

THE DESIGN OF A FORESTRY CHASSIS WITH AN ARTICULATED BODY WITH THREE DEGREES OF FREEDOM

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Carrying equipment into a forest is a difficult task for the common vehicle chassis because of the complex terrain conditions, which directly restricts the development of mechanization in the forests of southern China. In the present study, the design requirements for a vehicle chassis are analyzed according to the geomorphic conditions in the forests of southern China. A forestry chassis with an articulated body with three degrees of freedom is proposed. The performance when negotiating obstacles is enhanced by the articulated structure of chassis because the structure can achieve yawing, rolling, and pitching motion between the front and rear frames and the chassis can adapt to the complex variation of terrains timely and automatically. Furthermore, the wheels are also optimized by proposing a new method of maximizing the traction force and minimizing the sinkage. The results of the calculation of the turning radius, lateral stability and longitudinal stability indicate that this newly designed chassis could meet the requirements of the design well and adapt to the complicated roads in the forests of southern China.

Key words: all-wheel drive, articulated structure, forestry environment, wheel optimization.

1. Introduction

The forest terrains are different between the south and the north in China. The terrains of the northern forests are plain, which has a high level of mechanization. In contrast, the southern forests are mountainous, and conventional vehicles are difficult to drive in such forests. Therefore, it is particularly important to develop special vehicles that can adapt to the relevant terrain [1, 2]. In general, special vehicles are based on the following types of chassis: tracked, legged and special wheeled. The advantages of the tracked chassis are off-road crossing performance and driving stability. However, vehicles with tracked chassis cause significant damage to the soil and surrounding plants [3]. Another alternative to improve traction and adaptation to an uneven terrain is locomotion by legs; however, these

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vehicles have two major inherent drawbacks. The first drawback is the complexity of construction and control, and second is the very high energy consumption compared to a wheeled vehicle [4]. Vehicles with special wheeled chassis are more flexible and less troublesome than others, and trafficability can be improved by developing a key structure of body [5]. Therefore, the special wheeled chassis are more suitable for vehicles that must move on the rough terrains of a forest.

There are numerous papers in the literature regarding the design of special wheeled chassis. A variable ground clearance, variable wheel track self-leveling hillside vehicle power chassis (V2-HVPC) was designed by Beihang University [6]; in addition, the steering system design and stability analysis of V2-HVPC in rough terrains were also developed [7, 8]. The design of a lunar rover with a planetary wheel for surmounting obstacles can be found in the literature; in particular, a torsion bar spring with a magnetic spring absorber and planetary wheels are employed to surmount obstacles [9]; in addition, a high performance profiling terrain chassis was designed by Beijing Institute of Technology to enhance the adhesive force to terrain when operating in a hilly area, and the parameters of the chassis were calculated and analyzed [10].

Aside from the studies of the whole wheeled chassis, other methods have been adopted in the field to improve the performance by developing a key structure of body. Swingarms are simple mechanisms that ensure the contact of the wheels even on uneven terrain. A research study that showed excellent performance for swingarms was performed by X. Potau and M. Comellas [3, 4]; however, the chassis with swingarms were found to have an adverse effect on steering slippage, which causes serious wear to tires [11]. Considering that traditional four-wheel vehicles cannot cross high obstacles, Gengyu Ge and Junyu Wang proposed a model of a quadruped eccentric wheel-legged chassis, which enhanced the performance when crossing obstacles, even exhibiting the ability to climb stairs [12]; however, the low speed of the chassis has an effect on transportation. Articulated chassis have the characteristic of a small radius of steering, which is suitable for work in narrow spaces; however, a vehicle equipped with a conventional articulated structure similar to common vehicles cannot easily adapt to rough roads [13, 14].

Special wheeled chassis are widely applied in space exploration and agricultural applications, and research studies on such chassis mainly concentrate on improving the performance when climbing obstacles. As a result, the analysis of steering, climbing and stability is relatively scarce. At the same time, comparing southern China with a high level of mechanization country such as the United States, the difference between the forest terrains is large, which makes it necessary to improve the chassis application in the special terrains in southern China. Based on the above reasons, the current research into wheeled chassis suitable for the special terrains of southern forest of China is found to be

insufficient. Thus, in this paper, a forestry chassis with an articulated body with three degrees of freedom (3-DOF) is designed for the mountains and hills in the forests of southern China. A 3-DOF articulated chassis could achieve yawing, rolling and pitching motion between the front and rear frames; therefore, the chassis can adapt to the complex variation of terrains in the forests of southern China and is able to maintain good adhesion between the ground and the wheels, which means that it can overcome the slippage of wheels, thereby achieving a high trafficability on complex roads. In addition, the parameters of the wheels were optimized by proposing a new method of maximizing the traction force and minimizing the sinkage. The 3-DOF articulated chassis can be used not only for transport or carrying tools in a forest but also to install small items of mechanical equipment to replace manual operations, which helps to reduce the amount of physical labor and realize mechanization in the forests of southern China.

2. Design requirements

2.1 Design requirements for the size of the chassis

The proposed design requirements for the chassis in this article are based on the forest areas in Fujian province. The average gradient of the forest land is 26.6° in Fujian, according to the Forest Management Inventory in 1997 [15]. The ratio of forest land with different gradients to the total forest land in Fujian province is shown in Table 1.

Table 1

The ratio of forest land with different gradients to the total forest land in Fujian province

gradient scale	Unit of area: hm^2					
	I	II	III	IV	V	VI
area	124787	483087	3068291	4566075	976981	144371
ratio	1.33	5.16	32.77	48.77	10.43	1.54

(where: I $\leq 5^\circ$, II $6^\circ \sim 15^\circ$, III $16^\circ \sim 25^\circ$, IV $26^\circ \sim 35^\circ$, V $36^\circ \sim 45^\circ$, VI $\geq 46^\circ$)

As seen from Table 1, a gradient of less than 35° occupies 88.03% of the total forest area. Meanwhile, V ($36^\circ \sim 45^\circ$) and VI ($\geq 46^\circ$) is not suitable for mechanical operations as a result of the steep slope. Hence, the gradeability of chassis is up to 70% in this design. Moreover, the desired size of the chassis is 1600 mm in width and 3500 mm in length because the chassis is required to pass through the forest at a certain angle instead of vertically among 3000 mm row spacing and 5000 mm line spacing.

2.2 Design requirements for the turning radius

The vehicle makes a turn perpendicular to DE through tree A and tree E. At this time, the turning radius is $OA = \frac{AB}{\sin a} \approx 10000\text{mm}$, while $\cos a = \frac{DE}{AE} = \frac{2}{3}$ (shown in Fig. 1), where DE=2000 mm, AB=7500 mm (1.5 times the line spacing), d for a row spacing equal to 3000 mm, L for a line spacing equal to 5000 mm.

Turning is efficiently completed every third row when the radius is 10000 mm. Hence, the turning radius is limited to less than 10000 mm.

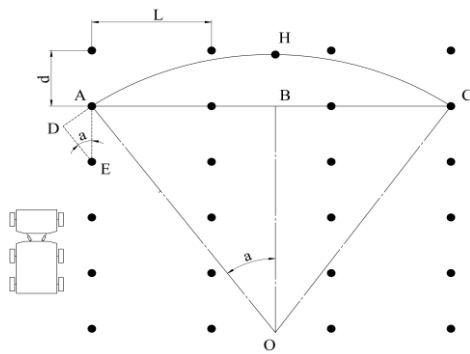


Fig. 1. Schematic of turning in the forest

Design parameter of chassis			
length	width	turning radius	gradeability
3500	1600	≤ 10000	70%
mm	mm	mm	

The design requirement for the equipment chassis is listed in Table 2, based on the above analysis.

3. Chassis

The primary structure of the chassis includes the front/rear asymmetric frame, leaf spring suspension, drive axle and articulated structure with 3-DOF. All-wheel drive is achieved by the powertrain, which transfers power produced by the engine to each wheel. The rated power of engine is 15.2 kW. In addition, the chassis is 1530 kg in mass (maximum load is 1.2t), 3500 mm in length and 1600 mm in width. The average velocity is 10 km/h on a smooth road. An assembly drawing of the chassis with Solidworks is shown in Fig. 2.

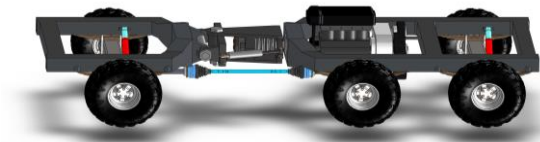
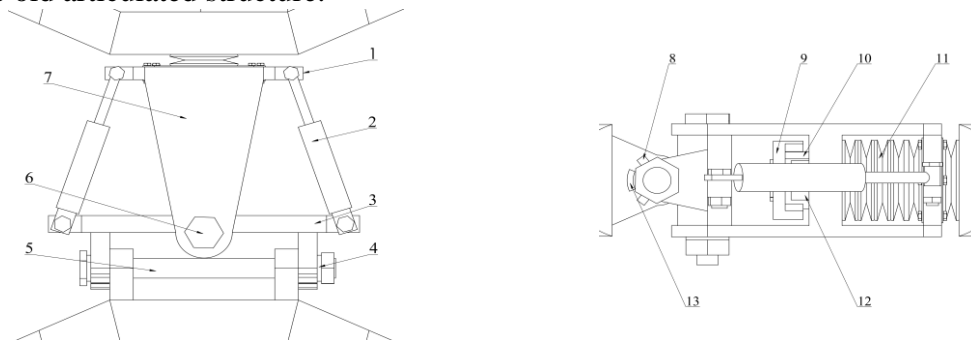


Fig. 2. Three-dimensional schematic of the chassis

4. Steering system and optimization for the wheels

4.1 3-DOF articulated structure

Reducing the radius of turning to increase flexibility as well as improving the adaptation to the rough terrain to enhance the vehicle's ability to surmount obstacles are two primary considerations [16, 17]. A new steering structure is proposed on this basis. More freedom is achieved between the front and rear frame in this structure, which addresses the lack of a single degree of freedom in the old articulated structure.



1. link span; 2. hydraulic cylinder; 3. articulation yoke; 4. elastic washer; 5. connecting pin; 6. articulated joint pin; 7. cover plate; 8. stop block; 9. stop rear cover; 10. stop block; 11. belleville spring; 12. nut; 13. stop block

Fig. 3. Schematic of the articulated structure with three degrees of freedom

Fig. 3 shows the 3-DOF articulated structure that is able to adjust itself in multiple directions and multiple angles between the front and rear frames. In this structure, the straight pin connects the rear frame with the link span 1, where a through hole is opened. The free end of the straight pin drills through the hole with a bearing equipped in it. Rolling motion is achieved by the relative rotation between link span 1 and the straight pin. The front frame rotates by means of link span 1 when a one-sided obstacle appears; meanwhile, the rear frame maintains its original motion. The independence of movement between the front frame and rear frame contributes to the stabilization and adaptability of the chassis (shown in Fig. 4). The connection between articulation yoke 3 and connecting pin 5 could achieve pitching motion. The comparison between the two structures when climbing is shown in Figure. 5. The chassis turns when articulation yoke 3 spins around articulated joint pin 6 during the movement of hydraulic cylinder 2.

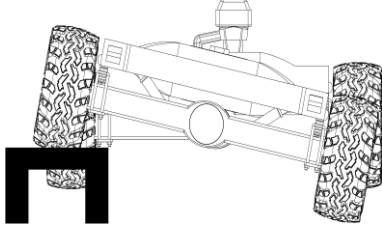
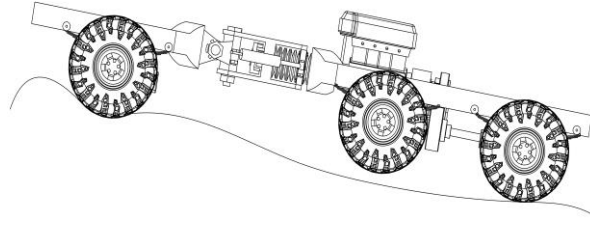
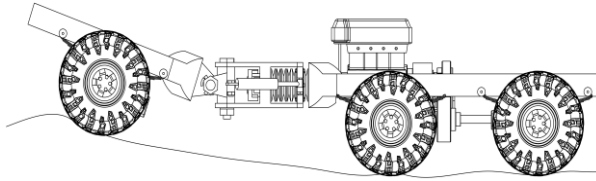


Fig. 4. Schematic during one-sided obstacle surmounting



(a) structure with a single degree of freedom
Fig. 5. Schematic of the chassis passing over bumps on a rough road



(b) structure with three degrees of freedom

Fig. 5. Schematic of the chassis passing over bumps on a rough road

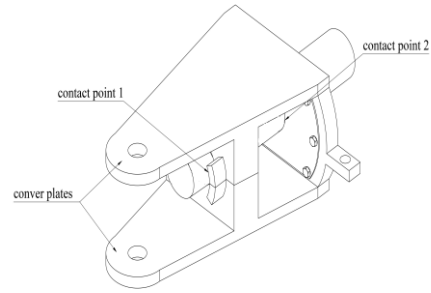


Fig. 6. Cover plate

The stop blocks limit both the angle of rolling to 30° and the angle of pitching to 35° to avoid large ups and downs (35° stems from 70% gradability). The rolling motion in the rear frame stops when rear cover 9 touches stop block 10. Similarly, pitching motion will not occur when stop block 8 touches block 13 (shown in Fig. 3). The cover plates, which have two contact points, limit the radial movement of the straight pin (shown in Fig. 6). Belleville spring 11 reduces vibration.

4.2 Optimization calculation for the wheels

Adequate traction is required for the vehicle to overcome resistance (including compaction resistance F_r and bulldozing resistance F_b) when traveling in the forest. Fig. 7 shows the details of a single-wheel vehicle model running on an inclined terrain with a slope angle of α . Assuming the vehicle is driving at a constant speed, the total force on the x-axis is balanced [6, 18, 19].

$$F_t - F_r - F_b - W_x = 0 \quad (1)$$

$$W_x = W \sin \alpha \quad (2)$$

Where,

F_t is the traction force.

W is the pressure on the wheel at full load.

W_x is the component pressure force of W in the x direction.

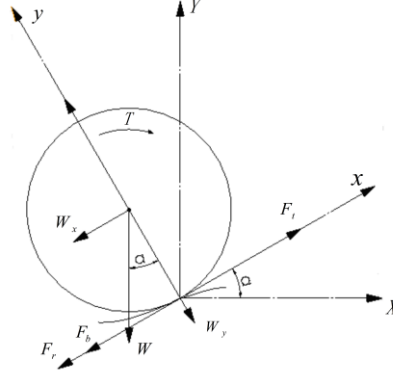


Fig. 7. Force analysis of the wheel when climbing

For simplicity, it is assumed that the tire is a rigid body and the road is elastic. Hence, the normal pressure acting on the wheel with a certain depth of soil sinkage is equal to the contact pressure on a plate at the same depth under the same loading condition; the sinkage of a rigid wheel in soft soil Z can be expressed as:

$$Z = \left[\frac{3W \cos \alpha}{(K_c + B \times K_\phi) \sqrt{D} (3-n)} \right]^{\frac{2}{2n+1}} \quad (3)$$

where,

K_c is the deformation modulus of cohesion;

K_ϕ is the deformation modulus of friction;

α is the slope angle of terrain; n is the soil deformation index number.

Equation (3) expresses the parametric relationship of the sinkage with the geometrical dimensions of the wheel, the width B , the diameter D , and the parameters of soil. As depicted in Equation (3), the larger and the wider the wheel, the smaller the sinkage will be.

Based on the simplified wheel model, the relationship of compaction resistance can be expressed as:

$$F_r = \frac{1}{(3-n)^{\frac{2n+2}{2n+1}} (n+1) (K_c + B \times K_\phi)^{\frac{1}{2n+1}}} \left[\frac{3W}{\sqrt{D}} \right]^{\frac{2n+2}{2n+1}} \quad (4)$$

When a wheeled vehicle gets into the field, it will shear the soil to obtain the necessary resistance to move forward, which develops the bulldozing resistance. The bulldozing resistance can be expressed as:

$$F_b = B(c \times Z \times K_{pc} + 0.5Z^2 \times \gamma_s \times K_{pr}) \quad (5)$$

Where,

$$K_{pc} = (N_c - \tan \phi) \cos^2 \phi;$$

$$K_{pr} = \left(\frac{2N_r}{\tan \phi} + 1 \right) \cos^2 \phi;$$

ϕ is the friction angle of soil;

N_c 、 N_r is the coefficients of soil bearing capacity.

According to this model, the bulldozing resistance increases rapidly with an increase of the wheel width, which means it is better to select a narrower wheel to decrease the bulldozing resistance.

To confirm the configuration of locomotion, an equation was deduced considering all essential factors of the wheel and the soil. When substituting Equation (2) through Equation (5) into Equation (1), the parameterized configuration equation can be expressed as:

$$F_t = \frac{1}{(3-n)^{\frac{2n+2}{2n+1}}(n+1)(K_c + b \times K_\phi)^{\frac{1}{2n+1}}} \left[\frac{3W}{\sqrt{D}} \right]^{\frac{2n+2}{2n+1}} + b(c \times Z \times K_{pc} + 0.5Z^2 \times \gamma_s \times K_{pr}) + W \sin \alpha \quad (6)$$

Equation (6) shows the function of traction force, the parameters of the wheel, and the properties of the soil conditions. To select the appropriate wheels for the chassis, Equation (6) was optimized using Matlab Optimization Toolbox and considering the constraint equations. The conditions of the forest soil and the availability of wheel dimensions that are constrained by the structure of the wheel and the feasibility of the wheel market are described below:

$$0 \leq \frac{Z}{D} \leq 1, \quad 2 \leq N \leq 8, \quad 0 \leq D \leq 1500, \quad 0 \leq B \leq 500$$

Where, N is the number of wheels.

Table 3

Initial parameter list

Parameter	Unit	Value
K_c	kN / m^{n+1}	16
K_ϕ	kN / m^{n+1}	1262
α	°	30
n		0.6
W	kN	6.499
ϕ	°	13
N_c		10
N_r		6
γ_s	kN / m^3	26.52

Table 4

Optimization results

Tire selection	Calculation
B	160 mm
D	457 mm
Z	13.12 mm
F_t	4310 N

The objective of this optimization function is to maximize the traction force F_t and minimize the sinkage Z . Under the conditions of a rigid wheel in the southern forest area of China, the initial conditions and the terrain parameters are listed in Table 3 and the optimum results are listed in Table 4.

5 Analysis of rideability in the chassis

5.1 Calculation of the turning radius

Turning is accomplished by the extension and drawback of the hydraulic cylinder on both sides, resulting in deflection of the frame. Calculation of turning radius (shown in Fig. 8):

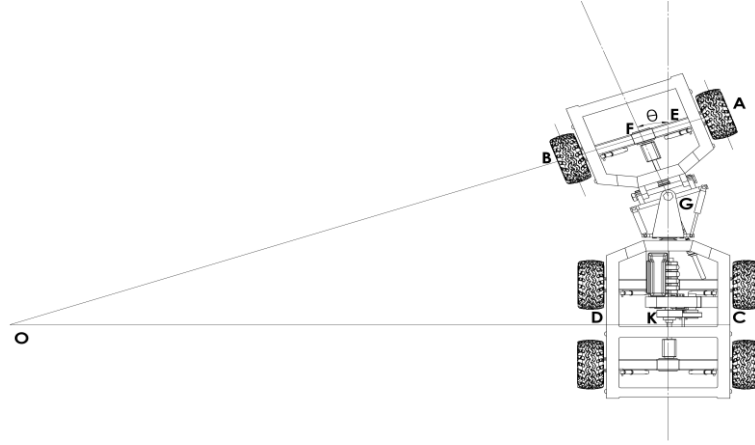


Fig. 8. Schematic of steering

$$EF = FG \tan \theta \quad (7)$$

$$EG = \frac{FG}{\cos \theta} \quad (8)$$

$$EA = \frac{1}{2} AB - EF \quad (9)$$

$$EK = EG + GK \quad (10)$$

$$OE = \frac{EK}{\sin \theta} \quad (11)$$

The turning radius is given by:

$$R = EA + OE \quad (12)$$

In the 3-DOF articulated structure, the maximum turn angle θ is 20° . In addition, the track $AB = 1600$ mm, $FG = 600$ mm, $GK = 1242$ mm. In this case, the turning radius is $R \approx 6080$ mm, which is smaller than the required 1000 mm.

5.2 The lateral stability of the chassis

Lateral stability refers to the reduction of the possibility of rollover or sideslip

when traveling on a lateral slope. Lateral stability is quantified by the critical angle of rollover and the critical angle of sideslip [7].

Rollover or sideslip induced by gravity will occur when the gradient of the lateral slope on which the vehicle is traveling is larger than a certain value. The ground reactive force on the right wheel is equal to 0 during rollover, and the critical angle of rollover satisfies (shown in Fig. 9):

$$hG \sin \alpha_{\max 1} = \frac{B}{2} G \cos \alpha_{\max 1} \quad (13)$$

Where, G is the force of gravity of the chassis

$$\alpha_{\max 1} = \arctan \frac{B}{2h} \quad (14)$$

The critical angle of rollover is dependent on the track B and height of the center of gravity h from equation (14). The center of gravity h of the vehicle is 339.68 mm (which could be rounded to $h=340$ mm), and $B=1600$ mm. We have $\alpha_{\max 1} \approx 67^\circ$ from the calculation. When sideslip occurs, the critical angle of sideslip $\alpha_{\max 2}$ satisfies:

$$\varphi G \cos \alpha_{\max 2} = G \sin \alpha_{\max 2} \quad (15)$$

φ is the coefficient of road adhesion

$$\alpha_{\max 2} = \arctan \varphi \quad (16)$$

It can be obtained from equation (16) that the critical angle of sideslip is dependent on the coefficient of road adhesion. The value of φ is 0.8 because forestry roads are usually dirt roads, and the value of $\alpha_{\max 2}$ calculated is 38° . As a result, $\alpha_{\max 2} < \alpha_{\max 1}$, which indicates that the road is passable by the vehicle when the gradient of the lateral slope is less than 38° .

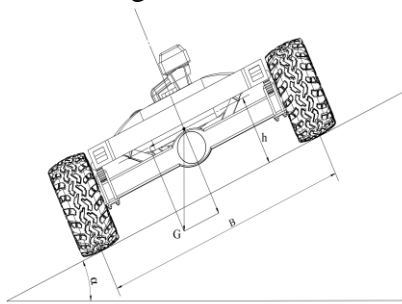


Fig. 9. Force analysis of vehicle on the cross fall

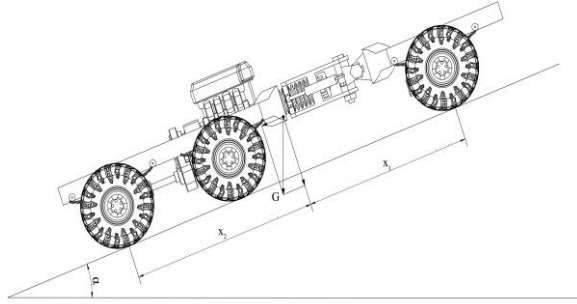


Fig. 10. Force analysis of vehicle on a longitudinal incline

5.3 The longitudinal stability of the chassis

Longitudinal stability is quantified by the critical angle of longitudinal turning and the critical angle of longitudinal-slip

When climbing (shown in Fig. 10), the critical angle of longitudinal turning $\alpha_{\max 3}$ satisfies:

$$hG \sin \alpha_{\max 3} = xG \cos \alpha_{\max 3} \quad (17)$$

Where, x is the distance between the center of gravity and the longitudinal turning point.

$$\alpha_{\max 3} = \arctan \frac{x}{h} \quad (18)$$

The critical angle of longitudinal turning is dependent on the distance x and the height of the center of gravity. The distance between the center of gravity and the front wheel is $x_1=1114\text{mm}$, and the distance between the center of gravity and the rear wheel is $x_2=1087\text{mm}$. Longitudinal turning is prone to occur when climbing as $x_1 > x_2$. We have $\alpha_{\max 3} = 71^\circ$ when $x = x_2$ and $h=340\text{ mm}$.

The brakes of all of the wheels are applied in the vehicle when the vehicle is parked on the slope, which causes longitudinal slip. The critical angle of longitudinal slip $\alpha_{\max 4}$ satisfies:

$$G \sin \alpha_{\max 4} = \varphi G \cos \alpha_{\max 4} \quad (19)$$

$$\alpha_{\max 4} = \arctan \varphi \quad (20)$$

The critical angle of longitudinal slip is dependent on the coefficient of forestry road adhesion. We have $\alpha_{\max 4} \approx 38^\circ$ when $\varphi = 0.8$. To avoid longitudinal slip when parking, the chassis is required to travel on slopes with gradients of less than 38° . This requirement satisfies the gradeability design (70%), which indicates the chassis is suitable for the southern forest area.

6. Conclusion

In this paper, a design requirement suitable for the terrain in the forest of southern China was proposed, and a 3-DOF articulated chassis was designed that could achieve yawing, rolling and pitching motion between the front and rear frames. This novel chassis is able to maintain good adhesion between the ground and the wheels, i.e., it can overcome the slippage of wheels, achieving a high trafficability in complex roads. In addition, the parameters of the wheels were also optimized via a new method of maximizing the traction force and minimizing the sinkage. The rideability of the chassis was analyzed at the end of the article. The 3-DOF articulated chassis was verified to satisfy the design requirement based on an analysis of its turning radius, gradeability and stability.

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