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DYNAMIC MODELING AND PREDICTION OF ROLLOVER STABILITY FOR ALL-TERRAIN VEHICLES

Haitao Luo[1,2,3], Zhimin Chen[1], Aryan Naveen[1], Bing Li[1]

[1]Department of Automotive Engineering, Clemson University International Center for Automotive Research (CU-ICAR), Greenville, SC 29607, USA
[2]State Key Laboratory of Robotics, Shenyang Institute of Automation, Chinese Academy of Sciences, Shenyang, 110016, China
[3]Institutes for Robotics and Intelligent Manufacturing, Chinese Academy of Sciences, Shenyang, 110169, China

ABSTRACT

With the particular passage capability, all-terrain vehicle (ATV) has been widely used for off-road scenarios. In this research, we conduct a lateral sway stability analysis for the suspension mechanism of a general vehicle and establish a mathematical model of static and dynamic stability based on the maximum lateral sway angle and lateral sway acceleration, by considering the combined angular stiffness of independent suspension, angular stiffness of the lateral stabilizer bar and vertical stiffness of tires. 3D point cloud data of a terrain environment is collected using an RGB-Depth camera, and a triangular topography map is constructed. The results in ADAMS show that the proposed stability model can accurately predict the critical tipping state of the vehicle, and the method deployed for real-world terrain modeling and simulation analysis is generalizable for the stability assessment of the interaction between ATV and real-world terrain.

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1. INTRODUCTION

All-terrain vehicle (ATV) maneuvering off-road terrains are susceptible to rollovers. Even the most experienced of human drivers fall prey to rugged terrains leading to rollovers. While some vehicle features like anti-roll stability help human-driven machines to tackle and prevent the problem of rollover for off-road terrains, they are not enough for autonomous vehicles. Therefore, autonomous vehicles need a different approach for preventing them from rolling over while deployed on any terrain in the world. The off-road autonomous vehicles need to perceive and evaluate the terrain surrounding them to navigate through them without tipping over.

Currently, according to the mathematical model building method, the calculation of the lateral leaning stability of a car can usually be divided into two types [1]. One method is to take the whole vehicle as a rigid body; another method is to consider the influence of the suspension. Maciej et al. [2] established a mathematical model of static side-tilt considering suspension angular stiffness and tire vertical stiffness and verified the model accuracy. Li et al. [3] derived a mathematical model of static side-tilt considering suspension side-tilt and different front and rear wheel spacing. Jadhao etc. [4] demonstrate the use of modeling and simulation in assessing the performance of the whole vehicle system using Multibody system software. Sert etc. [5] presents a method for systematic investigations on static and dynamic roll behavior of the suspension.

Aiming at understanding the environment, Marek, etc. [6] propose the use of a Simultaneous Localization and Mapping algorithm to generate local maps of forests by using Velodyne VLP-16 Lidar, a stereo camera, an inertial measurement unit (IMU), and a GPS. Stefan etc. [7] present easy to compute features for 3D point clouds using range and intensity values. Justin etc. [8] developed a general 3D tire-terrain traction model that operates on a novel deformable terrain representation that utilizes a soil compaction model is presented and discussed. Faniry etc. [9] describes a randomized incremental algorithm for computing the Delaunay triangulation of a set of point cloud. A high-quality rendering of three simple terrains is shown as a demonstration of the terrain generator.

This paper introduces suspension combined angular stiffness, tire vertical stiffness, and lateral stabilizer bar angular stiffness into comprehensive modeling for ATV static and dynamic rollover models. Those models are further integrated into the whole system to estimate the rollover stability. Ultimately, a wheel-ground contact dynamics model that takes into account tire parameters and pavement characteristics are built based on generating a fine triangular face sheet grid, and a dynamics simulation between ATV and real environmental terrain is performed.

2. ESTABLISHMENT OF MATHEMATICAL MODEL OF VEHICLE ROLLOVER

Figure 1 is a simplified diagram of the vehicle's static roll dynamics model. The vehicle is stationary on a slope with a slope of α , the contact points of the inner and outer tires, and the slope is points A and C, respectively. o_1 is the sprung mass centroid, o_2 is the unsprung mass centroid, and o is the roll center. F_1 and F_2 are normal forces at the contact points of the tire and the slope on inner and outer side tire; F_s is the tangential force at the contact point of the inner tire and the slope; G_u is the gravity of unsprung mass; G_s is the gravity of sprung mass; B is the Wheelbase; h_u is the equivalent height of the unsprung mass centroid; h_s is the equivalent height of the sprung mass centroid; h_r is the equivalent height of the roll centroid; h is the roll arm length; Φ_r is angle displacement of unsprung mass; Φ_u is angle displacement of sprung mass; δ_1 is vertical tire displacement caused by unsprung mass roll; δ_2 is vertical tire displacement caused by lateral stabilizer bar roll; S_1, S_2, and S_3 are unsprung mass centroid, roll centroid, and the distance between the sprung mass centroid and the longitudinal symmetry plane of the vehicle.

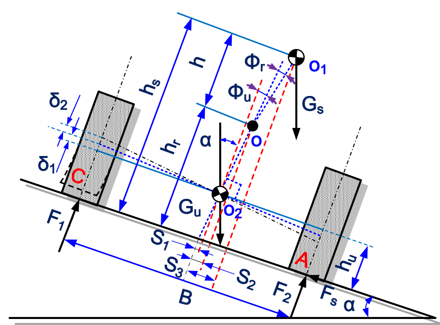


Figure 1: Dynamic model of vehicle Static rollover.

According to the geometric relationship in

$$S_1 = \Phi_u h_u \quad (1)$$

$$S_2 = \Phi_u h_r \quad (2)$$

$$S_3 = (\Phi_r + \Phi_u)h + S_2 = \Phi_u h_s + \Phi_r h \quad (3)$$

Take the moment of roll centroid o:

$$G_s \sin\alpha h + G_s \cos\alpha (S_3 - S_2) = (K_r + K_w)\Phi_r \quad (4)$$

Where K_r denotes roll stiffness of independent suspension, and K_w denotes the angular stiffness of lateral stabilizer bar.

Taking Eq. (3) into the above formula, obtaining the relationship of the roll angle of the spring-loaded mass centroid Φ_u and the roll angle of the unspring-loaded mass centroid Φ_r is:

$$\Phi_u = \left[\frac{K_r + K_w}{G_s \cos\alpha h} - 1\right]\Phi_r - \tan\alpha \quad (5)$$

2.1. Maximum Static Rollover Angle

When the gradient of the slope reaches the static maximum roll angle of the vehicle $\alpha_{\max}^{Static}$, the tire on the side of the slope just off the slope which means that $F_1 = 0$.

$$(G_u + G_s)(2\cos\alpha_{\max}^{static} - 1) = K_t(B\Phi_u + l\Phi_r) \quad (6)$$

Take Eq. (5) into Eq. (6), it can get:

$$\Phi_u = \frac{TG(2\cos\alpha_{\max}^{static} - 1) - LK_t \tan\alpha_{\max}^{static}}{(BT + l)K_t} \quad (7)$$

Here, G denotes the total gravidity of vehicle, T denotes the simplified calculation coefficient;

Take Eq. (5) and Eq. (6) into Eq. (1), it may have:

$$h_g(\Phi_u \cos\alpha_{\max}^{static} + \sin\alpha_{\max}^{static}) + \frac{G_s h_s(\Phi_u \cos\alpha_{\max}^{static} + \sin\alpha_{\max}^{static})}{G(T-1)} = \frac{1}{2}B\cos\alpha_{\max}^{static} \quad (8)$$

Here, $h_g = (G_u h_u + G_s h_s)/G$, which denotes the equivalent height of the center of mass.

Combining Eq. (7) and Eq. (8), we can easily obtain the relationship of the maximum static rollover angle with the suspension combined angular stiffness, tire vertical stiffness, and lateral stabilizer bar angular stiffness. By substituting the relevant geometric and structural parameters of the vehicle, the results can be obtained.

2.2. Vehicle Dynamic Roll Threshold

Taking car steady steering conditions as an example, the vehicle body is subjected to a certain degree of lateral acceleration. Take the same assumption as for static roll stability, and take the moment from the roll center point o:

$$G_s h \frac{a_y}{g} + G_s(S_3 - S_2) - K_r \Phi_r = 0 \quad (9)$$

Where, a_y — lateral acceleration.

Taking Eq. (2) and Eq. (3) into above, the spring-loaded mass roll angle is:

$$\Phi_r = \frac{\Phi_u + a_y / g}{K_r / G_s h - 1} \quad (10)$$

Due to the roll moment of the vehicle, the vertical normal loads acting on both sides' wheels are uneven. The inside wheel load F_i is gradually smaller than the outside wheel load F_o, the normal force from the road surface can be defined by the vertical stiffness of the tire, The amount of tire deformation is related to the roll angle of the unsprung mass:

$$\begin{cases} F_i = K_t(\delta_0 - \delta) \\ F_o = K_t(\delta_0 + \delta) \end{cases} \quad (11)$$

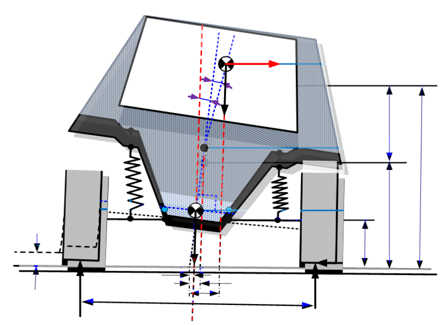


Figure 2: Dynamic model of vehicle dynamic rollover.

In the above formula, when δ_0 is the amount of deformation of the tire when $\Phi_u = 0$;

According to section 1.3, the vertical normal force of the inner tire can be obtained as:

$$F_i = \frac{1}{2}(G_u + G_s) - \frac{1}{2}BK_t\Phi_u \quad (12)$$

Take moment for outer wheel ground point A:

$$F_iB+(G_uh_u+G_sh_s)\frac{a_y}{g}-\frac{1}{2}B(G_u+G_s) \\ +G_s(h\Phi_s+h_s\Phi_u)+G_uh_u\Phi_u=0 \quad (13)$$

Combining Eq. (10), Eq. (12), and Eq. (13), it can get:

$$\Phi_u=\frac{G_sh_s+G_uh_u+\dfrac{G_sh}{\dfrac{K_r+K_w}{G_sh}-1}}{\dfrac{K_t}{2}B^2-G_sh_s-G_uh_u-\dfrac{G_sh}{\dfrac{K_r+K_w}{G_sh}-1}}\frac{a_y}{g} \quad (14)$$

$$\frac{a_y}{g}=\frac{\dfrac{K_t}{2}B^2-G_sh_s-G_uh_u-\dfrac{G_sh}{\dfrac{K_r+K_w}{G_sh}-1}}{\dfrac{K_t}{2\Phi_r}B^2(\dfrac{G_sh}{K_r+K_w\text{-}G_sh})} \quad (15)$$

The above Eq. (15) is the roll threshold value when the vehicle is in a steady-state turning condition which is used to evaluate the vehicle anti-rollover capability.

3. ROLLOVER STABILITY CALCULATION AND SENSITIVE ANALYSIS

3.1. Numerical Calculation of Roll Stability

Figure 3 shows the force analysis and posture description of ATV. The roll angle is as an indicator for the rollover stability, while ay is for rollover stability under dynamic. Fy is the force for vehicle lateral acceleration, which is a commentary factor for rollover stability analysis.

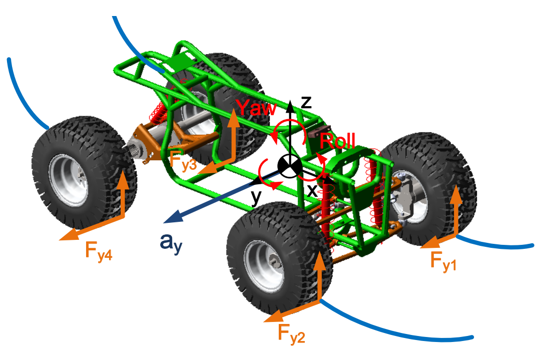


Figure 3: Force analysis and posture description of ATV.

The maximum static roll angle can be calculated by the Eq. (8) and obtained by Matlab calculation. Here list the geometric parameters and structural parameters of the ATV, as shown in Table 1. The maximum static roll angle of the vehicle can be obtained by numerical iteration.

Table 1: Simulation analysis parameters of ATV.

Parameter	Value	Unit
G_s	1220	N
G_u	400	N
h_s	0.51	m
h_u	0.14	m
h	0.37	m
B	1.12	m
K_r	35000	N·m/rad
K_w	180000	N·m/rad
K_t	750000	N·m/rad

For the vehicle dynamic roll threshold a_y/g, in the Matlab programming process, an initial value of 0 can be given and then increases it by 0.01rad each time to expand the iterative calculation of Eq. (12)-(15). The calculation will be terminated when $F_i=0$. At this point, the dynamic roll threshold of the vehicle a_y/g, roll angle of sprung mass Φ_r, and roll angle of unsprung mass Φ_u can be obtained.

Since the calculation is continuous, the resulting parameter changes are also continuous. They represent the parameter changes of each state. This method can be used to simulate the continuous state change of the entire vehicle from the beginning to the moment of rolling, and it can achieve the purpose of realistic simulation of car rollover.

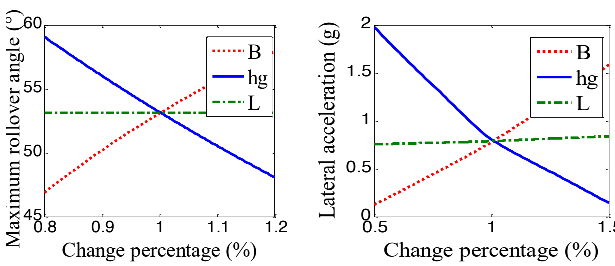
Based on the ATV parameters in Table 1, and using the iterative algorithm mentioned above, the final calculations show that the maximum static lateral tilt angle is 53.2 degrees and the dynamic lateral acceleration is 0.75 g.

3.2. Parameter Influence Analysis

It can be seen from Figure 4 that the impact of wheelbase B on the rollover angle is very significant, as the wheelbase increases the rollover angle increases rapidly when the wheelbase increases the vehicle's anti-rollover recovery moment increases, which makes the vehicle is not prone to rollover, so increasing the wheelbase can improve the vehicle's rollover stability.

The spring-loaded mass equivalent mass-centric height Hg also has a very large effect on the stability of the lateral tilt, the lower the Hg the greater the lateral tilt angle and lateral acceleration, the lower the spring-loaded mass-centric height makes the vehicle in the vertical direction of the lateral tilt force arm is reduced.

The lateral tilt moment is reduced thereby reducing the likelihood of vehicle rollover. Lateral tilt stability increases with increasing mid-length L of the stabilizer bar.



(a) Maximum rollover angle (b) Lateral acceleration

Figure 4: ATV roll stability changes with B, hg, and L.

From Figure 4, it can be obtained that for ATV vehicles under the above table parameters, the maximum static lateral tilt angle changes in the range of 47°-59°.

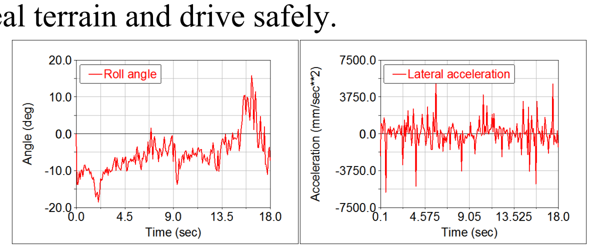
The dynamic lateral acceleration changes in the range of 0.1g-2g as the wheelbase B and the spring-loaded mass equivalent mass center height hg changes in the range of ±20%, while the middle length L of the lateral stabilizer bar has a negligible effect on both static and dynamic lateral stability.

4. DYNAMICS SIMULATION ON THE REAL TERRAIN ENVIROMENT

Based on the Adams/car virtual simulation environment, we build a typical ATV model and perform a system dynamic simulation and analysis. The four tire sizes of ATV are 205/55 R16, which is a tire model using Magic Formula and is suitable for the analysis of the roll stability of ATV. With four-wheel drive, the speed is set to 3 r/s, the tire diameter is 36 cm, and the forward speed of ATV is 3.39 m/s. The 3D equivalent volume pavement is selected as the simulation pavement. The real pavement elevation data collected by Lidar is made into the triangular topography map file [10], which is imported into Adams/View to generate the 3D pavement simulation model, as shown in Figure 5.

Figure 5: Simulation in ADAMS with field-collected data.

In the process of ATV driving, the curve of roll angle and lateral acceleration of the ATV center of mass position with time is shown in Figure 6. From Figure 6 (a), it can be seen that the change range of ATV's roll angle is about [-19°, +16°], which is less than its maximum roll angle of 53.2 degrees. From Figure 6 (b), it can be seen that the maximum lateral acceleration of ATV is about 0.5g, which is also less than its maximum roll acceleration threshold of 0.75g. In conclusion, it can be determined that the ATV can adapt to the real terrain and drive safely.



(a) Rollover angle (b) Lateral acceleration

Figure 6: Simulation results in ADAMS.

5. CONCLUSION

This paper establishes a mathematical model of automobile static and dynamic rollover that comprehensively considers the angular stiffness of the lateral stabilizer bar, the vertical stiffness of the tire, and the combined angular stiffness of the suspension. The model can more accurately reflect the rollover stability performance of ATVs and provide theoretical guidance and basis for the control of rollover stability in the vehicle design stage, which has high application value and practical significance. Based on the established model, the main geometric parameters affecting the maximum static tilt angle and dynamic lateral acceleration of an ATV model were analyzed.

The results show that in order to effectively improve the rollover stability of the ATV, the first consideration should be to increase the wheelbase and reduce the height of the center of mass. Besides, 3D point cloud data from the ATV's real terrain environment was acquired using RGB-Depth camera and triangular topography map file for ADAMS virtual simulation was generated using the Delaunay algorithm. It can be seen from the simulation analysis results that the ATV can successfully pass through the given real physical terrain environment without any risk of tipping.

6. REFERENCES

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