The Evolving Scholar Vol. 3, 2023 https://dx.doi.org/10.59490/6506799f4a8ec6d8f1c40fd6

Bicycle and Motorcycle Dynamics 2023
Symposium on the Dynamics and Control of Single Track Vehicles
18 – 20 October 2023, Delft University of Technology, The Netherlands

Type of the Paper: Conference Paper

A Study on Suspension Geometry for Personal Mobility Vehicles (PMVs) with Inward Tilt Mechanism

Tetsunori Haraguchi[1,2,*], and Tetsuya Kaneko[3]

[1]Institutes of Innovation for Future Society, Nagoya University; haraguchi@nagoya-u.jp; ORCID 0000-0003-4978-4005
[2]College of Industrial Technology, Nihon University; haraguchi.tetsunori@nihon-u.ac.jp
[3]Department of Mechanical Engineering for Transportation, Osaka Sangyo University; kaneko@ge.osaka-sandai.ac.jp

*corresponding author

Name of Editor: Jason Moore
Submitted: 17/09/2023
Accepted: 18/09/2023
Published: 19/09/2023
Citation: Haraguchi, T. & Kaneko, T. (2023). A Study on Suspension Geometry for Personal Mobility Vehicles (PMVs) with Inward Tilt Mechanism. The Evolving Scholar - BMD 2023, 5th Edition.

Abstract:

Since the concept of a Personal Mobility Vehicle (PMV) that tilts inward while turning is relatively new, there is currently a lack of theoretical considerations regarding the suspension mechanism. Therefore, this study aims to explore the theoretical relationship between suspension geometry and the pitching posture during turning in a PMV with two front wheels and one rear wheel that tilts inward during turns. Our findings suggest that a combination of a front telescopic suspension and a rear full trailing arm (swing arm) suspension is suitable for minimizing both the squatting pitching of the vehicle body during turns and the disturbances caused by changes in tread and tire camber angles during wheel strokes in the upright driving position from a static force balance perspective. From a dynamic perspective, there is no significant concern about pitching occurring even in cases where there may be a delay in active tilt angle tracking (PID) control when using the combination of front telescopic suspension and rear full trailing arm suspension. However, it is essential to note that a large sprung roll moment of inertia can still induce the squatting pitching.

Keywords: Personal Mobility, Suspension Geometry, Pitching Posture, Theoretical Considerations

ISSN: 2667-2812

1. Introduction

1.1. Background and Motivation

Personal Mobility Vehicles (PMVs) have a long history as small and highly efficient vehicles. In recent years, a completely new narrow PMV concept has been proposed [(1)-(4)], which leans inward when turning like a motorcycle to avoid overturning. However, due to the novelty of PMVs that lean inward during turns, there is still a limited body of prior research on such PMVs. The authors have conducted continuous studies on the social acceptability of PMVs (Figure 1), and the design of PMVs with two front wheels and one rear wheel, which can actively tilt inward according to the steering wheel angle [(3)-(20)]. Among them, recent efforts have been directed towards theoretical considerations of front-wheel steering axle configurations [(15)(16)(17)(19)] and suspension geometry settings [(18)(20)].

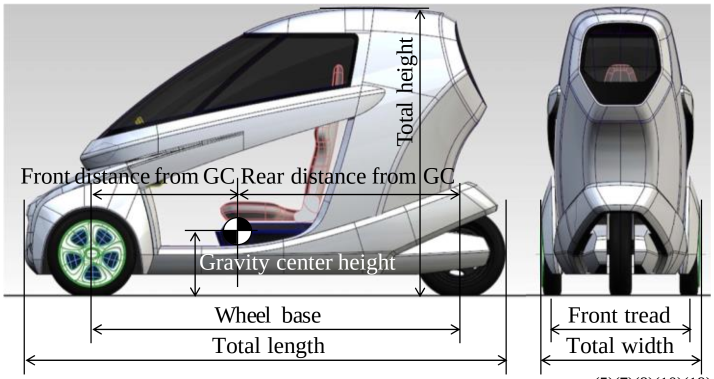
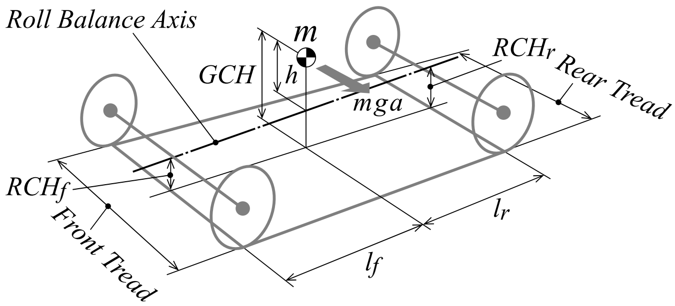


Figure 1. PMV with two front wheels and one rear wheel [(5)(7)(8)(10)(19)(20)]

Focusing on the pitching motion of passenger cars during turning, it is known that despite the relatively long wheelbase, the roll balance axis arrangement (Figure 2) causes forward-sloping pitching. However, this tendency is not observed in general motorcycles. Moreover, on a motorcycle, the rider can cancel out the vertical input from the road surface by bending and stretching both legs, which allows for a high natural frequency of the suspension. Therefore, it is possible to set the suspension's spring constant relatively high and suppress the pitching of the vehicle despite the short wheelbase.



RCH: Roll Center Height GCH: Gravity Center Height h: Equivalent roll moment height
Subscript f: front r: rear

Figure 2. Roll Center Height (*RCH*) and Gravity Center Height (*GCH*) [(20)(21)]

Considering the relatively short wheelbase (l) of the PMV shown in Figure 1, there is a greater concern about pitching during turns due to the roll balance axis arrangement than in automobiles. Additionally, unlike motorcycles, the occupants of PMVs are seated and are directly exposed to vertical input from the road surface. It is challenging to suppress pitching by setting the suspension spring constant (k) as high as in motorcycles. Therefore, the PMV, which tilts inward when turning, requires a suspension geometry that can take the pitching tendency under control as a design specification. However, there is no prior theoretical framework for the suspension mechanism of PMV.

1.2. Objective

When determining the suspension configuration for future PMV development, we aim to establish not just by applying existing suspension systems, but by defining the configuration with a theoretical necessity to avoid unnecessary pitching during turning.

ISSN: 2667-2812

 http://dx.doi.org/10.59490/6506799f4a8ec6d8f1c40fd6

Therefore, this paper proposes suspension requirements for taking the pitching tendency under control during cornering based on mechanistic theoretical considerations for PMVs with two front wheels and one rear wheel, which have an inward tilting mechanism.

The authors also proposed a method for setting the steering axis in order to minimize road disturbances in reference (19). In this paper as well, when suppressing pitching during turns, careful consideration is given to the suspension geometry to avoid unnecessary road disturbances. These requirements will serve as fundamental guidelines for future PMV development.

2. Vehicle Model for Mechanism Consideration

2.1. Assumed vehicle specifications

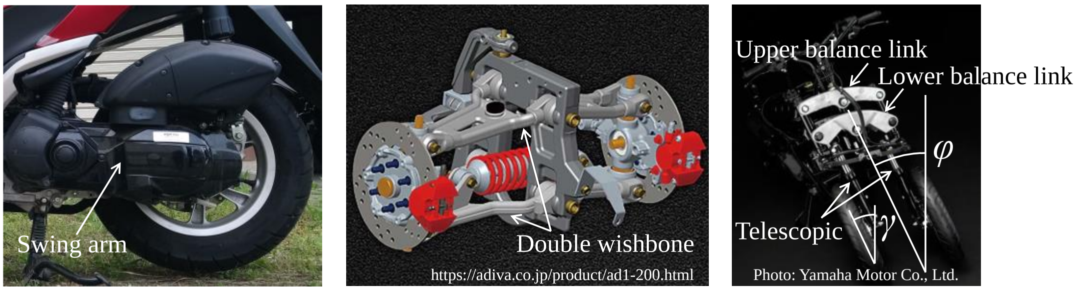
Table 1 presents the specifications of the PMV used in this study. To reduce vehicle pitching when driving on uneven road surfaces, it is typical to set the spring constant (wheel rate) of the front and rear suspensions in such a way that the resonant frequency of the sprung mass is lower at the front than at the rear. However, in this study, to keep things simple, the spring constants of the front and rear suspensions were set such that the sprung mass resonant frequency is similar to that of a very small passenger car and the front and rear frequency are equal.

Table 1. Assumed vehicle specifications [20]

Item	Unit	Value	Item	Unit	Value
Total length (L)	m	2.645	Total mass (m)	kg	370
Total width (W)	m	0.880	Front mass distribution (m_f)	kg	222
Total height (H)	m	1.445	Rear mass distribution (m_r)	kg	148
Wheel base (l)	m	2.020	Sprung roll inertia moment (I_x)	kgm^2	43.0
Front tread (Tr)	m	0.807	Sprung pitch inertia moment (I_y)	kgm^2	118
Gravity center height (GCH)	m	0.358	Front spring constant (k_f)	kN/m	7.20
Normalized camber stiffness (D_y)	10^{-2}/deg	1.49	Rear spring constant (k_r)	kN/m	10.0

2.2. Suspension Types

Although, a PMV with two front wheels and one rear wheel typically has a full trailing arm (swing arm) type suspension similar to a motorcycle (Figure 3 (a)) as the rear wheel, various suspension types such as typically double wishbone type (Figure 3 (b)) and telescopic type (Figure 3 (c)) are possible as the front wheels.



(a) full trailing arm **(b)** double wishbone **(c)** telescopic

Figure 3. Various Suspension Types [1][18][20]

2.3. Simplification of a Force and Moment Balance Model for Pitching Suppression

To achieve pitching suppression in a Personal Mobility Vehicle (PMV), the force balance formula related to the pitching posture of the car body during turning involves both the front and rear suspensions. However, in this report, we focus on comparing the balance of individual suspension forces between the front and rear of the vehicle to suppress pitching. Assuming no significant difference in the front and rear center of gravity height (GCH), we represent the pitching posture using separate front and rear back view models. In these models, the vehicle mass is divided according to the front and rear load distribution, as illustrated in Figure 4. The goal is to suppress the difference in the amount of sinking between the front and rear of the car body when turning. Specifically, we suppress the pitching of the PMV by matching the sinking of front center of gravity height (GCH_r) during turning with the sinking of rear center of gravity height (GCH_f) when the rear suspension is a full trailing arm type.

ISSN: 2667-2812

 http://dx.doi.org/10.59490/6506799f4a8ec6d8f1c40fd6

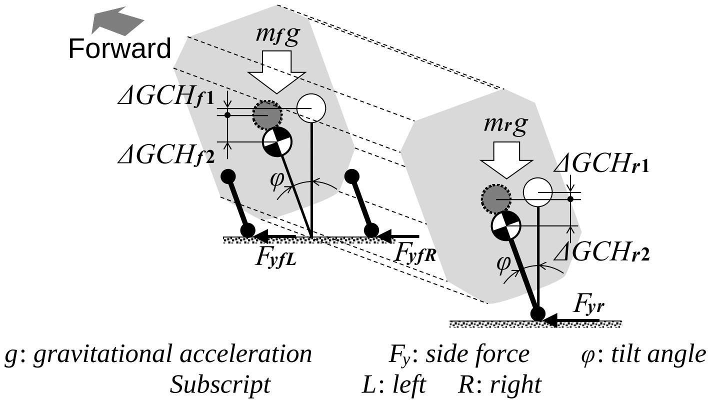


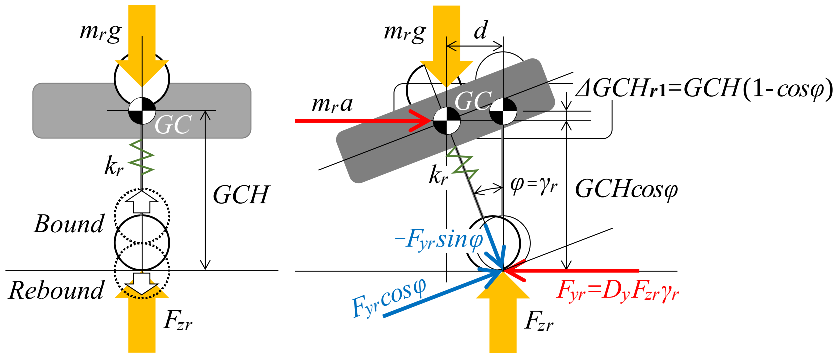
Figure 4. Force and moment balance model divided into front and rear [20]

3. Results on Mechanism Consideration

3.1. Full Trailing Arm Type Rear Suspension

Figure 5 (a) depicts a schematic view from the rear of the full trailing arm rear suspension. When the vehicle is upright, the grounding point of the tire moves up and down precisely in response to vertical input, and no unnecessary lateral force disturbance is generated from the road surface.

In Figure 5 (b), the force equilibrium state during turning is shown. As the vehicle tilts inward, the height of the center of gravity of the vehicle decreases by ΔGCH_{r1} as shown in Equation (1). The suspension reaction force ($F_{yr} cos\varphi$) that supports the lateral force (F_{yr}) on the ground is generated only in the direction of the virtual suspension arm axis, which is perpendicular to the movement trajectory of the tire grounding point. Therefore, a force ($-F_{yr} sin\varphi$) remains in the direction of the suspension stroke. By calculating Equation (2) using the spring constant of the rear wheel from Table 1, the relationship between the lateral turning acceleration and the amount of vertical car body sinking (ΔGCH_{r2}) is obtained, as shown in Figure 6. As the amount of vehicle body sink, the sinking due to force balance (ΔGCH_{r2}) adds to the decrease in the height of the center of gravity (GCH_r) due to inward tilting (ΔGCH_{r1}).



(a) on straight **(b)** during turning

Figure 5. Rear view of the full trailing arm type rear suspension [20]

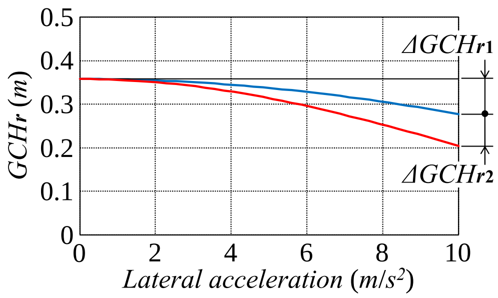


Figure 6. Rear body sinking for the full trailing arm type suspension [20]

ISSN: 2667-2812

 http://dx.doi.org/10.59490/6506799f4a8ec6d8f1c40fd6

$$\varDelta GCH_{r1} = (1 - \cos\varphi)GCH \tag{1}$$

$$\varDelta GCH_{r2} = \frac{F_{yr}\sin\varphi}{k_r}\cos\varphi \tag{2}$$

To suppress pitching during turning, it is essential to minimize the difference in sinking between the front and rear of the vehicle. The sinking of rear is represented by the sum of the changes in the rear center of gravity height ($\varDelta GCH_{r1} + \varDelta GCH_{r2}$).

3.2. Double Wishbone Type Front Suspension (for Cars)

The double wishbone type suspension is popular in automobiles. Therefore, let's first briefly organize the double wishbone type suspension for automobiles from a fundamental perspective. The mechanical equilibrium is illustrated in Figure 7. To simplify the illustration, Figure 7 demonstrates an example where the upper and lower suspension arms are parallel and equal in length, and there is no change in the tire camber angle due to suspension movement. Additionally, to avoid unnecessary complexity, the illustration assumes no roll angle is generated during turning, which implies that the torsional rigidity of the anti-roll bar is infinite.

Figure 7 illustrates the force balance of a double wishbone type suspension, commonly used in automobiles. In a conventional car, the virtual suspension arm axis (the line between the tire grounding point and the instantaneous center of the suspension) is typically angled (α) to reduce the vehicle body roll. The intersection of this axis and the vehicle center is called the roll center (RC), and its height (RCH_f) is used as a representative value for design. However, from the perspective of the tire grounding point trajectory, this angle (α) causes lateral movement of the tire along with its vertical stroke. This leads to unnecessary lateral force disturbance from the road surface. Thus, α is typically set to a small value as a compromise between roll suppression and disturbance suppression.

The suspension reaction forces (F^*_{yfL}, F^*_{yfR}) (Equations (2.1) and (2.2)) act only in the axial direction of the virtual suspension arm against the lateral forces (F_{yfL}, F_{yfR}) on the ground. Therefore, the forces in the suspension stroke direction (F^*_{zfL}, F^*_{zfR}) remain as shown in Equations (2.3) and (2.4). If there is no load shift between the left and right wheels during a turn, this residual force is purely a force couple in the direction to suppress vehicle roll, and does not cause the car body to lift or sink. However, load shifts occur from the inner to the outer wheel due to the roll moment balance during a turn. The influence of the outer wheel becomes dominant, and the effect of the body lifting corresponding to the angle (α) is obtained with the effect of suppressing roll.

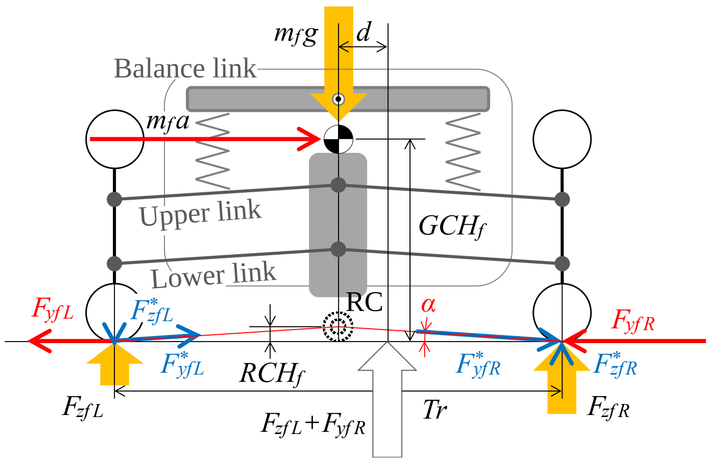


Figure 7. Force balance analysis for the double wishbone type front suspension [20]

$$m_f a = F_{yfL} + F_{yfR}$$

$$m_f g = F_{zfL} + F_{zfR}$$

$$F^*_{yfL} = F_{yfL}/\cos(-\alpha) \tag{2.1}$$

$$F^*_{yfR} = F_{yfR}/\cos\alpha \tag{2.2}$$

$$F^*_{zfL} = F_{yfL}\tan(-\alpha) \tag{2.3}$$

$$F^*_{zfR} = F_{yfR}\tan\alpha \tag{2.4}$$

F^*_{yfL}: Left suspension reaction force F^*_{yfR}: Right suspension reaction force
F^*_{zfL}: Left wheel lifting force in stroke direction F^*_{zfR}: Right wheel lifting force in stroke direction

ISSN: 2667-2812

 http://dx.doi.org/10.59490/6506799f4a8ec6d8f1c40fd6

3.3. Double Wishbone Type Front Suspension (for Inward Tilting PMVs)

Next, referring to the aforementioned fundamental consideration regarding the double wishbone type suspension on automobiles, Figure 8 depicts the force balance of the double wishbone type when the PMV tilts inward. In PMVs, inward tilt balances the rolling moment of the vehicle body during steady turning, eliminating the need to shift the load from the inner to the outer wheel in a steady state. However, Figure 8 shows the case where load shift occurs from the inner to the outer wheel along with the inward tilt, as a generalized relationship that includes the transient state. Additionally, a small initial α value is assumed, as in the case of automobiles.

During turning, the left and right wheels tilt inward by $\gamma = \varphi$ along with the vehicle body. Viewing the wheels from the vehicle body, the inner wheel strokes toward the bound direction, and the outer wheel strokes toward the rebound direction. As depicted in Figure 8, when the upper and lower suspension arms are equal in length (1/2 of the tread) and parallel, the roll moment balance point (RC) does not move. The reaction forces in the axial direction of the virtual suspension arm (F^*_{yfL}, F^*_{yfR}) are given by Equations (3.1) and (3.2), while the residual force (F^*_{zfL}, F^*_{zfR}) in the suspension stroke direction is expressed by Equations (3.3) and (3.4).

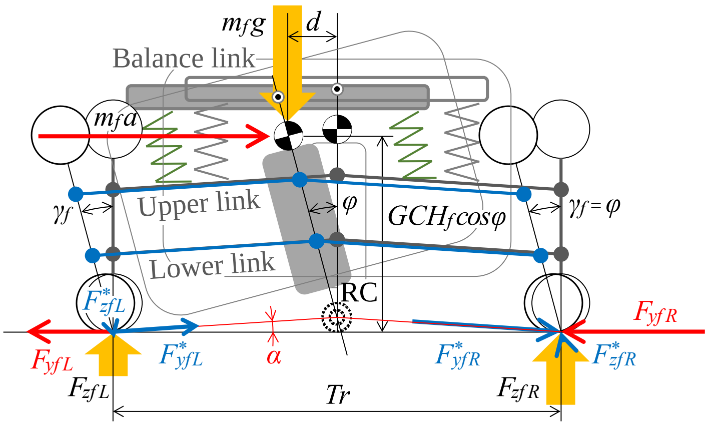


Figure 8. Front suspension with double wishbone type and inward tilting mechanism [20]

$$F^*_{yfL} = \frac{F_{yf\,L}\cos\varphi}{\cos\alpha\cos\varphi + \sin\alpha\sin\varphi} \tag{3.1}$$

$$F^*_{yfR} = \frac{F_{yfR}\cos\varphi}{\cos\alpha\cos\varphi - \sin\alpha\sin\varphi} \tag{3.2}$$

$$F^*_{zfL} = \frac{-F_{yfL}\sin\alpha}{\cos\alpha\cos\varphi + \sin\alpha\sin\varphi} \tag{3.3}$$

$$F^*_{zfR} = \frac{F_{yfR}\sin\alpha}{\cos\alpha\cos\varphi - \sin\alpha\sin\varphi} \tag{3.4}$$

In the case of a front double wishbone type PMV, if there is no load shift between the left and right wheels as in steady turning shown in Figure 8, the residual forces F^*_{zfL} and F^*_{zfR} cancel each other out regardless of the angle (α), and no lifting force is generated. Furthermore, even if there is a load shift between the wheels, no lifting force is generated if $\alpha = 0$. Although suppressing body roll is useful in automobiles, it is not necessary in PMVs. Therefore, to avoid unnecessary lateral disturbance, it is not essential to set the angle (α) to the virtual suspension arm at the practical vehicle height for PMVs. If α = 0, the vehicle height does not change when turning (see Figure 9). However, it should be noted that maintaining $\alpha = 0$ at all vehicle heights may not always be feasible.

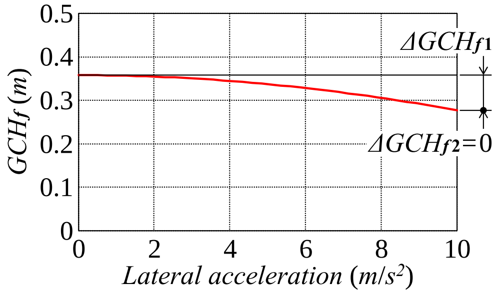


Figure 9. Squatting posture of the vehicle during tilting [20]

ISSN: 2667-2812

http://dx.doi.org/10.59490/6506799f4a8ec6d8f1c40fd6

As discussed in Section 3.1, when using a full trailing arm type rear wheel suspension, the vehicle body tends to sink during turning. Consequently, when combined with the double wishbone type front suspension described earlier, it becomes inevitable for the PMV to be in a squat position.

In general, it is difficult to make the actual arm length of the double wishbone suspension exactly 1/2 of the tread. Therefore, the balance point (RC) of the roll moment typically moves up and down along with the suspension stroke, as shown in Figure 10.

When the PMV tilts inward during turning, the roll moment balance point (RC) at the inner wheel moves in the direction opposite to the tilting angle (φ) by the angle θ_L, and the reaction force angle (β_L) to the lateral force generated by the tire becomes more negative. At the outer wheel, RC moves in the opposite direction by θ_R, and the reaction force angle (β_R) changes to a larger positive value. As a result, both F^*_{zfL} and F^*_{zfR} generate a lifting force, further strengthening the squat posture of the PMV.

The axial reaction forces (F^*_{yfL}, F^*_{yfR}) of the virtual suspension arm are expressed by Equations (4.1) and (4.2), and the residual force (F^*_{zfL}, F^*_{zfR}) in the stroke direction of the suspension is given by Equations (4.3) and (4.4).

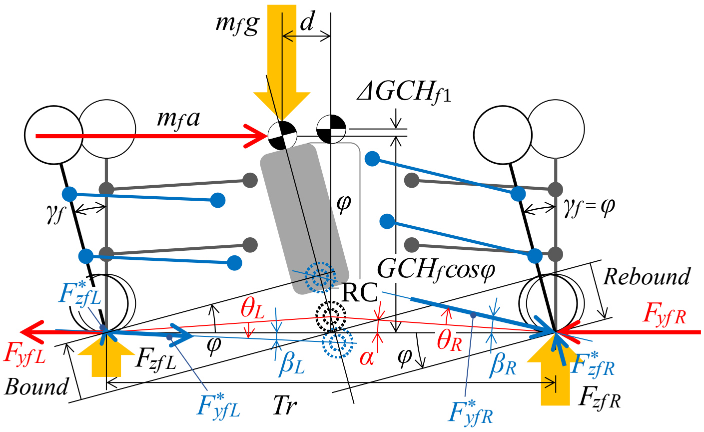


Figure 10. Movement of the instantaneous center during bound/rebound [20]

$$F^*_{yfL} = \frac{F_{yf\,L}\cos\varphi}{\cos\beta_L\cos\varphi + \sin\beta_L\sin\varphi} \tag{4.1}$$

$$F^*_{yfR} = \frac{F_{yfR}\cos\varphi}{\cos\beta_R\cos\varphi - \sin\beta_R\sin\varphi} \tag{4.2}$$

$$F^*_{zfL} = \frac{-F_{yfL}\sin\beta_L}{\cos\beta_L\cos\varphi + \sin\beta_L\sin\varphi} \tag{4.3}$$

$$F^*_{zfR} = \frac{F_{yfR}\sin\beta_R}{\cos\beta_R\cos\varphi - \sin\beta_R\sin\varphi} \tag{4.4}$$

$$\beta_L = \alpha + \theta_L \qquad \beta_R = \alpha - \theta_R$$

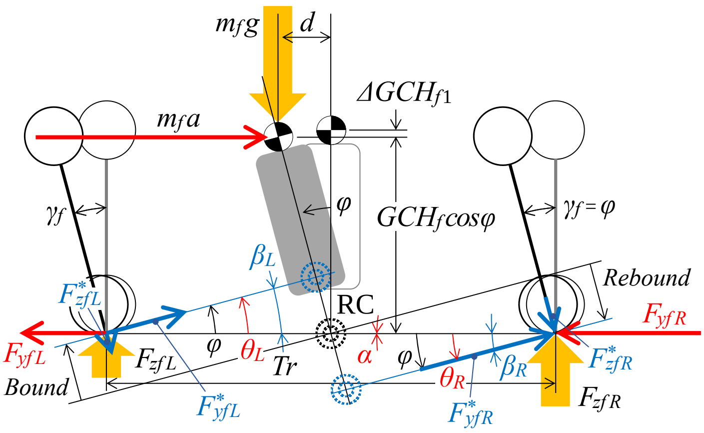


Figure 11. Same level of sinking as the full trailing arm type rear suspension [20]

ISSN: 2667-2812

 http://dx.doi.org/10.59490/6506799f4a8ec6d8f1c40fd6

To achieve the same sinking as the full trailing arm type without any unnecessary lateral disturbance, it is necessary to match the movement of the roll moment balance point (RC) to that of the full trailing arm type when tilted inward by setting the initial reaction force angle (α) to zero. This relationship is illustrated in Figure 11. Even if there is a load shift in both front wheels, the average sinking value of the front is the same as that of the rear, although there is some effect in suppressing the inward tilt.

To achieve the reaction force angle change shown in Figure 11 in a double wishbone type suspension, it is not easy in practice. One possible approach is to significantly shorten the length of the upper arms as shown in Figure 12. However, this can result in significant changes in the reaction force angle (α) and camber angle due to changes in vehicle height, and it may not be possible to avoid lateral force disturbance from the road surface. Therefore, it may not be possible to achieve both the avoidance of road disturbances and the suppression of squatting during cornering with a combination of double wishbone type front suspension and full trailing arm type rear suspension.

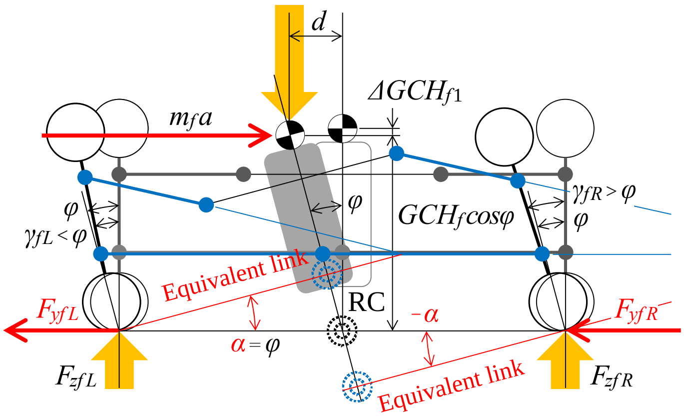


Figure 12. Arm arrangement to avoid the squatting posture [20]

3.4. Telescopic Type Front Suspension

Next, let us consider the upright state of a telescopic suspension, which is commonly used in motorcycles, for both front wheels of a PMV as shown in Figure 13. In the telescopic type, the reaction force angle (α) is always zero when the left and right wheels move up and down in phase. However, when turning as shown in Figure 14, the reaction force axis supporting the tire lateral force tilts with the inward tilt of the vehicle body, similar to the rear full trailing type.

The axial reaction forces (F^*_{yfL}, F^*_{yfR}) of the virtual suspension arm can be calculated using Equations (5.1) and (5.2), while the residual forces (F^*_{zfL}, F^*_{zfR}) in the suspension stroke direction can be expressed using Equations (5.3) and (5.4).

$$F^*_{yfL} = F_{yf\,L} \cos\varphi \tag{5.1}$$

$$F^*_{yfR} = F_{yfR} \cos\varphi \tag{5.2}$$

$$F^*_{zfL} = -F_{yfL} \sin\varphi \tag{5.3}$$

$$F^*_{zfR} = -F_{yfR} \sin\varphi \tag{5.4}$$

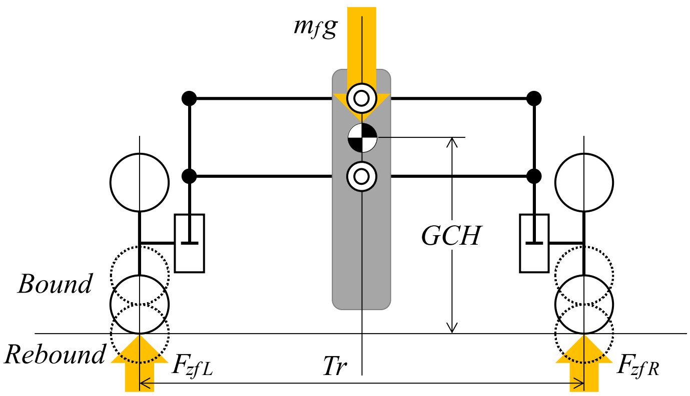


Figure 13. No road disturbances in the case of a telescopic type suspension [20]

ISSN: 2667-2812

 http://dx.doi.org/10.59490/6506799f4a8ec6d8f1c40fd6

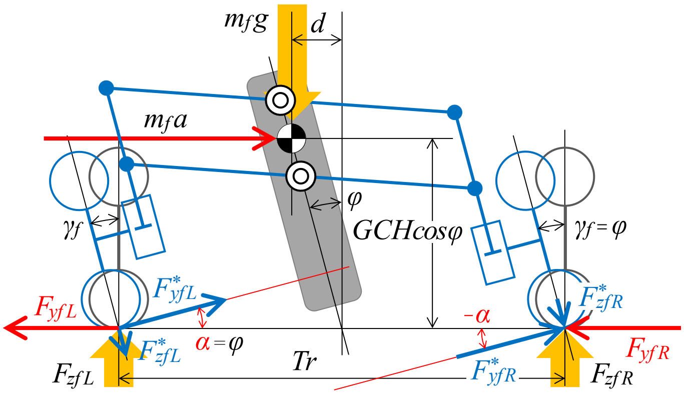


Figure 14. Same level of sinking as the full trailing arm rear suspension [20]

In a PMV that tilts inward when turning, there is no need for load shift between the left and right wheels during steady turning ($F_{zfL}=F_{zfR}=F_{zf}$), and the lateral forces generated by the tires on the left and right wheels are equal ($F_{yfL}=F_{yfR}=F_{yf}=D_y\gamma_f F_{zf}$). Due to the force ($F^*_{zfL}=F^*_{zfR}=-F_{yf}\,sin\varphi$) that causes the body to sink due to the suspension reaction force, both the left and right wheels have the same amount as the rear wheels of the full trailing arm type. Therefore, both the suppression of body pitching and the prevention of road disturbance can be achieved.

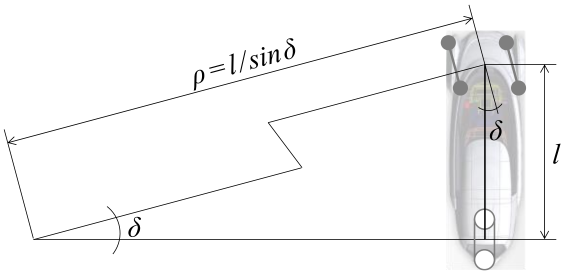
4. Additional Discussions from Dynamic Perspectives

In the previous chapter, as a fundamental theoretical consideration, we examined the vehicle's posture in a static balanced state and demonstrated that to suppress pitching during turns, it is preferable to use a telescopic type suspension for the front wheels. However, when transitioning to real-world analysis, it is necessary to consider the pitching during dynamic steering response in advance. As demonstrated in the authors' previous investigations on dynamic characteristics [(5)-(10), (13)] using CarMaker from a German company IPG, it is possible to obtain vehicle responses under dynamic conditions through multibody dynamics analysis e.g., Carmaker instead of real vehicle experiments. However, this is an outcome as just result and does not lead to a theoretically derived framework.

Hence, in this chapter, building upon the results obtained in the previous chapter, we extend the static analysis to systematically organize the pitching attitudes under dynamic conditions for the combination of front wheel telescopic suspension and rear wheel full trailing arm suspension using a theoretical approach.

4.1. Active Inward Tilt Tracking Control Model

For PMVs equipped with motorcycle tires, the lateral forces during turns primarily arise from camber thrust due to the tire's camber angle with respect to the ground (Figure 15). Assuming an extremely small tire slip angle, achieving an active inward lean angle (φ) involves following a target lean angle (*TTA*) as shown in Equation (6). An illustrative concept of a general PID control is shown in Figure 16. In the case of general PID control for achieving active inward lean angle tracking, the actual lean angle (φ) lags significantly behind the lateral acceleration (a) of the vehicle [(5)-(10), (13)]. Even during steady-state turns where there is no load transfer between the left and right wheels in a PMV, as required to balance the roll moment caused by lateral acceleration (a), transiently, there is a load transfer from the inner to the outer wheel similar to automobiles [(5)-(10)]. For simplicity, assuming no delay due to the body's roll inertia moment (I_x) shown in Equation (7), the load transfer between the left and right wheels solely depends on the delay of the active inward tilt angle tracking (PID control) behind to lateral acceleration (a).



δ: tire steered angle ρ: turning radius l: wheel base

Figure 15. Tire steering angle and turning radius [(10)]

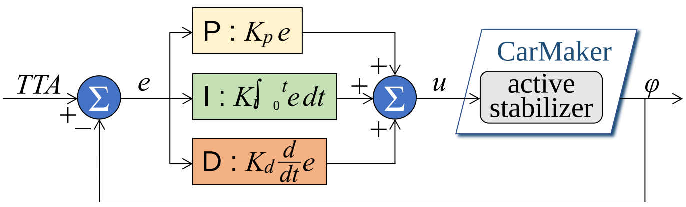
ISSN: 2667-2812

 http://dx.doi.org/10.59490/6506799f4a8ec6d8f1c40fd6

$$TTA = \tan^{-1}\frac{v^2 \sin\delta}{lg} \tag{6}$$

$$I_x\ddot{\varphi} = F_{zfR} - F_{zfL} \tag{7}$$

TTA: target tilt angle   *v*: vehicle speed
g: gravitational acceleration I_x: rolling inertia φ double dot: rolling acceleration



φ: actual tilt angle   *e*: error   *u*: output
K_p = 4000: proportional gain K_i = 50: integral gain K_d = 0: differential gain

Figure 16. PID tracking control [13]

4.2. Preliminary Dynamic Considerations of Delay in Active Inward Tilt Tracking Control

As an illustrative scenario, even if the vehicle has already responded (lateral acceleration is present), if the inward tilt angle (φ) has not yet occurred, load transfer between the left and right wheels will occur similarly to the case of an automobile described in Section 3.2. At this point, the tilt angle (φ) is zero. Consequently, as indicated in Figure 12, for telescopic suspension, $\alpha=\varphi$, which means the reaction angle (α) is also zero. By substituting $\varphi=0$ into Equations (5.3) and (5.4), $F^*_{zfL}=F^*_{zfR}=0$ is derived. This implies that no body sink occurs in either the front or rear suspensions, and consequently, no pitching takes place. Next, if the actual inward lean angle (φ) already matches the desired target lean angle (*TTA*) that corresponds precisely to the lateral acceleration (*a*) caused by the steering response due to the tracking control, load transfer between left and right wheels does not occur as Equation (8). As shown in Figure 5, tire lateral force (F_y) is shown in Equation (9). From Equation (8) and (9), tire lateral force of front both tires are described as Equation (10). By substituting Equation (10) into Equations (5.3) and (5.4), Equation (11) is obtained. Consequently, the body sinks but pitching does not occur. In this way, concerns regarding pitching induced by the delay in PID control are unnecessary.

$$F_{zfL} = F_{zfR} \tag{8}$$

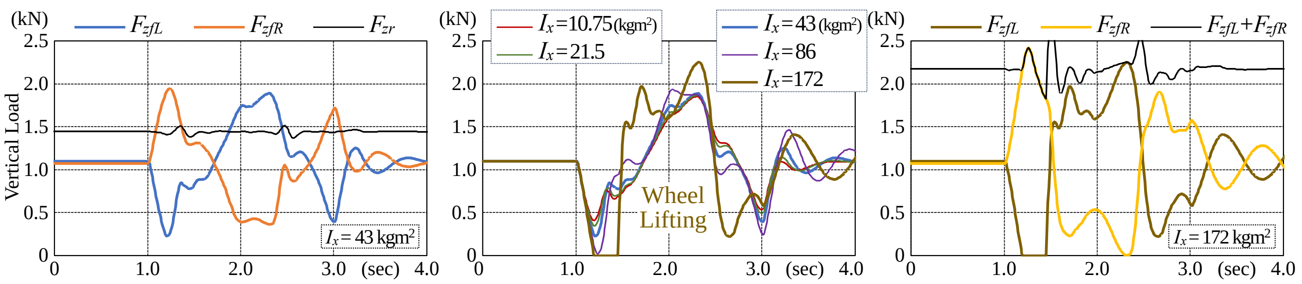
$$F_y = D_y F_z \gamma \tag{9}$$

$$F_{yfL} = F_{yfR} \tag{10}$$

$$F^*_{zfL} = F^*_{zfR} \tag{11}$$

4.3. Preliminary Dynamic Considerations on the Influence of Roll Inertia Moment

On the other hand, as indicated in references (5)-(10), sudden steering can lead to moments where the inner wheel lifts momentarily during a turn. An example of this phenomenon is shown in Figure 17. Vehicle specifications are given in Table 1.



(a) load transfer between front wheels **(b)** inner wheel load due to roll inertia **(c)** change in transient total vertical load

F_z: vertical load, subscript f: front, r: rear, L: left, R: right
v = 10m/s, sinusoidal steering input 0.5Hz, steering angle = ±60deg
I factor of PID control K_i = 50, standard sprung roll inertia moment I_x = 43

Figure 17. Difference in vertical load change due to roll inertia moment [8]

ISSN: 2667-2812

 http://dx.doi.org/10.59490/6506799f4a8ec6d8f1c40fd6

This phenomenon of the inner wheel lifting is influenced by the relationship between the body's roll inertia moment (I_x) and the control parameters (K_p, K_i, K_d) for the inward tilt angle tracking, as indicated in reference (8). Specifically, the larger the body's roll inertia moment (I_x) and the smaller the delay in the inward tilt angle tracking, the more pronounced the phenomenon becomes, resulting in a significant load transfer between the left and right wheels. The key point to consider here is that unlike the load transfer induced by the delay in PID control shown in the previous section, the load transfer becomes larger due to a smaller delay. In other words, when examining the significant load transfer between the left and right wheels induced by sharp steering, considering the effect of the body's roll inertia moment (I_x), rather than PID control, is essential.

When the body's roll inertia moment (I_x) is large, even if the intended stroke of the inner and outer wheels is achieved through tracking control, the required inward tilt angle cannot be achieved, leading to a significant rebound of the outer wheel and lifting of the inner wheel. As shown in Figure 17 calculated using CarMaker, while the inner wheel is lifted and the inner wheel's ground load remains zero in case with the largest I_x, the ground load on the outer wheel increases further. This indicates that the outer wheel is supporting the vehicle's mass and the front body is lifted. Thus, considering the influence of the body's roll inertia moment (I_x), it becomes evident that even with a front wheel telescopic suspension, during transient steering responses, concerns about the squatting pitching arise due to the inability to maintain a constant left-right sum of the vertical load.

5. Conclusions

After considering the suspension geometry of a PMV with an inward tilting mechanism, the following conclusions were drawn to take the pitching tendency under control during turning based on the mechanistic theoretical considerations:

- The rear wheel full trailing arm type suspension has the same sinking characteristic as a motorcycle, and in order to prevent body pitching, it is necessary to provide the front suspension with the same sinking characteristic as the rear suspension.

- The front double wishbone type suspension is incompatible with sinking when turning and avoiding road disturbances during straight running caused by tread and camber changes.

- The front telescopic suspension has sinking characteristics synchronized with the rear suspension and avoids road disturbances during straight running caused by tread and camber changes, regardless of whether there is load transfer between the left and right wheels due to the delay of the tracking control (PID control) for the active inward tilt.

- However, even with the front telescopic suspension, the delay in tilt angle due to the roll moment of inertia (I_x) of the vehicle body can cause the squatting pitching.

These findings indicate that sinking the vehicle body during turning with the front and rear suspensions of the PMV, which tilts inward when turning like a motorcycle, can take the squatting pitching tendency under control. These requirements can serve as fundamental guidelines for future PMV development. As future works, the transient phenomena that cannot be explained by static balance through dynamic model analysis on the time axis will be also investigated.

References

(1) Haraguchi, T. (2020). *4. Personal Mobility Vehicle (J)* (Mobility Service, Mobility Innovation Series, Vol. 1, pp. 92–123). Corona Publishing Co., Ltd., Tokyo, Japan. ISBN 978-4-339-02771-6.

(2) Haraguchi, T. (2020). Prospects of Future Personal Mobility (PMV) from a Historical Perspective (J). *Cutting-edge Automotive Technology, Monthly Automotive Technology, Technical Information Institute Co., Ltd., Tokyo, Japan.* ISSN2432-5694.

(3) Kaneko, T., Kageyama, I., & Haraguchi, T. (2016). A Study on the Harmonization of Personal Mobility Vehicle with a Lean Mechanism in Road Traffic (First Report) - Micro-Scale Traffic Flow Simulation in Consideration of Mutual Interference of Driver Risk Feeling - (J). *Proceedings of the JSAE Annual Congress (2016 Spring)*, Yokohama, Japan. No.55-16, p.1350-1354, 20165255.

(4) Haraguchi, T., Kaneko, T., Kageyama, I., Kuriyagawa, Y., & Kobayashi, M. (2017). Study of Tilting Type Personal Mobility Vehicle by the Immersive Driving Simulator with Five Large Screens (J). *Transactions of Society of Automotive Engineers of Japan*. Vol.48 No.3, p.693-698, 20174447.

(5) Haraguchi, T., Kaneko, T., Kageyama, I., Kobayashi, M., & Murayama, T. (2017). Obstacle Avoidance Maneuver of Personal Mobility Vehicles with Lean Mechanism - Comparison between Front and Rear Wheel Steering (FWS and RWS) - (J). *Proceedings of the JSAE Annual Congress (2017 Spring)*, Yokohama, Japan. No.18-17, p.494-499, 20175092.

ISSN: 2667-2812

(6) Kaneko, T., Kageyama, I., & Haraguchi, T. (2017). Dynamic Rollover Characteristics of Personal Mobility Vehicles with Lean Mechanism (J). *Proceedings of the JSAE Annual Congress (2017 Autumn)*, Osaka, Japan. No.163-17, p.1392-1398, 20176258.

(7) Haraguchi, T., Kaneko, T., & Kageyama, I. (2018). Study on Steering Response of Personal Mobility Vehicle (PMV) by Comparison of PMV with Passenger Cars and Motorcycles on the Obstacle Avoidance Performance (J). *Proceedings of the JSAE Annual Congress (2017 Autumn)*, Nagoya, Japan. No.113-18, p.1-6, 20186058.

(8) Haraguchi, T., Kageyama, I., & Kaneko, T. (2019). Inner Wheel Lifting Characteristics of Tilting Type Personal Mobility Vehicle by Sudden Steering Input (J). *Transactions of Society of Automotive Engineers of Japan*. Vol.50 No.1, p.96-101, 20194038.

(9) Kaneko, T., Kageyama, I., & Haraguchi, T. (2019). A Study on Characteristics of the Vehicle Response by Abrupt Operation and an Improvement Method for Personal Mobility Vehicle with Leaning Mechanism (J). *Transactions of Society of Automotive Engineers of Japan*. Vol.50 No.3, p.796-801, 20194345.

(10) Haraguchi, T., Kageyama, I., & Kaneko, T. (2019). Study of Personal Mobility Vehicle (PMV) with Active Inward Tilting Mechanism on Obstacle Avoidance and Energy Efficiency. *Applied Sciences, Multidisciplinary Digital Publishing Institute (MDPI),* 9(4737). https://doi.org/10.3390/app9224737.

(11) Kageyama, I., Kuriyagawa, Y., Haraguchi, T., Kobayashi, Y. Two-wheeled Vehicle Characteristics on Steady State Turning Considering Equivalent Compliance (J). *Transactions of Society of Automotive Engineers of Japan*. Vol.50 No.5, p.1402-1408, 20194696.

(12) Haraguchi, T., Kaneko, T., Kageyama, I. (2020). Steering Torque Characteristics of Personal Mobility Vehicle (PMV) with Inward Tilting Mechanism (J). *Transactions of Society of Automotive Engineers of Japan*. Vol.51 No.5, p.931-937, 20204471.

(13) Haraguchi, T., Kaneko, T., Kageyama, I. (2021). Steering Torque Hysteresis of Personal Mobility Vehicle (PMV) with Inward Tilting Mechanism (J). *Transactions of Society of Automotive Engineers of Japan*. Vol.52 No.5, p.987-993, 20214638.

(14) Haraguchi, T., Kaneko, T., Kageyama, I. (2022). A Study of Steer Characteristics of Personal Mobility Vehicle (PMV) with Inward Tilting Mechanism (J). *Transactions of Society of Automotive Engineers of Japan*. Vol.53 No.1, p.58-64, 20224033.

(15) Haraguchi, T., Kaneko, T., Kageyama, I. (2021). A Method to Minimize Steering Disturbance Caused by Vertical Load Reaction Force of Personal Mobility Vehicle (PMV) with Inward Tilting Mechanism (J). *Transactions of Society of Automotive Engineers of Japan*. Vol.52 No.2, p.462-468, 20214197.

(16) Haraguchi, T., Kaneko, T., Kageyama, I. (2022). Design Requirements for a Personal Mobility Vehicle (PMV) with an Inward Tilt Mechanism to Maintain Straight Running on Slant Roads and Rutted Roads (J). *Proceedings of the JSAE Annual Congress (2022 Spring),* Yokohama, Japan. No.66-22, p.1-6, 20225291.

(17) Kaneko, T., Haraguchi, T., Kageyama, I. (2022). Dynamic Performance Evaluation of a Personal Mobility Vehicle (PMV) with an Inward Tilt Mechanism on Non-flat Road Using Dynamic Motion Analysis Simulation (J). *Proceedings of the JSAE Annual Congress (2022 Autumn)*, Osaka, Japan. No.116-22, p.1-5, 20226116.

(18) Haraguchi, T., Kaneko, T., Kageyama, I. (2022). A Study of Suspension Geometry for Personal Mobility Vehicle (PMV) with Inward Tilt Mechanism (J). *Proceedings of the JSAE Annual Congress (2022 Autumn)*, Osaka, Japan. No.117-22, p.1-6, 20226117.

(19) Haraguchi, T., Kaneko, T. (2022). Design Requirements for Personal Mobility Vehicle (PMV) with Inward Tilt Mechanism to Minimize Steering Disturbances Caused by Uneven Road Surface. *Inventions*, *Multidisciplinary Digital Publishing Institute (MDPI)*, Inventions 2023, 8(37), DOI:10.3390/inventions8010037.

(20) Haraguchi, T., Kaneko, T., Kageyama, I. (2023). A Study of Suspension Geometry for Suppressing Pitching during Turning in Personal Mobility Vehicles (PMVs) with Inward Tilt Mechanism (J). *Proceedings of the JSAE Annual Congress (2023 Spring)*, Yokohama, Japan. No.167-23, p.1-6, 20235167.

(21) Abe, M. (2012). *Automotive Vehicle Dynamics Theory and Application (The second edition) (J)*. Tokyo Denki University Press. ISBN 978-4-501-41920-2.

ISSN: 2667-2812