction of the center of the turn
- (B) Resistance force that causes the vehicle to decelerate
- (C) Drive force that causes the vehicle to accelerate

4.3. Force Exerted on Contact Patches of the Four Wheels

When the wheel angle becomes larger while turning in the very low speed range, the direction of the tire slip angles on the left and right wheels will differ according to how the Ackermann geometry is incorporated. For that reason, F_y is generated in an opposing direction, in conjunction with which F_d is also generated in the respective tires. In order to maintain the vehicle at a constant speed, therefore, a drive force commensurate with F_d will also be needed. These forces cannot be ignored when driving in the very low speed

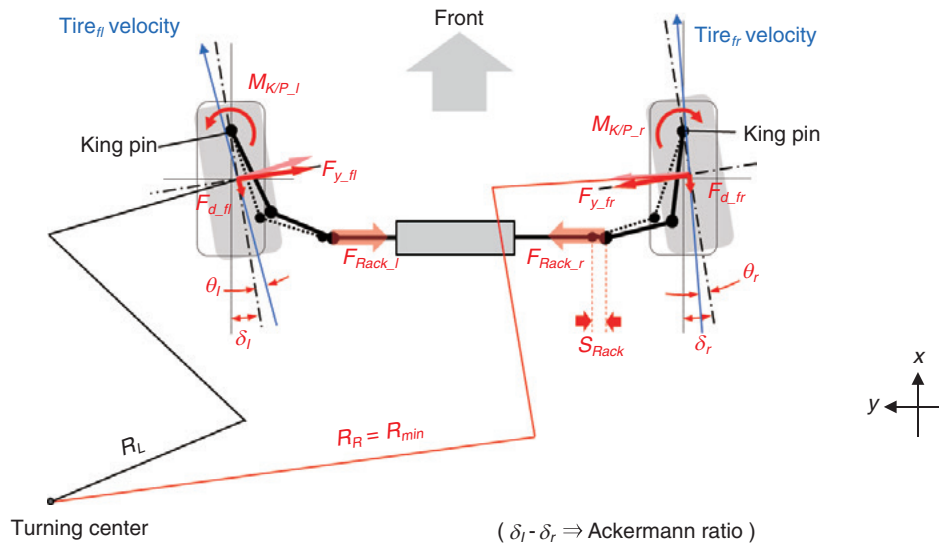


Fig. 8 Outline of model for calculating very low speed driving

range. All the forces generated in the tire contact patch were therefore classified under the following three headings and incorporated into the model.

(A) Lateral force generated in the direction of the center of the turn

(B) Resistance force that causes the vehicle to decelerate

(C) Drive force that causes the vehicle to accelerate

If the low speed state is so extreme that lateral acceleration can be ignored, the resultant force (A) that is generated in the vehicle as a whole can be treated as 0 N. When turning at a speed that does not allow lateral acceleration to be ignored, however, the force (B) will also be generated in each wheel according to the slip angle simultaneously with force (A). In order to continue turning at a constant speed at this time, a force (C) that is commensurate with force (B) will be needed.

Regarding force (B), when W is as the work rate for the vehicle as a whole, W_i is the work rate of each wheel, F_{d_i} is the resistance force defined by the coordinates for each wheel, R_i is the turning radius of each wheel, and r is the angular velocity at the vehicle center of gravity, then, as shown in Eq. (2), then W can be found from the summation of W_i .

$$\begin{aligned} W &= W_{fl} + W_{fr} + W_{rl} + W_{rr} \\ &= F_{d_{fl}} R_{fl} r + F_{d_{fr}} R_{fr} r + F_{d_{rl}} R_{rl} r + F_{d_{rr}} R_{rr} r \quad (2) \\ &= (F_{d_{fl}} R_{fl} + F_{d_{fr}} R_{fr} + F_{d_{rl}} R_{rl} + F_{d_{rr}} R_{rr}) r \end{aligned}$$

F_{d_i} will be in the following relationship, according to θ .

$$F_{d_i} = F_{y_i} \sin(\delta_i - \theta_g + \theta_i) + F_{x_i} \cos(\delta_i - \theta_g + \theta_i) \quad (3)$$

When F_{x_i} is the longitudinal force on each tire contact patch, F_{y_i} is the lateral force on each tire contact patch, δ_i is the wheel angle of each wheel, and θ_g is the slip angle of the vehicle center of gravity, then the slip angle θ of each tire is expressed by the relationship below.

$$\theta_i = \theta_g + l_i r + d_i r \quad (4)$$

The position vector for each wheel with respect to the center of gravity is (d_i , l_i). If the vehicle is turning to the left, then R_{min} corresponds to the lowest value of R_{fl} , and if the vehicle is turning to the right, then it corresponds to the lowest value of R_{fr} . The relational expressions Eqs. (2)-(4) will later become the conditions for determining R_{min} .

In cases where the influence of lateral acceleration cannot be ignored, a roll motion is generated in the vehicle and the state of the tire contact patch changes. The shift of load from the inside wheel to the outside wheel also affects the state of tire contact, so that the influence on those forces generated in the contact patch also needs to be taken into consideration. The influence of the load shifting from the inside wheel to the outside changes the force generated in the wheel contact patch.

The changes in F_x and F_y relative to the change in the moved load (F_z) are incorporated into Eq. (3) and the calculation is carried out.

4.4. Torque Around the King Pin Axis

Working from a conceptual approach like the above, the force generated in the contact patch is sought and on the basis of that force is converted into the torque around the king pin axis (M_{KIP}). F_{Rack} is generated by the M_{KIP} of the front left and right wheels. The M_{KIP} factors are classified under the following four headings.

- The factor of self-aligning torque that is generated in the tires and is converted to torque around the king pin axis
- The factor of lateral force that is generated in the tires and is converted to torque around the king pin axis
- The factor of vertical force that works on the tire-ground contact point and is converted to torque around the king pin axis
- The factor of force that is generated as spring reaction force by stabilizer or other such spring elements during steering and is converted to torque around the king pin axis

Items (a) and (b) can be converted into torque by the lateral force generated in the tire contact patch and by the moment arm length of the pneumatic trail portion and caster trail portion. When the pneumatic trail portion and caster trail portion of torque are designated M_{KIP_1} and M_{KIP_2} , respectively, then

$$\begin{aligned} M_{KIP_1} &= \Delta F_y \zeta_{pn} \cos \theta_{ch} \\ M_{KIP_2} &= \Delta F_y \zeta_{ct} \cos \theta_{ch} \end{aligned} \quad (5)$$

Here ζ_{pn} is the pneumatic trail, ζ_{ct} is the caster trail, ΔF_y is the lateral force generated while turning, and θ_{ch} is the caster angle. The values of ζ_{pn} and ΔF_y are mainly determined by the stand-alone tire characteristics, and the values of ζ_{ct} and θ_{ch} are mainly determined by the link geometry of the suspension and steering system. Figure 9 shows an overview of M_{KIP_1} and M_{KIP_2} .

With regard to item (c), the torque around the king pin axis resulting from the vertical force from the ground surface (M_{KIP_0}) can be found using Eq. (6).

$$M_{KIP_0} = F_z \zeta_z \sin \theta_z \quad (6)$$

F_z is the ground contact load, ζ_z is the distance between a normal line through the ground contact center and the king pin axis, and θ_z is the angle formed by a normal line through the ground contact center and the king pin axis. Therefore when the change in the ground contact load during a turn is ΔF_z , the torque around the king pin axis (M_{KIP_3}) according to item (c) is given by Eq. (7). Figure 10 shows an overview of M_{KIP_3} .

$$M_{KIP_3} = (F_z + \Delta F_z) \zeta_z \sin \theta_z \quad (7)$$

With regard to item (d), when elements that rotate around the king pin axis and spring elements are joined and when the vector of the deflection reaction force generated at the joint by spring elements does not intersect with the king pin, then the torque is that generated when the moment arm length is the distance between the vector of the deflection reaction force from the spring elements and the king pin axis. When this torque is designated M_{KIP_4} , then

$$M_{KIP_4} = \Delta F_s * \zeta_s * \sin \theta_s \quad (8)$$

Here ΔF_s is the reaction force of spring elements generated by suspension stroke changes or other such causes that occur while turning, ζ_s is the distance between the vector of the spring reaction force and the king pin axis, and θ_s is the angle formed by the vector of the spring reaction force and the king pin axis. Figure 11 shows an overview of M_{KIP_4} .

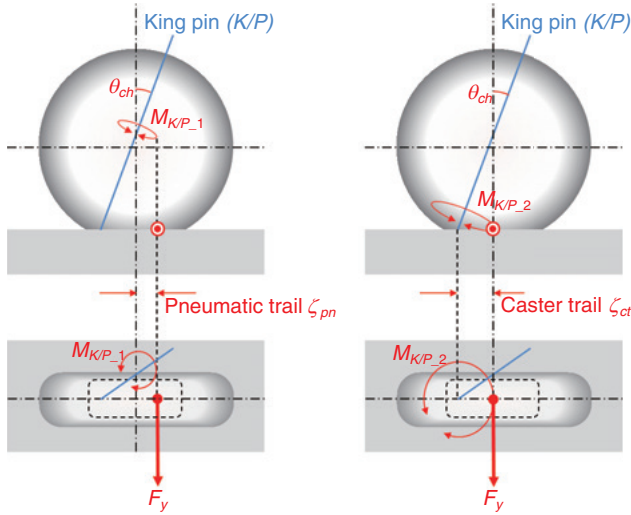


Fig. 9 Torque around king pin due to F_y

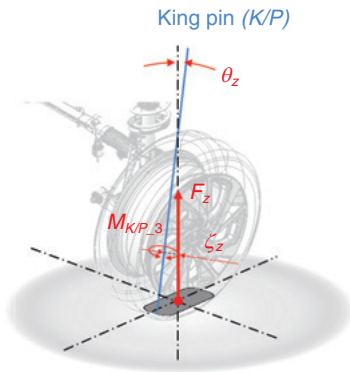


Fig. 10 Torque around king pin due to F_z

Figure 12 shows M_{KIP} changes with respect to S_{Rack} . The total of the torque around the king pin axis is M_{KIP} .

$$M_{KIP} = M_{KIP_1} + M_{KIP_2} + M_{KIP_3} + M_{KIP_4} \quad (9)$$

Through the link geometry, M_{KIP} becomes F_{Rack} .

4.5. Link Efficiency

Figure 13 shows a schematic representation of a steering system model. M_{KIP} reaches the rack through the link geometry of the steering system. The proportion of force transmitted changes according to changes in the angle of join in the tie rod link. When this happens, the relationship between the axial force on the link (F_{TR}), F_{Rack} , and M_{KIP} is described by Eq. (10).

$$\begin{aligned} F_{TR} &= M_{KIP} / (L_1 \cos \theta_a) \\ F_{Rack} &= F_{TR} \cos \theta_b \\ &= M_{KIP} \cos \theta_b / (L_1 \cos \theta_a) \end{aligned} \quad (10)$$

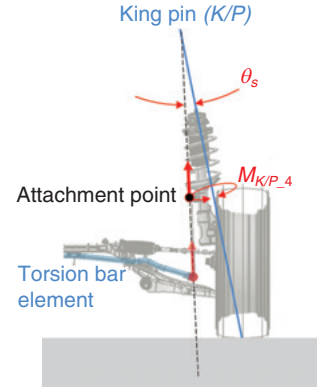


Fig. 11 Torque around king pin due to ΔF_s

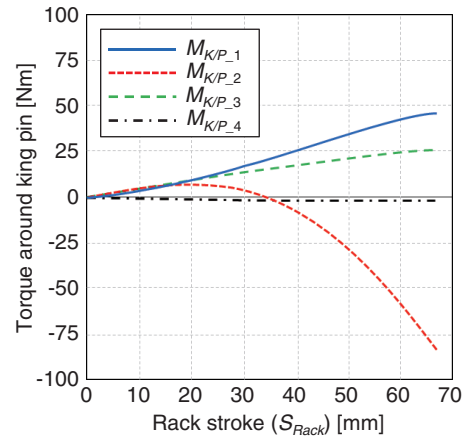


Fig. 12 Torque around king pin

In Fig. 13, θ_a and θ_b , are the angles of the join portion of the link, L_1 is the length of the knuckle arm, and $\cos\theta_b / (L_1 \cos\theta_a)$ is the link efficiency, a coefficient that determines the relationship of $M_{K/P}$ and F_{Rack} . The link efficiency of the inside and outside wheels displays a different change with respect to S_{Rack} .

As described above, the concepts of calculation of the force generated in the contact patch of each tire, calculation of $M_{K/P}$ by each factor, and calculation of the link efficiency were successfully incorporated into the model. Incorporating these concepts into the model enables description of the S_{Rack} - F_{Rack} characteristic from the results in Fig. 12 and the relationship in Eq. (10). As a result, it became possible to investigate the S_{Rack} - F_{Rack} characteristic from the high speed range to the extreme low speed range working only from the information available at the initial stages of development.

4.6. Prediction of Minimum Turning Radius

The created steering system model is capable of deriving the left and right wheel angle changes individually in order to predict the link efficiency, and as a result it can also simultaneously predict the Ackermann ratio. The turning radius of each wheel was derived by convergence calculation from the Ackermann ratio and S_{Rack} highest value as well as from the relationships in Eqs. (2)-(4). Then the turning radius of the outside front wheel was defined as R_{min} .

4.7. Prediction Results

Figure 14 shows a comparison of prediction results for turning radius and the measurement results for minimum turning radius. The measurement results have a margin because they include the measurement error. The prediction results have sufficient accuracy, and they confirmed that changes in the turning radius could simultaneously be predicted across vehicle speeds from 2 km/h to the vicinity of 10 km/h.

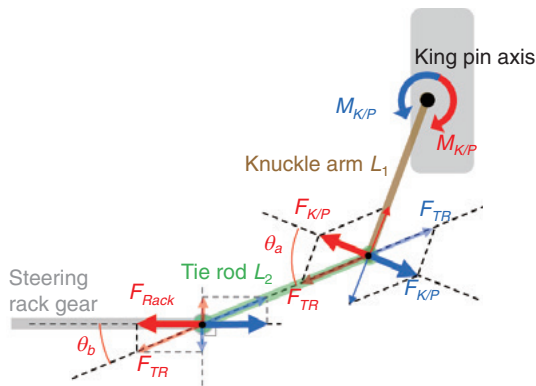


Fig. 13 Diagram of steering system model

Figure 15 shows changes in link efficiency with respect to S_{Rack} . Comparison was made with prediction results from a full vehicle model, and it was confirmed that the differing characteristics of the inside and outside wheels were reproduced satisfactorily.

Figure 16 shows the prediction results for the S_{Rack} - F_{Rack} characteristic. Comparison was made with prediction results from a full vehicle model, and it was confirmed that the F_{Rack} peak value, S_{Rack} value at peak F_{Rack} , changes in F_{Rack} value in the vicinity of the greatest S_{Rack} , and other such characteristics that are needed for investigation were largely successfully reproduced.

Figure 17(a) shows prediction results for very low speed range vehicle handling indices and Fig. 17(b) shows indices for vehicle dynamics characteristics predicted simultaneously while driving at a speed of 120 km/h. The legend indicates the different specifications for which

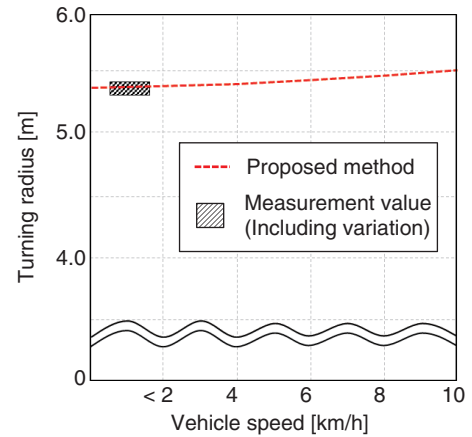


Fig. 14 Comparison of minimum turning radius

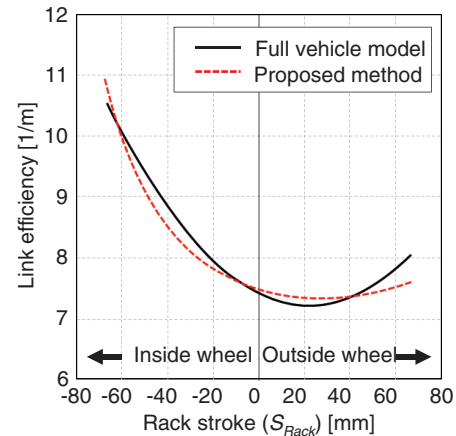


Fig. 15 Comparison of link efficiency

prediction results were obtained. It was confirmed from Fig. 17 that the investigation of specifications could proceed while comparing performance driving at the very low speed range and at high speed.

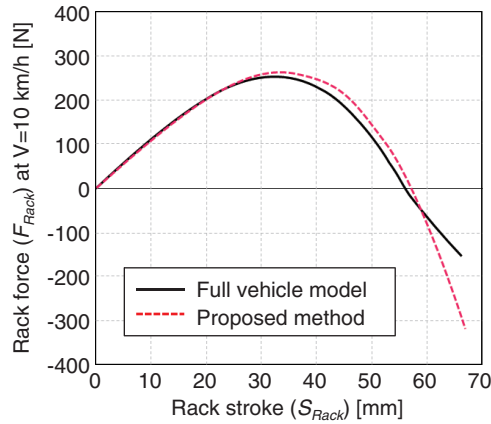


Fig. 16 Comparison of axial force of rack

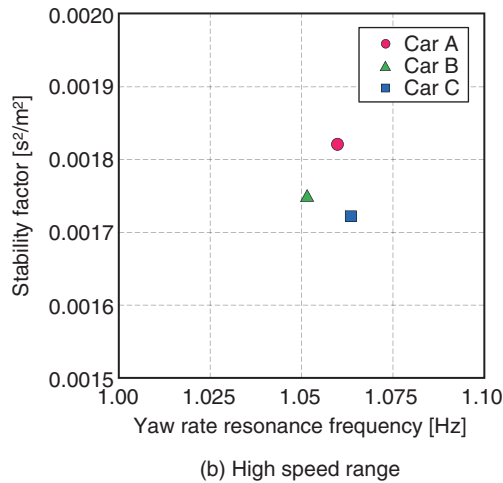
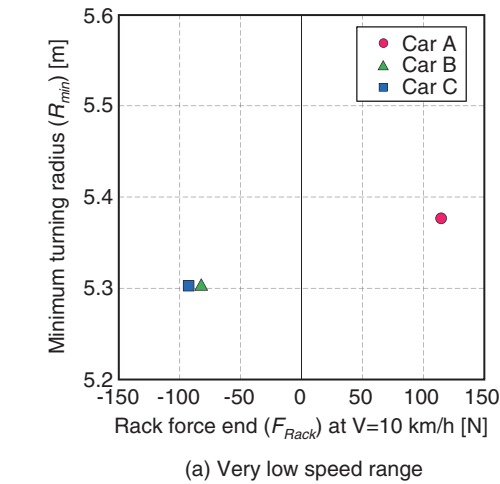


Fig. 17 Performance design including very low speed area

5. Cases of Application

Figure 18 shows examples of proposed countermeasures for chassis elements investigated on the desk after analyzing actual factors with regard to M_{Steer} . These examples show a breakdown of M_{KIP} , and the extent of influence that the specification changes for each chassis element part have on M_{Steer} is apparent. Given these analysis results, the investigation of specifications for chassis elements can be carried out efficiently. It is anticipated that the prediction results for F_{Rack} characteristic can be used not only in investigating the specifications for chassis elements, but also for examining the specifications for the control motors used to assist steering torque. This kind of investigation now provides the means to obtain a view of prospective chassis specifications early in the process. This technique was applied in the performance design of the chassis platform for compact vehicles that has been adopted from the 2014 model year FIT.

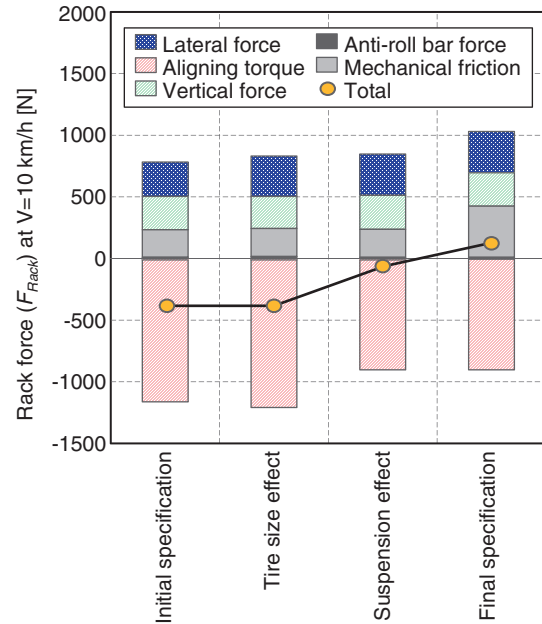


Fig. 18 Factor analysis calculation result

6. Conclusion

The simulation model used for performance design in the development of mass-production vehicles was taken as a base for creating a simulation technique that is capable of predicting indices for vehicle handling at the very low speed range together in bulk, where these were previously calculated separately by creating multiple models. The substance of this created technique is summarized in the following.

- (1) The elements that provide for investigation of very low speed range vehicle handling were incorporated into the conventional model and a model that is capable of seamless prediction from high speeds to the very low speed range was created.
- (2) It was confirmed that analysis of the contribution to very low speed vehicle handling by chassis elements could proceed simultaneously with investigation of the specifications for performance at other vehicle speed ranges.
- (3) This simulation technique was applied in development of the 2014 model year FIT, it was used in actual mass-production vehicle development, and the technique was confirmed to be useful.

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■ Author ■



Takayuki TOYOSHIMA



Yoshitomo AZEKATSU



Yoshiaki KOBAYASHI



Naofumi IKEDA



Hiroki TAKANO



Isao OBATA