

Proposal of New Steering Mechanism Enabling Ideal Ackermann Steering Characteristics

Shota YAMAZAKI¹

ABSTRACT

A new steering mechanism is proposed to achieve ideal Ackermann steering characteristics that conventional steering mechanisms cannot perfectly achieve, with zero slip angles of the inner and outer front wheels at large steering angles. The proposed mechanism applies an angular phase offset to the left and right tie rod joint points of a Pitman-arm-type steering system to asymmetrically control the tie rod strokes. This method was theoretically demonstrated to realize the steering angle difference required for ideal Ackermann steering characteristics at the target turning radius. Applicability to a wide range of specifications such as a wide range of wheelbases and tread widths was also confirmed.

1. Introduction

To mitigate tire wear, tire-squeal noise, and a lack of steering reaction torque experienced by the user when steering, Specifically, if an inner wheel arc smaller than that of the outer wheel and an alignment where the center of turning of the inner and outer wheels overlap⁽¹⁾ can be achieved as shown in Fig. 1, the lateral slip angle of the tires can be reduced to 0 deg. This kind of geometric relationship is named the ideal Ackermann steering geometry^{(2), (3)}.

In terms of actual steering mechanisms, a symmetric linkage-type mechanism (hereinafter, linkage type) has been widely used in combination with a recirculating ball-and-nut-type rotary mechanism since the mid-20th century. Later, the rack-and-pinion steering system (hereinafter, rack-and-pinion type) was practically applied, and both types are currently used according to the vehicle application⁽⁴⁾.

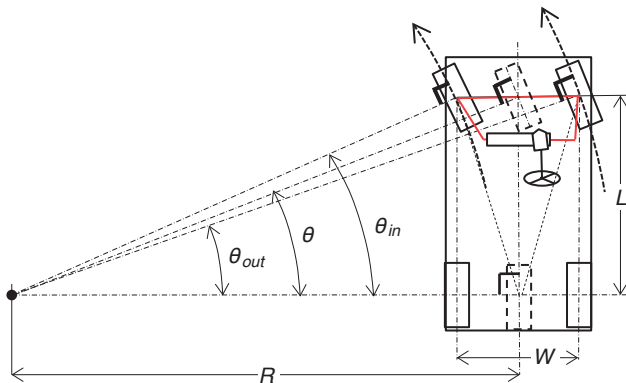


Fig. 1 Ackermann geometry

The linkage type has been widely adopted due to its high durability and low steering kickback from the road surface input⁽⁵⁾. With the introduction of power steering, however, the rack-and-pinion type, which provides a more direct feel in steering, is becoming more widespread⁽⁶⁾. Figure 2 shows a schematic image of the rack-and-pinion type.

Both linkage and rack-and-pinion types have the same fundamental challenge: as the left and right tie rods are connected by a single relay rod or rack, the left and right wheel movement is the same and an inner-outer wheel steering angle difference cannot be produced. To address this issue, current steering mechanisms have incorporated a geometric structure (hereinafter, geometry) comprising a four-bar closed linkage in an isosceles-trapezoid shape to turn the inner wheel more and the outer wheel less⁽⁷⁾. However, due to spatial constraints around the front wheel suspension, the degree of freedom of the joint point positions is low. As a result, there is the challenge that the steering angle difference corresponding to the ideal Ackermann steering geometry cannot be fully achieved.

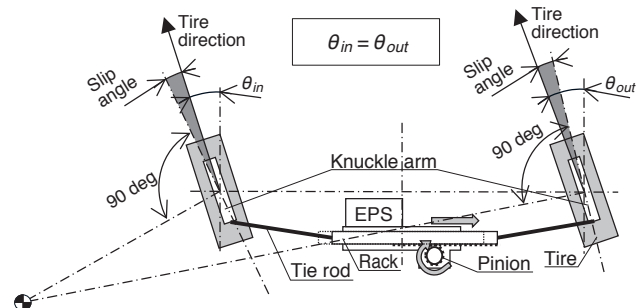


Fig. 2 Rack and pinion steering system

¹ Automobile R&D Center

To fundamentally address this challenge, it is effective to realize independent movement for the right and left wheels in the steering system.

This proposal addressed the challenge of achieving the ideal Ackermann steering geometry with an inner and outer wheel slip angle of 0 deg by introducing a structure that applies a phase offset to the rotary-to-translational linkage mechanism used in the steering mechanism. This enabled the design of a mechanism that can vary the left and right tie rod movement, and the effectiveness of this mechanism was confirmed.

2. Current Challenges

First, the inner-outer wheel steering angle difference of the ideal Ackermann steering geometry shown in Fig. 1 was obtained. The inner and outer wheel actual steering angles θ_{in} and θ_{out} are given by Eq. (1) and Eq. (2)⁽⁹⁾, respectively, using the turning radius R , wheelbase L , and tread width W ⁽⁸⁾. The ideal inner-outer wheel steering angle difference $\Delta\theta_{ideal}$ is expressed by Eq. (3) and is the target steering angle difference.

$$\tan\theta_{in} = \frac{L}{R - \frac{W}{2}} \quad (1)$$

$$\tan\theta_{out} = \frac{L}{R + \frac{W}{2}} \quad (2)$$

$$\Delta\theta_{ideal} = \theta_{in} - \theta_{out} \cong LW \cdot \frac{1}{R^2 - \frac{W^2}{4} + L^2} \quad (3)$$

Equation (3) shows that as the turning radius R decreases, the ideal steering angle difference $\Delta\theta_{ideal}$ needs to be increased to achieve the ideal Ackermann steering geometry.

Next, the inner-outer wheel steering angle difference of actually used steering mechanisms is considered.

Figure 3 shows the case with the steering gearbox located behind the front axle (hereinafter, steering gearbox behind the front axle) to accommodate the transverse engines used in front-wheel-drive passenger cars.

With the steering gearbox behind the front axle, the inner-outer wheel steering angle difference $\Delta\theta_{actual}$ is given

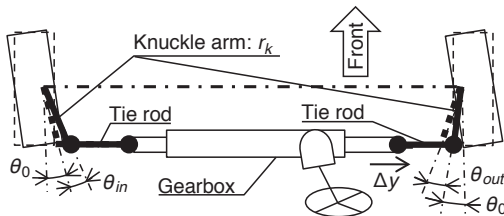


Fig. 3 Schematic image of steering system behind front axle

by Eq. (4) using the inner wheel steering angle θ_{in} , outer wheel steering angle θ_{out} , and knuckle arm angle θ_0 .

$$\begin{aligned} \Delta\theta_{actual} &= \theta_{in} - \theta_{out} \\ &= \theta_{in} - \theta_0 + \arcsin\{2\sin\theta_0 - \sin(\theta_0 + \theta_{in})\} \end{aligned} \quad (4)$$

Next, Fig. 4 shows the case with the steering gearbox located ahead of the front axle (hereinafter, steering gearbox ahead of the front axle) to accommodate the longitudinal mounted engines of rear-wheel-drive passenger cars.

With the steering gearbox ahead of the front axle, parallel steering with a knuckle arm angle θ_0 of 0 deg⁽¹⁰⁾ is widely used due to spatial constraints, so the inner-outer wheel steering angle difference $\Delta\theta_{actual}$ becomes as shown in Eq. (5).

$$\Delta\theta_{actual} = \theta_{in} - \theta_{out} = 0 \quad (5)$$

Figure 5 shows a comparison between the changes of the outer wheel steering angle θ_{out} relative to the inner wheel steering angle θ_{in} for steering gearbox ahead of the front axle and behind the front axle with ideal Ackermann steering characteristics. The Ackermann ratio κ_A , which indicates the achievement rate of the actual steering angle difference $\Delta\theta_{actual}$ relative to the steering angle difference

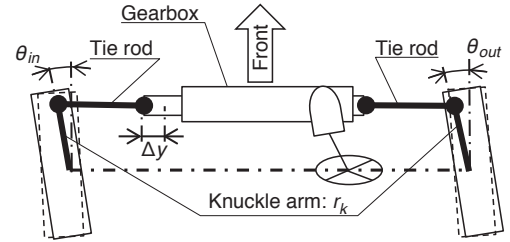


Fig. 4 Schematic image of steering system ahead of front axle

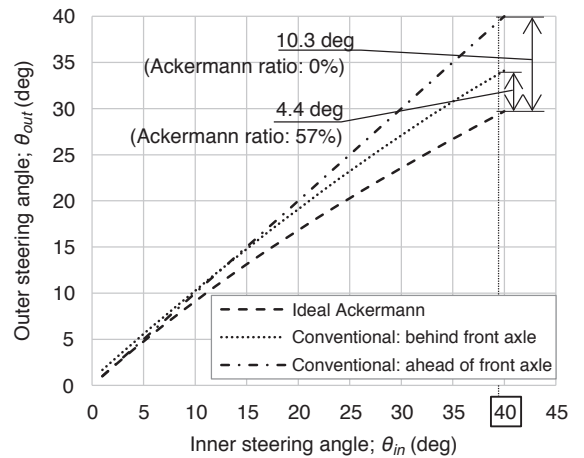


Fig. 5 Comparison of Ackermann ratio of conventional steering systems with ideal Ackermann characteristics

Table 1 Parameters for C-category passenger sedan

Wheelbase	L (m)...	2.735
Tread	W (m)...	1.535
Knuckle arm length	r_k (m)...	0.142
Knuckle arm angle	θ_0 (deg)...	13.0

$\Delta\theta_{ideal}$, is defined as $\kappa_A = 100 \times \Delta\theta_{actual} / \Delta\theta_{ideal}^{(11)}$. These calculations used the compact vehicle specifications listed in Table 1.

In the figure, the inner wheel steering angle is plotted up to 40 deg, corresponding to a turning radius of 4.0 m. In the case of the steering gearbox behind the front axle indicated by the dotted line, the larger the inner wheel steering angle, the more the outer wheel steering angle deviates from ideal Ackermann steering characteristics. If the deviation of this angle is defined as $\theta_{Err} = \Delta\theta_{ideal} - \Delta\theta_{actual}$, then at an inner wheel steering angle of 40 deg, the steering angle deviation $\theta_{Err} = 4.4$ deg and the Ackermann ratio at this time is 57%.

Similarly, in the case with the steering gearbox ahead of the front axle, $\theta_{Err} = 10.3$ deg and the Ackermann ratio becomes 0%.

Assuming, for example, that the average cornering stiffness of 235/40R18 size tires is 1400 N/deg, this steering angle difference produces opposing inward tire forces that do not contribute to turning, the left and right sum total of which is 6160 N for steering gearbox behind the front axle or 14420 N for steering gearbox ahead of the front axle. This results in reduced steering reaction torque as well as tire wear and tire-squeal noise⁽¹²⁾.

The above clarified that currently widespread steering mechanisms cannot provide a sufficient inner-outer wheel steering angle difference, with the result that excess tire lateral force is generated and impacts steering performance. To address these challenges, a new steering mechanism is needed that can independently realize this inner-outer wheel steering angle difference.

3. Achievement Methodology

3.1. Proposed Mechanism

A new linkage mechanism based on the widely used linkage type has been proposed to freely achieve the inner-outer wheel steering angle difference, which is challenging with current steering mechanisms. This mechanism applies an angular phase offset to the left and right tie rod joint points on the Pitman arm⁽¹³⁾ to asymmetrically control the left and right tie rod strokes. Figure 6 shows an overview of this mechanism when applied to the geometry with the steering gearbox ahead of the front axle that is widely adopted such as in current off-road SUVs.

Through this new linkage mechanism, the Pitman arm moves the relay rod and tie rods (hereinafter, tie rod, etc.) in the tread direction according to the rotation angle, that movement being specified by a sine function. By properly

designing the joint positions of the linkage mechanism in consideration of this relationship, an arbitrary left-right difference can be produced in the movement.

3.2. Basic Principle

Figure 7 shows the mechanism by which the proposed mechanism produces a left-right difference in the movement. Here, the Pitman arm length is r , and the phase angle applied to the linkage joint position is α . In addition, to achieve an equal left-right difference in the movement when turning either left or right, an idler arm is also positioned and the relay rod and tie rod lengths are adjusted accordingly.

In the proposed linkage mechanism, the inner end positions of the left and right tie rods are line symmetric, which has the effect of producing an equal inner-outer wheel steering angle difference when steering to either the left or right.

The left and right tie rod, etc. movements Δy_{in} and Δy_{out} relative to the rotation angle δ of the sector shaft transmitted from the steering wheel are relatively expressed by Eq. (6) and Eq. (7) using the inner and outer wheel actual steering angles θ_{in} and θ_{out} and the phase angle α .

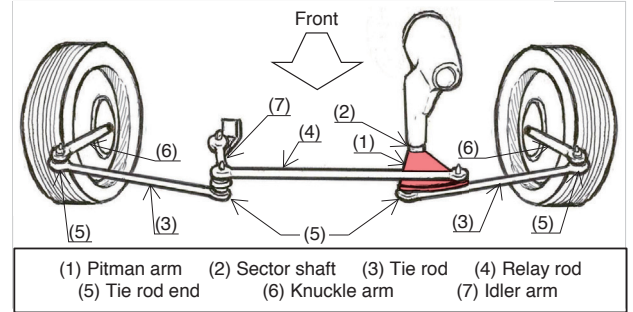


Fig. 6 Front view of steering system with proposed pitman arm

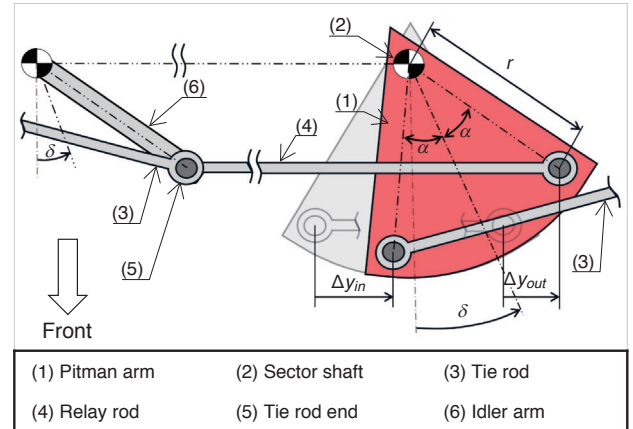


Fig. 7 Mechanism for producing asymmetric tie rod stroke using angled pitman arm system

$$\begin{aligned}\Delta y_{in} &= r_k \sin \theta_{in} \\ &= r \{ \sin \alpha - \sin(\alpha - \delta) \}\end{aligned}\quad (6)$$

$$\begin{aligned}\Delta y_{out} &= r_k \sin \theta_{out} \\ &= r \{ \sin(\alpha + \delta) - \sin \alpha \}\end{aligned}\quad (7)$$

By using a trigonometric addition formula to expand these equations, the inner-outer wheel steering angle difference for the proposed mechanism $\Delta\theta_{proposed}$ can be calculated by Eq. (8). Here, since α and δ are small enough for practical purposes, the approximations $\sin\theta \doteq \theta$ and $\cos\delta \doteq 1 - \delta^2/2$ are used.

$$\begin{aligned}\Delta\theta_{proposed} &= \theta_{in} - \theta_{out} \\ &= \frac{r}{r_k} \{ 2\sin\alpha(1 - \cos\delta) \} \\ &\cong 2 \frac{r}{r_k} \cdot \alpha \cdot \frac{\delta^2}{2} \\ &= \frac{L^2 r_k \alpha}{r} \cdot \frac{1}{R^2}\end{aligned}\quad (8)$$

$$\begin{aligned}\text{where } \sin\alpha &\cong \alpha, \quad \cos\delta \cong 1 - \frac{\delta^2}{2} \\ r\sin\delta &= r_k \sin\theta \Rightarrow \delta \cong \frac{r_k}{r} \theta \\ \theta &= \tan^{-1} \frac{L}{R} \cong \frac{L}{R}\end{aligned}$$

From the above, the inner-outer wheel steering angle difference for the proposed mechanism $\Delta\theta_{proposed}$ is derived based on the vehicle specifications and parameters (α , r) and the turning radius R .

3.3. Method of Deriving the Phase Angle

The appropriate α and r for Eq. (8) are determined based on Eq. (3), which expresses ideal Ackermann steering characteristics. Equation (8) depends on the turning radius R , so the method of performing interval integration with the turning radius R and obtaining the parameter α that results in the smallest area can be used to derive the optimal parameters for the target turning radius or turning radius range.

In Fig. 8, the horizontal axis is R , the vertical axis is the left-right wheel steering angle difference $\Delta\theta$, the

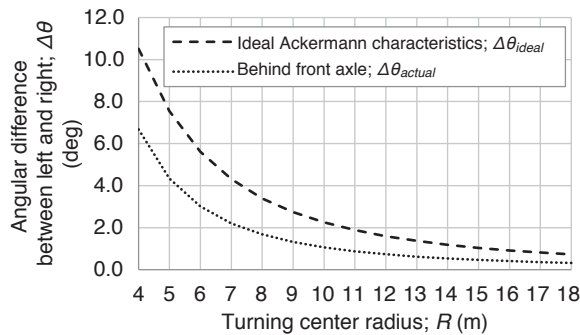


Fig. 8 Schematic image of characteristics of wheel angle difference $\Delta\theta$ vs. turning center radius R

steering angle difference $\Delta\theta_{ideal}$ of ideal Ackermann steering characteristics is shown with a dashed line, and the steering angle difference $\Delta\theta_{actual}$ for the steering gearbox behind the front axle is shown with a dotted line. This figure shows that as the turning radius R decreases, the ideal steering angle difference $\Delta\theta_{ideal}$ increases, but at the typical $\Delta\theta_{actual}$, the deviation from ideal Ackermann steering characteristics increases.

Therefore, a method of setting the phase angle α and length r of the proposed Pitman arm to bring $\Delta\theta_{proposed}$ closer to $\Delta\theta_{ideal}$ is described. As an example, r was set to 0.1 m based on the packaging constraints, and α was considered as a design degree of freedom. Figure 9 shows the relationship between $\Delta\theta_{proposed}$ and $\Delta\theta_{ideal}$ and their difference e .

To enable ideal Ackermann steering characteristics with any turning radius R , it is sufficient to determine the parameter f so that the solid and dashed lines in Fig. 9 overlap at the turning radius R . That is to say, a steering angle difference that enables ideal Ackermann steering characteristics can be obtained by calculating the optimal α from Eq. (9) so that the difference e is 0 deg at the given R .

Assuming the Pitman arm length r , turning radius R , tread width W , wheelbase L , and knuckle arm length r_k , the equation for obtaining α is derived as shown in Eq. (10).

$$e(\alpha, r, R) = \frac{4LW}{4R^2 - W^2 + 4L^2} - L^2 r_k \frac{\alpha}{r} \cdot \frac{1}{R^2} = 0 \quad (9)$$

$$\alpha = \frac{4R^2 W r}{(4R^2 - W^2 + 4L^2) L r_k} \quad (10)$$

This method of enabling Ackermann steering characteristics for any turning radius is called point optimization.

Additionally, to approach as close as possible on average to ideal Ackermann steering characteristics over the target turning radius range R_1 to R_2 , the parameters that minimize E^2 , which is the integrated value of the squared difference between the solid and dashed lines, can be determined using Eqs. (11) and (12).

$$e^2(\alpha, r, R) = \left(\frac{4LW}{4R^2 - W^2 + 4L^2} - L^2 r_k \frac{\alpha}{r} \cdot \frac{1}{R^2} \right)^2 \quad (11)$$

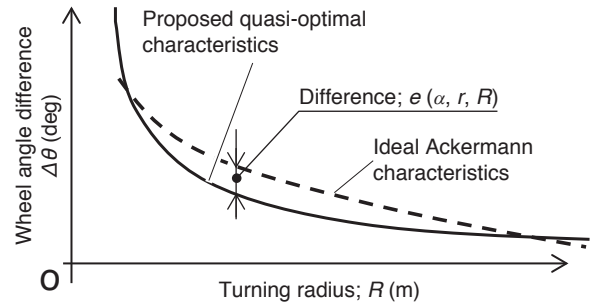


Fig. 9 Schematic image of target difference

$$E^2(\alpha, r, R) = \int_{R_2}^{R_1} e^2 dR \quad (12)$$

Figure 10 shows a schematic image of this optimization method.

Therefore, the condition for minimizing E^2 is given by Eq. (13).

$$\frac{d}{d(\alpha/r)} E^2(\alpha, r, R) = 0 \quad (13)$$

By solving this, the α that results in the smallest difference is derived by Eq. (14).

$$\alpha = \frac{3BR_1^2 R_2^2 r \{A(R_1 - R_2) - R_1 R_2 (\tan^{-1} \frac{R_1}{A} - \tan^{-1} \frac{R_2}{A})\}}{A^3 C (R_1^3 - R_2^3)} \quad (14)$$

$$\text{where } A^2 = -\frac{W^2}{4} + L^2, \quad B = LW, \quad C = L^2 r_k$$

This method of enabling Ackermann steering characteristics on average over a specific turning radius range is called range optimization.

This mechanism changes the left and right tie rod, etc. movements by setting the phase angle α and length r of the Pitman arm as described above to artificially enable Ackermann steering characteristics. It was therefore decided to name it an angled Pitman arm (APA) mechanism.

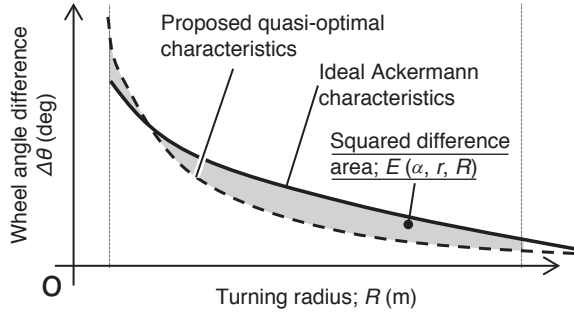


Fig. 10 Schematic image of squared difference to the target characteristics

4. Results

4.1. Verification of Effect in Small-radius Turning

To verify that the proposed mechanism enables Ackermann steering characteristics, Fig. 11 compares ideal Ackermann steering characteristics with the characteristics provided by the proposed method with point optimization for adequate low-speed maneuverability at $R = 4$ m. The characteristics for steering gearbox behind the front axle and ahead of the front axle are also plotted to confirm the degree of enhancement over the conventional mechanisms.

The point-optimized characteristics shown by the solid

line are close to ideal Ackermann steering characteristics shown by the dashed line, and at an inner wheel steering angle of 40 deg for $R = 4$ m, the two curves match and the Ackermann ratio is 100%, confirming the effectiveness of point optimization for small-radius turning ($R = 4$ m).

4.2. Confirmation of Effect of Optimization Method

To show that the proposed optimization method is effective under various conditions, Fig. 12 shows the results of point optimization for turning radii of 4 m and 6 m and the results of range optimization for a turning radius range of 4 m to 10 m.

This figure shows the characteristics provided by the proposed method with solid lines, ideal Ackermann steering characteristics with dashed lines, and the characteristics of the conventional mechanism with dotted lines.

This shows that in all three cases, the proposed method achieves characteristics closer to ideal Ackermann steering characteristics than those of the conventional mechanism.

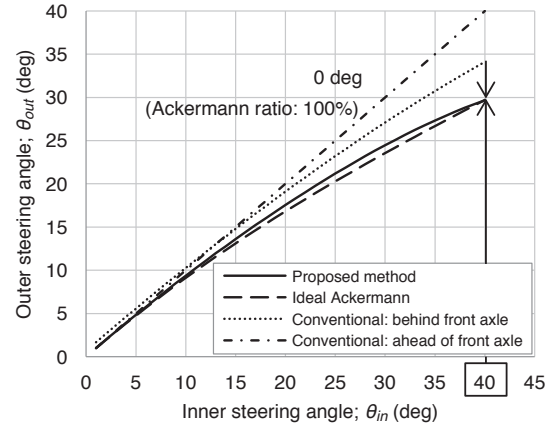


Fig. 11 Comparison of Ackermann characteristics of proposed - method with ideal Ackermann characteristics

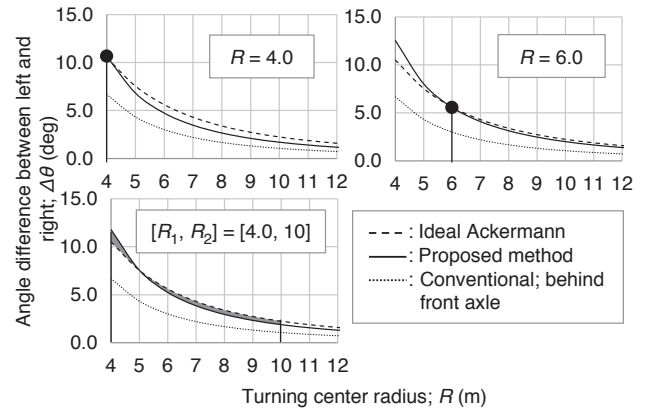


Fig. 12 Area of squared difference between functions each of ratio a/r in 3 cases of target turning radius

The above confirmed that use of the proposed mechanism and its parameter setting method enables control of the inner-outer wheel steering angle difference close to the ideal, both when setting the turning radius to a pinpoint value and targeting average optimization over a specific turning radius range.

4.3. Confirmation of Effect with Arbitrary Vehicle Specifications

To confirm that the proposed model is effective for models with various specifications, a case study of point optimization for a turning radius of 4 m was conducted to confirm whether ideal Ackermann steering characteristics can be achieved.

Table 2 shows the specifications and optimized parameters for the vehicles considered, from compact vehicles to large SUVs, for steering gearbox behind the front axle. In Fig. 13, the actual steering angle characteristics of the inner and outer wheels when the APA mechanism and its optimization method are applied to each specification are shown with solid lines, ideal Ackermann steering characteristics are shown with dashed lines, and the characteristics of the conventional mechanism are shown with dotted lines.

From Fig. 13, it can be confirmed that the steering angle difference obtained by the APA mechanism is closer to ideal Ackermann steering characteristics than the current steering gearbox behind the front axle arrangement used in various vehicle specifications. This confirmed that the proposed method can be applied in typical vehicle specifications ranging from compact vehicles to large SUVs.

4.4. Evaluation of Deviation

Since α and δ are sufficiently small, the proposed

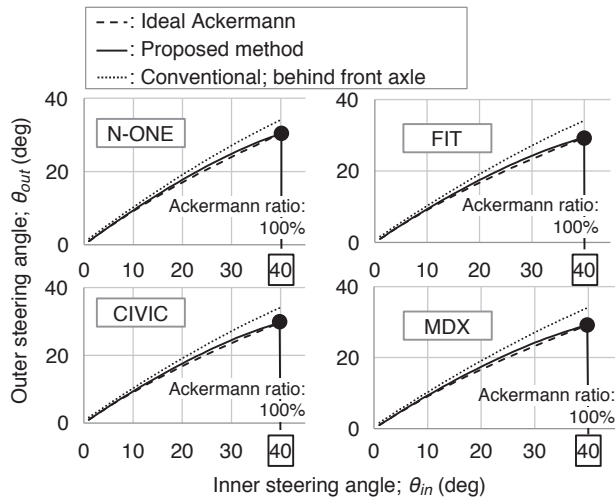


Fig. 13 Transition of inner and outer steering angle and best a/r ratio in each model category (Target $\theta_{in} = 40$ (deg))

Table 2 Specifications from 4 models, including parameters α and r achieving Ackermann ratios of 100%

Model	N-ONE	FIT	CIVIC	MDX
L (m)	2.520	2.530	2.735	2.891
W (m)	1.305	1.484	1.535	1.720
r_k (m)	0.142			
r (m)	0.10			
	↕			
α (deg)	14.3	16.6	16.0	16.6
Ackermann ratio (%)	100	100	100	100

method derives the optimization approach using a trigonometric approximation formula. The deviation from the exact solution was therefore compared to confirm the rationality of the approximations in the proposed method.

This exact solution can be calculated using geometry in a Cartesian coordinate system.

First, the coordinates of each joint point are set for the tread width direction and wheelbase direction components. The Cartesian coordinates of the virtual kingpin axis O of the front wheel suspension, the joint point S between the knuckle and the tie rod, etc., and the linkage joint point G between the tie rod, etc. and the APA mechanism are defined in Fig. 14, assuming the inner side of a left turn.

The initial positions of each point are denoted S_0 and G_0 , the respective positions after movement by the rotation angle δ are S and G , the lengths of the knuckle arm and the tie rod, etc. are respectively r_k and r_j , and the actual steering angles of the inner and outer wheels are respectively θ_{in} and θ_{out} . The outer side of the turn is defined symmetrically left-right relative to the body center line via the relay rod.

The coordinates of each point before APA rotation are expressed by the phase angles α , r_k , and r_j , and the relational expression between δ and θ_{in} , θ_{out} can be derived by expressing in terms of the coordinates when the APA is rotated by the angle δ . This allows obtaining the exact solution of the inner-outer wheel steering angle difference $\Delta\theta_{exact}$.

First, if the coordinates of points S and G after applying the rotation angle δ to the APA mechanism are obtained, they correspond to the coordinates of points S and G in

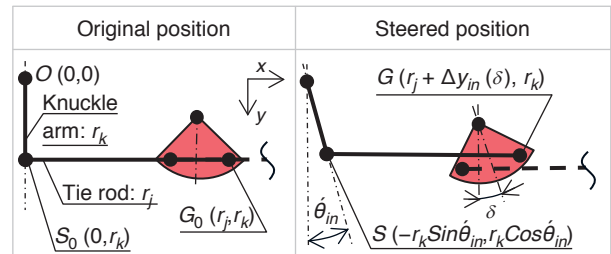


Fig. 14 Position of using points in Descartes coordinates

Fig. 14. From this, the inner and outer wheel side tie rod movements Δy_{in} and Δy_{out} , without using approximation, are relatively given by Eqs. (15) and (16).

$$\Delta y_{in}(\delta) = -r \sin(\alpha) + r \sin(\delta - \alpha) \quad (15)$$

$$\Delta y_{out}(\delta) = -r \sin(\delta + \alpha) + r \sin(\alpha) \quad (16)$$

Here, since the r_k between points S and G does not change even after movement, Eqs. (17) and (18) can be expressed using the Pythagorean theorem. For simplification of the equations, the inner and outer wheel point G coordinates are relatively expressed as $G_{in}(X_i(\delta), Y_i(\delta))$ and $G_{out}(X_o(\delta), Y_o(\delta))$.

$$(X_i(\delta) + r_k \sin \theta_{in})^2 + (Y_i(\delta) - r_k \cos \theta_{in})^2 = r_j^2 \quad (17)$$

$$(X_o(\delta) + r_k \sin \theta_{out})^2 + (Y_o(\delta) - r_k \cos \theta_{out})^2 = r_j^2 \quad (18)$$

Trigonometric synthesis is applied to these equations to obtain Eqs. (19), (20), and (21).

$$\theta_{in} = \sin^{-1} \left(\frac{X_i(\delta)^2 + Y_i(\delta)^2 + r_k^2 - r_j^2}{2r_k \sqrt{X_i(\delta)^2 + Y_i(\delta)^2}} \right) + \tan^{-1} \frac{Y_i(\delta)}{X_i(\delta)} \quad (19)$$

$$\theta_{out} = \sin^{-1} \left(\frac{X_o(\delta)^2 + Y_o(\delta)^2 + r_k^2 - r_j^2}{2r_k \sqrt{X_o(\delta)^2 + Y_o(\delta)^2}} \right) + \tan^{-1} \frac{Y_o(\delta)}{X_o(\delta)} \quad (20)$$

$$\Delta \theta_{exact} = \theta_{in} - \theta_{out} \quad (21)$$

Based on the above, Fig. 15 shows the result of obtaining the exact inner wheel steering angle θ_{in} and outer wheel steering angle θ_{out} from the APA rotation angle δ , and

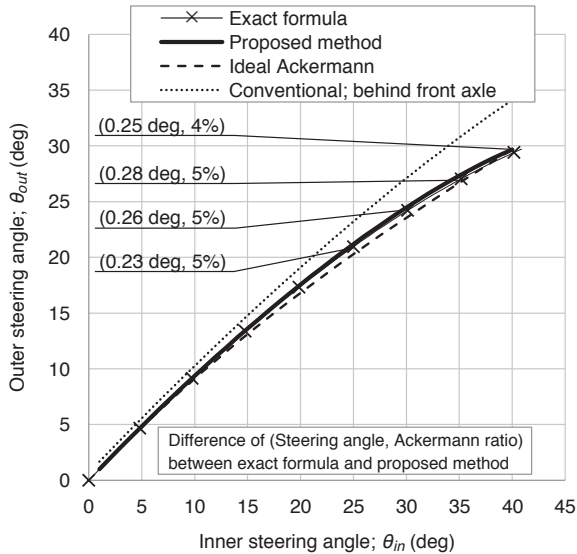


Fig. 15 Comparison of Ackermann characteristics from exact formula method with both proposed quasi-optimal method and ideal characteristics

calculating the inner-outer wheel steering angle difference $\Delta \theta_{exact}$.

In the figure, $\Delta \theta_{proposed}$ obtained by the proposed method using approximation is shown by the solid line, ideal Ackermann steering characteristics are shown by the dashed line, and the characteristics of the conventional steering gearbox behind the front axle are shown by the dotted line.

Compared to the conventional mechanism, the proposed method using point optimization for an inner wheel steering angle of 40 deg is able to retain the deviation to a slip angle of 0.28 deg or less. This indicates that the proposed method using approximation has high approximation accuracy.

Although the method of calculating the exact solution has high accuracy, it has the characteristic of being unable to perform the algebraic integration needed for range optimization. This means that it is not easy to apply, making it difficult to use for early-stage performance design in vehicle development. As the proposed method can theoretically set an Ackermann ratio of 100% while keeping the deviation small, it can be considered a simple method that provides adequate results.

4.5. Evaluation of Versatility for All Turning Radii

Having confirmed the effect of the proposed mechanism, lastly the feasibility under the current spatial constraints was confirmed. Here, assuming $r = 0.1$ m, each α corresponding to the range from a sufficiently small turning radius of $R = 4$ m to $R = \infty$ m, which indicates straight running, was calculated and the largest and smallest possible angles were confirmed.

Figure 16 shows the appropriate phase angle α for the turning radius R calculated using the point optimization Eq. (10).

This shows that for the proposed method to enable ideal Ackermann steering characteristics on a turning radius of $R = 4$ m, α needs to be 15.8 deg, and the calculated α converges to 22.6 deg as the turning radius R increases. By using the point optimization Eq. (10) to derive α with a turning radius of $R = \infty$, it can be shown algebraically that it converges as in Eq. (22).

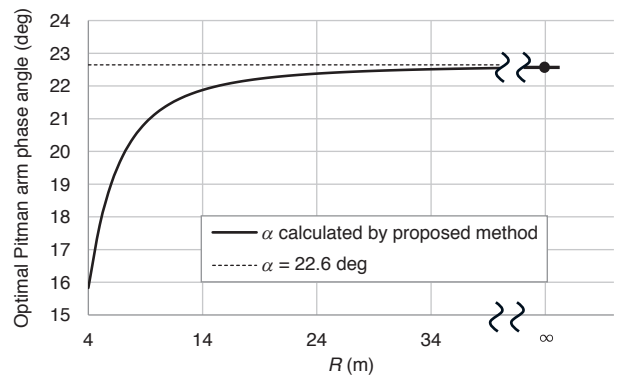


Fig. 16 Optimal pitman arm phase angle "α" vs. target "R"

$$\alpha|_{(R \rightarrow \infty)} = \frac{0.42}{1.1} \cdot \frac{R^2}{R^2 + 6.9}|_{(R \rightarrow \infty)} = 22.6(\text{deg}) \quad (22)$$

These results show that by setting an α range of 15.8 to 22.6 deg, ideal Ackermann steering characteristics can be approximately enabled from the smallest turning radius to straight running, confirming that the proposed mechanism can accommodate various specifications and conditions within the spatial constraints.

This confirmed that ideal Ackermann steering characteristics can be enabled by using the proposed new steering mechanism to achieve independent movement for the left and right wheels.

5. Conclusion

This report proposed an angled Pitman arm mechanism that enables different left and right tie rod strokes to achieve Ackermann steering characteristics at the target turning radius. The obtained insights are summarized below.

- (1) A new steering mechanism was proposed that enables ideal Ackermann steering characteristics for various vehicle specifications.
- (2) A relational expression was derived for the inner-outer wheel steering angle difference produced by the knuckle arms connected to the tie rods and relay rod at various positions, and an optimization method was established that achieves zero slip angles of the inner and outer wheels during turning.
- (3) The proposed mechanism was theoretically demonstrated to achieve an Ackermann ratio of 100% for specifications ranging from small passenger cars to large SUVs and over a wide range of turning radii.

Application of the proposed method to a vehicle enables the realization of optimal front-wheel slip angles during turning, and can be expected to have effects such as enhancing steering reaction torque and helping to reduce tire wear and tire-squeal noise.

References

- (1) Mitchell, W. C., Staniforth, A., Scott, I.: Analysis of Ackermann Steering Geometry, SAE Technical Paper 2006-01-3638, (2006), doi: 10.4271/2006-01-3638
- (2) Suzuki, Y., Sato, T., Tachiiri, M., Takahashi, T., Kuramitsu, S., Hijikata, M.: Four-Wheel Independent Steering Control That Extends Ackerman Theory, DENSO TECHNICAL REVIEW, Vol. 28, (2023) (in Japanese)
- (3) Abe, M.: Automotive Vehicle Dynamics: Theory and Applications, 2nd ed., Tokyo Denki University Press, p. 64-69, (2012) (in Japanese)
- (4) Matsuoka, H.: Development and Future Outlook of Steering Systems, JTEKT ENGINEERING JOURNAL, No. 1013, p. 10-15, (2015) (in Japanese)
- (5) Hashimoto, T.: Reduction of Kickback by Attaching

a Check Valve to Hydraulic Power Steering, KOYO Engineering Journal, No. 157, p. 66-70, (2000) (in Japanese)

- (6) Kumari, N. L., Faraz, M. S. A., Syed, F. H.: Structural Modelling And Stress Analysis Of Rack And Pinion Steering Mechanism, IOSR Journal of Mechanical and Civil Engineering, Vol. 22, Issue 2, p. 15-25, (2025)
- (7) Bose, V. C., Anand, P., Kumar, S. B., Darshan, C., Hariprasanna, C. T.: Design and Implementation of Steering System for All Terrain Vehicle, International Journal of Research in Engineering, Science and Management, Vol. 3, Issue 6, p. 96-98, (2020)
- (8) JIS D 0102:1996, Road vehicles -- Dimensions, masses, weights and performances -- Vocabulary
- (9) Vogel, J.: Tech Explained: Ackermann Steering Geometry, Racecar Engineering, <https://www.racecar-engineering.com/articles/tech-explained-ackermann-steering-geometry/>, (accessed 2026/4/23)
- (10) Deshmukh, A., Tawhare, G., Kochat, S.: Anti-Ackermann Steering System of Formula Student car, International Research Journal of Engineering and Technology (IRJET), Vol. 8, Issue 05, p. 3540-3549, (2021), doi: 10.13140/RG.2.2.27730.48320
- (11) Ozaki, A.: An Analysis of Influencing Factors on Stability Factor of 4-Wheel Vehicle by Planar Dynamic Equation with CG Height of Body, Proceedings of JSAE Annual Congress, No. 8-14, p. 13-18, (2014) (in Japanese)
- (12) CHMODKINN: Ackermann Steering Geometry: Principles, Correction & Performance Tuning, Online Article, <https://www.chmodkinn.com/ackermann-steering-geometry-principles-correction-performance-tuning/>, (accessed 2026/4/23)
- (13) JASO Z213: 2019, Glossary of terms relating to relating to steering systems of automobiles

■ Author ■



Shota YAMAZAKI