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Research Paper

Electric Delta Trike Stability Characteristic and Maneuverability Analysis: Experiment and Multi-Body Dynamic Simulation

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Abstract

This paper reveals the stability characteristics of an electric-powered delta trike (e-trike), which is developed for goods delivery services. The changeable center of gravity position and weight due to electric component placement and the carried good weight can cause instability of the e-trike. Three main parameters are firstly evaluated on the e-trike: 1) geometry, 2) center of gravity, and 3) stiffness and damping coefficient of the suspensions. Single Lane-Change (SLC) and Double Lane-Change (DLC) tests were then conducted following ISO 14791:2000 and ISO 3888-1:2018 standards, respectively. An e-trike model was created and simulated using SIMPACK, a multi-body dynamic software. The simulation results showed that the developed e-trike model can replicate SLC and DLC tests, indicating the model was valid. A parametric study with the validated model was then conducted with various e-trike weights, center of gravity position, and suspension stiffness and damping ratio values. The results showed additional weight and higher center of gravity position can decrease threshold velocity to avoid rollover. The low suspension stiffness also contributed to lower the threshold velocity. However, the damping coefficient value did not change the threshold velocity significantly. These results can be a guideline in designing a delta trike with better performance in stability and maneuverability.

Keywords: Electric vehicle; Delta trike; Rollover; Single lane-change; Double lane-change; Multi-body dynamic

1. Introduction

A tricycle vehicle or trike has two kinds of configurations, 1) configuration which has two wheels on the front axle (tadpole trike), and 2) configuration which has one wheel on the front axle (delta trike). Tadpole trike requires a relatively complicated steering system design due to wheel configuration whereas the delta trike configuration is usually preferred due to the simplicity of the design and low-cost production [1], [2]. However, the delta trike tends to be more unstable compared with the tadpole trike [3]. Comparison study of stability characteristics with other vehicle configurations, such as a four-wheel vehicle, velomobile, recumbent bike, and tadpole trike, acclaims delta trike as the vehicle with high instability aspect [4].

Lateral stability greatly depends on the value of the vehicle understeer coefficient which may be either directionally stable at a certain velocity or become unstable above the threshold velocity [5], [6]. Many factors could affect the lateral stability, such as the vehicle dimension, center of gravity, turning radius, etc [7]. The change in those factors could lead to changes in stability characteristics [8]. The study by J.C Huston et al. indicates that lateral stability increases when the delta trike center of gravity is closer to the rear axle [5].

Rollover stability is another major concern that needs to be evaluated on the delta trike [9], [10]. Bunte et al [4] reported that 58.3% of trikes would experience an accident when passing the roundabouts infrastructure, compared with velomobile at 6.3% and recumbent bikes at 3%. Rollover stability is a measure of a vehicle's tendency to tip over sideways when a lateral body force is applied, e.g. during cornering. In the common vehicle dynamics control systems, one suitable way to control the vehicle's lateral motion when the severe maneuver is applied is by controlling the yaw moment [11]. According to direct yaw moment control comparison, the trike has a higher tendency to be unstable due to the odd number of wheels in comparison to the even number of wheels [12]. There is also a comprehensive study by Van Valkenburgh et al. [10] which reported that tadpole trike tends to understeer more than four-wheelers whereas delta trike configuration tends to oversteer.

Several studies discussed various ways to improve the stability of a three-wheeled electric vehicle. Ataei et. al. [13] developed a reconfigurable actuator system for three- and four-wheeled vehicles to improve stability. The system was applied to the mathematical model of the vehicles and improved the maneuverability of the vehicles in following a desired yaw rate. However, implementing a comprehensive control system give additional complexity to the vehicle and the study lacks experimental tests to validate the calculation result. Furuichi et. al. [14] designed a three-wheeled vehicle with a tilt controller for all wheels to improve the vehicle stability during cornering. The stability performance of the vehicle was analyzed from the simulation and experiment with the condition of normal driving state where all wheels were on the ground with a speed of 2.2 m/s and 8 m/s and sudden turn state to make one wheel elevated above ground with a speed of 2.7 m/s. The study obtained a range of safe angular velocity to steering angle graph for the speed of 2.2 m/s normal driving with a maximum value around 1.25 rad/s and 2.7 m/s sudden turn with a value around 0.7 rad/s. The vehicle still gives a considerably low threshold angular velocity given the low-speed driving. A study to improve three-wheeled electric vehicle stability which implementing simulation and experiment analysis was needed to improve the threshold velocity of the vehicle to an adequate value for operation.

The stability of an electric trike (e-trike) having delta configuration for goods delivery is evaluated in this study. The e-trike prototype was created to fulfill goods mobility due to e-commerce growth [15]. The stability must be evaluated properly because the e-trike brings various packages during service, which can directly determine the center of gravity depending on the package arrangement. Furthermore, due to the nature of electric components such as electric motor, inverter, battery, and control system, which are mostly connected to cables, the electric components can be positioned more flexibly, but it causes the center of gravity point alters significantly [16]. Design failure of the e-trike can increase the risk of accidents due to instability. Considering the number of accidents and all related fundamental parameters, a comprehensive study on the stability characteristics in delta trike needs to be learned and observed thoroughly to achieve better performance and to better define the threshold velocity.

The objective of this study was to reveal the stability characteristic of electric delta trike for good delivery which can be useful for improving stability during the design process. Both experimental and muti-body dynamic analyses were conducted to comprehensively understand the delta e-trike characteristics for good delivery. The evaluated e-trike in this study was previously designed and manufactured in our laboratory. Center of gravity and suspension stiffness and damping coefficient were chosen as the main parameters analyzed in this study since they are customizable for e-trike and significantly affect the vehicle stability. The e-trike geometry, center of gravity, and equivalent suspension were firstly evaluated experimentally. On-road tests, Single Lane-Change and Double Lane-Change were then conducted following ISO 14791:2000 and ISO 3888-1:2018, respectively. Lateral, rollover, and maneuverability aspects of the e-trike were also analyzed. A model for multi-body dynamic simulation was then developed in SIMPACK software. The simulation replicated on-road testing to assure the validated model is obtained.

Lateral and rollover stabilities obtained from experiments and simulations are then compared. Furthermore, the correlation between center of gravity, suspension system, and threshold velocity was revealed from simulation results. Maneuverability was also evaluated using multi-body dynamic simulation which was preferred as an optimum method for evaluating the stability characteristic. The influences of center of gravity and suspension on threshold velocity were then comprehensively discussed.

2. Methods

2.1. E-Trike Geometry, Equivalent Suspension, and Center of Gravity

Three parameters to evaluate the e-trike stability characteristic, i.e., dimension, equivalent suspension, and the center of gravity, were firstly measured. The prototype of e-trike had been manufactured and is currently under performance validation before unit commercialization. The direct measurement of those parameters on the e-trike prototype was preferred than collecting data from the 3D CAD model because it already considers data error due to the manufacturing process of the prototype. Incompatibility data between actual prototype and 3D modeling may appear due to the manufacturing process such as raw material properties, machining activities, weld process, painting process, and other applied processes. The dimensional values obtained by the direct measurement on the prototype including the front and rear suspension angles are shown in **Figure 1**.

WitMotion HWT901B sensor was placed at the bottom of the driver seat to record acceleration and angular velocity in three axes. The position of the acceleration sensor relative to the wheels was also measured. In this study, the required data that can be obtained from the sensor are accelerations in three axes (A_X, A_Y, A_Z), angles ($\angle_X$, $\angle_Y$, $\angle_Z$), and rotational speeds in roll, pitch, yaw (ω_X, ω_Y, ω_Z). To assure validated measurement results, the sensor was firstly calibrated before used for measurement.

Testing the suspension characteristics is done by passing the vehicle on a track with a speed bump. The vehicle was given an initial speed of 5 km/h which was kept constant throughout the test. Characteristic testing is carried out separately between the front suspension and rear suspension (see **Figure 2a** and **Figure 2b**). The data recorded by the WitMotion HWT901B sensor was the vertical acceleration of the vehicle on the Z-axis (Az) as shown in **Figure 2c**. From the recorded data, the e-trike suspension was categorized as an underdamped system. The data went through the sampling rate increase process to give better observation of the phenomenon and then the data was analyzed using Matlab® 2021b with the Discrete Meyer Wavelet method. The recorded acceleration was transformed by using Fourier transformation and plotted in the frequency domain as shown in **Figure 2d**. It was found that the natural frequency of the front and rear suspensions were 2.916 Hz and 2.72 Hz, respectively, while the initial data acquisition frequency of the sensor was 10 Hz which satisfies

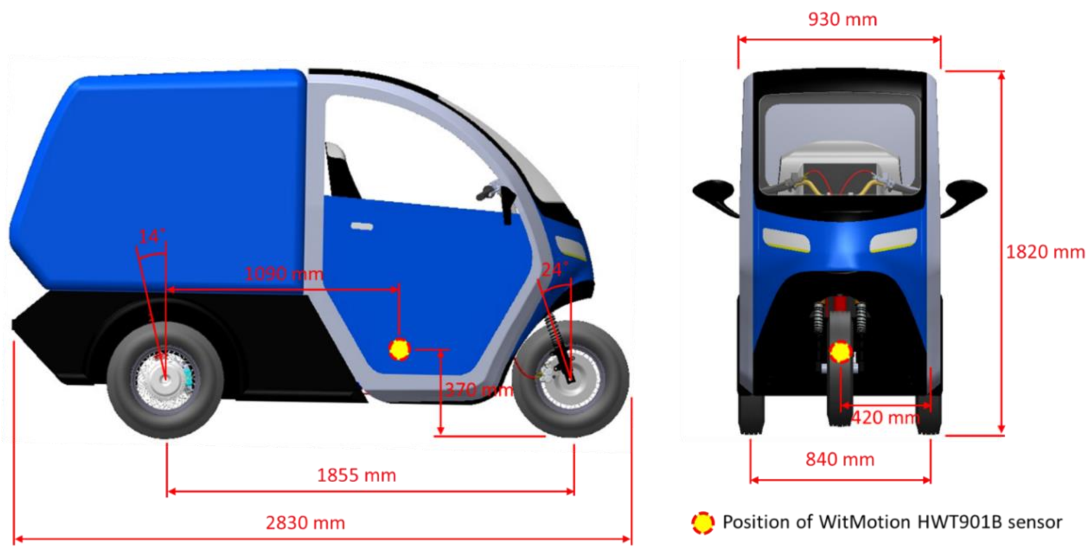


Figure 1. Delta trike prototype geometry

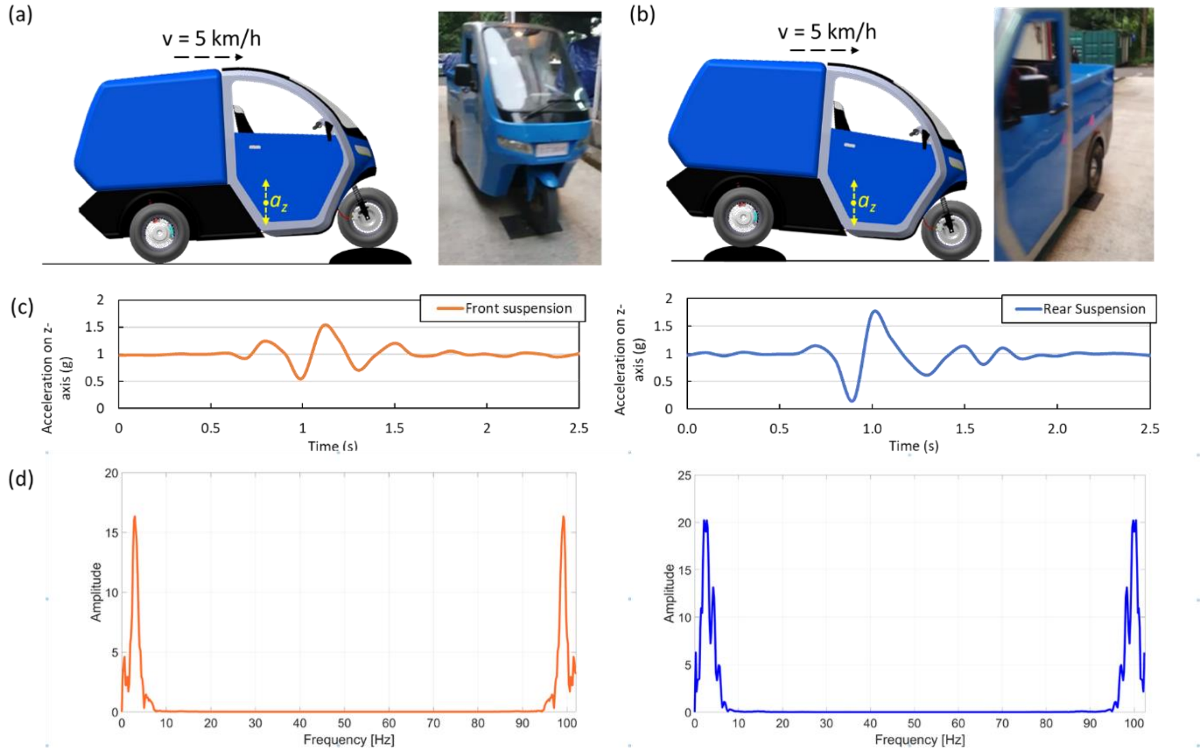


Figure 2. Schematic and photograph: (a) front suspension, (b) rear suspension evaluation, (c) acceleration curves recorded in sensor, and (d) Fourier transformation of the recorded acceleration

Shannon's Sampling theorem, since the Nyquist frequency is higher than the natural frequencies. The sampling frequency was determined based on the average over time and the number of samples obtained from the test data. Evaluation of suspension characteristics was carried out using a simplified quarter-car model, where the dynamics of the vehicle suspension was considered to have two degrees of freedom consisting of the sprung mass (M) representing the vehicle body component and the unsprung mass (m), in this case representing the wheels and suspensions of the vehicle. The sprung mass and the unsprung mass are related to the suspension stiffness (Ks) and the suspension damping coefficient (Cs). Meanwhile, the wheel of the vehicle which in this case connects the unsprung mass with the road has stiffness (Kt), which was assumed to be negligible in every driving condition [17].

Based on the quarter-car model, the value of the logarithmic decrement (δ) was determined using Eq. 1 from the oscillation data obtained from measurements of the initial wave crest (X_0) and the nth wave crest (Xn) on the number of waves observed (n) obtained from the measurement curve in the time domain. Then the damping ratio (ξ) was determined using Eq. 2 as the relationship between logarithmic decrement (δ) and the system that works on the observed data.

$$\delta = \frac{1}{n} \cdot ln\left(\frac{X_0}{X_n}\right) \tag{1}$$

$$\xi = \frac{\delta}{\sqrt{(4 \cdot \pi^2 + \delta^2)}} \tag{2}$$

Calculation of suspension stiffness (K) was obtained using Eq. 3, where f_n is the natural frequency of the suspension. The suspension damping coefficient (C) was obtained using Eq. 4 as the relationship between suspension stiffness (K) and damping ratio (ξ). Eq. 3 and Eq. 4 use the specific weight (M_s) obtained from the measurement of the load acting on each suspension system.

$$K = 4 \cdot \pi^2 \cdot {f_n}^2 \cdot M_s \tag{3}$$

$$C = 2 \cdot \xi \cdot \sqrt{K \cdot M_s} \tag{4}$$

The results of the suspension characteristics evaluation were obtained and presented in **Table 1**. The results of this evaluation were used as the basis for input in multibody dynamic simulations which greatly influence the multibody dynamic simulation developed for delta trike vehicles.

Table 1. E-Trike suspension characteristics

Parameter	Symbol	Value
Damping ratio of front suspension	ξ_d	0,20
Front suspension stiffness	K_d	80,29 kN/m
Damping coefficient of front suspension	C_d	183,83 N s/m
Damping ratio of rear suspensions	ξ_b	0,29
Rear suspensions stiffness	K_b	45,79 kN/m
Damping coefficient of rear suspensions	C_b	160,71 N s/m

Based on the results of the suspension characteristics test, it was found that the stiffness and damping coefficients were different for the front and rear suspension due to the use of different types of suspension. The front suspension of the e-trike prototype vehicle used a telescopic fork type which was commonly used in two-wheeled vehicles while the rear suspension used a dual shock dual swing arm type, which consisted of 2 arms equipped with 2 springs mounted on the rear axle.

Calculation of the damping ratio was carried out on the extension and compression cycle received by each suspension which was then calculated on the average. It was observed from the results of the average value of the front suspension damping ratio (ξ_d) and the rear suspension damping ratio (ξ_b) showed good results and were in the range of the ideal damping ratio category for reference passenger vehicles [18]. Mahyuddin and Nurprasetio [18] stated that the ideal damping ratio (ξ) for four-wheeled passenger vehicles is 0.2 – 0.4.

Body mass (M_B) of 485 Kg was measured by neglecting the mass of all wheels from the value of the prototype total mass (M) of 508 Kg. Experimental work was performed to acquire the center of gravity by replicating the free body diagram of the prototype as indicated in **Figure 3**. The center of gravity in the X-axis and Y-axis was determined by direct measurement on the prototype as shown in **Figure 3a** and **Figure 3b** respectively. The center of gravity location in the X-axis was calculated from the position of force applied on the rear left wheel (F_L) to the location of prototype weight (W). The location of the center of gravity in the Y-axis was calculated from the location of the force applied in the front wheel (F_F) to the rear wheel indicated as the total force of the rear left wheel and rear right wheel ($F_L + F_R$).

The location of the center of gravity in the Z-axis was calculated from the ground level, and thus required several adjustments. The assessment of the center of gravity in the Z-axis was performed by indirect measurement of the applied variation of height (H_1, H_2, and H_3) on the prototype as shown in **Figure 3c**. The implication of variation of height leads to the observation of variant values of angle (θ), front-wheel force (F_F'),rear-left wheel force (F_L'), and rear-right wheel force (F_R'). Results of the center of gravity experimental work were 240.1 mm, 16 mm, and 380.4 mm from x, y, and Z-axis origin, respectively. Experimental work on the center of gravity indicated that the prototype was not balanced in the Y-axis, which was identified from different forces observed in the rear left wheel and the right rear wheel. The unbalance might occur because the positions of electric components were not distributed properly. This issue might significantly influence the stability of the e-trike. Considering the electric components can be positioned with flexibility, it was strongly recommended to rearrange the electric components to achieve balance conditions on the Y-axis.

2.2. *Evaluation of Lateral and Rollover Stabilities*

Stability and maneuver test of the prototype vehicle was needed to ensure the prototype vehicle was in a stable condition when performing maneuvers before entering the vehicle commercialization stage. This was deemed necessary to reduce the risk of anything unwanted happening when the vehicle is operating. The results of the validation will later be used as guidelines when compiling operating standards and vehicle use. The tests were carried out experimentally on the e-trike prototype vehicle by performing single lane-change test and double lane-change test (see **Figure 4**).

The single-lane change test was conducted to evaluate the maneuvering characteristics of the e-trike prototype vehicle. In principle, the test was carried out by running the vehicle on the track by

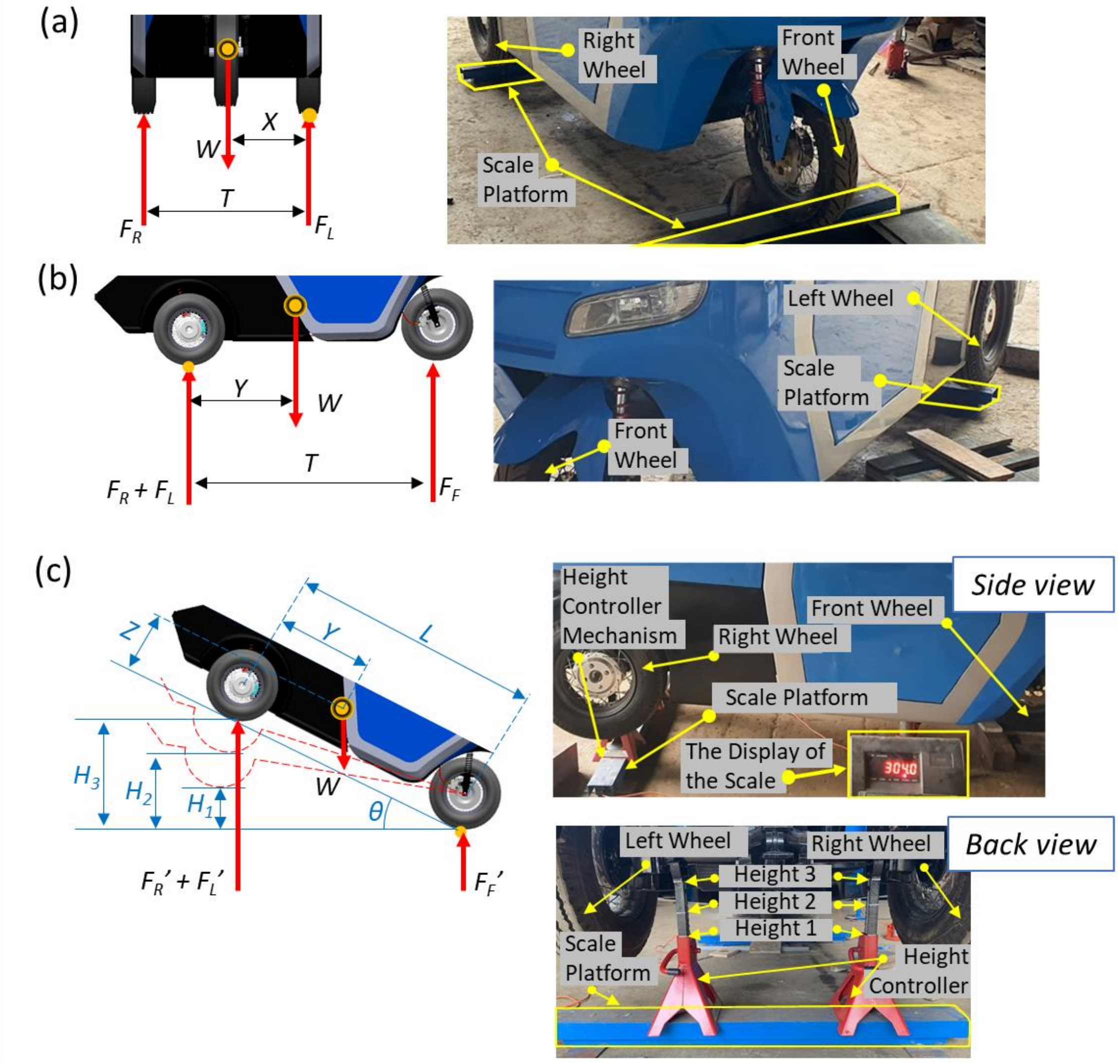


Figure 3. Free body diagram and photograph for evaluating center of gravity: (a) X-axis, (b) Y-Axis, and (c) Z-axis.

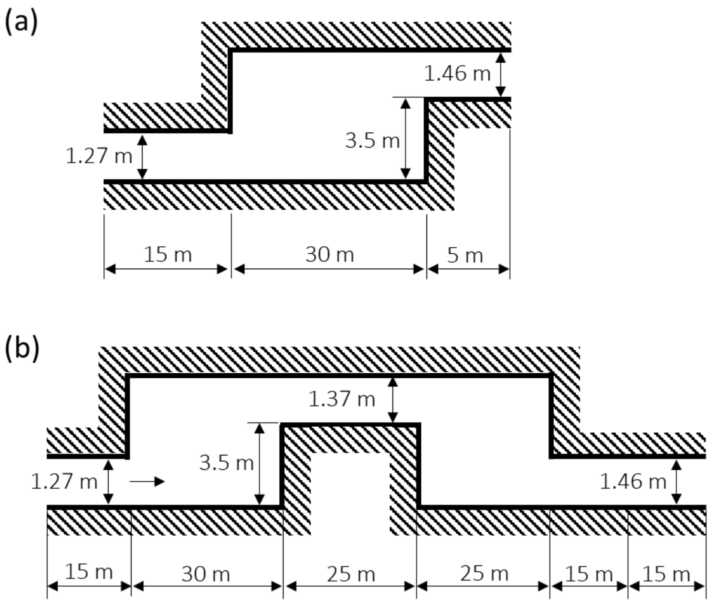


Figure 4. Schematic testing roads: (a) single lane change ISO 14791:2000 and (b) double lane change ISO 3888-1:2018.

moving once to the left lane, then staying in the same lane until the completion of the test. Figure 4a shows the path used when performing a single-lane change test. The e-trike speed was kept constant during maneuvering single-lane change. The data of the test was collected using the WitMotion HWT901B sensor, in which the evaluation of the test results, observations were made on the lateral acceleration data on the Y-axis (A_Y) and yaw rate on the Z-axis (ω_Z).

The double-lane change test was carried out based on the ISO 3888-1:2020 document, where the vehicle changes lane to the left lane, stayed in the lane for a while, and then returns to the starting lane. Figure 4b shows the path used when performing the double-lane change test. The e-trike vehicle speed was controlled by holding the gas throttle position steady using a mark prepared on the throttle in the steering system of the e-trike prototype vehicle. The double-lane change test data was collected using the WitMotion HWT901B sensor, the evaluation of the test results was made by observing the lateral acceleration data on the Y-axis (A_Y) and yaw rate on the Z-axis (ω_Z).

The rollover stability condition was defined by calculating the rollover index (RI) using Eq. 5 where the FR and FL are the vertical force of the right wheel and left wheel, respectively. Kazemian et. al. [19] analyzed vehicle rollover condition by operating a right-hand turnabout trial on threshold velocity when the left wheel lost its vertical force, or FL equal to zero ($F_L = 0$) and RI was defined equal to one.

$$RI = \frac{F_R - F_L}{F_R + F_L}; \quad -1 \le RI \le 1 \qquad (5)$$

The acceptable condition of RI is reached if the value is in the range between minus one to plus one ($-1 \le RI \le 1$) as observation results from the presence of vertical forces applied at the left and right wheels. Condition outside the range value of RI indicates the system has lost vertical force on one of the wheels. The closer the value of RI to zero, i.e. the vertical force of the left wheel is approximately the same as the vertical force of the right wheel, indicates higher rollover stability of a vehicle.

2.3. *Multi-Body Dynamic Simulation*

The results of single-lane change dan double-lane change tests were utilized to validate the numerical simulation model of e-trike dynamics to guarantee its accuracy. The purpose of this simulation was to investigate the effect of suspension factors and carried freight mass on the stability of e-trike especially the threshold velocity by conducting a parameter study. The simulation was conducted using a multi-body dynamic software named SIMPACK 2020. A multi-body dynamic model of e-trike was developed based on a topology as shown in Figure 5. The e-trike vehicle model consists of three main components as the main structure of the vehicle model formation: 1) a vehicle body component which had 6 degrees of freedom, 2) a suspension component which had 2 degrees of freedom for the front suspension and 1 degree of freedom for the rear suspension, 3) wheel components of the vehicle which had 1 degree of freedom each. All components were connected using the connections set in the system software. Several joints which locked different numbers of degrees of freedom were defined in the vehicle model depending on the interactions within the structure of each component. The relationship between the wheel and the road uses the Pacejka Similarity type [20], which has taken into account the stiffness and damping ratio of 200 kN/m and 75 Ns/m, along with other parameters. Research by Eichberger and Hofmann [21] shows the wheel relationship and becomes the main factor in conducting vehicle modeling in multibody dynamic simulations.

Figure 6a shows the formation of a 3D e-trike model in a dynamic multibody simulation. The parameters used as inputs in the multibody dynamic simulation were based on data obtained from the results of tests carried out on the prototype of the e-trike vehicle. These parameters include the dimensions of the vehicle, the location of the vehicle's center coordinates, the center of gravity, the value of the vehicle suspension stiffness and damping coefficients. In addition, the speed parameters and the shape of the cartographic trajectory in the simulation match the parameters obtained when testing single-lane change and double-lane change as shown in Figure 6b and Figure 6c, respectively.

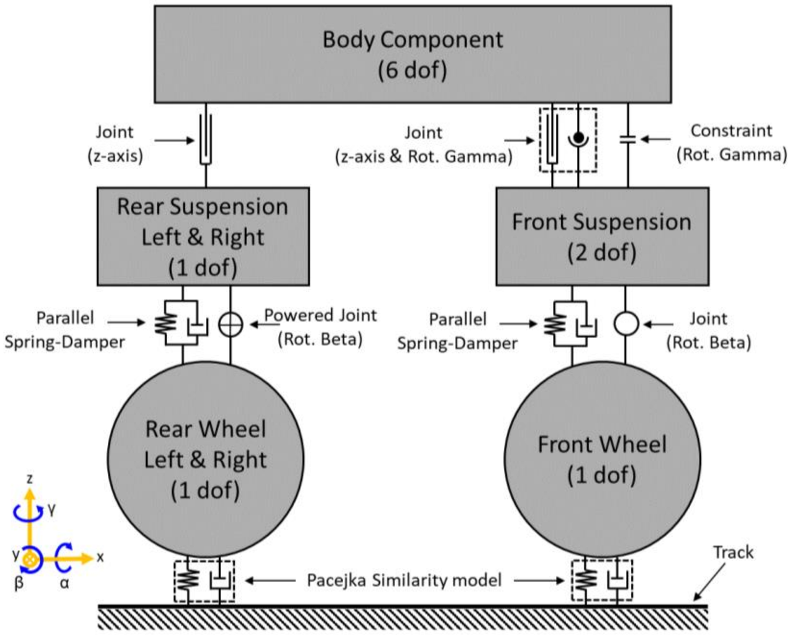


Figure 5. Model topology on the software

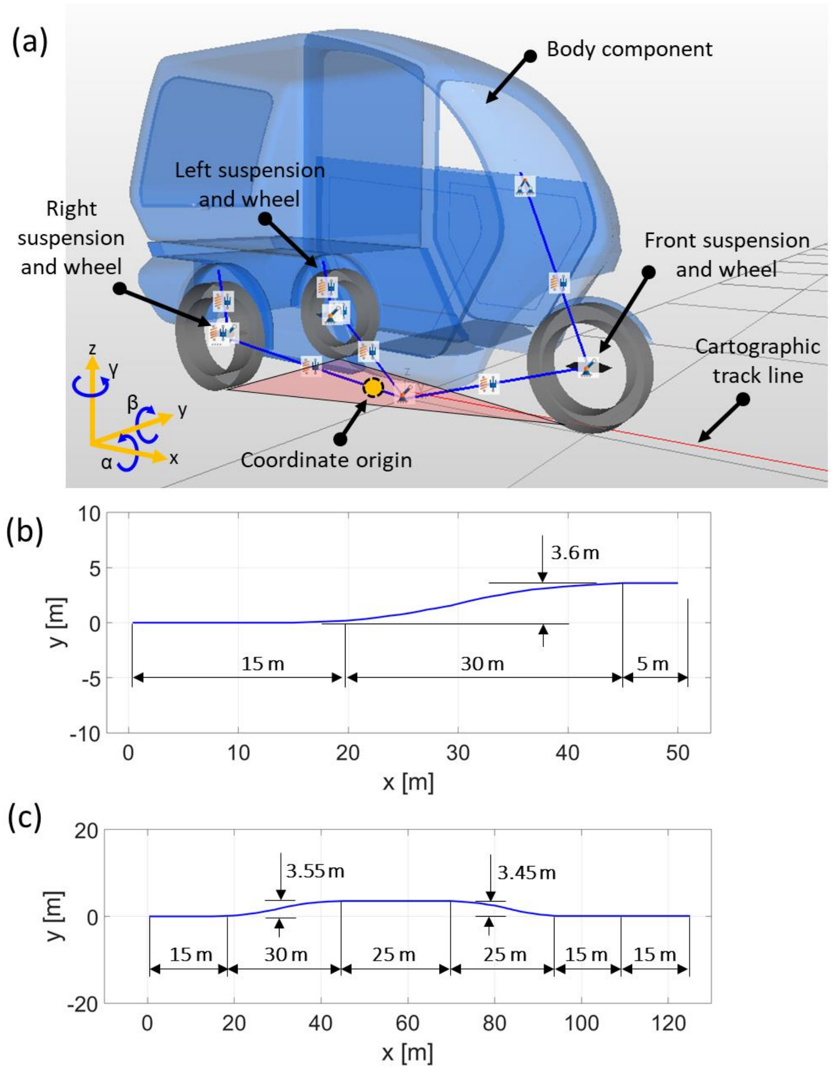


Figure 6. The multi-body simulation: (a) E-trike model, (b) single lane-change (ISO 14791:2000), and (c) double lane-change ISO 3888-1:2020

3. Results and Discussion

3.1. Experimental Validation

The experimental and simulation results of e-trike performance tests of Single Lane-Change and Double Lane-Change were compared to validate the simulation accuracy. Validation is an important step since the simulation was intended to perform a parameter study to investigate the effect of suspension and freight weight on e-trike performance. The parameters being compared were longitudinal velocity, lateral acceleration, and yaw rate. These parameters represent the stability of e-trike during a maneuver. The results of the single lane-change test are shown in **Figure 7**.

Figure 7a shows the comparison of e-trike longitudinal velocity during the maneuver in the single lane-change test and simulation. The simulation longitudinal velocity plotted in the figure is an approximation to the experimental longitudinal velocity and was used as the simulation input. **Figure 7b** and **Figure 7c** show the e-trike lateral acceleration and yaw rate during the test. From both figures, it can be seen that the experiment and simulation results had a good agreement. The experiment started having fluctuation which was suspected due to the vehicle that initially had not reached the steady state. The slight difference in amplitude might be caused by the difference in steering behavior between the driver in the experiment and the algorithm in the simulation. The vehicle reached steady state in terms of lateral acceleration and yaw rate after performing single lane-change which indicated good lateral stability.

The experimental and simulation results of the Double Lane-Change test are shown in **Figure 8**. The longitudinal velocity of e-trike during double lane-change maneuver is increasing and approximated as linear for the simulation input as

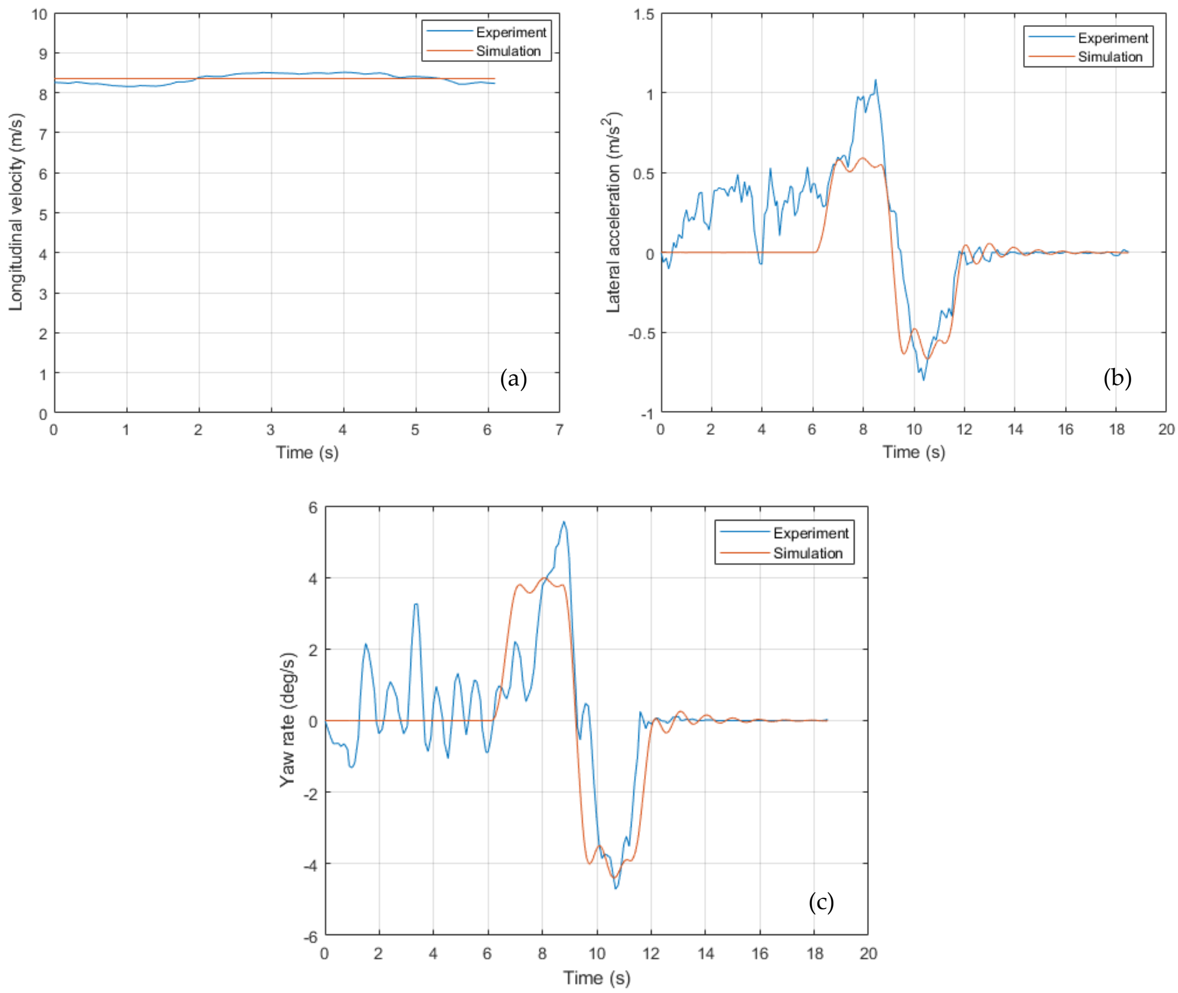


Figure 7. Single lane-change experiment vs simulation results: (a) longitudinal velocity, (b) lateral acceleration, and (c) yaw rate

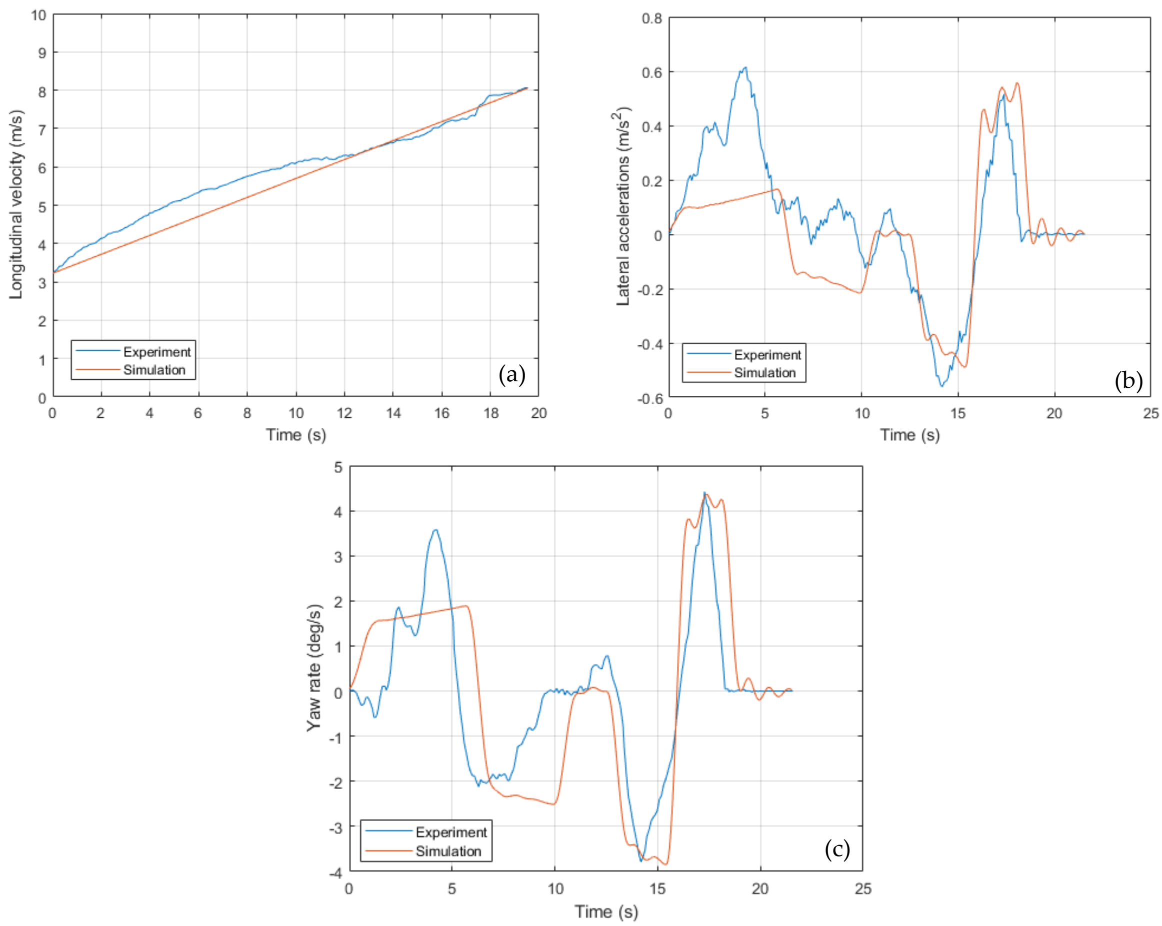


Figure 8. Double lane-change experiment vs simulation results: (a) longitudinal velocity, (b) lateral acceleration, and (c) yaw rate

shown in **Figure 8a**. The lateral acceleration and yaw rate of the Double Lane-Change test from experiment and simulation also showed a good agreement as shown in **Figure 8b** and **Figure 8c**. Both parameters reached steady state after the maneuver which indicated good lateral stability.

The results of the simulation were in good agreement with the experiment results, both Single Lane-Change and Double Lane-Change, which meant the simulation model is validated and can be used for further study of the characteristics of e-trike. In the following section, the simulation model was utilized to perform a parametric study to investigate the effect of suspension factors and freight mass on e-trike stability.

The validated simulation model of e-trike was analyzed for its rollover stability during Single Lane-Change and Double Lane-Change by its rollover index which was calculated using Eq. 5. The results are shown in **Figure 9**. The rollover stability of e-trike during steady state is shown by the dashed red line which is not zero since the center of gravity is slightly on the left side of e-trike due to the placement of e-trike components. The rollover index during Single Lane-Change and Double Lane-Change were considerably low meaning the vertical force of the left and right wheels were close and indicated good rollover stability.

3.2. *Parametric Study*

A parametric study was done to investigate the effect of suspension stiffness and damping coefficient and freight mass on e-trike stability which was represented by threshold velocity. The threshold velocity used here was the velocity just below a velocity in which one of the wheels lost its grip off the road, i.e. vertical force reach zero during Single Lane-Change and Double Lane-Change maneuver. The suspension stiffness and damping coefficients were varied into 80%, 100%, and 120% from their original value to determine the effect of suspension stiffness and damping coefficients on e-trike stability. The stiffness and damping coefficient values are shown in **Table 2**.

Besides the suspension factors, the freight mass carried by e-trike was also varied to determine its effect on threshold velocity since e-trike was designed for goods transportation. The freight was placed in the back trunk and varied into 0 kg, 50 kg, 100 kg, 150 kg, and 200 kg hence affecting the overall e-trike center of gravity. The freight mass and the corresponding center of gravity position are shown in **Table 3**.

The parametric study was performed in two parts as shown in **Figure 10**. First, the threshold velocity was observed for the varied suspension stiffness and the freight mass as the damping coefficient was set to constant. The results of the first parametric study are shown in **Figure 10a**. It can be seen that the freight mass was inversely proportional to the threshold velocity, and higher suspension stiffness coefficient led to higher threshold velocity. The second part of the parametric study was to observe threshold velocity while the damping coefficient and the freight mass varied. The results of the second parametric study are shown in **Figure 10b**. The freight mass effect was consistent with the first parametric study, while the change of damping coefficient had a smaller effect on threshold velocity compared to the stiffness coefficient. Heavier freight mass led to higher center of gravity which results in higher moment and more weight shift during a turn. These make the vehicle more prone to rollover, and therefore lower the threshold velocity. Higher suspension stiffness and damping coefficients make the suspension faster in responding to the weight shift during a turn and therefore give better support and stability to the vehicle, and thus increasing the threshold velocity. The parametric study using the simulation model facilitate the calculation for the complex multibody system of e-trike and successfully gave the exact amount and description of the phenomenon.

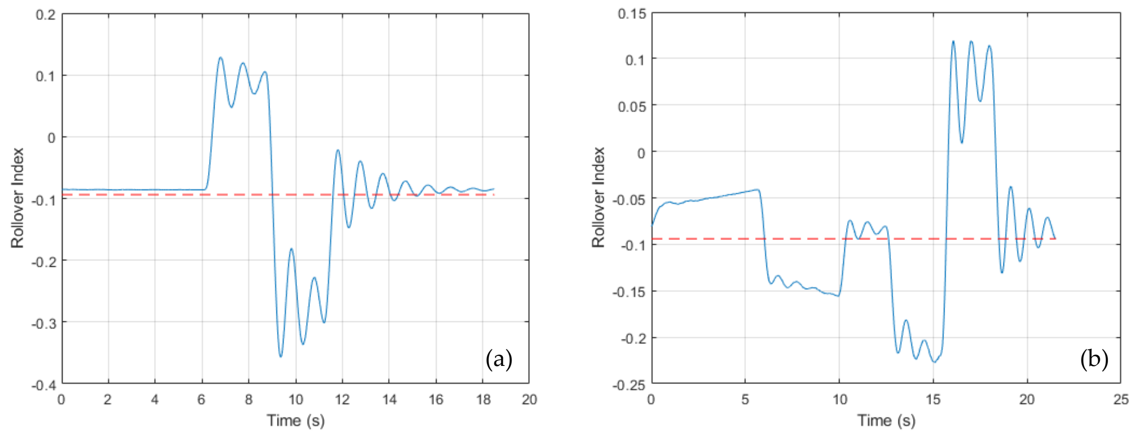


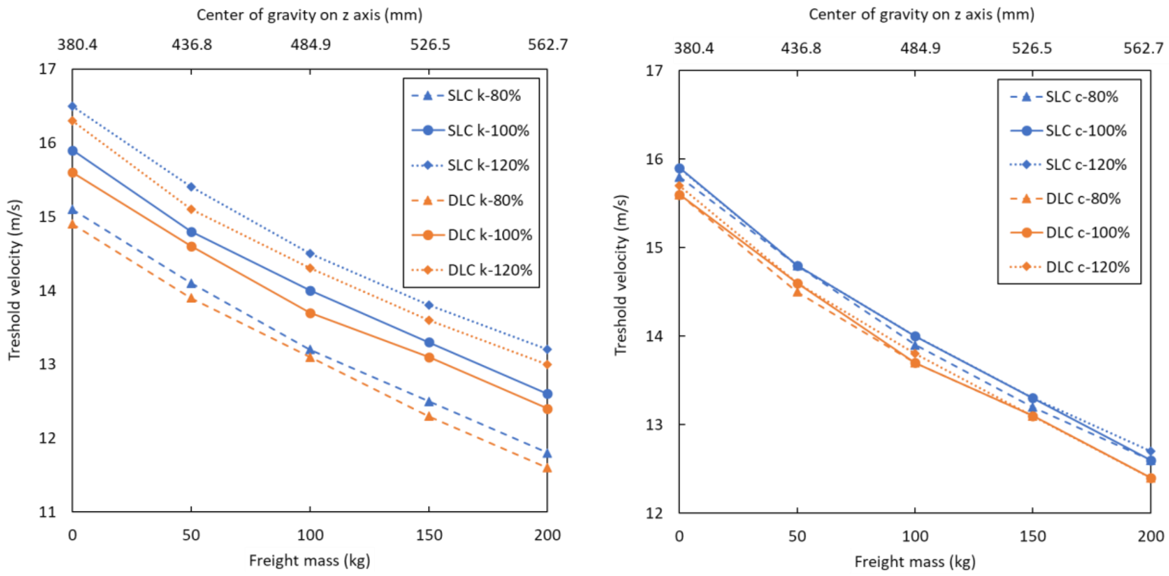
Figure 9. The multi-body simulation rollover index: (a) single lane-change and (b) double lane-change

Table 2. The variation of suspension stiffness and damping coefficients

Wheel	Variable Parameter					
	k (N)			c (Ns/m)		
	80%	100%	120%	80%	100%	120%
Front wheel	64 231.20	80 289.00	96 346.80	147.07	183.83	220.60
Right wheel	36 630.28	45 787.85	54 945.42	128.56	160.71	192.85
Left wheel	36 630.28	45 787.85	54 945.42	128.56	160.71	192.85

Table 3. The variation of freight mass and corresponding center of gravity

Freight mass (kg)	Center of Gravity (mm)		
	x	y	z
0	240.1	16.0	380.4
50	185.9	14.8	436.8
100	139.7	13.7	484.9
150	99.7	12.8	526.5
200	64.9	11.9	562.7



(a) (b)

Figure 10. Parametric study of freight mass: (a) stiffness and (b) damping coefficient to threshold velocity

4. Conclusion

Experiments and simulations of Single Lane-Change and Double Lane-Change based on ISO 14791:2000 and ISO 3888-1:2018 had been conducted to test e-trike stability and maneuverability. The lateral acceleration and yaw rate which represent e-trike stability during maneuver were chosen as comparison parameters for experiment and simulation. The simulation results showed a good agreement with the experiment hence validated and utilized for parametric study to investigate the effect of freight mass and suspension factors on e-trike stability. It was found that e-trike stability was decreasing as the freight mass increased. The suspension factors also considerably affect e-trike stability. Higher suspension coefficients resulted in higher e-trike stability. The stiffness coefficients are more significant in affecting e-trike stability compared to the damping coefficients.

The results of this study indicated several parameters as a guideline for improving stability during the design process of the e-trike. The prototype had the advantage of using the electrical infrastructure for the main power sources, which allows rearranging the electrical module to get the appropriate location of the center of gravity as evaluated in this paper. Arrangement guidelines also may be prepared as stability anticipation during operational e-trike as goods delivery. Threshold velocity may be proposed as the consideration for the handling guidelines of the e-trike and as information for related government parties to establish corresponding regulations. Following the improvement of e-trike stability, the analysis regarding the intended efficiency and reliability improvement for e-trike mass production system is considered for future studies.

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Author's Declaration

Authors' contributions and responsibilities

The authors made substantial contributions to the conception and design of the study. The authors took responsibility for data analysis, interpretation and discussion of results. The authors read and approved the final manuscript.

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Availability of data and materials

All data are available from the authors.

Competing interests

The authors declare no competing interest.

Additional information

No additional information from the authors.

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