

Numerical Simulation of Sideways Overturning Dynamics for Agricultural Vehicles on a Slope

CHAORAN SUN

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Chapter 1

Introduction

1.1 Background

Safety is an important design factor for farm vehicles. For example, a wheeled farm tractor has a characteristic structure of a centre-pivoted front axle, which can easily result in overturning on slopes (Reynolds and Groves, 2000). In Japan in 2016, 217 fatal accidents occurred during the use of agricultural vehicles, including 53 cases of overturning or falling accidents (Ministry of Agriculture Forestry and Fisheries, 2018).

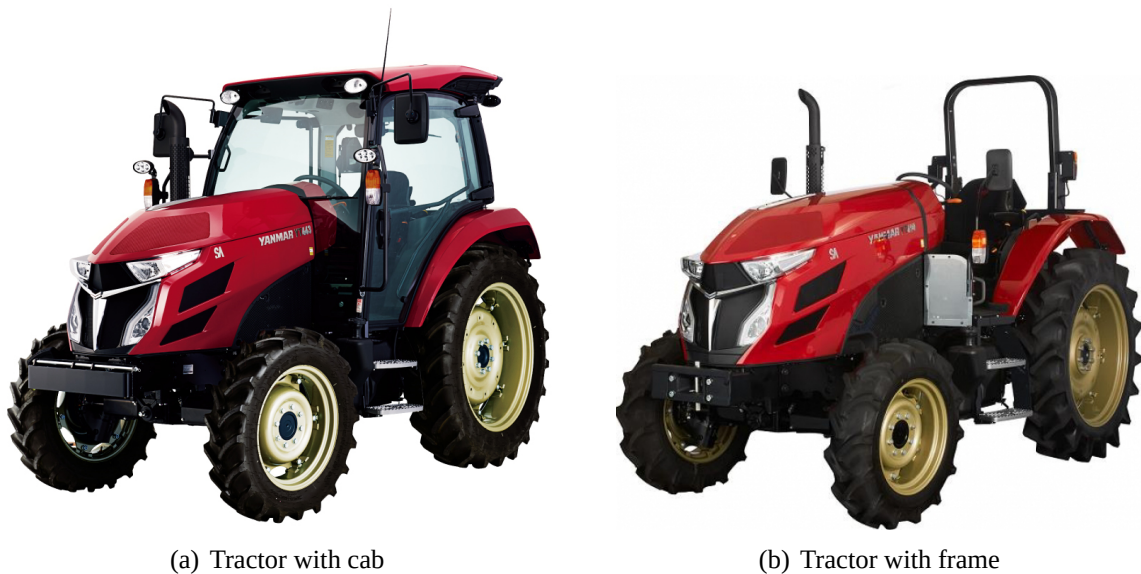


Figure 1.1: Examples of agricultural tractors with ROPS (YANMAR, 2018).

Japanese society also faces its own challenges in agriculture, where the farm workers become elder and their work becomes more dependent on agricultural machinery, which a fatal accident becomes even more unacceptable. Ministry of Agriculture Forestry and Fisheries (2018) reported that 81 percent of all fatal accidents are farm workers over 65 years old, in which 38 percent of elder farm workers involved in fatal accidents are over 80 years old. Because of the mountainous terrain, wheeled agricultural tractors (Fig. 1.1) have been operated not only on horizontal fields but also sloped fields. Agricultural tractors are the most popular agricultural wheeled vehicles on Japanese market with 36,881 units built in 2017 (Japan Agricultural Machinery Manufacturers Association,

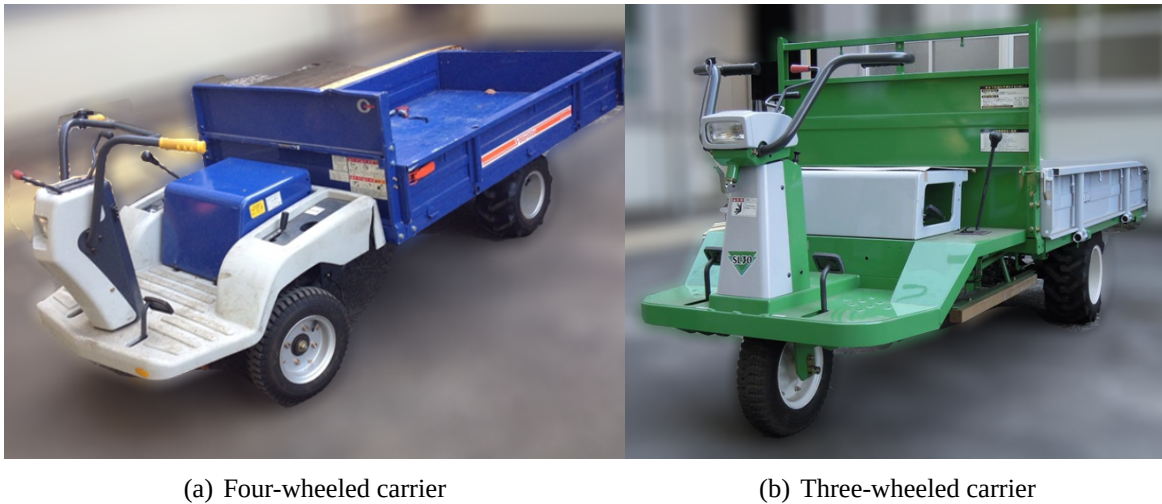


Figure 1.2: Examples of agricultural carriers with ROPS.

2017). The total number of fatal accidents involving wheeled agricultural tractors was 87 and the cases caused by overturning were 53. Moreover, small agricultural carriers (Fig. 1.2) have also been widely used to the farm workers' labor of transporting farm products or implements over various slope, which are the secondly popular agricultural wheeled vehicles on Japanese market with 11,483 units built in 2017 (Japan Agricultural Machinery Manufacturers Association, 2017). The total number of fatal accidents involving small agricultural carriers was 37 and the cases caused by overturning were 12 (Ministry of Agriculture Forestry and Fisheries, 2018). Not only mistaken operations of vehicles during farm work which might happen easily for an elder operator but also their characteristic structure such as a center-pivoted front axle of the tractor can cause fatal accidents. For this reason, rollovers on slopes have been a main cause of fatal accidents for farm tractors (Reynolds and Groves, 2000). The investigation on overturning characteristics of agricultural tractor and carrier becomes crucial for reducing the fatal accidents during farm work.

A rollover protection structure (ROPS), such as a safety cabin (Fig. 1.1(a)) and safety frame (Fig. 1.1(b)), designed to absorb the impact energy of overturning while maintaining sufficient space for an operator (Scarlett et al., 2006), has gradually been equipped on farm tractors since the 1960s. In the late 1960s, OECD released their official testing of ROPS on agricultural and forestry tractors as Code 3. In July 1970, SAE J333a standard concerning operator protection for wheel-type agricultural and industrial tractors was introduced, which was the predecessor of the current standard SAE J2194 (Stockton et al., 2002). The safety issues of the farm tractors drew a lot of attention to the development of ROPS specially in the 1970s, when the researchers engaged in the experiments and calculation focused on the forward speed (Schnieder and Baker, 1972), the energy balance of ROPS (Chisholm and Seward, 1976), and the dynamics of tractor (Chisholm, 1979). Consequently, the introduction of these structures to farm tractors has been effective for reducing fatalities. In Japan, 69 percent of tractors have already been equipped with ROPS and the rate of fatal accidents of tractors with ROPS (3%) became significantly lower than that without ROPS (25%) (National Agriculture and Food Research Organization, 2006). Moreover, the injury rate of tractor accidents after equipped with ROPS could be lowered to 27 percent. Figure 1.3 shows that the tendency of the fatal accidents related to the wheeled agricultural vehicle in recent years. The total number of fatal accidents tends to decrease mainly because that fatal accidents by wheeled agricultural tractors have been reduced. Relatively, the proportion of agricultural carriers becomes higher, where the introduction of

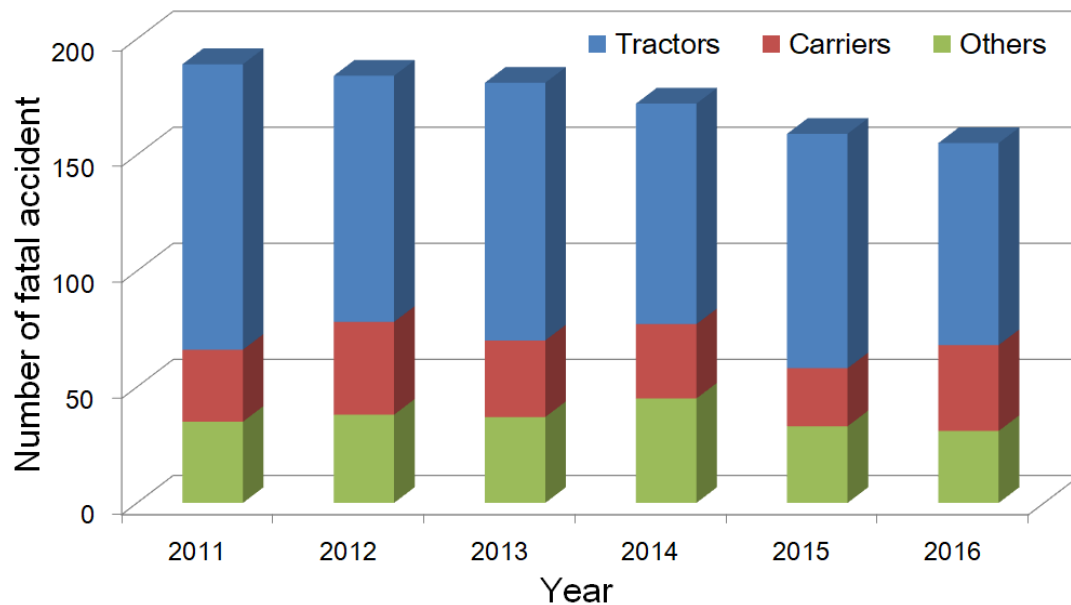


Figure 1.3: Fatal accident survey of wheeled agricultural vehicle in last 6 years by Ministry of Agriculture Forestry and Fisheries (2018).

the safety structure for agricultural carriers should be encouraged. Therefore, National Agriculture and Food Research Organization, Japan, starts making safety consultations with small manufacturers for agricultural carriers to estimate the effectiveness of a prototype overturning protective structure to contribute to the reduction of fatal accidents caused by the continuous rollover of agricultural carriers.

With the development of computer technologies, previous studies have focused on a prediction of overturning behaviour of agricultural tractors with ROPS working on a slope (Schwanghart, 1973). A simple computer code written in BASIC became available to predict the effectiveness of ROPS during overturning of a tractor known as the Code 6 by OECD (2017). The OECD Code 6 has been broadly accepted as a method to calculate the lateral overturning dynamics of an agricultural tractor with ROPS as shown in Fig. 1.1(b). To promote the safety of other agricultural machinery, it is a natural idea to extend the applicability of the Code 6 for farm tractors as described above. In the US, the standard (ANSI/ASAE S547 and now withdrawn) was published by the Turf and Landscape Equipment Systems Committee of ASAE to regulate test procedures for a tip-over protective structures, or TOPS, designed for turf keepers and other vehicles for landscape maintenance work (ASAE, 2003). Rollover of a front driven turf keeper based on the OECD code was also discussed with respect to S547 (Comer et al., 2003).

As a preliminary investigation of preventing fatal accidents of agricultural carrier, the Bio-oriented Technology Research Advancement Institution (BRAIN) of the National Agriculture and Food Research Organization, Japan, conducted an experiment using a small four-wheeled agricultural carrier with TOPS as shown in Fig. 1.2(a), which revealed that the deformation of a TOPS frame under a fully loaded condition was 58.6 mm (Tsukamoto et al., 2008). The overturning dynamics whether the four-wheeled agricultural carrier rolls continuously or not could be predicted by applying the input parameter of carriers agreeing with the OECD Code 6 algorithm to evaluate the efficiency of protecting the operator from fatal injury (Tsukamoto et al., 2008). Tsukamoto et al. (2008) reported the required height of TOPS of a Japanese commercially available four-wheeled agricultural carrier for the cases of zero load, only operator and operator with 611 kg payload. However, a three-wheeled agricultural carrier as shown in Fig. 1.2(b) which has one front wheel with high flexibility, can also

commonly be seen to transport farm products or implements. BRAIN planned to develop a new program which could predict the overturning dynamics of three-wheeled agricultural carriers. The program for three-wheeled agricultural carrier was also expected to evaluate the efficiency of protecting operator from fatal injury as Tsukamoto et al. (2008) did for four-wheeled agricultural carrier. Thus, whether or not the OECD Code 6 program can be modified to compute a similar protective performance for three-wheeled agricultural carrier with TOPS compared to Tsukamoto et al. (2008) should be discussed.

On the other hand, physics engine is a computer program that calculates the collision detection and dynamics of contacting objects to predict impacts and movements incorporated laws from the physical world (Featherstone, 2008). With the capability of graphics which are drawn to portray the current motion of simulated objects, the physics engine can provide the production of animation data. Recently, with the development of video games or computer graphics, the real-time programming tool called Bullet which provides a three-dimensional rigid body dynamics including collision detection has become popular (Erwin, 2013). Bullet generally outputs perceptually acceptable footage of collision models rather than a list of numerical data. NASA (2015) used Bullet to build robots based on the provided rigid components with constraint connecting elements and enable the rapid co-exploration of structures and controls in a physics based simulation environment. A comparison on numerical dynamics data and experimental result should be investigated. Nevertheless, it is expected that the simulation of the dynamics for the colliding pair of agricultural vehicle and slope can be realised by using the physics engine.

1.2 Research Objective and Scope

Firstly, this study is to develop a tool which can predict the critical height of TOPS for both four-wheeled and three-wheeled agricultural carriers. Tsukamoto et al. (2008) studied a solution for four-wheeled simulation and the additional part for three-wheeled needed to follow the same algorithm as Tsukamoto et al. (2008) based on the OECD code 6 by adjusting calculation loop and contact points, which could generally achieve coherence in calculating disposition. However, the accuracy of the developed tool for both four-wheeled and three-wheeled agricultural carriers should be comprehensively evaluated not only by comparing the difference in critical height of TOPS between experiment and simulation but also by the responses to restitution and moment of inertia consistent with previous researches using OECD Code 6. The similarities and differences between four-wheeled simulation and three-wheeled simulation should also be investigated.

Secondly, another purpose of this study is to introduce the physics engine as an improvement of three-dimensional simulation for overturning dynamics of both four-wheeled and three-wheeled agricultural carriers. The perceptual accuracy of calculation on collision detection and dynamics can be easily judged through the natively produced animation compared with experimental footage. The difference in predicting critical height of TOPS with the coefficient of friction and the responses to various carrier and slope setting will be discussed.

Physics engine allows linear or angular displacement between two connected rigid components using constraint library. Therefore, connecting basic rigid components by constraint portray another possibility that some parts can have relative displacement against the main vehicle body. A cylinder component passively rolls around a box component might be the beginning of modelling an agricultural tractor with a pivoted front axle, and a cylinder component actively rolls around some other components should be an expression of driving an agricultural tractor forward. The application of physics engine to simulate the overturning dynamics drove to model the colliding pair of an agri-

cultural tractor model and slope model. Comparing to the previous experiments on the overturning dynamics of an agricultural tractor over a slope, the simulation result will be evaluated by a statistical technique which measured the errors in separated numerical data.

The relative displacement of the parts against the main tractor body grounded the feasibility on quantifying the dynamics of an accurate point, which specifically runs toward the measurement of impact force upon the protective structure of the modelled agricultural tractor. The lateral impact force on the contacting point of ROPS along with the dynamics data of different location on tractor during overturning would be observed in our cooperating experiment with Hunan Agricultural University. Based on the comparison of impact force and dynamics data between experiment and simulation, the investigation of the calculating accuracy in dynamics and constraint will be examined.

1.3 Outline of the Dissertation

This Thesis consists of eight chapters as follows.

In Chapter 1, the background and purpose of this study were summarised.

In Chapter 2, a numerical program was developed to predict the effective height and width of a safety frame, or tip-over protective structure (TOPS), for a three-wheeled agricultural carrier (Sun et al., 2017a). To facilitate program development, the main algorithm of computation in the original program for a tractor with protective structure laterally rolling over (OECD, 2017) was not modified. Based on catalogue data of commercially available carrier, the accuracy of the developed program was also investigated.

In Chapter 3, a physics engine was introduced to find out whether or not it can be applied to predict the sideways overturning behaviour for an agricultural carrier with TOPS (Sun et al., 2017b).

In Chapter 4, the overturning dynamics of agricultural model carrier were analysed by the physics engine: Bullet (Sun et al., 2018a).

In Chapter 5, we will simulate the overturning behaviour of agricultural tractors with a roll over protective structure (ROPS) on a bank slope and on a uniform slope using Bullet physics engine (Sun et al., 2018b). Visual behaviour data and velocity transition during overturning processes are discussed based on the earlier reported experimental data.

In Chapter 6, overturning dynamics were measured using a tractor with ROPS (Xiao et al., 2018). The target is a small-sized tractor which weights near 1200 kg and is widely used in orchard and terrace field. Two cases of sideways condition were considered: no travelling speed and speed of 0.77 m s^{-1} .

In Chapter 7, calculation accuracy of impact force acting on ROPS of an overturning tractor using a physics engine was investigated comparing to the observed results in our own experiment (Sun et al., 2018c). Angular velocity and rotating angle around rolling axis of the tractor were examined by experiments and by numerical simulation.

Chapter 8 summaries the present study, and conclusions obtained in the present computer simulations are provided. Suggestions for improvement toward a soft body simulation using physics engine and the approaches of introducing a soft body into a rigidly colliding pair are also presented for future studies.

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Chapter 2

OECD-based Simulation of Agricultural Carriers

2.1 Introduction

In this chapter, a numerical simulation program that can estimate the height of TOPS for a three-wheeled carrier (TWC) on a slope is developed. The numerical simulation program can ascertain whether or not a designed agricultural carrier has sufficient height or width of TOPS to prevent continuous roll over on a slope surface. To investigate the accuracy of numerical analyses, experiments using small model carriers were also conducted. Then the results of the maximum height of TOPS are compared.

2.2 Methods

A computer program for four-wheeled carriers (FWC) was developed previously (Tsukamoto et al., 2008), following a similar approach for evaluation of TOPS of a turf keeper (Comer et al., 2003). Because the development of program for TWC is based on an FWC program, the essence of methods in the computer program for FWC and for TWC is summarised in this section.

2.2.1 Common assumptions

The vehicle's center of gravity is assumed to be on a center line of the half width of a carrier for simplicity. As described in an earlier report (OECD, 2017), the following points are assumed in the development of numerical programs for a three-wheeled agricultural carrier.

- A carrier begins to roll on a flat slope surface with a gradient of 1:1.5 (33.7 deg) as soon as the carrier's center of gravity passes beyond the vertical line of the edge through the axis of rotation when the mountainside under the vehicle is statically lifted.
- The deflection of tyres is negligible.
- Contact points can be chosen based on the carrier shape. Two contact points will define a candidate of rotation axis. For example, the first axis of rotation will be formed by two contact points at the center of the contact surfaces of the front and rear tyres on the slope.
- A carrier does not slide on a slope after overturning.

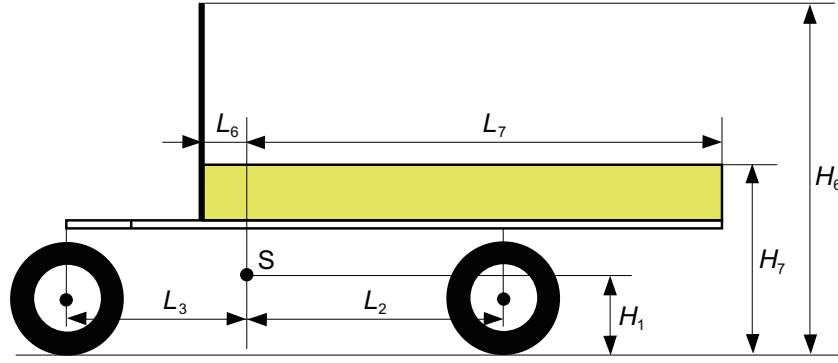


Figure 2.1: Schematic side view of an agricultural carrier with TOPS.

- Normal impact on the surface of slope by a carrier is partially inelastic, with a coefficient of elasticity of 0.2.
- The total amount of the deformation of TOPS including the sinkage into the slope surface does not exceed 0.2 m. It is noteworthy that the proportional reduction of this value will be applied based on the total mass of a carrier vehicle so that the behaviour of the small model carrier is calculable using the same program.
- No carrier component other than the contacting point on TOPS will penetrate into the slope surface.

2.2.2 Correspondence of reversed parameters

Definitions of main parameters for FWC model (Fig. 2.1) were modified from those for tractor model (OECD, 2017) as a reverse use of the same variables on the front and rear of the vehicle. Because target carriers have no suspension and because their front and rear axles are fixed to the body, the center-pivot angle of rear axle originally effective as the front axle of the tractor model is set to be fixed by assuming the angle data of 0 degrees. The tractor model engine bonnet can be regarded as the bed in the case of a carrier. Parameters modified in the four-wheeled carrier model are presented in Table 2.1.

2.2.3 Calculation of angular velocity on a rotation axis for a four-wheeled carrier

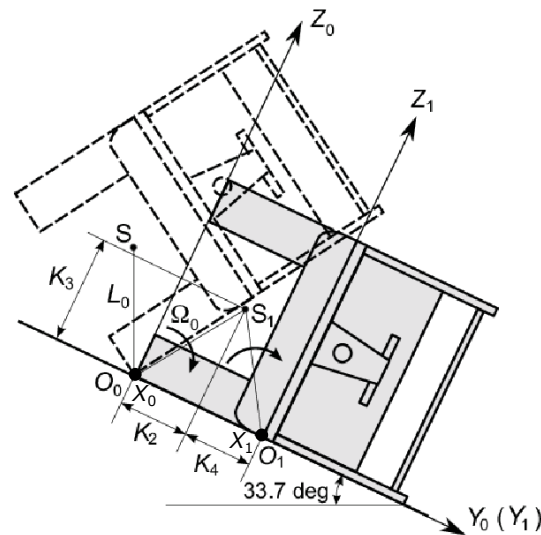
The calculation of angular velocity for a three-wheeled carrier is the same as that for a four-wheeled carrier (Tsukamoto et al., 2008), based on the same algorithm developed previously for a tractor (OECD, 2017). The fundamental part is summarised here in this subsection.

The local coordinate system is definable on the slope surface, such that the rotation axis is the X -axis, the normal axis is the Z -axis. The perpendicular axis to X -axis on the slope is the Y -axis. Rollover behaviour can be traced by the equations of motion for the center of the gravity of the vehicle. Moreover, according to the law of conservation of energy, the kinetic energy used for successive rollover behaviour of the vehicle is expected to derive from the initial potential energy of the vehicle rested on a slope with the center of gravity S lying on the vertical plane extended through the expected rotation axis I from the horizon as shown on the dashed line in Fig. 2.2.

Table 2.1: Correspondence of parameters between a four-wheeled carrier model and a tractor model.

FWC (Fig. 2.1)		Tractor (OECD, 2017)
D_0	Rear-axle swing angle	Front-axle swing angle
D_2	Height of rear tyres	Height of front tyres
D_3	Height of front tyres	Height of rear tyres
H_0	(Not used)	Height of front axle
H_7	Height of upper edge of cargo bed	Height of engine bonnet
L_2	Horizontal distance between S and the rear axle	Horizontal distance between S and the front axle
L_3	Horizontal distance between S and the front axle	Horizontal distance between S and the rear axle
L_7	Horizontal distance between S and the rear end of cargo bed	Horizontal distance between S and the front corner of engine bonnet

N.B.) Other parameters of FWC are the same as those of the tractor model (OECD, 2017).

**Figure 2.2:** Rollover of a four-wheeled carrier on the slope around rotation axes of X_0 and X_1 .

Then, the carrier will overturn around X_0 -axis with the origin of O_0 in the figure, with its angular velocity from zero at rest to the value of Ω_0 immediately before the impact onto the surface of slope, based on the principle of conservation of total energy, as shown below.

$$\frac{1}{2}Q_0\Omega_0^2 = Mgh \quad (2.1)$$

Therein, the left-hand side shows the kinetic energy before the impact and the right-hand side represents the consumed potential energy before the impact on the slope, h is the vertical height of fall of the center of gravity S from the position where the rollover starts and is known according to the geometrical relation between the rotation axis I and S , M is the carrier mass, Q_0 is the roll moment of inertia of the carrier with respect to the rotation axis I , and g is the acceleration of gravity.

In Eq. (2.1), the roll moment of inertia Q_0 around rotation axis I can be determined using the parallel axis theorem as shown below.

$$Q_0 = Q + ML_0^2 \quad (2.2)$$

In that equation, Q is the roll moment of inertia about the center of gravity S , the reference axis of which is parallel to the X_0 -axis defined on the slope. In addition, L_0 is the shortest distance from S to X_0 -axis, which is the radius of rotation of the vehicle with respect to the axis, as shown in Fig. 2.2.

Consequently, the angular velocity Ω_0 immediately before the impact of carrier body to the sloped surface with respect to X_0 -axis can be calculated from Eq. (2.1) using the roll moment of inertia Q_0 such that the following is true.

$$\Omega_0 = \sqrt{\frac{2Mgh}{Q_0}} \quad (2.3)$$

Similarly, the momentum loss between the rotation axes, X_0 -axis and X_1 -axis, in the figure can be expressed by two directional components: loss in Y_0 -direction as ΔP_y , and loss in Z_0 -direction as ΔP_z as follows.

$$\Delta P_y = -M(v_{y_1} - v_{y_0}) \quad (2.4)$$

$$\Delta P_z = M(v_{z_1} - v_{z_0}) \quad (2.5)$$

In those equations, v_{y_1} and v_{z_1} respectively represent the velocity components of Ω_1 in the Y_1 -direction and Z_1 -direction on the coordinate system. Also, O_1 - $X_1Y_1Z_1$ and v_{y_0}, v_{z_0} respectively denote those of Ω_0 in the Y_0 -direction and Z_0 -direction on the coordinate system O_0 - $X_0Y_0Z_0$.

Moreover, the angular momentum loss is calculable as

$$Q(\Omega_1 - \Omega_0) = \Delta P_y K_3 - \Delta P_z K_4, \quad (2.6)$$

where K_3 is the Z -component of distance L_0 between the center of gravity S and X_0 -axis, and K_4 is the Y -component of L_1 between S and X_1 -axis (see Fig. 2.2).

We can also obtain the following relations on velocity components ($v_{y_0}, v_{y_1}, v_{z_0}, v_{z_1}$) using angular

velocities of ω_0 and ω_1 as shown below.

$$v_{y_0} = \frac{K_3}{L_0}(\Omega_0 L_0) = \Omega_0 K_3 \quad (2.7)$$

$$v_{y_1} = \frac{K_3}{L_1}(\Omega_1 L_1) = \Omega_1 K_3 \quad (2.8)$$

$$v_{z_0} = \frac{-K_2}{L_0}(\Omega_0 L_0) = -\Omega_0 K_2 \quad (2.9)$$

$$v_{z_1} = \frac{K_4}{L_1}(\Omega_1 L_1) + (K_2 + K_4)\Omega_0 e \quad (2.10)$$

$$= \Omega_1 K_4 + (K_2 + K_4)\Omega_0 e \quad (2.11)$$

Those are applicable considering the coefficient of restitution e in the case of the Z_1 component of velocity v_{z_1} .

By substituting these velocity components from Eq. (2.7) to Eq. (2.11) into Eqs. (2.4) and (2.5), one can obtain ΔP_y and ΔP_z .

Therefore, from Eq. (2.6), the rotation (or angular) velocity Ω_1 immediately after the impact on the slope surface by X_0 -axis is obtainable as shown below.

$$\Omega_1 = \left[\frac{Q + MK_3^2 - MK_4^2 e - (e + 1)MK_2 K_4}{Q + MK_3^2 + MK_4^2} \right] \Omega_0 \quad (2.12)$$

In this way, the calculation of angular velocities before and after impact will continue until all the rotation axes are evaluated. If necessary, the transform of rotation axis will be applied beforehand in the derivation of an angular velocity before the impact with respect to a next rotation axis.

The roll moment of inertia on the center of gravity S is assumed to be transformed with a simple weighted transformation using an ellipsoid of inertia as OECD (2017)

$$Q' = Q \cos^2 \phi_i + 3Q \sin^2 \phi_i, \quad (2.13)$$

where Q' stands for the roll moment of inertia after coordinate transformation on the rotation axis, ϕ_i signifies the angle between the current axis, and the previous axis. It is noteworthy that the roll moment of inertia Q' is approximately converted using Eq. (2.13) instead of precisely applying the transformation law of the roll moment of inertia Q for simplicity.

2.2.4 Initial posture and Cartesian coordinate system for a three-wheeled carrier

Fig. 2.3 depicts the comparison of initial posture of FWC and TWC on the slope surface. Overturning of TWC is assumed to start from the position with the steered front wheel on the slope, as shown in Fig. 2.3(b). As shown in the figure, the first rotation axis I is defined by contact points at the edge of dead bottom center of steered front wheel and that of the rear wheel (National Agency of Vehicle Inspection, 2014). The rotation axis I is therefore set as parallel to the slope contour line.

The slope surface is expressed as X - Y plane with outward normal direction as the Z -axis, based on the right-hand system. Because rotation axis I is no longer parallel to the longitudinal axis of the vehicle, a steering, or yaw, angle is generated between the axis I and the longitudinal direction of the vehicle. The shortest projected line on to the rotation axis I from the center of gravity S defines an intersection point on the rotation axis I, which then becomes origin O of coordinate system for TWC, as shown in Fig. 2.4.

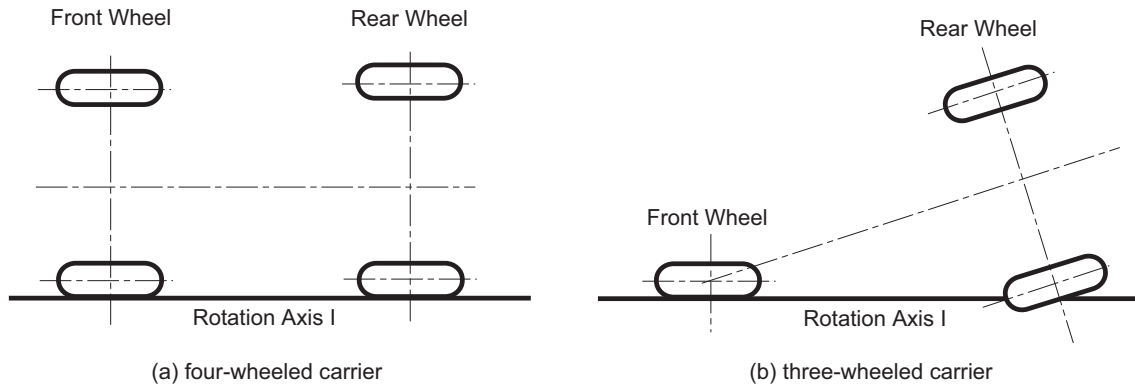


Figure 2.3: Difference of initial posture of an agricultural carrier on a slope against rotation axis I parallel to the contour line of slope.

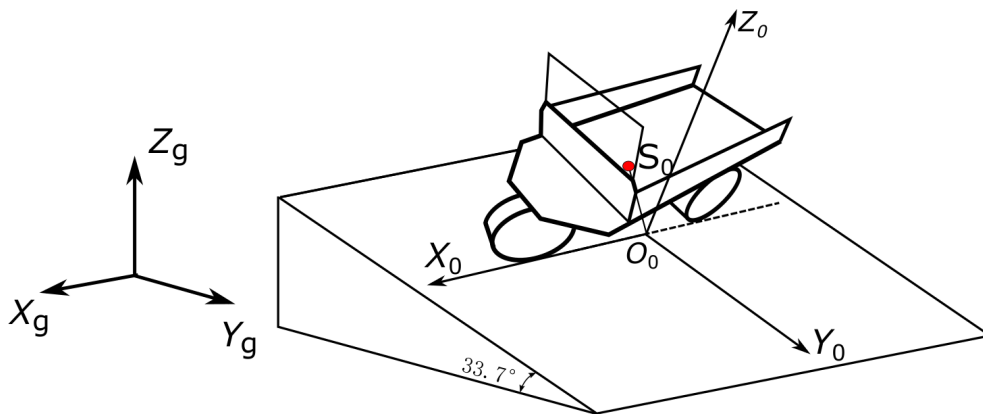


Figure 2.4: Initial Cartesian coordinate system defined on the slope surface with the posture of three-wheeled carrier.

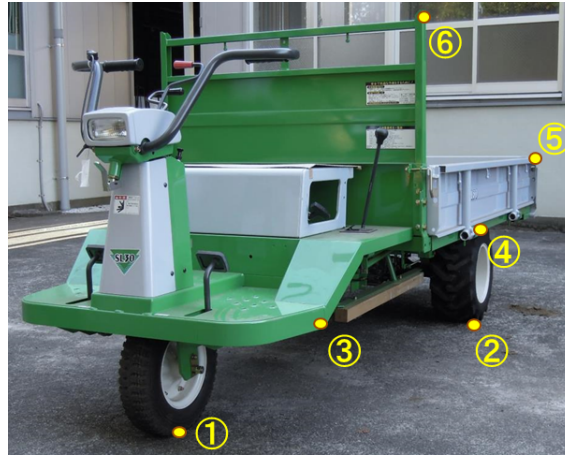


Figure 2.5: Possible contact points of a three-wheeled carrier.

Table 2.2: Relation of angular velocity and rotation axis for a three-wheeled carrier.

Angular velocity	Variable	Axis	Note
ω_0	O0	I	Rotation velocity before impact on axis I
ω_1	O1	I	Rotation velocity after impact on axis I
ω_{20}	O20	II	Initial rotation velocity on axis II
ω_{21}	O21	II	Rotation velocity before impact on axis II
ω_{22}	O22	II	Rotation velocity after impact on axis II
ω_2	O2	III	Initial rotation velocity on axis III
ω_3	O3	III	Rotation velocity before impact on axis III
ω_4	O4	III	Rotation velocity after impact on axis III
ω_5	O5	IV	Initial rotation velocity on axis IV
ω_6	O6	IV	Rotation velocity before impact on axis IV
ω_7	O7	IV	Rotation velocity after impact on axis IV
ω_8	O8	V (V')	Initial rotation velocity on axis V (V')
ω_9	O9	V (V')	Rotation velocity when CG reaches at its highest position

2.2.5 Possible rotation axes during rollover of a three-wheeled carrier on the slope

New parameters are introduced into TWC, such as those for the front wheel width, which is not required in the program for FWC (Tsukamoto et al., 2008). The procedure of computation for TWC is, in principle, the same as in the program for FWC summarised in the previous subsection.

Development of the computer program for TWC emphasises the calculation of coordinates on possible contact points from ① to ⑥ shown in Fig. 2.5. Consequently, the total number of rotation axes in TWC is assumed to be five, as summarised below. Evaluation of angular velocities from ω_0 to ω_9 is applied to the case of TWC as presented in Table 2.2. As in the table, variables for angular velocities on rotation axis II are added as a modification of the previously developed program for FWC (Tsukamoto et al., 2008) while maintaining the same use of variables to the greatest degree possible.

Rotation axis I: Contact points ① and ② of those two sides of contacting surfaces of tyres on the slope (see Fig. 2.5) will define the rotation axis I of overturning, which is parallel to the contour line but which is not parallel to the longitudinal axis of the vehicle. The height of fall of S can be obtained similarly as in the case of the four-wheeled carrier. The angular velocity about rotation axis I (ω_0) before the impact of contacting point ③ to the ground can be calculated. It is noteworthy that the contact by the side of rear tyre is neglected in the rotation around the axis I. From the center of gravity S , one can draw a perpendicular line to axis I, for which the intersection point becomes the origin O_0 of the coordinate system of X_0 , Y_0 , and Z_0 . Angular velocity calculation on this axis is for ω_0 before the impact, and ω_1 after the impact.

Rotation axis II: Contact points at a side of the front part of the body ③ and the lower side edge of rear tyre ② will form rotation axis II. Then, we can calculate the angular velocity on this axis II, considering the conservation of energy and momentum caused by the impact on the slope. Angular velocity calculations on this axis are expected to include ω_{20} which is the initial angular velocity on axis II by transforming the component of ω_1 , ω_{21} which is the updated angular velocity around axis II before the impact of contacting point ④, and ω_{22} which is the updated angular velocity after the impact on axis II. This rotation axis does not exist in the program for FWC (Tsukamoto et al., 2008). Introduction of rotation axis II is the main modification in this study to realise the smooth transition of rotation axes.

Rotation axis III: Contact points at the side of front part ③ and at the upper-side edge of the rear tyre on slope ④ will form rotation axis III. Similarly, angular velocities of ω_2 , ω_3 , and ω_4 can be calculated on this axis with possible impact on the contact point ⑥.

Rotation axis IV: Possible contact points on the upper-side edges of the rear tyre on slope ④ and on the top corner point of the safety frame ⑥ will form rotation axis IV. During program development for TWC, it is assumed that the outside width between the two rear tyres has the same as that of the cargo floor. Angular velocities of ω_5 , ω_6 , and ω_7 are calculated similarly on this axis with possible impact on the contact point ⑤.

Rotation axis V (or V'): Two rotation axes can be assumed depending on the location of the center of gravity S . The rotation axis V will be obtained by the point of the upper tail-end of the cargo bed ⑤ and the top corner point of safety frame ⑥. Alternatively, the rotation axis V' is definable as the point of the front part of the cab body ③ and the top corner point of the safety frame ⑥ if the center of gravity S is located in front of the safety frame. In this situation, the carrier would not roll around the rotation axis III, but the calculation about axis V' continues after the calculation about axis III. After calculating the initial angular velocity ω_8 , angular velocity ω_9 on the axis V (or V'), which implies that the rotation velocity at the possible highest location of the center of gravity S_5 with respect to the global vertical axis Z_g (see Fig. 2.4) during the rotation around the axis, is inferred from the conservation of energy and momentum.

An example of a trace of rotation axes on the slope surface is shown schematically in Fig. 2.6 for a case of continuous rollover with contact points explained above. It is noteworthy that each rotation axis becomes the locally defined X-axis of which the origin is defined as the normal shortest length of the center of gravity S to the rotation axis. Translation of the projected center of gravity S is also depicted in the figure. Those rotation axes II, II, and IV might share almost identical projected points of the center of gravity S_2 , S_3 , S_4 , as in the figure. As the figure shows, if the center of gravity S_5 with respect to the rotation axis V is located below axis 5, then the carrier will continue rolling because of the unstable equilibrium condition on the slope surface.

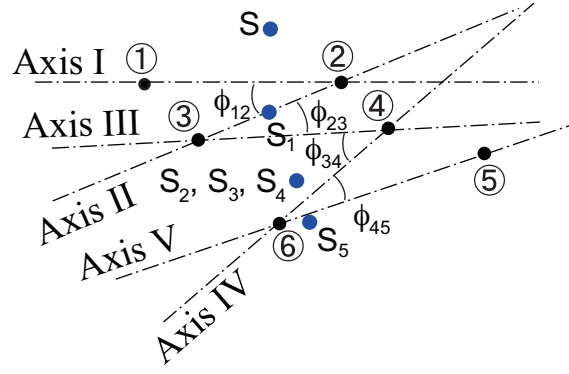


Figure 2.6: Trace of rotation axes of a three-wheeled carrier on the slope surface, observed on Z-axis (case of successive overturning).

The vehicle will not rotate further. It is understood that the height of TOPS is sufficiently high to prevent the vehicle from successive rollover on the slope if ω_9 (O9) is equal to or less than zero. If ω_9 (O9) becomes larger than zero, then the vehicle will continue rotating on the rotation axis V (or V'), which implies that the current height of TOPS cannot resist continuous rolling over on the slope.

2.2.6 Calculation flow for lateral rollover of a three-wheeled carrier on the slope

Let the angle between the first rotation axis and the longitudinal axis of the carrier, which corresponds to a steered angle of front wheel of TWC defined by the geometry of TWC, be ϕ_0 . Then, we can obtain the following two equations for the length relation of common PP', as shown in Fig. 2.7.

$$2 \times PP' = 2 \times (L_2 + L_3) \sin \phi_0 \quad (2.14)$$

$$2 \times PP' = (B_{rw} + B_0) \cos \phi_0 - B_{01} \quad (2.15)$$

In those equations, P stands for the middle point of rear axle, P' signifies that of the apexes of right triangle, B_{rw} denotes the distance between two rear wheels measured at the center of wheel width, B_0 represents the width of the rear wheel, and B_{01} is the width of a front wheel, as shown in Fig. 2.7.

Consequently, from Eqs. (2.14) and (2.15), we have the following.

$$\begin{aligned} 2 \times (L_2 + L_3) \sin \phi_0 &= (B_{rw} + B_0) \cos \phi_0 - B_{01} \\ \text{or, } 2 \times (L_2 + L_3) \sin \phi_0 + B_{01} &= (B_{rw} + B_0) \cos \phi_0 \\ \therefore 2 \times (L_2 + L_3) \sin \phi_0 + B_{01} &= (B_{rw} + B_0) \sqrt{1 - \sin^2 \phi_0} \end{aligned} \quad (2.16)$$

By multiplying each side by itself of Eq. (2.16), we can finally obtain a quadratic equation on $\sin \phi_0$, as

$$a \sin^2 \phi_0 + b \sin \phi_0 + c = 0, \quad (2.17)$$

where $a = 4(L_2 + L_3)^2 + (B_{rw} + B_0)^2$, $b = 4(L_2 + L_3)B_{01}$, and $c = B_{01}^2 - (B_{rw} + B_0)^2$.

Therefore, by solving Eq. (2.17) and taking the positive sign, the angle ϕ_0 is obtainable as

$$\phi_0 = \sin^{-1} \frac{-b + \sqrt{b^2 - 4ac}}{2a}, \quad (2.18)$$

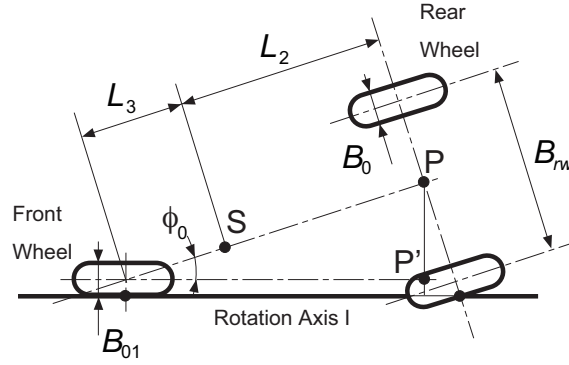


Figure 2.7: Angle between the longitudinal axis of a three-wheeled carrier and the first rotation axis.

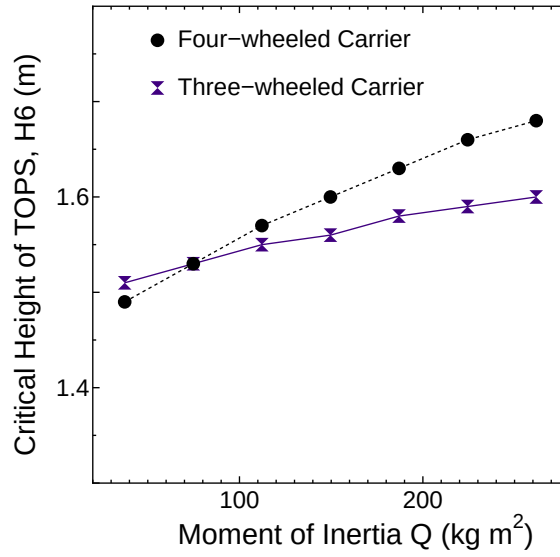


Figure 2.8: Effect of roll moment of inertia on the height of TOPS.

where coefficients a, b, c are as presented above.

Radius L_0 of rotation on rotation axis I, or length SO_0 (see Fig. 2.4) can then be calculated using the angle obtained from Eq. (2.18), as shown below.

$$L_0 = \sqrt{(L_3 \sin \phi_0 + 0.5B_{01})^2 + H_1^2} \quad (2.19)$$

In that equation, L_3 stands for the distance between the center of front axle and the center of gravity, $L_3 \sin \phi_0 + 0.5B_{01}$ represents the projected length from the center of gravity to the first rotation axis in the Y_0 -axis direction, and H_1 denotes the height of the center of gravity in the Z_0 -axis direction.

A similar approach of obtaining rotation axes and corresponding angular velocities is useful for calculations for TWC as in those for FWC based on the overturning code for farm tractors.

2.2.7 Parametric analysis of the program

Similarly to a previous report (Comer et al., 2003), the effects of moment of inertia and the coefficient of elasticity on the necessary (or critical) height of TOPS were investigated so that the program developed for TWC can yield similar results.

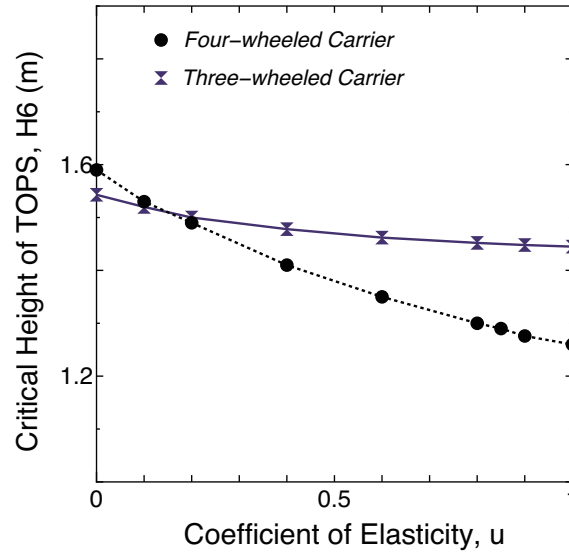


Figure 2.9: Effect of the coefficient of elasticity on the height of TOPS.

Figure 2.8 presents the effect of roll moment of inertia on the necessary height of TOPS. The roll moment of inertia around the center of gravity for both TWC and FWC are input as 37.4, 74.8, 112.2, 149.6, 187, 224.4, and 261.8 kg m². Other parameters such as the distance from center of gravity to rotation axis, or mass of vehicle are kept unchanged. Similar behaviour of increasing the height of TOPS as the increase in roll moment of inertia is apparent in TWC and FWC cases. This increasing trend coincides with the case for a turf keeper (Comer et al., 2003).

The effect of the coefficient of elasticity on the necessary height of TOPS is presented in Fig. 2.9. The definition of the coefficient of elasticity u in the original OECD program is understood as being equivalent to the coefficient of restitution e , where $e = 1.0$ denotes a fully elastic response and $e = 0.0$ implies non-elastic response with no repulsion (Comer et al., 2003).

However, the result in Fig. 2.9 explicitly portrays the opposite behaviour with respect to the definition of the coefficient of elasticity u , where the height of TOPS should be decreased in the case of non-elastic impulse (e.g. $u = 0.0$) to the slope surface because the impact energy is expected to be dissipated at the time of impact. The small value of u implies the increased dissipation of impact energy. This behaviour is similar to the result for the case of a turf keeper (Comer et al., 2003).

From these investigations, it can be inferred that the program developed for a three-wheeled agricultural carrier might be sufficiently accurate to follow a similar algorithm to that adopted in the original OECD program for farm tractors.

2.3 Results and discussion for commercially available three-wheeled carriers

To investigate the accuracy of rollover simulation for a three-wheeled carrier working on a slope, the overturning of a commercially available three-wheeled carrier (SL30; Atex Co. Ltd., Japan) was analyzed. Using the dimension data of the carrier from its catalogue (Atex Co., 2014) and by assuming that the carrier is unloaded without an operator, the mass of the vehicle was found to be 310 kg. Although the roll moment of inertia of the TWC (SL30) was not measured, the roll moment of inertia of a FWC (ES60; CANYCOM), which has similar size and weight to that of SL30 was

Table 2.3: Input data of three-wheeled carrier SL30.

Parameters	Vehicle only	With operator	With operator and cargo
B_f , m	1.21	1.21	1.21
B_{rw} , m	1.01	1.01	1.01
B_0 , m	0.2	0.2	0.2
B_{01} , m	0.15	0.15	0.15
B_6 , m	varied*	1.08	1.08
B_7 , m	1.21	1.21	1.21
H_f , m	0.471	0.471	0.471
H_1 , m	0.387	0.447	0.54
H_2 , m	0.471	0.471	0.471
H_6 , m	varied‡	varied‡	varied‡
H_7 , m	0.76	0.76	0.76
L_f , m	0.869	0.749	1.213
L_2 , m	0.597	0.717	0.253
L_3 , m	1.228	1.108	1.572
L_6 , m	0.519	0.399	0.853
L_7 , m	1.442	1.562	1.098
M , kg	310.0	370.0	970.0
Q , kg m ²	37.4	45.38	89.22

*Cases of B_6 : 0.9, 1.0, 1.05, 1.08 m.

‡Cases of H_6 : increased until O_9 becomes zero.

measured using BRAIN. Therefore, it is assumed that the roll moment of inertia of TWC (SL30) is the same as that of FWC (ES60), which is $Q = 37.4 \text{ kg m}^2$. From the BRAIN measurement, results showed that the center of gravity S of TWC was located 387 mm above the ground, 597 mm from the rear axle of the wheel, and 1228 mm from the front axle of the wheel. The maximum safety frame width was set as 1080 mm because the original carrier had a frame for protective purposes with this width behind the operator seat.

All the input data for TWC are presented in Table 2.3. To observe the effects of increased vehicle mass for an operator and for an operator with additional load on the cargo floor, corresponding data are modified. They are also shown in the table.

Fig. 2.10 portrays results of simulation for SL30 with various widths of TOPS. The increased height of the safety frame has a strong influence on the protection for avoiding successive overturning because the angular velocity does become zero from the height of 1.5 m or larger for largest TOPS width of 1.08 m. The figure also shows the rational tendency of increased height of TOPS for avoiding successive overturning when the TOPS width is decreased from 0.9 to 1.08 m.

Fig. 2.11 presents the effects of additional loads, such as an operator and cargo load, on the height of TOPS for TWC, where the TOPS width was fixed to the vehicle width of 1.08 m. Input data are listed in Table 2.3, where the operator mass is assumed to be 60 kg. That of cargo load is assumed to be 600 kg.

As the figure shows, the effective TOPS height increases from 1.504 m up to 1.527 m for a vehicle with an operator. The necessary height of TOPS reaches 1.769 m, which seems unrealistically large as the additional structure for an agricultural carrier if a further load is added to the cargo bed. For

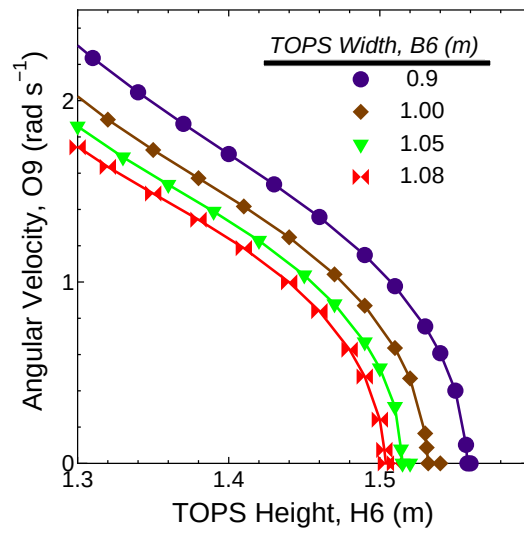


Figure 2.10: Simulation results of the three-wheeled carrier SL30 (vehicle only).

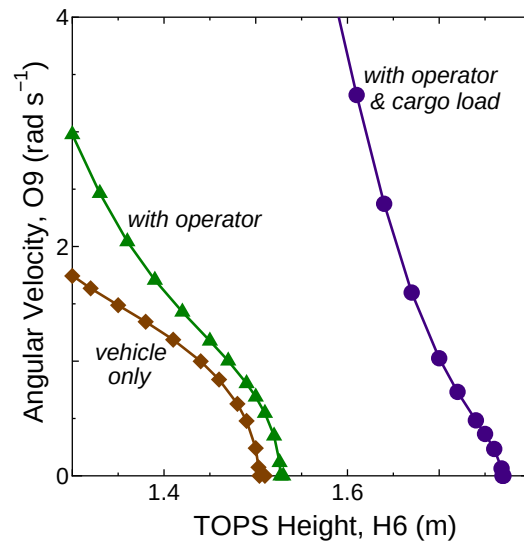


Figure 2.11: Simulation results of the three-wheeled carrier with additional loads.

reference, the results of the effective TOPS height for FWC are 1.411 m (vehicle only), 1.456 m (vehicle with operator), and 1.746 m (vehicle with operator and cargo load) using catalog data of a four-wheeled agricultural carrier by the currently developed program for FWC. The higher height of TOPS obtained in TWC might derive from the fact that the TWC vehicle mass is greater than that of FWC while maintaining the same assumed roll moment of inertia.

It is noteworthy that the result shown in Fig. 2.11 is based on parameters obtained mostly from the available catalogue and with a assumed moment of inertia without comparing with experimental result of overturning using an actual carrier vehicle. However, overturning experiments using real vehicles are time-consuming and expensive. To confirm the validity of the developed program cost-effectively, such model-based experiments on three-wheeled or four-wheeled carriers should be conducted with all known parameters.

2.4 Model-based verification of developed program

2.4.1 Introduction

Table 2.4: Parameters of three-wheeled and four-wheeled models.

Parameters	TWC Model	FWC Model
B_f , m	0.12	none*
B_{rw} , m	0.09	0.09
B_0 , m	0.03	0.03
B_{01} , m	0.02	none*
B_6 , m	0.11	0.11
B_7 , m	0.12	0.12
D_2 , m	0.062	0.62
D_3 , m	0.062	0.62
H_f , m	0.062	none*
H_1 , m	0.055	0.053
H_6 , m	varied [‡]	varied [‡]
H_7 , m	0.074	0.74
L_f , m	0.062	none*
L_2 , m	0.043	0.072
L_3 , m	0.111	0.082
L_6 , m	0.067	0.028
L_7 , m	0.123	0.152
M , kg	0.714	0.850
Q , kg m ²	0.001697	0.002490

*For the FWC model, these parameters are unnecessary

[‡]Cases of H_6 : increased until O9 becomes 0

A three-wheeled model of about one-tenth scale of a real carrier was prepared to assess the accuracy of result obtained using the developed program for a three-wheeled agricultural carrier. It is noteworthy that the consideration of the similitude law was not applied to the model because construction of the model is based on the ease of manufacturing. All the necessary parameters of these models can be measured for use in computer programs.

The center of gravity S of model vehicles is designed to become close to the that of real carriers by changing the material for the upper parts of the model by aluminum and for wheels by vinyl chloride. By changing the total number of front wheels, a four-wheeled model can also be constructed. Parameters of these models are listed in Table 2.4. The roll moment of inertia was measured by application of a method of two string-supported torsional pendulum to the model. The model frame height can be increased from 117 mm to 227 mm on a 5 mm basis.

The slope was covered by plates of low rebound material (Hanenite GP60L; Naigai Rubber Industry Co. Ltd.), each of which was 5 mm thick $\times$ 350 mm wide $\times$ 500 mm long, with repulsion elasticity of 2% (Naigai Rubber Industry, 2014).

2.4.2 Experiments

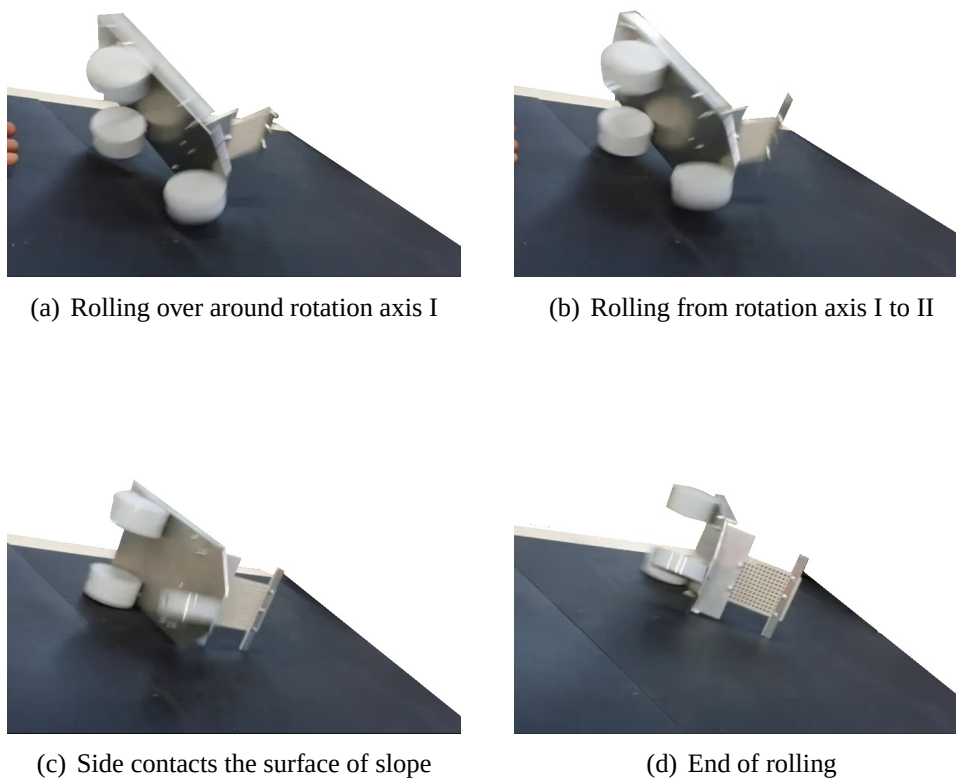


Figure 2.12: Example of a rollover of TWC model on the slope in the experiments ($H_6=172$ mm).

The model vehicle was rolled over a slope of 1:1.5 by its own potential (or gravitational) energy after manually and slowly upsetting its mountain-side edge of the vehicle. As the initial posture on the slope surface, the front wheel of the model vehicle was steered so that the side-edge point of left rear wheel can be set on a parallel contour line on the slope, as shown in Fig. 2.3. The TOPS frame height can be increased with an increment of 5 mm in the experiment. The experiment for both TWC and FWC was started from the set of the lowest frame height (117 mm). Three repetitions of experiments for the same condition were applied to ascertain whether the vehicle was rolling over continuously, or not. The frame height was increased on a 5 mm basis up to 227 mm (23 cases), and the video was taken in each experiment. In all, 69 experiments were designed for each model vehicle of TWC and FWC. When the TOPS frame of a model vehicle becomes sufficiently high to prevent continuous rolling on the surface of slope, the frame height is chosen as the necessary height for model TOPS.

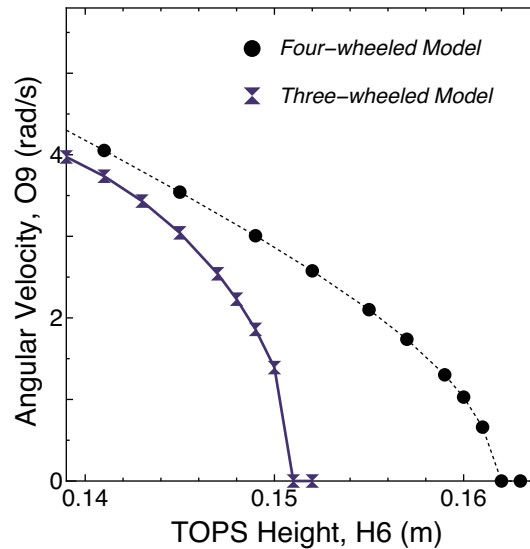


Figure 2.13: Simulation result of protective height for TWC and FWC models.

Fig. 2.12 presents an example of successive photographs extracted from the video data taken during overturning of a model vehicle on the slope.

From the experiments, results showed that the necessary protective height of frame was 172 mm for the TWC model and 217 mm for the FWC model.

2.4.3 Results of simulation

Using the parameters listed in Table 2.4, the overturning of both three-wheeled and four-wheeled model carriers can be analyzed using the developed program.

Figure 2.13 presents results of simulation for the FWC and the TWC models. In the figure, the height of TOPS necessary to prevent successive overturning of model vehicles is observable. For the TWC model, the angular velocity does become zero when the frame height is set as 151 mm. Moreover, the necessary height of frame of the FWC model was found as 162 mm.

2.4.4 Discussion

By comparing the results of model experiments with those of simulations, the numerical result shows the same tendency by which the protective frame height for the three-wheeled model becomes smaller than that for the four-wheeled model. Consequently, it was verified qualitatively that the developed program has sufficient capability to explain the difference of TOPS height between TWC and FWC.

However, a quantitatively large difference of TOPS height exists in the model experiments and that in the numerical analysis using all the measured parameters listed in Table 2.4. Because the developed program for a three-wheeled carrier is based on a program for arm tractor rolling on a slope with reverse definition of parameters, there might be limited applicability of the tractor program to small agricultural carriers such as those observed in this study. For further detailed comparison between experiments and numerical analysis, observations related to the real angular velocity might be helpful to elucidate the carrier vehicle behaviours in rolling over, although this point is beyond the scope of this study.

The original program OECD (2017) cannot calculate the displacement in a tangential direction during calculation of angular velocity when the potential energy is converted to kinetic energy, which implies ignorance of the energy loss caused by sliding over the slope surface in the current program. Moreover, some bounces of model vehicles observed in these experiments would generate great deviation of rotation axes and cause changes in the potential energy and released kinetic energy. In the current program, the rollover behaviour of a vehicle is not a result of pure dynamical behaviour but a result of rotation axis defined by two contact points preassigned by the geometry of the vehicle without consideration of the effect of friction between the surface of slope and contact points.

For precise prediction of necessary TOPS height, new development of computer program following the physics of rolling or sliding of three-dimensional vehicle models should be introduced.

2.5 Conclusion

A program for overturning processes of a three-wheeled agricultural carriers was developed based on the program for analysis of farm tractor overturning on a slope (OECD, 2017). The program development was realised by modifying the program of four-wheel carrier by adding the calculation on the initial steered posture of the vehicle.

The program can estimate the necessary height of the tip-over protective structure for both three-wheeled carriers and four-wheeled carriers once the data for dimensional specifications and the mass and the roll moment of inertia were obtained. The current edition of the program can be used as a safety instruction tool for the design of optimum height of a safety frame for the manufacture of agricultural carriers.

For more precise prediction of necessary TOPS height for three-wheeled agricultural carriers, a computer program based on contact physics, including rolling or sliding of vehicle models, should be developed.

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Chapter 3

Verification of Collision Detection by Physics Engine

3.1 Introduction

In Chapter 2, the developed program for a three-wheeled carrier followed the same algorithm originally developed for the analysis of overturning of farm tractor (OECD, 2017). Moreover, all the contacting coordinates should be prepared a priori, and relative tangential velocities and Coulomb's friction law at contact points could not be considered. For more precise, or realistic, description of overturning behaviour and estimation of critical TOPS height, a new computer program should be introduced. A physics engine is expected to be applied to simulate the dynamic with Coulomb's friction of a carrier model on a slope.

In this chapter, an alternative approach of applying the physics engine is attempted.

The purpose is to examine whether or not the physics engine: Bullet is applicable for the analysis of sideways overturning behaviour of small agricultural carrier models with various TOPS height.

3.2 Outline of the physics engine: Bullet

3.2.1 Rigid body dynamics

Bullet provides various rigid shapes including sphere, box, cylinder and so on for collision dynamics. Moreover, soft body shapes (cloth and rope) are supported in the engine. Figure 3.1 shows the data structures and computation stages in the physics pipeline of Bullet (Erwin, 2013). This pipeline is executed from left to right, starting a simulation by applying gravity, and ending by position integration.

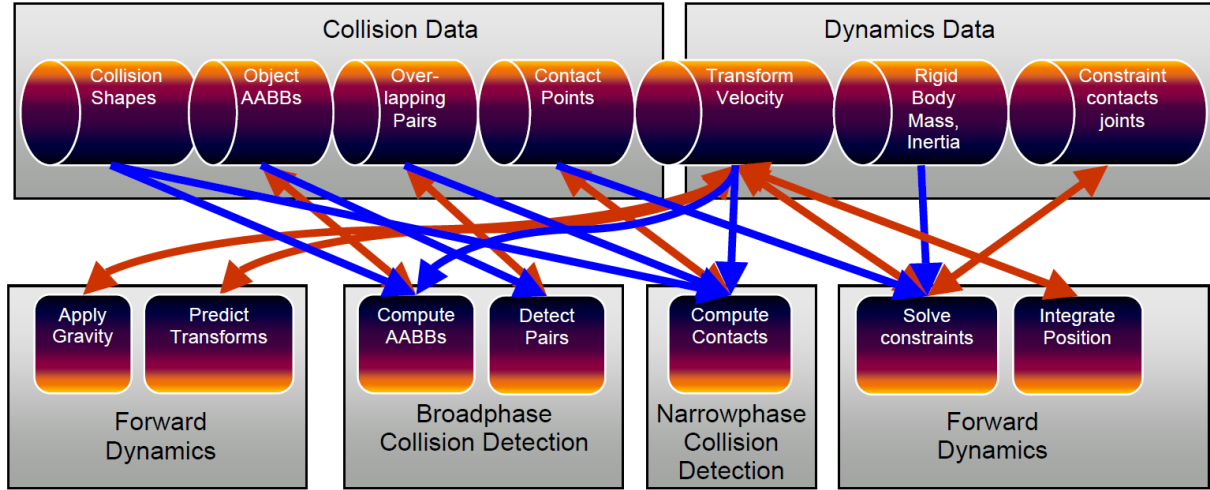


Figure 3.1: Rigid body physics pipeline in Bullet (Erwin, 2013).

3.2.2 Collision model

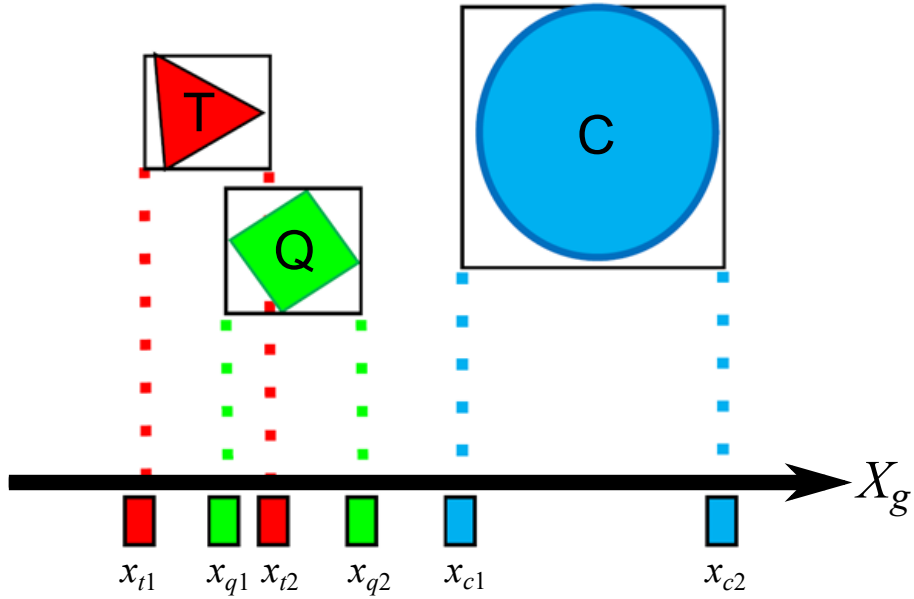


Figure 3.2: Collision models like triangle, quadrangle and circle with broad-phase and feedbacks on axis (Erwin, 2013).

Simple collision shape, such as box and cylinder, can have its own broad-phase and narrow-phase, which maximises the performance and quality in one calculation time step.

After the construction of collision model, the broad-phase as the box can be drawn for each time step, as shown in Fig. 3.2.

3.2.3 Collision detection

Collision detection is an important aspect of physics engine that calculates the process of comparing two rigid-bodies, detecting whether they penetrate or will penetrate and then calculating the exact points of contact. To shorten the computing time of collision detection, the calculation is break down into two basic phases: a broad-phase and a narrow-phase (Ballanytyne, 2007). The broad-phase is designed to reduce the number of intersections required by using various culling techniques. The narrow-phase then points out which pairs of objects are intersecting, and calculates the point of intersection in another step.

During a collision, the following four assumptions are applicable to rigid bodies (Erleben et al., 2005):

- The collision duration is zero.
- Only impulsive forces need be used to avoid penetration.
- Positions of contact coordinates are the same before and after the collision.
- Non-impulsive forces have no effect during the collision.

Broad-phase

This is an effective way to calculate whether the broad-phase of two objects had been intersecting or not rather than calculating the bounding pair.

Bullet physics engine uses the Axis-Aligned Bounding Boxes method, which is simplicity and effective culling that only computes the set of intersecting pairs along the aligned axis, as called “sweep and prune” algorithm (Eberly, 2003). The maximum and minimum coordinates of bounding boxes on the global X_g -axis as shown in Fig. 3.2 are updated to the feedback system, Bullet prunes those un-connected pairs and keeps the X_g -axis intersecting pairs. In the feedback system, as shown in Fig. 3.2, the triangle and quadrangle are the intersecting pair and circle is unconnected to each other along the X_g -axis. Then, Bullet repeats the same process that checks the coordinates of bounding boxes on the global Y_g -axis in those X_g -axis intersecting pairs. Finally, Bullet wipes out the pairs which have no intersected coordinates on Z_g -axis in the remaining $X_g \times Y_g$ -axis intersecting pairs. The survived $X_g \times Y_g \times Z_g$ -axis intersecting pairs will be marked as the potential contact pairs.

Narrow-phase

After the broad-phase selected the intersecting pairs which means those two objects is contacting or will be contacting, the narrow phase calculates a separating plane between two objects. The contact points can only appear on this separating plane. If the objects are penetrating, then the collision detection will reverse to the last calculated contact points.

3.2.4 Collision dynamics in Bullet

Bullet uses an impulse-based method (Mirtich and Canny, 1995) that is characterised by the fact that the method represents every interaction between bodies in the world through collision. Although some other approaches exist to model contact dynamics such as penalty-based methods and the constraint-based methods, the impulse-based method is simpler to implement. During a collision, the following four assumptions are applicable to rigid bodies (Erleben et al., 2005):

- The collision duration is zero.
- Only impulsive forces need be used to avoid penetration.
- Positions of contact coordinates are the same before and after the collision.
- Non-impulsive forces have no effect during the collision.

Friction

We can introduce the stick–slip phenomenon by setting the coefficient of friction μ appropriately. By ascertaining the force in the normal direction, $\mathbf{F}_n$, and the relative tangential non-zero velocity $\mathbf{u}_t$ on a contact point, we then have dynamic or sliding friction force $\mathbf{F}_t$ as

$$\mathbf{F}_t = -\mu \frac{\mathbf{u}_t}{\|\mathbf{u}_t\|} \|\mathbf{F}_n\|. \quad (3.1)$$

Moreover, two cases of static friction exist: stable and unstable. In a stable static friction, the time derivative of the tangential contact velocity becomes zero. We can obtain Eq. (3.2) as shown below.

$$\dot{\mathbf{u}}_t = 0 \quad , \quad \|\mathbf{F}_t\| \leq \mu \|\mathbf{F}_n\| \quad (3.2)$$

The unstable static friction can be expressed as

$$\mathbf{F}_t \cdot \mathbf{u}_t \leq 0 \quad , \quad \|\mathbf{F}_t\| = \mu \|\mathbf{F}_n\|. \quad (3.3)$$

The difference between dynamic friction and unstable static friction is that we cannot describe the direction of slide except for the fact that $\mathbf{u}_t \neq 0$ in cases of unstable static friction.

The coefficient of friction μ usually has different values for static friction and dynamic friction. For our dynamics model, the same value of μ is assumed for simplicity.

Restitution

After impact of a vehicle against the rigid slope surface, normal velocity u'_n is obtainable using the coefficient of restitution as

$$u'_n = -eu_n \quad , \quad 0 \leq e \leq 1, \quad (3.4)$$

where u_n stands for the normal velocity at the center of gravity of an object before impact. Also, u'_n signifies the normal velocity after impact. In addition, e denotes the coefficient of restitution.

If $u'_n \neq 0$, then the collision will result in bouncing of the object after impact. If $u'_n = 0$, then the object will stick to the slope surface as a result of perfect inelastic collision.

Applied impulse and velocity

Because the restitution and friction are expected to affect the dynamic contact behaviour of the simulated vehicle, those impulses must be applied to each collision model (Featherstone, 2008). Applied impulse $\mathbf{J}$ causes a change in both relative linear velocity $\Delta \mathbf{v}$ and relative angular velocity $\Delta \boldsymbol{\omega}$ for two colliding models, i.e., vehicle and slope, as presented below.

$$\Delta \mathbf{v}_v m_v = \mathbf{J}_v \quad , \quad \Delta \boldsymbol{\omega}_v \mathbf{I}_v = \mathbf{r}_v \times \mathbf{J}_v \quad (3.5)$$

$$\Delta \mathbf{v}_s m_s = \mathbf{J}_s \quad , \quad \Delta \boldsymbol{\omega}_s \mathbf{I}_s = \mathbf{r}_s \times \mathbf{J}_s \quad (3.6)$$

In those equations, m_v and m_s respectively denote the masses for the model vehicle and slope, $\mathbf{I}_v$ and $\mathbf{I}_s$ are the inertia tensor of model vehicle and slope, and $\mathbf{r}_v$ is a position vector that points from the respective center of mass of the model vehicle to the collision point. Because the slope is a static model, m_s , and $\mathbf{I}_s$ are not regained: $\Delta \mathbf{v}_s$, and $\Delta \boldsymbol{\omega}_s$ become zero.

Therefore, the change in contact velocity of model vehicle can be expressed as

$$\Delta \mathbf{u}_v = \Delta \mathbf{v}_v + \Delta \boldsymbol{\omega}_v \times \mathbf{r}_v, \quad (3.7)$$

where $\Delta \mathbf{u}_v$ is used to define the after-contact velocity $\mathbf{u}_{v|a}$ with the before-contact velocity $\mathbf{u}_{v|b}$ associated, as

$$\mathbf{u}_{v|a} = \mathbf{u}_{v|b} + \Delta \mathbf{u}_v. \quad (3.8)$$

Substituting Eq. (3.5) into Eq. (3.7) for $\Delta \mathbf{v}_v$ and $\Delta \boldsymbol{\omega}_v$, we have

$$\Delta \mathbf{u}_v = \frac{\mathbf{J}_v}{m_v} + (\mathbf{I}_v^{-1}(\mathbf{r}_v \times \mathbf{J}_v)) \times \mathbf{r}_v. \quad (3.9)$$

Using the identity matrix $\mathbf{1}$ as $\mathbf{M}_v = m_v \mathbf{1}$, the diagonal mass matrix $\mathbf{M}_v$ can be imported. Also, Eq. (3.9) might be transformed as

$$\Delta \mathbf{u}_v = (\mathbf{M}_v^{-1} + \mathbf{r}_v^{\times T} \mathbf{I}_v^{-1} \mathbf{r}_v^{\times}) \mathbf{J}_v, \quad (3.10)$$

where we used the expression for cross product as in Lee and Hashash (2015).

Because $\Delta \mathbf{u}_s = 0$, the change in relative contact velocity between the model vehicle and slope becomes the following.

$$\Delta \mathbf{u} = \Delta \mathbf{u}_v - \Delta \mathbf{u}_s = (\mathbf{M}_v^{-1} + \mathbf{r}_v^{\times T} \mathbf{I}_v^{-1} \mathbf{r}_v^{\times}) \mathbf{J}_v \quad (3.11)$$

Let the collision matrix $\mathbf{K}$ be defined as

$$\mathbf{K} = \mathbf{M}_v^{-1} + \mathbf{r}_v^{\times T} \mathbf{I}_v^{-1} \mathbf{r}_v^{\times}. \quad (3.12)$$

Then, from Eq. (3.11),

$$\Delta \mathbf{u} = \mathbf{K} \mathbf{J}_v. \quad (3.13)$$

To import both restitution and friction into our dynamics modeling, we assume that the model vehicle will adhere to the point of impact when a static situation, where the after-contact tangential velocity $u_{t|a}$ and the after-contact normal velocity $u_{n|a}$ are described, using unit normal vector $\mathbf{n}$ of slope surface, as Guendelman (2003)

$$u_{t|a} = 0 \quad , \quad u_{n|a} = -e u_{n|b} \mathbf{n}. \quad (3.14)$$

We compute the normal and tangential components of the impulse $\mathbf{J}$ as

$$\mathbf{J}_n = (\mathbf{J} \cdot \mathbf{n}) \mathbf{n} \quad , \quad \mathbf{J}_t = \mathbf{J} - \mathbf{J}_n. \quad (3.15)$$

If $\|\mathbf{J}_t\| \leq e \|\mathbf{J}_n\|$, then the model vehicle adheres to the slope surface because of static friction. The explanation can be found in Erleben et al. (2005). Alternatively, let a unit vector in sliding direction be $\mathbf{t} = \mathbf{u}_{t|b} / \|\mathbf{u}_{t|b}\|$. We can compute the sliding friction with the impulse as

$$\mathbf{J} = j_n \mathbf{n} + \mu j_n \mathbf{t}, \quad (3.16)$$

where j_n represents the magnitude of the normal impulse $\mathbf{J}_n$.

The after-contact velocity $\mathbf{u}_a$ and the before-contact velocity $\mathbf{u}_b$ of the model vehicle are known as

$$\mathbf{u}_a = \mathbf{u}_b + \Delta \mathbf{u}. \quad (3.17)$$

The after-contact tangential velocity $u_{t|a}$ is zero as Eq. (3.14), the after-contact normal velocity $u_{n|a}$ can be expressed as follows by taking the dot product of Eq. (3.13) with $\mathbf{n}$ from the left.

$$\mathbf{n} \cdot \Delta \mathbf{u} = \Delta u_n = u_{n|a} - u_{n|b} = \mathbf{n}^T \mathbf{K} \mathbf{J} \quad \text{or} \quad u_{n|a} = u_{n|b} + \mathbf{n}^T \mathbf{K} \mathbf{J} \quad (3.18)$$

Using Eq. (3.14),

$$-e u_{n|b} = u_{n|b} + \mathbf{n}^T \mathbf{K} \mathbf{J}. \quad (3.19)$$

Substituting the definition of $\mathbf{J}$ of Eq. (3.16) into Eq. (3.19), then

$$-e u_{n|b} = u_{n|b} + \mathbf{n}^T \mathbf{K} (j_n \mathbf{n} - \mu j_n \mathbf{t}). \quad (3.20)$$

Finally we can find j_n such that

$$j_n = \frac{-(1+e)u_{n|b}}{\mathbf{n}^T \mathbf{K} (\mathbf{n} - \mu \mathbf{t})}. \quad (3.21)$$

After computing $\mathbf{J}$ by Eq. (3.16) using the result of j_n from Eq. (3.21), the changes in both linear and angular velocities are obtainable by Eq. (3.5). Then, the after-contact linear velocity and the after-contact angular velocity with respect to the center of gravity of model vehicle can be described as

$$\mathbf{v}_a = \mathbf{v}_b + \Delta \mathbf{v} \quad , \quad \boldsymbol{\omega}_a = \boldsymbol{\omega}_b + \Delta \boldsymbol{\omega}. \quad (3.22)$$

3.3 Numerical experiments on sideways overturning of agricultural carriers

3.3.1 Specification of small model vehicle

A small TWC model of about 1/10 scale of a real agricultural carrier was constructed to observe the behaviour of sideways overturning on a slope (Sun et al., 2017). By changing the front parts of the small TWC model, a small FWC model is also made as shown in Fig. 3.3.

The mass of each accessory are listed in Table 1 with their definition in Fig. 3.3. The height of model frame can be adjusted from 117 mm to 227 mm on 5 mm basis in the experiments and on 1 mm basis in the simulation.

In Bullet, the carrier model is built by provided components, box shape for chassis, cargo bed and frame along with cylinder shape for wheel. After setting up the mass and size of those accessories, Bullet calculated the roll moment of inertia around gravity center for both TWC and FWC, which had similar result to the measurement as shown in Table 3.1. More details of parameters for small TWC model can be found in Sun et al. (2017).

Overturning process of three-wheeled carrier has to start from the position with the steered front wheel with angle ϕ_0 on the slope of 1:1.5 as shown in Fig. 3.4 (National Agency of Vehicle Inspection, 2014). The parameters of slope surface in Bullet for both small TWC model and FWC model simulations were decided by the measurement as $\mu = 0.5$ and $e = 0.35$.

3.3.3 Initial condition of rollover of vehicle model for three-wheeled carrier

The starting line of overturning shown in thick line in Fig. 3.4 is parallel to the global X_g -axis. When the gravity center of carrier model S exceeds the border plane defined by the extended vertical line parallel to Z_g -axis and intersects with the starting contour line (considering the dry friction or sliding friction situation) by the applied roll angle β_0 , the model will begin to roll over the surface of slope. As for the TWC model, the initial setting of quaternion on the gravity center of carrier model S is $\tilde{q} = \tilde{q}_2 \tilde{q}_1$ where $\tilde{q}_1 = (0, 0, 1, \phi_0)$, $\tilde{q}_2 = (0, 1, 0, \beta_0)$.

3.4 Results and discussion

3.4.1 Result of small TWC model

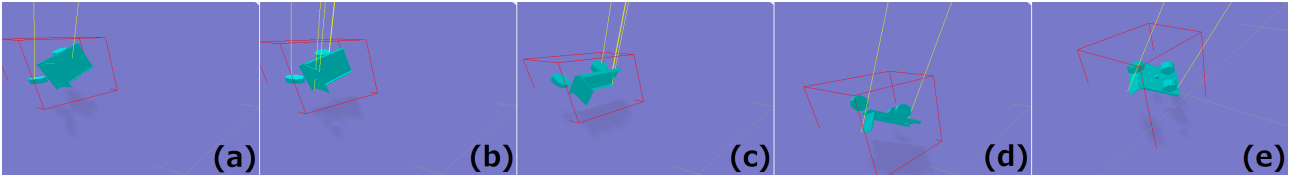


Figure 3.5: Continuous overturning behaviour in Bullet of small TWC model (TOPS height: 141 mm).

Figure 3.5 shows the sideways overturning behaviour of Bullet simulation for the small TWC model when the TOPS height is 141 mm. The red box is the broad-phase of carrier model and the yellow line vertical to the slope surface portrays the contact point. The result of rollover of calculated model can be obtained in every 1/60 s in the simulation.

The contacting process can be observed by the changing of contacting point:

- Firstly, front and rear wheels contacting ground;
- Front and rear wheels along with the front corner of chassis contacting ground momentarily;
- Secondly, rear wheel and front corner of chassis contacting ground (not present in Fig. 3.5);
- Side part of carrier model contacting ground momentarily;
- TOPS and rear side of cargo bed contacting ground;
- Upside-down.

According to ASAE (2003), a vehicle with TOPS rolling upside-down means the TOPS do not have enough protective capability to prevent the vehicle from continuous overturning. Even the small TWC model with 141 mm height TOPS kept this upside-down posture and slipping along the ground

to the bottom in Bullet simulation. Thus, the protective effect of TOPS would still be regarded as NOT SAFE.

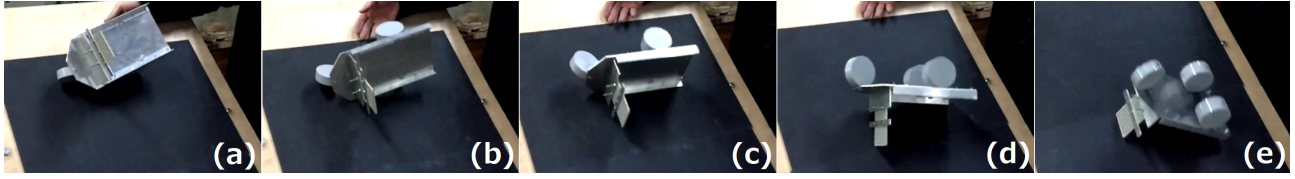


Figure 3.6: Continuous overturning behaviour in experiment for small TWC model (TOPS height 137 mm).

Figure 3.6 is the experimental images by high-speed camera recording, where the overturning behaviour can be separated into several steps by the accessories which were contacting the ground: firstly front and rear wheels; secondly front corner of chassis; thirdly side part of carrier; finally TOPS. Then, the height of TOPS was not large enough to inhibit the rolling of vehicle, and became upside-down.

Since the carrier with 137 mm resulted in continuous overturning, the critical height of TOPS for this carrier should be larger. Comparing to this result of experiment, the overturning behaviour by Bullet in Fig. 3.5 is similar to the experiment behaviour in each contacting step.

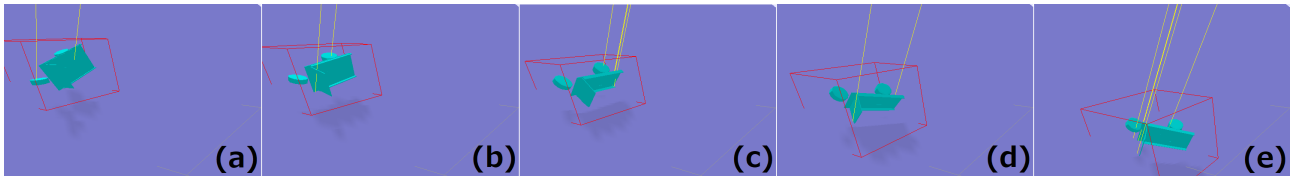


Figure 3.7: Non-continuous overturning behaviour in Bullet for small TWC model (TOPS height 142 mm).

Figure 3.7 shows the simulated sideways dynamic of the small TWC model when the TOPS height is 142 mm with broad-phase and narrow-phase.

The contacting process is the same as the case of 141 mm TOPS height before the side part of carrier model contacting ground, because the increasing height of TOPS has scarcely effect on the dynamic behaviour before TOPS impacting slope itself. When the TOPS made contact to the surface of slope, the contacting process became:

- TOPS and rear side of cargo bed contacting ground;
- Vehicle rolling around the axis produced by the two contacting points above, toward the bottom of slope;
- Vehicle rolling around the axis produced by the two contacting points above, toward the top of slope (not present in Fig. 3.7);
- Vehicle back to the posture that side part of carrier model contacted ground.

This time, the vehicle stepped into the non-continuous overturning situation, which also means the TOPS stopped the vehicle from rolling into upside-down posture. Moreover, the small TWC model would not be continuous overturning anymore if the height of TOPS was not lower than 142 mm in Bullet simulation, which means the critical height of TOPS for this TWC model is 142 mm.



Figure 3.8: Non-continuous overturning behaviour in experiment for small TWC model (TOPS height 142 mm).

The experimental dynamic recording of small TWC carrier with 142 mm TOPS is presented in Fig. 3.8. The overturning process is the same as the Bullet simulation in Fig. 3.7 as follows: firstly front and rear wheels; secondly front corner of chassis; thirdly side part of carrier; then TOPS; finally side part of carrier.

Because the overturning tendency of this carrier had been successfully stopped by TOPS, so the critical height of TOPS for this carrier should be no larger than 142 mm. Since the height in experiment can be adjusted by only 5 mm basis, the critical height of TOPS for small TWC carrier was detected at 142 mm combining the results in Fig. 3.6 and Fig. 3.8.

3.4.2 Result of small FWC model



Figure 3.9: Continuous overturning behaviour in Bullet of small FWC model (TOPS height: 152 mm).

Figure 3.9 shows the continuous overturning behaviour of the small FWC model conducted by Bullet when TOPS height is 152 mm. The overturning process can be concluded as:

- Firstly, front and rear wheels contacting ground;
- Side part of carrier model contacting ground;
- TOPS contacting ground;
- Vehicle rolling around the contacting point on TOPS, toward the bottom of slope;
- Upside-down.

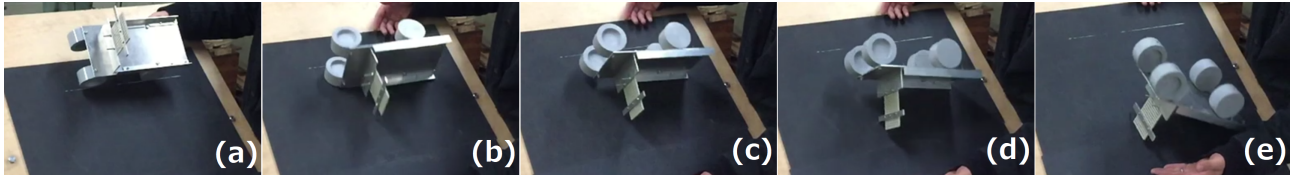


Figure 3.10: Continuous overturning behaviour in experiment for small FWC model (TOPS height 157 mm).

Figure 3.10 is the overturning behaviour in experiment with the TOPS height 152 mm. Similar to the result of simulation, the contacting process can also be partitioned: firstly front and rear wheels; secondly side part of carrier model; then TOPS. The same as Fig. 3.9, the TOPS is not high enough to stop the continuous overturning, so the carrier kept rolling over into upside-down.

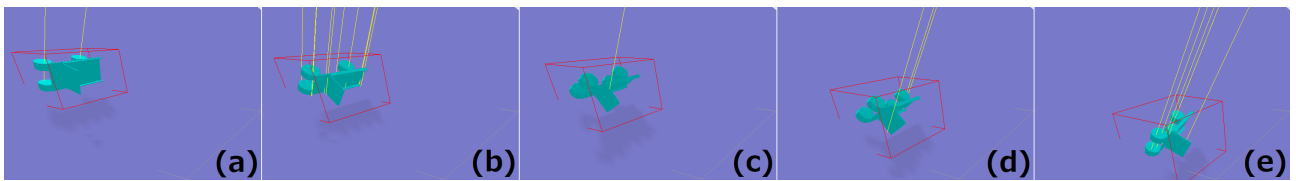


Figure 3.11: Non-continuous overturning behaviour in Bullet for small FWC model (TOPS height 153 mm).

As the height of TOPS of Bullet model being increased to 153 mm, the dynamic of the small FWC model became a non-continuous overturning in Fig. 3.11 as follows:

- Vehicle rolling toward the bottom of slope, around the axis conducted by front and rear wheels contacting ground;
- Side part of carrier model contacting ground;
- Vehicle rolling around the contacting point of TOPS, toward the bottom of slope;
- Vehicle rolling around the contacting point of TOPS, toward the top of slope;
- Vehicle back to the posture that side part of carrier model contacted ground.

If the height of TOPS was 1 mm lower than 153 mm, the vehicle will still keep rolling after TOPS contacting the ground. Therefore, 153 mm may be said the critical height of TOPS to prevent the small FWC model from continuous overturning.

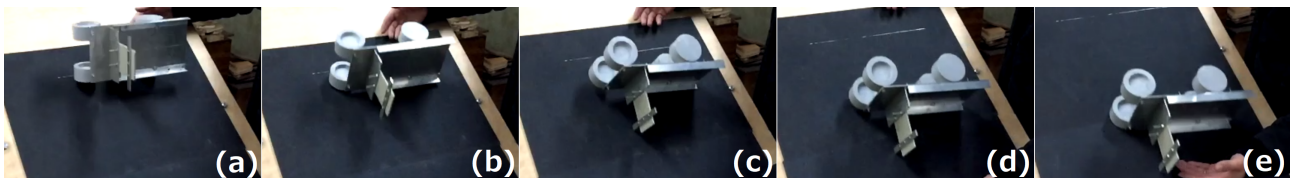


Figure 3.12: Non-continuous overturning behaviour in experiment for small FWC model (TOPS height 162 mm).

Table 3.2: Comparison of critical TOPS height between simulation and experiment

	Small TWC Model	Small FWC Model
Critical height in simulation	0.142 m	0.153 m
Critical height in experiment	0.142 m	0.162 m

The experimental sequence recording is presented in Fig. 3.12, when the TOPS height of small FWC model is 162 mm. The overturning process is the same as the Bullet simulation in Fig. 3.11 as follows: vehicle rolling toward the bottom of slope; TOPS contacting the surface of slope; vehicle rolling back toward the top of slope; side of vehicle contacting ground.

Because the adjustment for the height of TOPS can only be made by 5 mm basis, so the critical height for this small FWC carrier in experiment is 162 mm.

3.4.3 Discussion

Table 3.2 summarises the result of critical TOPS height. The result of FWC model in simulation is 9 mm lower than the experiment one, although similar critical height is obtained for TWC model.

As shown in Figs 3.5, 3.6, 3.7 and 3.8, the front corner of chassis impacted the ground earlier than impacts of the side of vehicle in case of the small TWC model. But in Figs 3.9, 3.10, 3.11 and 3.12, there had no extra part for the small FWC model to contact the ground before the side of the vehicle. Thus, the TWC model always has one additional contacting process than the FWC model, and releases more energy before the TOPS contacting ground. This might explain the difference of result in Table 3.2 that the simulated critical TOPS height of TWC became closer to the experimental result than that of FWC.

The simulated sideways overturning behaviours using physics engine are in accordance with that in experiment by comparing those images above. The collision detection also make the correct decisions about those contacting points, which matches the calculating flowchart using OECD program with contacting coordinates. With the introduction of friction, the result of Bullet simulation tends to be more visually acceptable than previous research (Sun et al., 2017).

3.5 Conclusion

Numerical simulation using a physics engine can be applied with sufficient accuracy to simulate sideways overturning behaviour for both small three-wheeled carrier model and small four-wheeled carrier model over a slope. Thus, the prediction of critical TOPS height for agricultural carriers will easily be realised when the physics engine is applied to our previously developed in-house code (Sun et al., 2017).

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Chapter 4

Overturning Behaviour of Agricultural Carrier by Physics Engine

4.1 Introduction

In Chapter 3, a physics engine called Bullet, which could easily express three-dimensional rigid body dynamics (including collision detection) as well as soft body dynamics (Erwin, 2013), was applied to successfully simulate sideways overturning of an agricultural carrier on a slope.

In Chapter 4, the numeralization of sideways overturning dynamics of agricultural carriers is examined. Moreover, the three dimensional simulation results using Bullet are evaluated comparing to the previous researches using OECD code in relation to the coefficient of friction and coefficient of restitution.

4.2 Collision model of agricultural carriers in Bullet

Bullet combined the collision detection with the dynamics calculation to simulation the impact performances of the rigid models as the description of Section 3.2 (Page 27).

The collision models for both TWC as shown in Fig. 4.1 and FWC as shown in Fig. 4.2 were able to point the contact points as the yellow lines in those figures, which would be compared to the prepared contact points according to Tsukamoto et al. (2008) and (Sun et al., 2017) using OECD code.

4.2.1 Output

The calculation loop as shown in Fig. 3.1 (Page 28) is modified to intercept and record the dynamic data of collision models.

The dynamic data of collision model can be intercepted in **OpenGLGuiHelper.cpp**. The dynamics of all collision models would be updated every timestep as **for (int i = 0; i < numCollisionObjects; i++)**, as the dynamic of our vehicle model is the only demanded data for recording, the CollisionObjects Number 2 as **if (i==2)** can call the dynamic data recording function.

- Use **colObj->getInterpolationAngularVelocity().getX()**; to read the angular velocity around the center of gravity at X (also Y and Z) which is pitch (also roll and yaw) direction, then the data would be written into **Carrier-AngelVelocity.txt**.

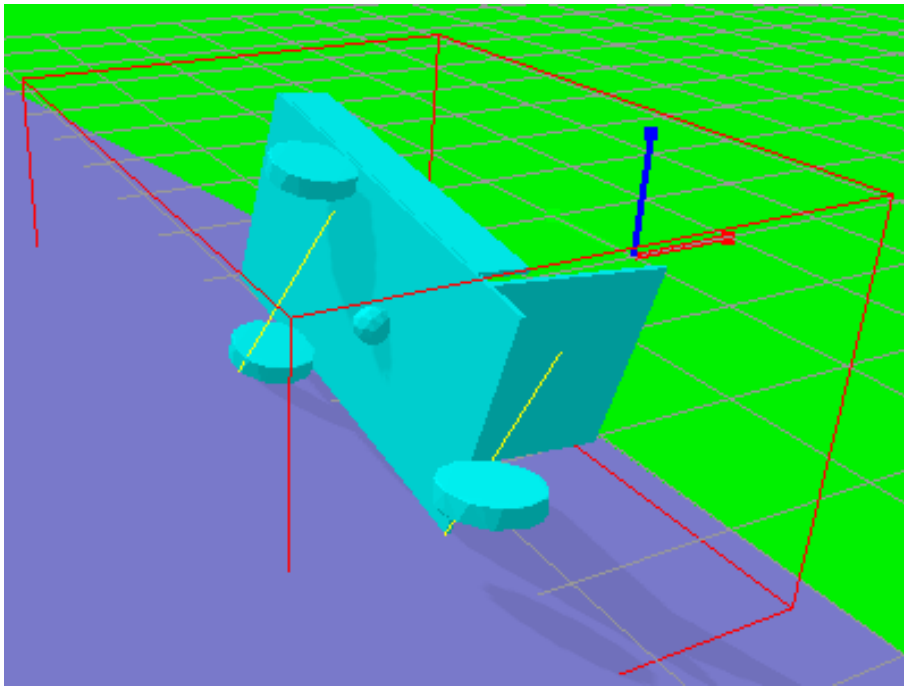


Figure 4.1: Example of Bullet model for a three-wheeled carrier.

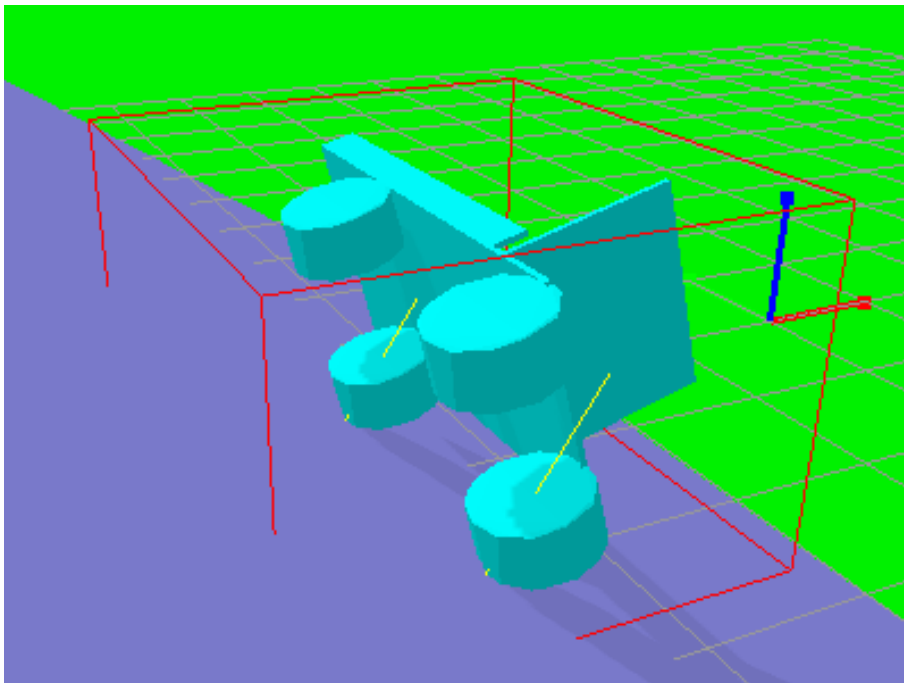


Figure 4.2: Example of Bullet model for a four-wheeled carrier.

- Use `colObj->getInterpolationLinearVelocity().getX()`; to read the linear velocity of the center of gravity at X (also Y and Z) direction, then the data would be written into **Carrier-LinearVelocity.txt**.
- Use `colObj->getWorldTransform().getOrigin().getX()`; to read the coordinate of the center of gravity at X (also Y and Z) direction and `colObj->getWorldTransform().getRotation().getX()`; to read the quaternion around the center of gravity at X (also Y, Z, and ω) direction, the data would be written into **Carrier-Cor.txt**.

The data of collision detection can also be intercepted in **btCollisionWorld.cpp**. Marking the colliding pair number 0 as **if (i==2)** after all colliding pairs has been scanned as **for (int j=0;j<numContacts;j++)**, the data of this colliding pair which not only occurred between slope model and vehicle model but also is the only pair in our Bullet model can be recorded.

- Use `cp.m_positionWorldOnB.getX()` to read the coordinate at X (also Y and Z) direction and save it into **ContactPoint.txt**.
- Use `normal.angle(btVector3(1,0,0))` to read the impulse angle at pitch (also roll (0,1,0) and yaw (0,0,1)) direction and save it into **Carrier-Impulse.txt**.
- Use `contactPoint.m_appliedImpulse` to read the amount of impulse and save it into **Carrier-ResOutTEST.txt**, the pitch component of impulse can be calculated as `contactPoint.m_appliedImpulse * normal.angle(btVector3(1,0,0))`.

It is noticeable that the colliding pair only reported when the contact between slope model and vehicle model did happen. Otherwise, the calculate loop of Bullet would skip this part of calculation for efficiency.

4.2.2 Time step

Even through the manual of Bullet suggested that the time step should be fixed at 1/60 second, the description of **virtual int stepsimulation(timestep, maxSubSteps, fixedTimestep)** in **btDiscreteDynamicsWorld.h**, **btDynamicsWorld.h**, and **btSimpleDynamicsWorld.h** provided the ability to adjust the time step. The example of time step setting would be shown as follows:

- **timestep = 3** means the 0, 1, 2 is calculated with out output, the first output step is 3.
- **masSubStep = 3** means the 0, 3, 6, ... $3n$ is calculated and outputted, the others (1, 2, ... $3n+1$, $3n+2$) is calculated with out output.
- **fixedTimestep = 1/60** means the calculation loop runs every 1/60 second, 1/122 is the boundary for refined calculation as out test.

4.2.3 Setting of contact condition

In this study, the pair of contacting model include the three-wheeled carrier model on slope model in **BasicExample.cpp**, and the four-wheeled carrier model on slope model in **RollingFriction-Demo.cpp**. The contact condition between the carrier model and slope model can also be adjusted as follows:

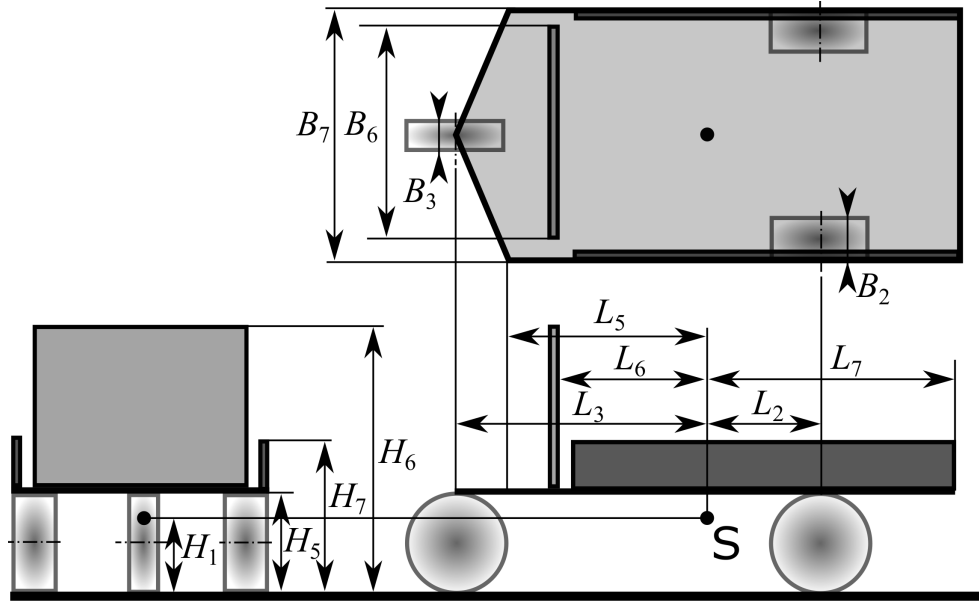


Figure 4.3: Definition of parameters for a three-wheeled carrier (Sun et al., 2017).

- **body->setFriction(CarrierFriction);** enables the friction coefficient of the carrier model to be a specified number. In our study, the “(CarrierFriction)” is set at 0.2 – 0.6 in various group of simulations, which can be simply replaced with a number as **body->setFriction(0.1);**
- **body->setFriction(CarrierRestitution);** means the restitution coefficient of the carrier model to be a specified number.
- **body->setFriction(RoadFriction);** means the friction coefficient of the slope model can also be adjusted. In our study, the “(RoadFriction)” is always set at 1.
- **body->setFriction(RoadRestitution);** means the restitution coefficient of the slope model can also be adjusted. In our study, the “(RoadRestitution)” is always set at 1.

It is noticeable that the friction coefficient between carrier model and slope model would be 0.2 only if the (CarrierFriction) is 0.2 at the same time (RoadFriction) = 1. The setting of restitution coefficient between carrier model and slope model should obeys the same rule stated above.

4.3 Numerical experiments

4.3.1 Specifications of the small model vehicle

A small TWC model of about one-tenth scale of a real agricultural carrier was constructed to observe the sideways overturning behaviour on a slope (Sun et al., 2017).

Parameters of the model are listed in Table 4.1 with their definitions in Fig. 4.3. The model frame height (H_6) can be adjusted from 117 mm to 227 mm on 5 mm basis in the experiments (Sun et al., 2017) and on a 1 mm basis in the simulation using Bullet.

Parameters for small FWC model are also presented in Table 4.1 with their respective definitions in Fig. 4.4.

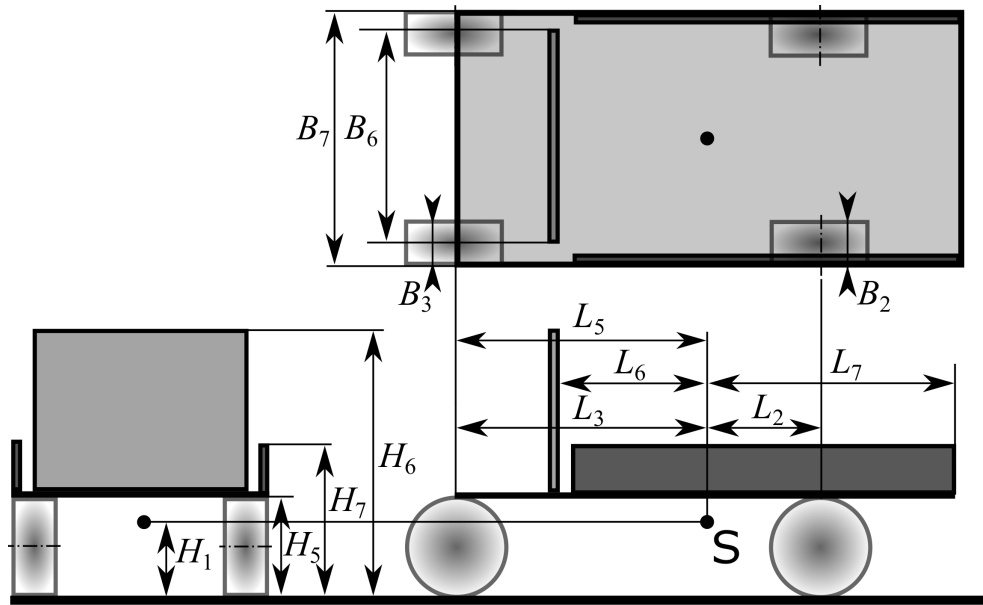


Figure 4.4: Definition of parameters for a four-wheeled carrier (Sun et al., 2017).

Table 4.1: Parameters of a small carrier model.

Parameters	Small TWC Model	Small FWC Model
B_2 , m	0.03	0.03
B_3 , m	0.02	0.03
B_6 , m	0.11	0.11
B_7 , m	0.12	0.12
H_1 , m	0.055	0.053
H_5 , m	0.062	0.062
H_6 , m	varied [‡]	varied [‡]
H_7 , m	0.074	0.074
L_2 , m	0.043	0.072
L_3 , m	0.111	0.082
L_5 , m	0.089	0.082
L_6 , m	0.067	0.028
L_7 , m	0.123	0.152
M , kg	0.714	0.850
Q_{roll}^* , kgm ²	0.001697	0.002277
Q'_{roll}^{**} , kgm ²	0.001725	0.002260

*For both TWC and FWC models: when the height of H_6 is 0.172 m

**Calculated results from Bullet in cases of $H_6 = 0.172$ m

[‡]Height can be adjusted from 0.117 m to 0.227 m on 0.005 m basis

To adjust the density distribution of the model vehicle in correspondence with the experiment, the densities of respective parts (such as wheels, chassis, frame in Fig. 3.3 (Page 33)) of the model were decided. The wheels of both TWC and FWC models were made of vinyl chloride. The chassis and the upper part of small carrier were modeled as aluminum.

After preparation in Bullet, similar roll moments of inertia for the small TWC and FWC models were obtained as presented in the table.

4.3.2 Specifications of full-scale agricultural carriers

Carrier data available in the manuals for the TWC (SL30; Atex Co. Ltd., Japan) and FWC (ES60; CANYCOM, Japan) were used as inputs for numerical simulation using Bullet. The position of the center of gravity and roll moment of inertia were measured at the Institute of Agricultural Machinery (IAM), National Agriculture and Food Research Organization, Japan.

Table 4.2: Full-scale carrier parameters.

Parameters	SL30 TWC	ES60 FWC
B_2 , m	0.105	0.295
B_3 , m	0.1	0.295
B_6 , m	1.08	1.072
B_7 , m	1.21	1.185
H_1 , m	0.387	0.387
H_5 , m	0.471	0.569
H_6 , m	varied $\ddagger$	varied $\ddagger$
H_7 , m	0.76	0.736
L_2 , m	0.597	0.712
L_3 , m	1.228	0.829
L_5 , m	0.869	0.829
L_6 , m	0.519	0.371
L_7 , m	1.442	1.508
M , kg	310	315
Q_{roll}^* , kgm ²	42.6	37.4
Q'_{roll}^{**} , kgm ²	41.1	36.4
Q''_{roll}^{***} , kgm ²	43.3	38.6

*Measured by IAM without TOPS

**Calculated result by Bullet for $H_6 = 1.40$ m

***Calculated result by Bullet for $H_6 = 1.60$ m

$\ddagger$ The height is adjustable in the simulation

4.3.3 Slope condition and initial configuration for the three-wheeled carrier

Overturning of a three-wheeled carrier starts from the position with the steered front wheel at an angle ϕ_0 against the global X_g -axis on the slope surface of 1:1.5, as shown in Fig. 3.4 (Page 33) (National Agency of Vehicle Inspection, 2014). Slope surface parameters in Bullet for both the small carrier model and full-scale carrier model simulations were decided by measurements as $\mu = 0.5$ and $e = 0.35$ as in 3.3.2.

When the center of gravity of carrier model S exceeds the border plane defined by the extended vertical line parallel to Z_g -axis and intersects with the starting contour line (considering the dry friction or sliding friction situation) by the applied roll angle β_0 , the model will begin to roll over the

slope surface. As for the TWC model, the initial setting of the quaternion on the gravity center of carrier model S is $\tilde{q} = \tilde{q}_2 \tilde{q}_1$ where $\tilde{q}_1 = (0, 0, 1, \varphi_0)$, $\tilde{q}_2 = (0, 1, 0, \beta_0)$.

When the frame of a model is sufficiently high to prevent continuous rolling on the slope surface, the frame height becomes the necessary, or safe, height for model. It is called the critical TOPS height (CTH).

4.3.4 Overturning process

The dynamics of overturning behaviour are computed using Bullet for every 1/60 s. Figure 4.5 portrays a series of sideways overturning behaviour of the small TWC model when the TOPS is at CTH. The contacting processes are visible as the change of contacting point.

- First, front and rear wheels contact the slope surface (Fig. 4.5(a)).
- Secondly, a rear wheel and front edge of chassis contact to the slope surface (Fig. 4.5(b)).
- The side part of carrier model contacts the slope surface momentarily (Fig. 4.5(c)).
- TOPS and the rear side of cargo bed contact the slope surface (Fig. 4.5(d)).
- The vehicle reverts to the previous posture, where the side part of carrier model contacts the slope surface (Fig. 4.5(e)).

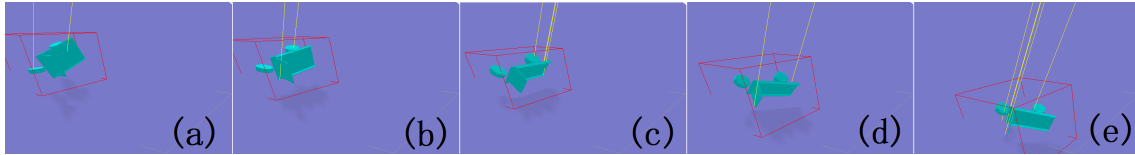


Figure 4.5: Non-continuous overturning behaviour for the small TWC model in Bullet.

It is noteworthy that the animation data presented in Fig. 4.5 are readily obtainable in Bullet.

Figure 4.6 presents an example of continuous overturning behaviour of the small FWC model when the TOPS height is 1 mm below its CTH. The contact process can be summarised as

- First, front and rear wheels contact the slope surface (Fig. 4.6(a)).
- The side part of carrier model contacts the slope surface (Fig. 4.6(b)).
- TOPS contacts the slope surface (Fig. 4.6(c)).
- The vehicle is rolling around the contacting point on TOPS, toward the bottom of the slope (Fig. 4.6(d)).
- The vehicle becomes inverted (Fig. 4.6(e)).

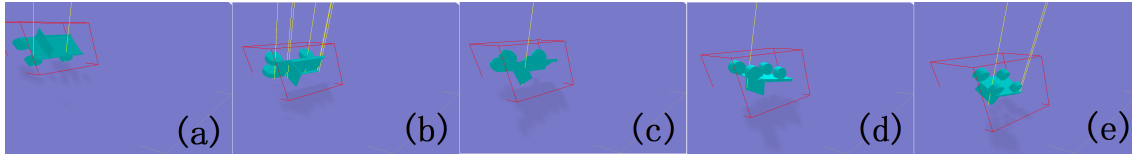


Figure 4.6: Continuous overturning behaviour for the small FWC model in Bullet.

4.4 Results and discussion

4.4.1 Small agricultural carrier model

Result of angular velocity for small TWC model

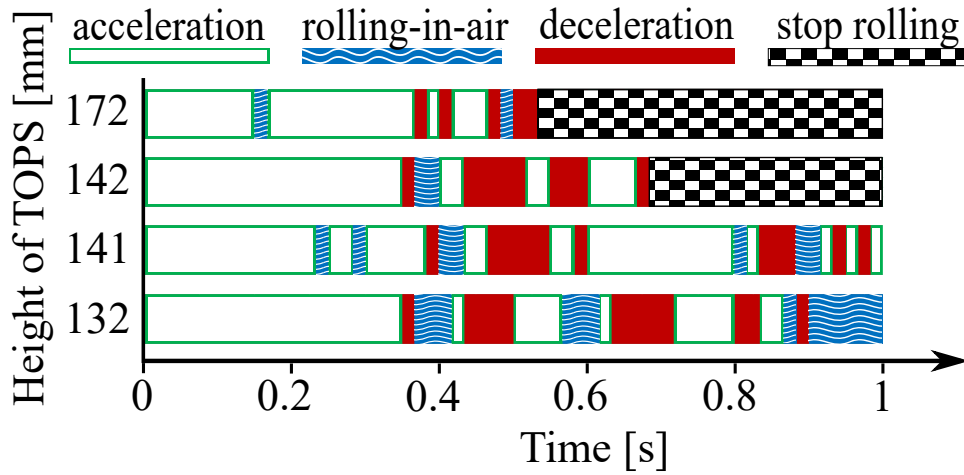


Figure 4.7: Schematic distribution of angular velocity for the small TWC model.

Figure 4.7 portrays a schematic distribution of angular velocity around the center of gravity S of carrier model when TOPS height H_6 is at a different setting. The model vehicle rollover result is obtainable in every $1/60$ s in the simulation. In the figure, the acceleration stage is shown in green, deceleration in red, and rolling-in-air in blue. The checkerboard pattern for 142 mm and 172 mm indicates that the vehicle would not overturn further, but it still had small impacts of up to 3 rad s^{-1} to the slope surface because of the impulse-based method of Bullet calculation, which is understood as negligible.

Case of $H_6 = 132$ mm: The front edge of the chassis contacted the slope surface at 0.35 s. Then the vehicle bounced back and rolled in the air. At 0.43 s, the upper side of the vehicle impacted the slope surface, which decreased the rolling speed. At 0.5 s, the vehicle accelerated again because of the impact of the lower side of the vehicle, or the wheel, until 0.57 s, when the vehicle was off the slope surface. Finally, TOPS contacted the slope surface at 0.63 s, but failed to stop the overturning tendency. Subsequently, the vehicle accelerated to the bottom of the slope from 0.7 s.

Case of $H_6 = 141$ mm: The vehicle was off the slope surface after the front edge of the chassis contacted the slope surface at 0.38 s. Then, the side of the vehicle contacted the slope surface

from 0.45-0.55 s. After brief acceleration, TOPS contacted the slope surface momentarily at 0.58 s. From 0.6 s, the vehicle accelerated to the bottom of the slope.

Case of $H_6 = 142$ mm: This behaviour resembles that of the case of $H_6 = 141$ mm summarised above. However, the time at which TOPS spent on stopping the overturning (0.55-0.6 s) was longer than in the case of $H_6 = 141$ mm. The vehicle accelerated toward the top of the slope from 0.6 s. The side part of the vehicle was back to the slope surface again at 0.68 s.

Case of $H_6 = 172$ mm: The vehicle bounced in the air from 0.37 s, impacting the lower side from 0.42 s, with powerful deceleration by TOPS from 0.47 s. It finally rolled back toward the slope surface.

Result of angular velocity for the small FWC model

Figure 4.8 portrays a schematic showing the distribution of angular velocity around the gravity center S for the small FWC model.

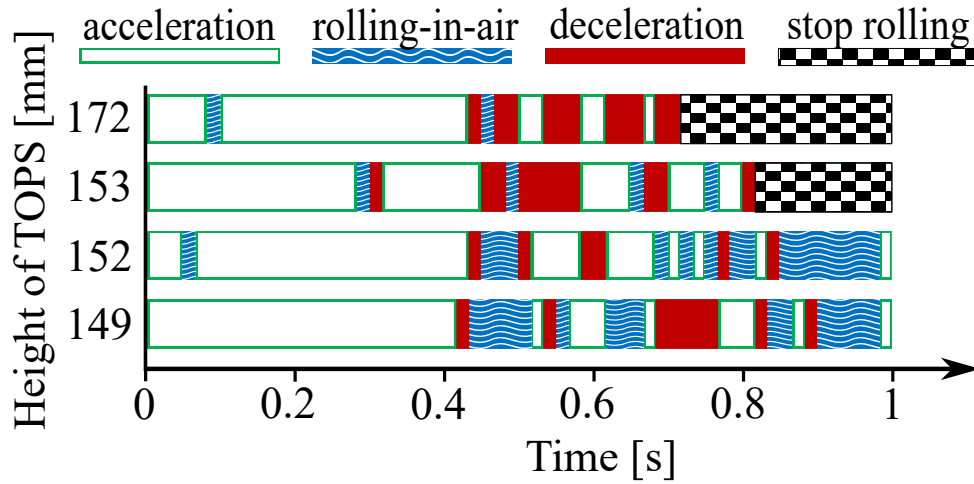


Figure 4.8: Schematic distribution of angular velocity for the small FWC model.

Case of $H_6 = 149$ mm: The side of the vehicle impacted the slope surface at 0.42 s. Then the vehicle bounced back and rolled in the air until 0.53 s. Then, TOPS contacted the slope surface and slowed its rolling behaviour. At 0.58 s, the vehicle rolled over its rotation center and accelerated to the slope bottom side. Finally, it was off the slope surface and rolling inverted at 0.62 s.

Case of $H_6 = 152$ mm: Bouncing and off the slope surface from 0.43 s. The side of the vehicle fell back to the slope surface and remained in contact from 0.5 s. Then TOPS impacted the slope surface at 0.58 s and tried to stop overturning, but failed. The vehicle continued rolling down the slope from 0.62 s.

Case of $H_6 = 153$ mm: The side of the vehicle contacted the slope surface at 0.45 s. The rolling speed decreased to 0.58 s. Then, vehicle accelerated again by gravity until TOPS contacted the slope surface at 0.67 s. The overturning tendency halted at 0.7 s. Then the vehicle was rolling back toward the top of the slope. The side part of the vehicle was back to the slope surface at 0.82 s. Its overturning was halted.

Case of $H_6 = 172$ mm: The vehicle went bouncing in air from 0.43 s, impacting the side from 0.47 s, acceleration by gravity from 0.5 s, deceleration by TOPS from 0.53 s and finally was rolling back to the slope surface and not overturning anymore from 0.73 s.

Discussion

Table 4.3 summarises the result of critical TOPS height. The result of FWC model in simulation is 9 mm lower than the experimental one, although similar critical height is obtained for the TWC model.

Table 4.3: Comparison of critical TOPS height between the simulation and experiment.

	Small TWC Model	Small FWC Model
Critical Height in simulation	0.142 m	0.153 m
Critical Height in experiment	0.142 m	0.162 m

In physics, the kinetic energy of an object is the energy attributable to its motion (Jain, 2009). In Bullet, the linear velocity at the gravity center of the vehicle parallel to the slope surface or each axis of coordinate system can also be recorded respectively, which enables the kinetic energy E_v to be summarised as the summation of linear and rotational energies. Assuming that the potential energy E_p is zero at the origin of world coordinate system, the potential energy of the model carrier can be computed by calculating the change of height on the global Z_{world} -axis. Then, the energy balance in each time step, $E_b (= E_p - E_v)$, can be depicted as presented in Fig. 4.9(a) for the small TWC model and in Fig. 4.9(b) for the small FWC model.

By analyzing the result of E_b in Fig. 4.9, the energy change of FWC at each impact (side contact, TOPS contact, back to side) can be recognised clearly in Fig. 4.9(b). Unlike the tetragonal section shape of FWC, the TWC has a pentagonal section shape to contact the slope surface before its impact of TOPS to the slope. Therefore, the energy change when TOPS impacted the slope surface at 0.54 s did not become distinct in Fig. 4.9(a). Consequently, the smooth rolling behaviour of TWC might explain the result presented in Table 4.3 showing that the simulated critical TOPS height of TWC became equal to the experimentally obtained result.

4.4.2 Full-scale agricultural carrier model

Result of angular velocity for full-scale TWC model

Figure 4.10 presents a schematic distribution of angular velocity with the TOPS height increasing from 138 cm to 147 cm for SL30 (TWC).

For all cases of TOPS height, those vehicles went through the same process before TOPS contacts the slope surface. First, rolling around the axis defined by front and rear wheels was observed until the front edge of the chassis contacted and slowed. Secondly, rolling around the axis defined by the rear wheel and the front edge of chassis was followed until the side part of the vehicle made contact and slowed. Finally, the vehicle underwent rolling around the axis defined by the side part of itself until the highest end of TOPS was contacted.

Case of $H_6 = 138$ & 141 cm: The vehicle accelerated down the slope and finally became inverted.

Case of $H_6 = 142$ & 147 cm: The vehicle accelerated up the slope. Then, the side of the vehicle contacted the slope surface again and did not overturn any further in the checkerboard area in Fig. 4.10.

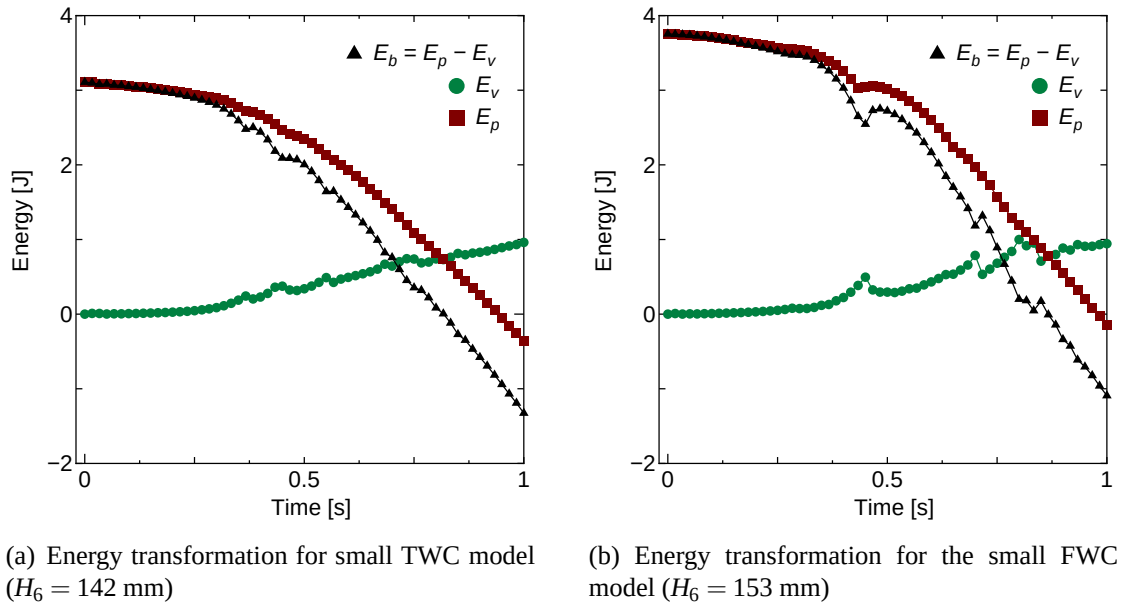


Figure 4.9: Examples when TOPS is set at critical height.

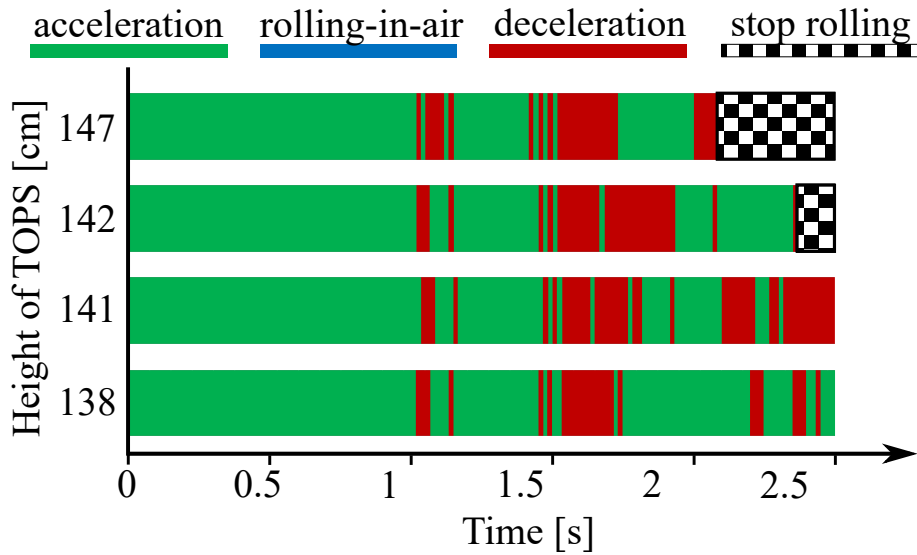


Figure 4.10: Schematic distribution of angular velocity for the full-scale TWC model.

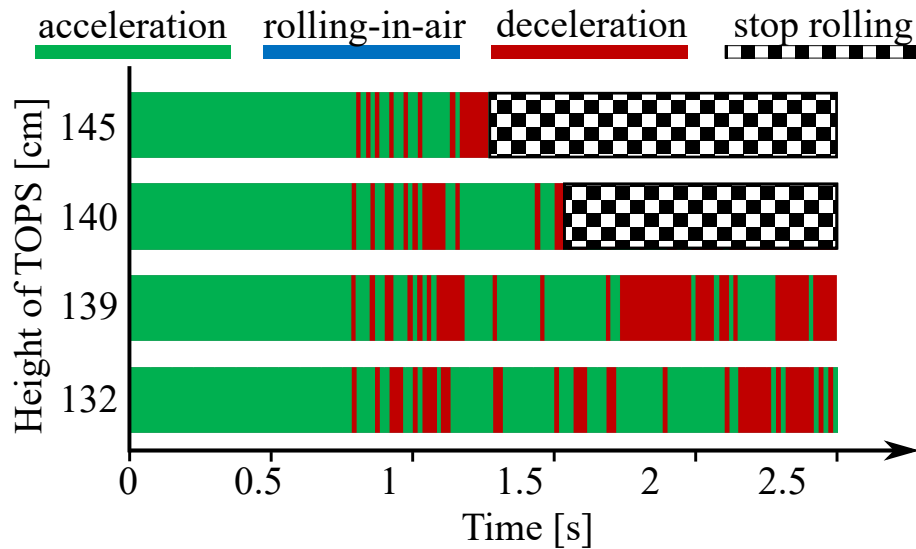


Figure 4.11: Schematic distribution of angular velocity for the full-scale FWC model.

Result of angular velocity for the full-scale FWC model

Figure 4.11 portrays a schematic distribution of angular velocity for the four-wheeled carrier ES60 with various heights of TOPS. The slowing of ES60 (FWC) is not continuous because the contact points continued to slip along the slope surface.

Case of $H_6 = 132$ cm: The side part of the vehicle contacted the slope surface at 0.88 s. The vehicle slowed momentarily by TOPS contacting the slope surface (0.97–0.98 s). Then it accelerated until it became inverted.

Case of $H_6 = 139$ cm: The deceleration time (0.8–0.88 s) by the impact of the side part of the vehicle was longer than in the case of $H_6 = 132$ cm, as was the deceleration time (1–1.2 s) by TOPS. However, the vehicle continued to accelerate down the slope.

Case of $H_6 = 140$ cm: After TOPS impacted the slope surface at 0.93 s, continuous overturning was prevented. The vehicle rolled back up the slope from 1.12 s.

Case of $H_6 = 145$ cm: The side part of the vehicle impacted the slope surface first at 0.82 s. Then, the vehicle repeated the following process during 0.82–1.05 s: TOPS impacted the slope surface; the vehicle rolled back; the side part impacted the slope surface; then TOPS impacted the slope surface. The vehicle rolled back up the slope from 1.05 s.

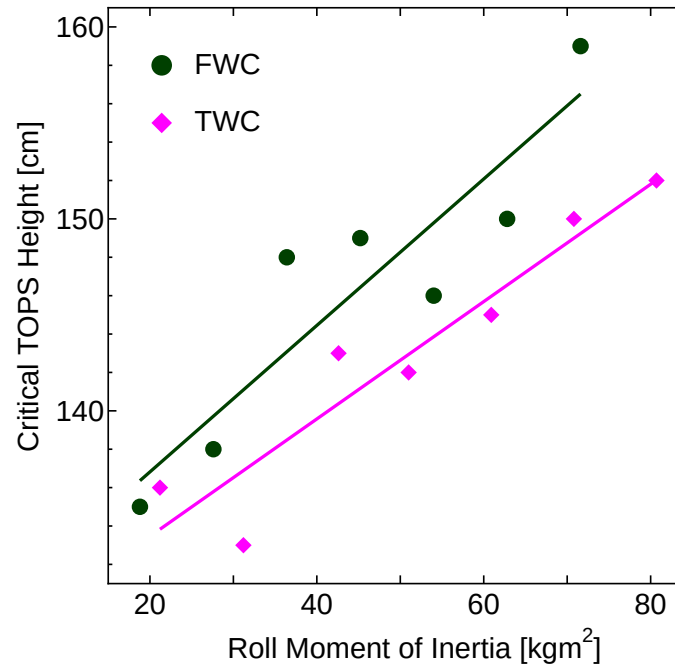
Discussion

As presented in Table 4.4, the critical height of TOPS for TWC is 142 cm using the Bullet physics engine, which is lower than the result of the previous study (153 cm) using OECD algorithm (Sun et al., 2017). The critical TOPS height of FWC obtained using the Bullet engine (140 cm) is similar to the result obtained using BRAIN (142 cm), which also uses the OECD code (Tsukamoto et al., 2008). It might be readily apparent that the critical TOPS height of TWC with larger moment of inertia becomes higher again than that of FWC with a lower moment of inertia.

Table 4.4: Comparison of the critical height of TOPS between Bullet simulation and earlier research.

	Full-scale TWC Model	Full-scale FWC Model
Critical Height using Bullet	1.42 m	1.40 m
Critical Height by OECD code*	1.53 m	1.42 m

*Sun et al. (2017)

**Figure 4.12:** Relation between critical TOPS height and roll moment of inertia.

4.5 Parametric analysis of Bullet simulation for the full-scale carrier model

4.5.1 Effect of roll moment of inertia

To investigate the effects of the roll moment of inertia on the height of TOPS, the mass of components was adjusted in Bullet to compute a lower or higher value of the roll moment of inertia without changing the center of gravity. The roll moments of inertia are 18.8, 27.6, 36.4, 45.2, 54, 62.8 and 71.6 kgm² for FWC, 21.2, 31.2, 41.1, 51, 60.9, 70.8 and 80.7 kgm² for TWC when the height of TOPS is 140 cm.

Figure 4.12 presents the effect of roll moment of inertia on the critical TOPS height (CTH), whereas the coefficient of friction is 0.8 and coefficient of restitution is 0.2.

Similar increasing behaviour of CTH to the increase in the roll moment of inertia is apparent in both the TWC and FWC cases. Those increasing trends also coincide with the results obtained using the OECD program for agricultural carrier models (Sun et al., 2017), for turf keepers (Comer et al., 2003), and for tractors (Ayers et al., 2016). As the figure shows, if the TWC model and the FWC model were the same roll moment of inertia, then the CTH of TWC might be lower than FWC because the TWC model has a pentagonal section shape to experience more potential impacts.

4.5.2 Effect of coefficient of restitution

The effect of coefficient of restitution on the CTH is presented in Fig. 4.13. The increase in coefficient of restitution results in increasing CTH, which might be reasonable in terms of the definition of restitution.

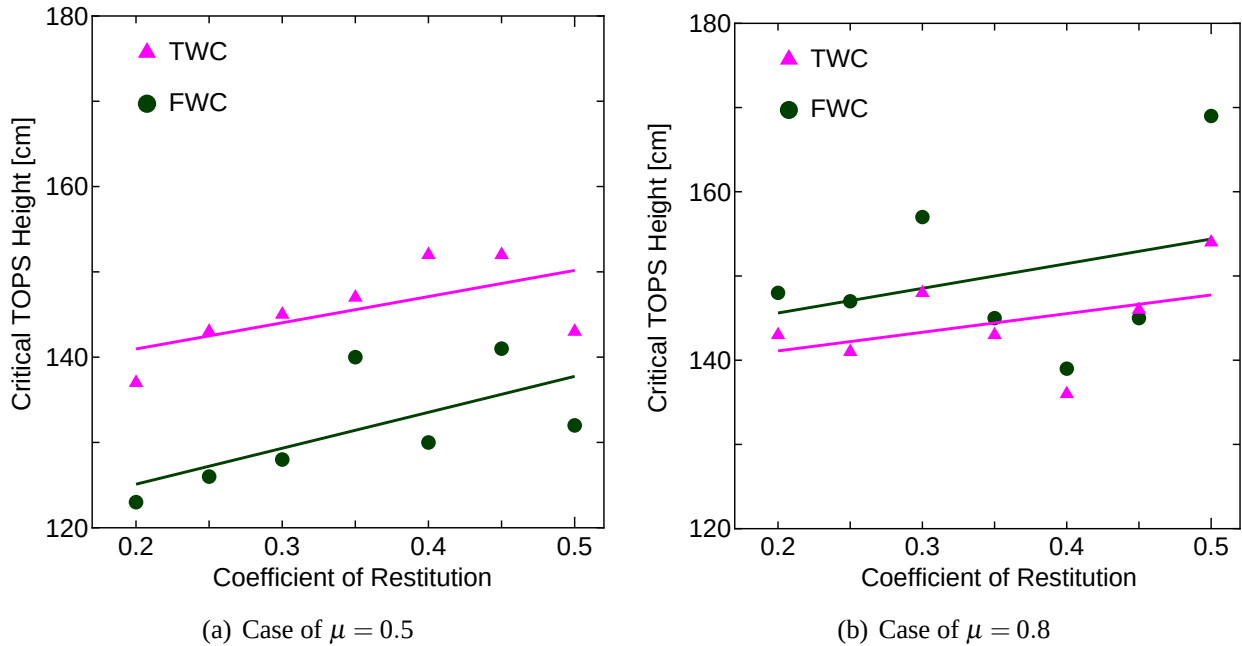


Figure 4.13: Relation between critical TOPS height and coefficient restitution.

Regarding earlier research using the OECD program, such as simulation of the turf keeper (Comer et al., 2003) and agricultural carrier (Sun et al., 2017), an opposite behaviour by which the larger coefficient of restitution resulted in a lower critical height of TOPS was reported. The simulation using Bullet engine can apply and reflect real-world physics under Newton's law.

It is noteworthy that the CTH for TWC does not depend clearly on the coefficient of restitution in the figure. However, for FWC, the CTH might be characterised as dependent on both the coefficients of restitution and the coefficient of friction.

4.5.3 Effects of friction coefficient

The effect of friction coefficient on CTH cannot be summarised clearly because the initial posture of the carrier changes as the friction condition is changed. A small friction coefficient generates a slippery surface of the slope. In other words, the carrier requires a larger uplift angle to start rolling with a sufficient normal load, or the carrier will begin to slip along the slope surface without rolling. Figure 4.14 presents the difference in initial posture of carrier at the time of starting to roll for three cases of friction coefficient: 0.3 (Fig. 4.14(a)), 0.5 (Fig. 4.14(b)), and 0.8 (Fig. 4.14(c)).

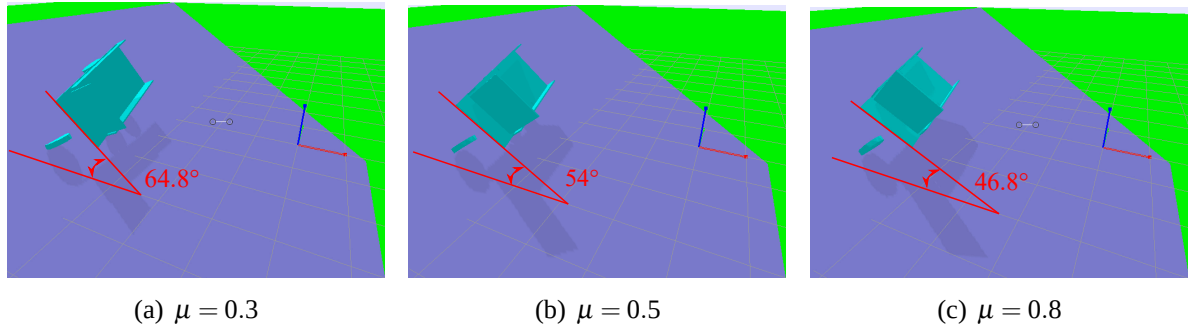


Figure 4.14: Difference of initial roll angle settings of the TWC.

By summarizing the angular energy and linear energy of carrier, the kinetic energy for both the FWC and TWC with various cases of friction coefficient (0.4, 0.6, 0.8) is portrayed as Fig. 4.15, with the critical height of TOPS and the coefficient of restitution being 0.35.

As the friction coefficient becomes higher, the first impact of the carrier for both the FWC and TWC appears to be delayed. The decrease in kinetic energy caused by the FWC impact is greater than that of TWC because the pentagonal section shape of TWC provides a more continuous and constant contacting process. For $\mu = 0.8$ for FWC, the carrier had no slipping, but some rotation around its rotation axis during 0–2 s. The motion of this vehicle stopped after TOPS prevented the carrier from continuously rolling-over.

Assuming that the highest coordinate of initial posture is the origin of the potential energy of vehicle, results clarify that the potential energy is decreasing from zero as the vehicle starts rolling on the slope. As shown in Fig. 4.15, the higher friction can weaken the decreasing tendency of potential energy by dissipating the friction loss of energy.

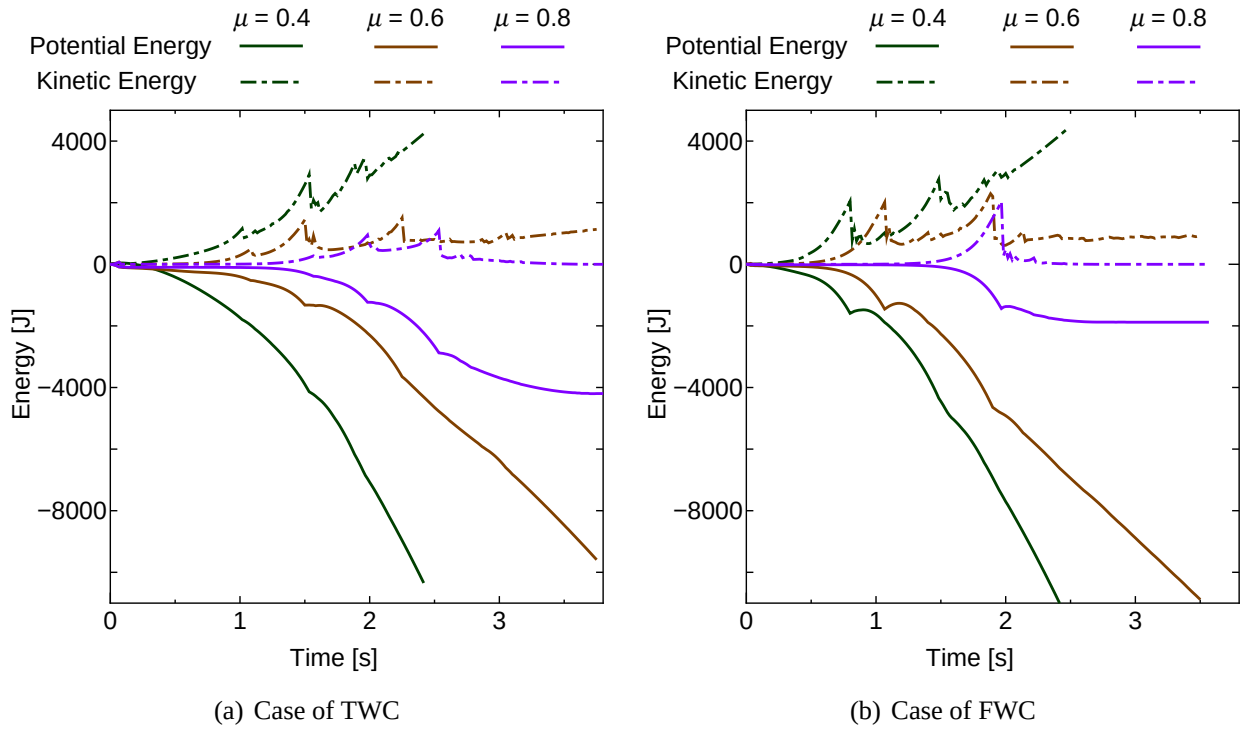


Figure 4.15: Results of kinetic energy and potential energy.

4.6 Conclusion

Numerical simulation using a physics engine can be applied with sufficient accuracy to simulate dynamic behaviour of sideways overturning on a slope for both small models and full-scale models of three-wheeled and four-wheeled agricultural carriers. The current simulation using the physics engine is applicable to the design of optimum height of a safety frame for wheeled agricultural carriers on a slope surface.

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Chapter 5

Application of Physics Engine for Overturning Tractor

5.1 Introduction

Compared to those results obtained using the same algorithm reported in OECD (2017) (Sun et al., 2017), the predictive accuracy of the critical height of a tip-over protective structure (TOPS) for agricultural carrier models on a slope using Bullet could be confirmed in Chapter 4 (Sun et al., 2018).

Collision detection by Bullet is expected to obtain those contacting points correctly during the overturning of a tractor, which might coincide with the results of calculation using the OECD program with contacting coordinates. With the introduction of friction, the Bullet simulation results will be more visually acceptable than those of earlier studies using the OECD program.

The sideways overturning behaviour of tractors using the Bullet physics engine for two cases of damage by ROPS (Chisholm, 1973): a case in which a tractor fell over the edge of a bank slope of 2 – 4 m above the level surface, and a case of multiple rolling of tractor over a uniform slope, is simulated in Chapter 5. Applicability of Bullet to the overturning tractor over a bank slope is evaluated by Root Mean Square Error (RMSE). Moreover, because OECD (2017) reported that ROPS should be able to prevent a tractor from continuous rolling or multiple rolls on a 33.7 deg uniform slope surface, the critical height of ROPS for the example of a tractor in OECD (2017) was investigated using Bullet.

5.2 Outline of Bullet

Bullet can manage basic rigid components such as a sphere, box, or cylinder and compute the collision dynamics of the components. By adding a constraint to connect those rigid components, an object model of any shape can be assembled in which each component obeys the individual physical properties such as mass and moment of inertia in collision dynamics.

5.2.1 Collision model for tractor

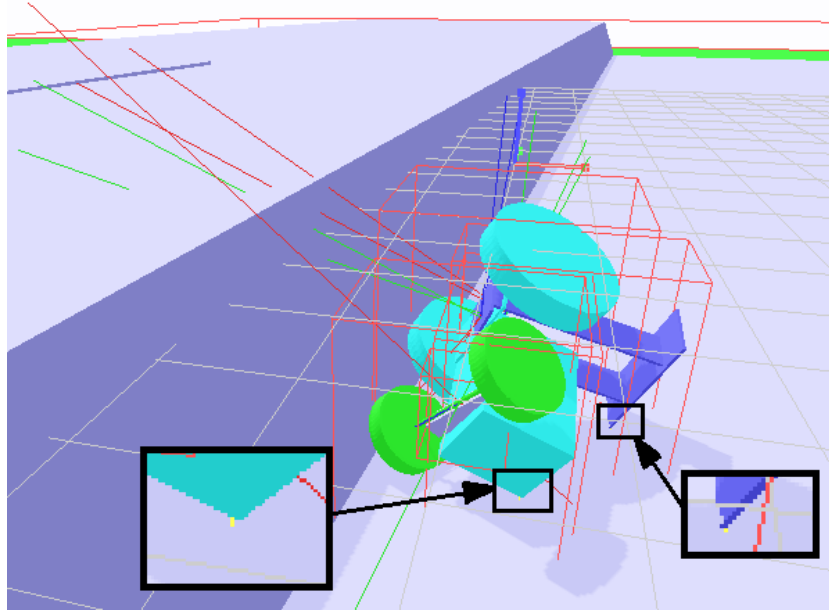


Figure 5.1: Tractor model and bank slope model.

Bullet provides the convex primitives of collision detection component to customise the collision model shape. Figure 5.1 presents an example of the tractor model with a pivoted front axle and a bank slope model. For the tractor model, cylinder components are used to construct the wheels. Box components are applied to express the engine bonnet, pivoted front axle and ROPS.

All those collision components are detected by broad phases as shown by red boxes and narrow phases as normal vectors originating at a contact point with the yellow line, as shown in the two close-up views in Fig. 5.1. Similarly to other physics engines, broad-phase detects candidate impacting components; narrow-phase calculates a particular contact point for the target colliding pair of components in Bullet (Erwin, 2013).

5.2.2 Constraint condition

Three basic constraints exist: point to point, hinge, and slider, as shown in Fig. 5.2, implemented in Bullet (Erwin, 2013). In our vehicle model, the hinge constraint, which restricts spatial rotation into one rotational degree of freedom, is useful to simulate the rotating behaviour of wheels around the front and rear axles, in addition to the behaviour of centre pivoting at the front axle, and resistance behaviour of ROPS against lateral impact on to ROPS.

By applying torque on the hinge constraint for wheel axles, two linked rigid-bodies can become rotated around the hinge (or axle), which realises the generation of forward thrust by driven wheels (Weinstein et al., 2006).

If necessary, aligning the torque and angular limit at a pivot angle on the hinge constraint can be considered for simulating the front axle of tractor with a suspension system. The impact energy to the ROPS when the tractor is overturning over a surface can be recorded as the reactive torque or moment. Even if the geometry and dynamics of each component are computed individually, the behaviour of the tractor model is influenced by those components through the constraint. For instance, a front axle

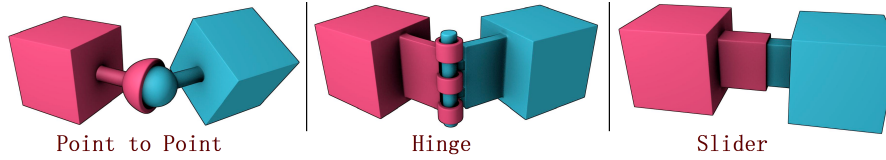


Figure 5.2: Constraint types in Bullet (Erwin, 2013).

can pivot around the tractor freely but still provide linear support and angular vibration (Matsui et al., 2017).

Import hinge constraint

The tractor model in our study is a relatively simple combination by hinge constraint which allows one rigid body rotate around another rigid body by only one degree of freedom Fig. 5.2. Moreover, the limit and control between two rigid bodies can be set as follows.

btHingeConstraint* pHinge1 = new btHingeConstraint(*F_pBodyA,*F_pBodyB,F_pivotInA, F_pivotInB,F_axisInA,F_axisInB,false); set a hinge constraint called “**pHinge1**” between the main tractor body (***F_pBodyA**) and the rear axle (***F_pBodyB**). **F_pivotInA** means the limit of this hinge on the main tractor body (***F_pBodyA**), which can only be set as (1,0,0) at pitch direction, (0,1,0) at roll direction or (0,0,1) at yaw direction, the limit **F_pivotInB** obeys the same rule for ***F_pBodyB**. **F_axisInA** decides the connecting point on ***F_pBodyA**, so do **F_axisInA** for ***F_pBodyB**. The last command **false** means that this two rigid model ***F_pBodyA** and ***F_pBodyB** have relative mechanical effect through the hinge, which means there is no collision detection calculated between this two body.

Use **pHinge1->enableAngularMotor(true,-2,30);** to import a relative torque between ***F_pBodyA** and ***F_pBodyB**. The command **true** means the power is activated. The following command **-2** means the maximum relative angular velocity is limited at 2 rad s^{-1} . The last command **30** means the maximum torque acted on **pHinge1** is limited at 30 N m.

Use **pHinge1->setLimit(-1, 2);** to limit the rotating angle between ***F_pBodyA** and ***F_pBodyB** from -1 to 2 which can also be adjusted if necessary. In our tractor model, this function had been used to set the pivot angle of the front axle.

5.3 Numerical experiment

5.3.1 Tractor and bank slope models

Sideways overturning experiments of the tractor were reported by Chisholm (1979), where a bank slope was used.

In the experiments, the tractor was driven by a remote control system to a predetermined point where the vehicle became unbalanced (Chisholm, 1979). Figure 5.3(a) presents a coordinate system of the tractor with a bank slope model, where the X-axis is parallel to the longitudinal axis of tractor in the forward direction, and a pivot of front axle about the X-axis is in the roll direction. The lateral direction of tractor is defined as the Y-axis in Fig. 5.3(a). The vertical direction of tractor is the Z-axis in Fig. 5.3(a) based on the right hand system.

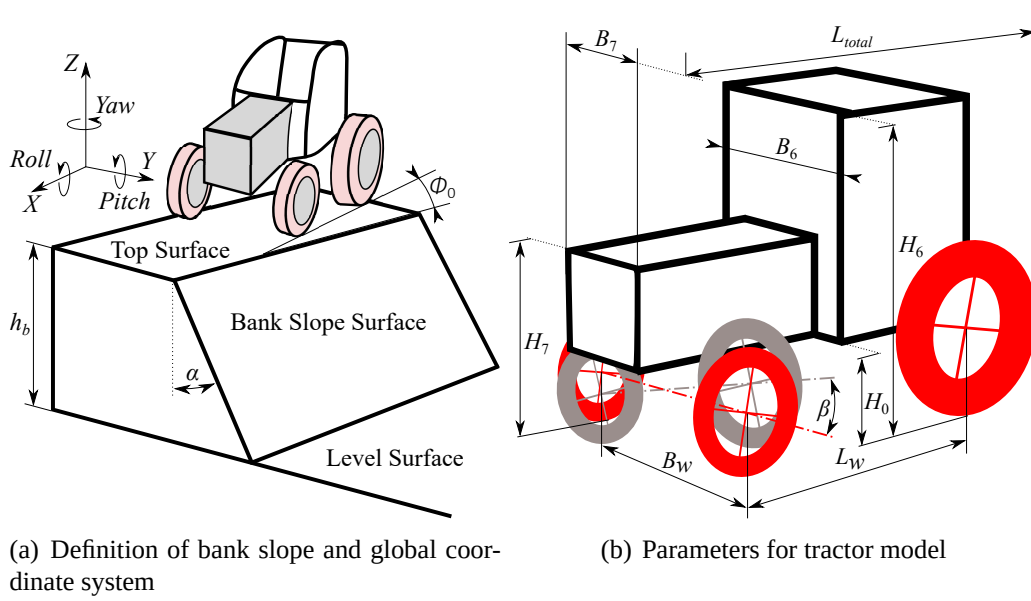


Figure 5.3: Definitions of parameters.

Parameters for the tractor model used in the Bullet simulation are presented in Fig. 5.3(b) and are presented in Table 5.2. Most values were reported by Chisholm (1979), but undefined values such as the coefficient of restitution or the tractor bonnet size was determined according to examples presented in subsequent report in the relevant literature (OECD, 2017). It is assumed that the maximum distances between wheels at the front and rear axles (B_w) are the same, which defines the tractor width. The pivot angle of front axle β ($\neq 0$) means that the front axle can be rotated at its centre pin with β . When $\beta = 0$, the front axle is fixed rigidly to the body of tractor.

As for the bank slope condition, the angle of slope α was adjustable from 0 to 37.5 deg. The coefficient of friction μ for the slope surface can vary from 0.95 (dry condition) to about 0.15 (detergent flooded condition).

Table 5.1: Initial condition in Chisholm (1979)

Examples	α (deg)	B_6 (m)	μ	β (deg)
#1	22.5	1.37	0.95	± 26.79
#12	22.5	1.37	0.95	0.0
#4	15	1.37	0.95	± 26.79
#6	37.5	1.37	0.95	± 26.79
#7	30	1.37	0.95	0.0
#16	0.0	1.37	0.95	0.0
#19	22.5	1.67	0.95	0.0
#20	7.5	1.67	0.95	0.0
#22*	22.5	1.67	0.15†	0.0

* With graphs of displacement published

† Level surface is 0.95

The graphs of linear and angular velocities of nine groups of observed data and the graphs of linear and angular displacements of one group of observed data were reported by Chisholm (1979).

The difference in initial settings for respective groups is presented in Table 5.1. Therefore, the accuracy of sideways overturning behaviour with velocity data using Bullet can be evaluated using the experimentally obtained results.

Table 5.2: Parameters and conditions related to bank slope simulation

Bank slope angle to vertical, α	0, 7.5, 15, 22.5, 30, 37.5 deg
Bank friction, μ	0.95, 0.15
Bank restitution‡, e	0.2
Bank height, h_b	2.5 m
Total mass of tractor, M_{total}	3065 kg
Mass on front tyre, M_{front}	1105 kg
Mass on rear tyre, M_{rear}	1960 kg
Roll moment of inertia, Q_{roll}	831 kgm ²
Lowest point of bonnet‡, H_0	0.445 m
Centre of gravity height, H_1	0.894 m
Front tyre diameter, H_2	0.857 m
Rear tyre diameter, H_3	1.44 m
ROPS height, H_6	2.26 m
Bonnet height‡, H_7	1.35 m
Front tyre width, B_2	0.2 m
Rear tyre width, B_3	0.29 m
Bonnet width‡, B_7	0.8 m
ROPS width, B_6	1.37, 1.67 m
Tractor width, B_w	1.54 m
Wheelbase, L_w	1.621 m
Tractor length, L_{total}	2.597 m
Pivot angle of front axle, β	0, ± 26.79 deg
Forward speed, v_0	1.5 m s ⁻¹
Approach angle to bank edge, ϕ_0	6 deg

‡ No definition in Chisholm (1979)

5.3.2 Tractor and uniform slope models

American Society of Agricultural Engineers (ASAE) Standard S478 and Society of Automotive Engineers (SAE) Standard J2194 do not include the continuous overturning test for wheeled tractors. Because ASAE S547 Standard (withdrawn now) related to continuous lateral roll of small tractors followed the same description as that in OECD Code 6 (Wang et al., 2007), OECD Code 6 appeared to be the typical demonstration of continuous overturning using Bullet simulation.

The overturning simulation for a tractor on a uniform slope of 33.7 deg as recommended by OECD (2017) is also planned for investigation whether or not a ROPS fitted to the tractor can satisfactorily prevent continuous rolling. The tractor is tilted laterally from the initial orientation where the tractor's longitudinal axis is parallel to the contour line of slope, with its left front and rear wheels kept contacting the valley-side slope surface. The tractor would begin overturning as soon as the centre of gravity passed through the vertical plane above the axis of rotation defined by two contact points at the dead bottom centre of the left front and rear wheels. In the OECD code 6, all coordi-

nates of possible contact points must be prepared a priori. The angular velocity of the tractor around the rotation axis defined by those contacting points was calculated (OECD, 2017). According to 87/402/EEC (EEC, 1987) and the OECD Code 6 (OECD, 2017), if the tractor did not rotate further or rotated momentarily but fell back toward the surface of the slope after the impact of ROPS against the slope surface, the height of ROPS can be regarded as sufficiently large for protecting the tractor from continuous overturning.

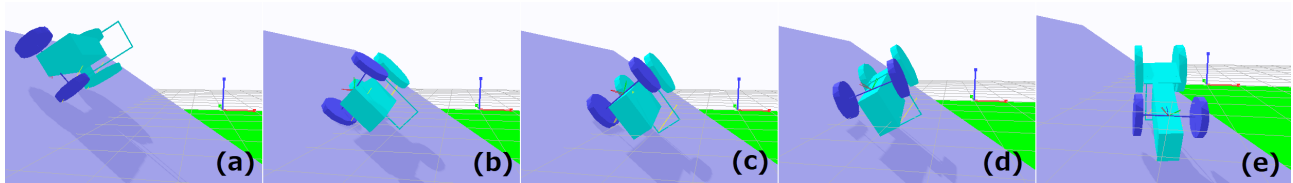


Figure 5.4: Example of overturning process over a uniform slope using Bullet.

Figure 5.4 portrays the overturning process by collision detection in Bullet. The yellow line in the figure is narrow-phase generated from the collision detection in Bullet. The overturning process can be summarised as follows:

- First, the left front and rear wheels of tractor contact the slope surface (Fig. 5.4(a));
- Secondly, the upper edge point of the rear wheel and the side of the front wheel contact the slope surface (Fig. 5.4(b));
- Upper edge points of the front and rear wheels along with ROPS contact the slope surface (Fig. 5.4(c));
- The upper edge point of the front wheel and ROPS along with the engine bonnet contact the slope surface (Fig. 5.4(d));
- Both ROPS and bonnet contact the slope surface. The tractor rotates toward the top or bottom side of the slope (Fig. 5.4(e)).

The transition of contacting points for an overturning tractor is similar to that reported in OECD (2017). It is noticeable that those contacting points in the Bullet model were generally sliding along the ground surface, which means the dynamic behaviour was generated by a combination of linear velocity and angular velocity. For example, in Fig. 5.4(e), if the contacting point of ROPS were sliding faster than the lateral displacement of the centre of gravity of the tractor, the angular velocity would be updated toward the top side of the slope.

The parameters of tractors examined in this study are the same as those reported in OECD (2017). The ROPS design follows the latest standards summarised in Ayers et al. (2018) where the height is adjustable to find the critical ROPS height in the simulation. The definition of parameters for the tractor is the same as in Fig. 5.3(b). The main parameters of each example are presented in Table 5.3. Other parameters for overturning simulation on the uniform slope are listed in Table 5.4.

Table 5.3: Significant parameters for tractors (OECD, 2017)

Cases of tractor	B_w (m)	B_6 (m)	L_w (m)	M_{total} (kg)	Q_{roll} (kgm ²)
Example 6.1	1.310	0.78	2.046	2565	295
Example 6.2	1.310	0.70	1.946	1800	250
Example 6.3	1.215	0.64	1.959	1780	280
Example 6.4	1.215	0.64	1.970	1780	280
Example 6.5	2.025	0.70	1.946	1800	250
Example 6.6	1.310	0.70	1.946	1800	275
Example 6.7	1.310	0.78	1.946	1800	200
Example 6.8	1.215	0.64	1.970	1780	280
Example 6.9	1.310	0.77	1.946	1800	300
Example 6.10	1.310	0.70	1.946	1800	275
Example 6.11	1.310	0.70	1.946	1800	250

Table 5.4: Parameters and conditions related to uniform slope simulation

Uniform slope angle to level, α	33.7 deg
Uniform slope friction, μ	0.6
Uniform slope restitution, e	0.2
Weight distribution on front axle	40.8%, 41.0%, 43.8%, 45.1%
Clearance, H_0	0.440, 0.445 m
Centre of gravity height, H_1	0.718, 0.762, 0.7653, 0.766 m
Front tyre diameter, H_2	0.702, 0.880, 0.90 m
Rear tyre diameter, H_3	1.217, 1.293, 1.380, 1.480, 1.520, 1.550 m
Bonnet height, H_7	1.212, 1.337, 1.350, 1.370, 1.550, 1.639 m
Rear tyre width, B_3	0.195, 0.315, 0.91 m
Bonnet width, B_7	0.36, 0.49, 0.8, 0.875, 0.89, 0.95 m
Pivot angle of front axle, β	$\pm 18, \pm 20$ deg

5.4 Results and discussion

5.4.1 Tractor and bank slope models

In the Bullet simulation, the velocities of the centre of gravity of the tractors were updated by their own calculation pipeline (Fig. 3.1 (Page 28)) at intervals of 1/60 s. In the experiment (Chisholm, 1979), several data points were designed to be recorded using a camera. Therefore, the linear and angular velocities could be measured by calculating the displacement of each data point. By separating the experimental curve into several sampled data during overturning ($n = 60 \times \text{time}$, where n is the number of each data and time is the travelling time from $n = 0$), the experimental data were rearranged along the time scale of the simulation data. To ensure the data were comparable, the total overturning time was shortened to 2.25 s (from the beginning of acceleration of velocity to the end) in both the experimental and the simulation data. As for Example #4, the beginning of acceleration of velocity in the experimental results shown in Fig. 5.5(a) is at 1.75 s (t_0^{exp}). Therefore, the overturning period of the experiment is 1.75 – 4 s separated into 135 ($= 60 \times 2.25$ s) uniformly spaced data.

Table 5.5: Comparison of RMSE between the simulation result and the experimentally obtained result (Chisholm, 1979).

Examples	RMSE of ω^r (rad s ⁻¹)	RMSE of v^l (m s ⁻¹)	RMSE of v^v (m s ⁻¹)
#1	0.785	0.618	0.899
#4	0.658	0.537	0.709
#6	0.651	0.682	0.472
#7	0.586	0.812	0.739
#12	0.624	0.582	0.648
#16	0.837	0.679	0.953
#19	0.622	0.553	0.719
#20	0.633	0.621	0.698
#22	0.444	0.678	0.885
Average	0.649	0.640	0.747

Moreover, the marked beginning of acceleration of velocity in the simulation result shown in Fig. 5.5(b) is at 0.5 s (t_0^{sim}). The overturning period in the simulation will thus be 0.5 – 2.75 s. Therefore, the lateral velocity v_n^l , vertical velocity v_n^v , and roll angular velocity ω_n^r of the simulation data at $n/60 + t_0^{sim}$ s became comparable to that of the experimental data at $n/60 + t_0^{exp}$ s ($n = 1, 2, \dots, 135$).

The Root Mean Square Error (RMSE) is utilised to investigate the differences between the simulation result and the experimental result. Compared to other statistical methods, RMSE is more sensitive to extreme values, which makes it capable of measuring the differences in the sequence of velocity between experiment and simulation. The RMSE of the uniformly spaced simulation and experimental data is calculated as

$$RMSE = \sqrt{\frac{\sum_{n=1}^{135} (v_n^{exp} - v_n^{sim})^2}{135}}, \quad (5.1)$$

where v_n^{exp} is the experimental data and v_n^{sim} is the simulation data. The RMSE of the lateral velocity v^l , vertical velocity v^v , and roll angular velocity ω^r indicate the accuracy of each simulation example during tractor overturning modelled using Bullet. The result of RMSE for overturning behaviours in the 2.25 s from the beginning of acceleration of velocity to the end of overturning is presented in Table 5.5.

In Table 5.5, the RMSE of lateral velocity is similar to that of roll angular velocity. Considering the range of lateral velocity (-1 to 2 m s⁻¹), which is smaller than the range of roll angular velocity (-3 to 1 rad s⁻¹), the performance along the lateral direction is inferior to that along the roll direction in Bullet simulation. Although the RMSE of vertical velocity is relatively higher than those of the velocities, Bullet simulation along the vertical direction might reach the highest fitness to the experimental result because the range of vertical velocity is -4 to 1 m s⁻¹. The overturning dynamics of Example #4, in which the front axle can pivot at its centre pin, exhibits medium RMSE values. The differences between the experimental and the simulation results in Example #16 lead to larger RMSE values of ω^r and v^v .

Figure 5.5 is the result of linear velocity and angular velocity for Example #4 of bank slope ($\alpha = 15$ deg). Figure 5.5(a) portrays the result of experiment (Chisholm, 1979). Figure 5.5(b) exhibits the result obtained using Bullet. The behaviours of curves in linear and angular velocities in Fig. 5.5(a) and Fig. 5.5(b) are similar. However, the overturning process of the Bullet simulation begins earlier and ends 1.5 s earlier than that of the experiment.

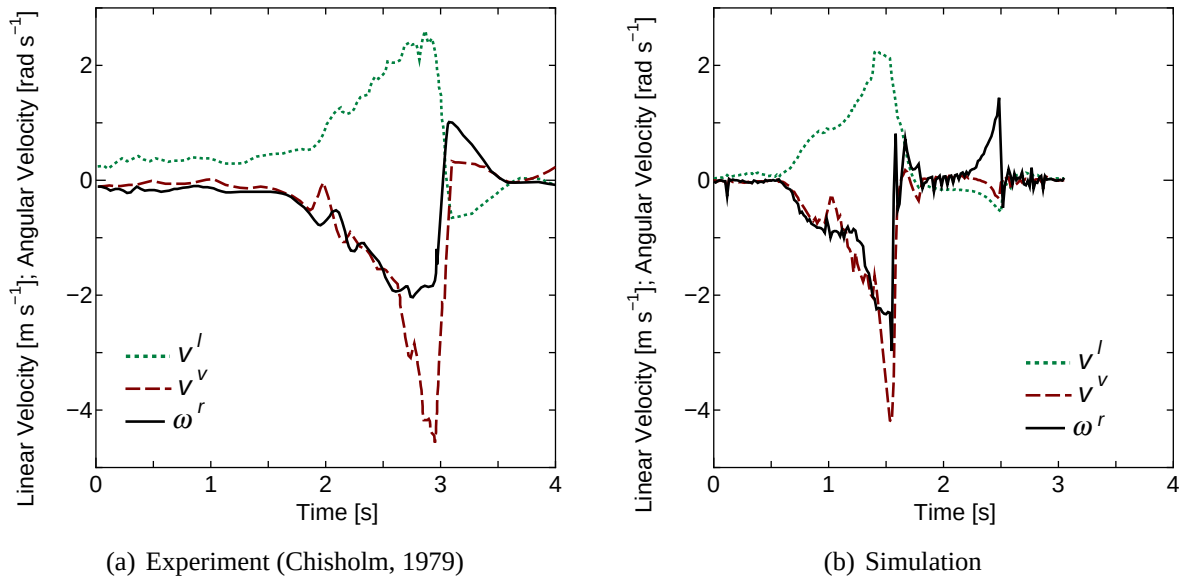


Figure 5.5: Result of linear velocity and angular velocity for Example #4.

Figure 5.6 is the sequence of overturning process for the tractor of Example #4 falling from a bank slope. As described in an earlier report (Chisholm, 1979), the “initial behaviour” known as the nominal start of overturning process is defined as the front left tyre travelling through the edge of the bank slope as shown in Fig. 5.6(a), which is also the origin of the time scale in Fig. 5.5.

The coordinate system of the experiment in Chisholm (1979), in which the lateral direction pointed to the horizontally normal direction of the edge of top surface, deviated by 0.6 deg from our coordinate system in Fig. 5.3(a). That might be the reason why the initial lateral velocity in that experiment was 0.1 m s^{-1} , at 0 s in Fig. 5.6(a), whereas that in our initial lateral velocity was 0 m s^{-1} , at 0 s in Fig. 5.6(b). The deviation influenced negatively the RMSE result, albeit the influence does not seem to be critical.

After the left front wheel had fallen completely, the tractor body was slightly shaken by the momentary balance loss at 0.2 s in Fig. 5.5(b) as shown in Fig. 5.6(b). However, the angular velocity returned to almost zero again from 0.2 to 0.5 s because the other rigid wheels still contacted the top surface and provided strong support for the tractor.

When the rear left tyre fell off the edge of the bank slope, the tractor lost its support from the valley side and accelerated to eventually fall from the top surface at 0.5 s in Bullet (Fig. 5.6(c)). In the experimental result (Fig. 5.5(a)), the obvious change in velocity started at 1.5 s, which is later than that in the simulation result, and this delay might be ascribed to the fact that the load on the left tyres increased as soon as the front left tyre and rear left tyre travelled toward the valley side and elastic deformation of left tyres could generate a relatively larger contact area with bank surface supporting tractor travel further on the bank surface than in the Bullet rigid model.

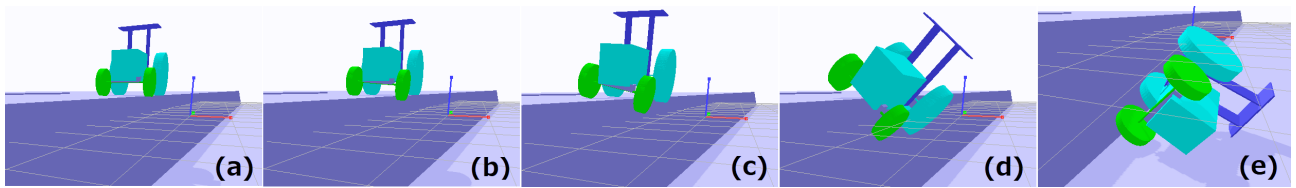


Figure 5.6: Overturning process at (a) 0, (b) 0.2, (c) 0.5, (d) 1 and (e) 1.5 s for Example #4 using Bullet.

All velocities started to increase markedly after 0.5 s, when the tractor began falling from the top surface toward the valley side. Then, the lower parts of the tractor (front axle or transmission) contacted the edge of the bank slope, as shown in Fig. 5.6(d), which interrupted the continuous fall of the tractor momentarily and influenced the trend of dynamic simulation data at 1 s, as shown in Fig. 5.5(b). The peak at 2 s in Fig. 5.5(a) might have been caused by contact of the lower parts of the tractor after half a second of fall from the top surface. Finally, the tendency to fall was stopped by the contact of ROPS with the level surface, as shown in Fig. 5.6(e), at 1.5 s in Fig. 5.5(b).

Figure 5.7 presents results of linear velocity and angular velocity for Example #16 of bank slope ($\alpha = 0$ deg). The visible acceleration of velocity for the experimentally obtained result was 1.5 s later than the origin of time in Fig. 5.7(a). It is noteworthy that the time period from the origin to the marked acceleration of velocity for the simulation result, which was 0.8 s in Fig. 5.7(b), is still shorter than that of the experimental result.

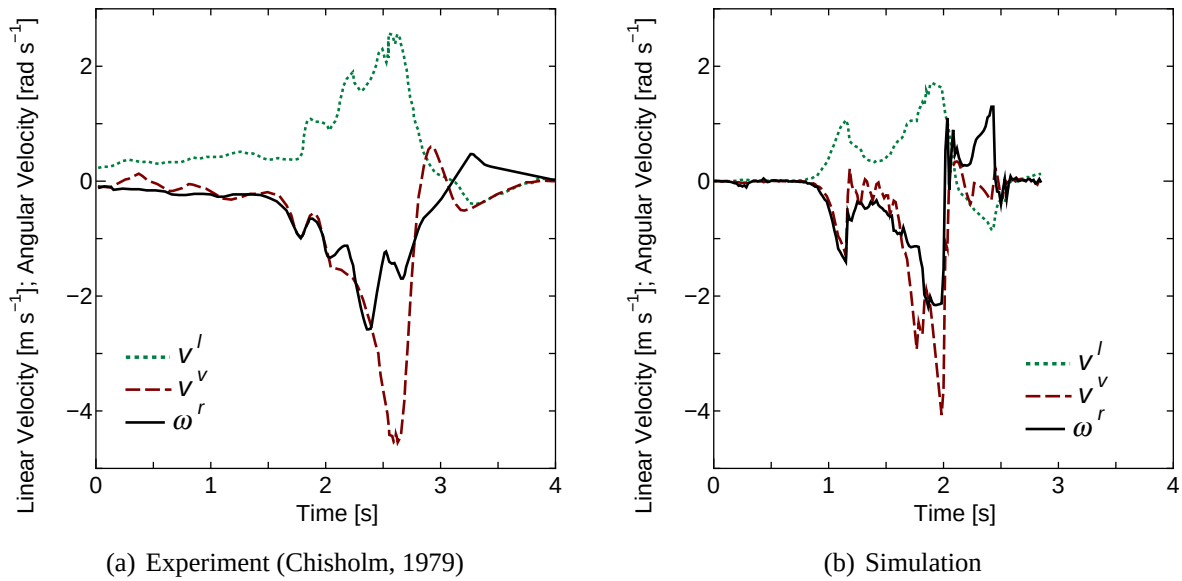


Figure 5.7: Results of linear velocity and angular velocity for Example #16.

Figure 5.8 shows the overturning behaviour in the case of Example #16, where the pivot angle of the front axle β is 0. Compared to the posture in Example #4, shown in Fig. 5.6(b), the front left tyre in Example #16, shown in Fig. 5.8(b), did not fall off the top surface but hung in the air because the front axle could not pivot and the front left tyre always remained at the same height as the other tyres. The result of velocities at 0.2 s in Fig. 5.7(b) show that vibration of the tractor was not obvious when the front left tyre travelled through the edge of the top surface. The strong impact of the lower parts of the tractor in Fig. 5.6(d) deferred the tendency to fall and changed the velocities to almost 0 m s^{-1} (rad s^{-1}) at 1.3 s in Fig. 5.7(b) because the centre of gravity of the tractor happened to be located vertically above the contact points. In the experimental result shown in Fig. 5.7(a), contact of the lower parts of the tractor did not have such an effect on the velocity value, which might be the reason why the error in case of Example #16 in Table 5.5 is larger than those in cases of Example #4 and other groups. Finally, the ROPS impacted the level surface and stopped falling of the tractor at 2 s, as shown in Fig. 5.8(e). Then the tractor rotated back momentarily and the left wheels fell back to level ground.

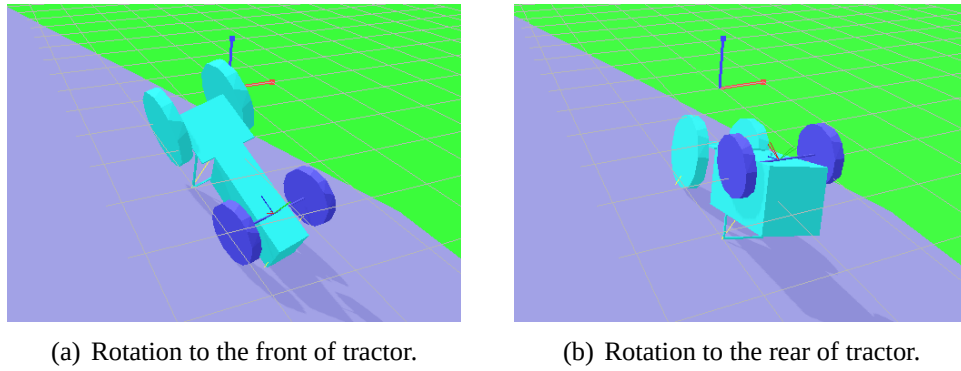


Figure 5.9: Possible difference of tractor behaviour in rotation direction.

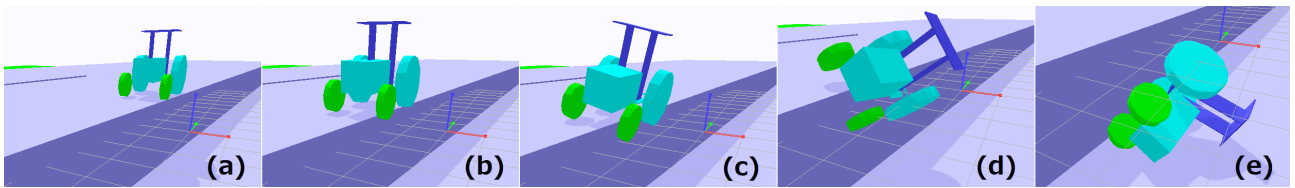


Figure 5.8: Overturning process at (a) 0, (b) 0.2, (c) 0.8, (d) 1.3 and (e) 2 s for Example #16 using Bullet.

Not only cases of Examples #4 and #16 but also other cases of examples exhibited similar phenomenon by which the time period of simulation result during the left wheels travelling through the bank edge became shorter than that of the experimentally obtained result. Chisholm (1979) reported that the left tyres deformed when it travelled through the edge between the top surface and the bank surface and supported the tractor even when it was travelling on the bank slope momentarily, which might explain the slower change in the experimental velocity from 0 s to 2 s. However, Bullet simulation uses a rigid cylinder component to simulate the dynamics of wheels, which is incapable of replicating the realistic behaviour of the soft tyre envelope over the sharp edge of the bank slope. Consequently, the overturning process of all examples in the simulation might become faster than that of experiments.

5.4.2 Tractor and uniform slope models

As defined in OECD (2017), non-continuous rolling behaviour means that the tractor might bounce off the slope surface after initial rotation about the contacting points of ROPS without successive rolling over. Consequently, ROPS can prevent the tractor from rolling upside down and provide a non-continuous rolling of the tractor. The lowest height of ROPS for non-continuous rolling behaviour is definable as the critical ROPS height (CRH).

Figure 5.9 shows the possible direction of rotation when ROPS is contacting the slope surface, and the behaviour of tractor can be separated into two cases: rotation to the front of the tractor, and rotation to the rear of the tractor. When the centre of gravity of the tractor lies in the forward position, the front part of the tractor, or the bonnet, would contact the slope surface as shown in Fig. 5.9(a). Otherwise, the rolling motion of the tractor would be backward. The upper edge of the rear wheel would contact the slope surface as shown in Fig. 5.9(b).

Table 5.6 summarises the results of the Bullet simulation for those examples (6.1 - 6.11) reported in OECD (2017). The critical ROPS height (CRH) obtained using Bullet clearly becomes smaller

than that of the OECD result for each example. If the coefficient of friction were increased to 1.0 in the Bullet model, then the slipping distance of contact points would become shorter, which also makes the CRH using Bullet closer to that of the OECD result. However, the result of the coefficient of friction 0.6 should be adopted because this setting corresponds more closely to reality from the result of preliminary simulation by Bullet. The direction of rotation of each example using Bullet also coincides with that of the OECD result.

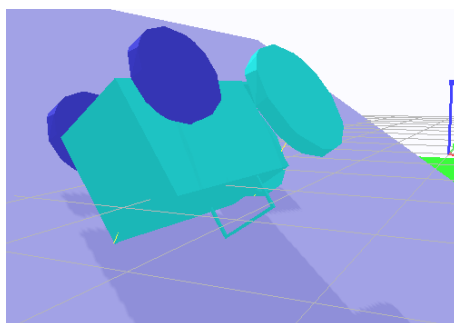
Table 5.6: CRH for Bullet result and OECD results

Cases of tractor from OECD (2017)	Simulation using Bullet		Result of OECD (2017)	
	CRH (cm)	Rotate direction	CRH (cm)	Rotate direction
Example 6.1	204	Front	212	Front
Example 6.2	198	Front	200	Front
Example 6.3	179	Front	No continuous rolling*	
Example 6.4	167	Rear	189	Rear
Example 6.5	242	Front	252	Front
Example 6.6	185	Rear	207	Rear
Example 6.7	192	Rear	Calculation not feasible†	
Example 6.8	168	Rear	189	Rear
Example 6.9	184	Rear	201	Rear
Example 6.10	187	Rear	192	Rear
Example 6.11	186	Rear	No continuous rolling*	

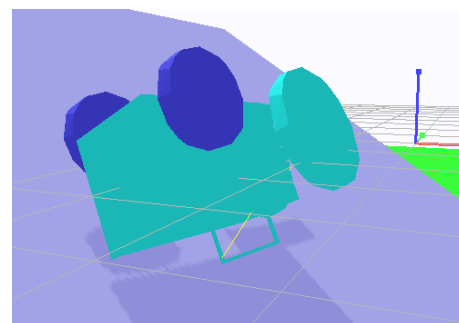
* Rolling stopped before ROPS hits the slope surface

† Engine bonnet hit the slope surface ahead of ROPS

In Examples 6.3 and 6.11 (Table 5.6), the simulation results of OECD demonstrated that the tractor stopped its rolling tendency after the side of the wheels contacted the slope surface, as shown in Fig. 5.4(b). Consequently, the tractor would not begin continuous overturning behaviour whatever the height of ROPS in the OECD program. Consequently there was no CRH for the tractors in Example 6.3 or 6.11. By contrast, the Bullet simulation can show the result of CRH even for those cases of Examples 6.3 and 6.11. Because the friction calculation in Bullet enabled the tractor in Examples 6.3 and 6.11 to slide toward the bottom side of the slope based on the friction of the wheels along the reverse direction, it is possible for the tractor to rotate temporarily after sliding along the slope surface.



(a) Engine bonnet hit the slope surface first.



(b) ROPS hit the slope surface later.

Figure 5.10: Overturning dynamics of a tractor in Example 6.7.

According to the sequence of overturning dynamics obtained using Bullet for a tractor in Example 6.7, as shown in Fig. 5.10, the overturning process differs from that of Fig. 5.4, where the tractor bonnet contacted the slope surface ahead of that of ROPS. As described earlier, calculation by the OECD algorithm was feasible if and only if the contact points were prepared a priori. The overturning process of the tractor must obey this precondition exactly. Otherwise, the calculation could not continue and would cause “Calculation not feasible” in the case of Example 6.7.

In the original OECD algorithm, the change in height of ROPS had no influence on the height of the centre of gravity or the roll moment of inertia of the object vehicle, which had no effect on the vehicle rolling behaviour (Sun et al., 2017). However, in the simulation using Bullet, the geometric data of each component such as the position of the centre of gravity and the moment of inertia are calculable individually. Consequently, with the increased height of ROPS, the dynamics of vehicle model as an assembly of components can also be influenced, which is one of the reasons that the results are obtainable using Bullet for those Examples 6.3 and 6.11.

Collision detection of Bullet not only enables the calculation of friction but also ensures its applicability for an unprepared case of overturning behaviour, such as ROPS of the vehicle in Example 6.7, unexpectedly affected the slope surface ahead of the effects of the bonnet.

From Table 5.6, a tendency is apparent by which CRH and the rotation direction of each example using Bullet converge to coherence compared to the simulation of OECD (2017). It is noteworthy that the CRH obtained using Bullet is about 6.6 percent lower than the CRH in OECD (2017), which might result from the introduction of friction. The contacting points would slip over the slope surface with energy loss, as reported for an agricultural carrier model by Sun et al. (2018).

5.5 Conclusion

This study was conducted to elucidate the ability of the Bullet physics engine to simulate overturning behaviour of agricultural tractors with a roll over protective structure (ROPS) on a bank slope and on a uniform slope. Although the duration of overturning motion was shorter in simulation, compared to reported experiments, probably because of the assumption of rigid-body contact, Bullet simulation results were in reasonably good agreement with experimental results with small RMSE values for the rigid model. Moreover, the simulation results were sufficiently accurate to predict the critical ROPS height (CRH) for various agricultural tractors.

For future work, a soft model for tyre or surface of slope may be utilised in the Bullet simulation, so that the dynamics of the tractor during the transition from translation to rotation would become comparable to the experiments and thus contribute to further reducing the RMSE values in Table 5.5.

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Chapter 6

Overturning Experiment of Tractor with ROPS

6.1 Introduction

The experiment in this chapter is cooperated with Hunan Agricultural University to measure the angular velocity and the rotating angle of a tractor and quantify impact of the ROPS during the lateral overturning. The purpose of this experiment is to investigate the relationship between the dynamic of tractor with the impact of ROPS during an overturning. Moreover, the dynamics of a tractor which is vertically towed to overturn is compared to that of a tractor which is longitudinally along a bank slope at idling speed and overturned itself.

6.2 Method

6.2.1 Main body of tractor

Table 6.1: Parameters of Dongfanghong SA-250 tractor

	Specification	Weight
Front wheel	Track width 1.0 m; Tread width 0.11 m; Diameter 0.62 m	15 kg
Rear wheel	Track width 1.02 m; Tread width 0.22 m; Diameter 1.04 m	55 kg
Bonnet	Length 1.01 m; Width 0.75 m; Height 0.83 m	10 kg
Engine	Length 0.65 m; Width 0.40 m; Height 0.80 m	260 kg
Transmission	Length 1.30 m; Width 0.24 m; Height 0.31 m	676* kg
Frame	Thickness 0.08 m; Width 1.19 m; Height 1.39 m	56 kg

* including all mechanical components after engine, such as rear transaxle, power take-off

The small-sized SA-250 tractor (1100kg) manufactured by Dongfanghong Industry (Luoyang, China) did not equip with ROPS originally (YTO, 2018). The geometry of the ROPS was redesigned to be wider than any vertical side components of the tractor and the bottom of the ROPS was fixed on the rear part of the transmission.

The tractor is of simple structure. The main components of this tractor are wheels, transmission, engine, bonnet and ROPS, which are regular shapes. On the assumption that the whole tractor kept lateral symmetry and the center of gravity of each component was their geometrical center, the weight



Figure 6.1: Example of assembly model.

of each component could be easily calculated as shown in Table 6.1 after the weight distribution on front and rear wheels (39% on front and 61% on rear) and weight of each wheel had been measured.

After the weights of all components had been determined as presented in Table 6.1, the height of center of gravity of the tractor was calculated as 0.64 m off the ground.

6.2.2 Measuring device

Two JY-901 gyroscopes (WIT-Motion) were attached on the left and right fender of rear wheels as shown in Fig. 6.1, which can record the acceleration including gravity, angular velocity and rotating angle every 1/10 second (WIT, 2018).

An electronic force sensor (MEACON, MIK-LCLY-G) was assembled between the main frame of ROPS and the widen impact bar as shown in Fig. 6.1. The widen impact bar on the right side of the main frame would laterally impact the electronic force sensor so that only lateral impact force could be measured. The instant value would be displayed and only extreme value could be recorded, so the only maximum impact force during overturning and the instant impact force after all the motion of tractor stopped would be collected. The sensor measures the lateral impact force 10 times per second (MEACON., 2018).

6.2.3 Overturning setting

Figure 6.2 shows the coordinate of the gyroscope on the right fender, the longitudinal axis follows the forward direction of the tractor, the lateral is toward the left side of the tractor, and the vertical axis points to the up direction of the tractor. The coordinate rotates along with the overturning of the tractor.

In the case of the overturning experiment without traveling speed, the ROPS was laterally towed to overturn the tractor on the level ground. For the overturning experiment with traveling speed, the tractor traveled at 0.77 m s^{-1} as an idling speed, and the right wheels were downhill along the 4.4 deg bank slope until the tractor became overturned.

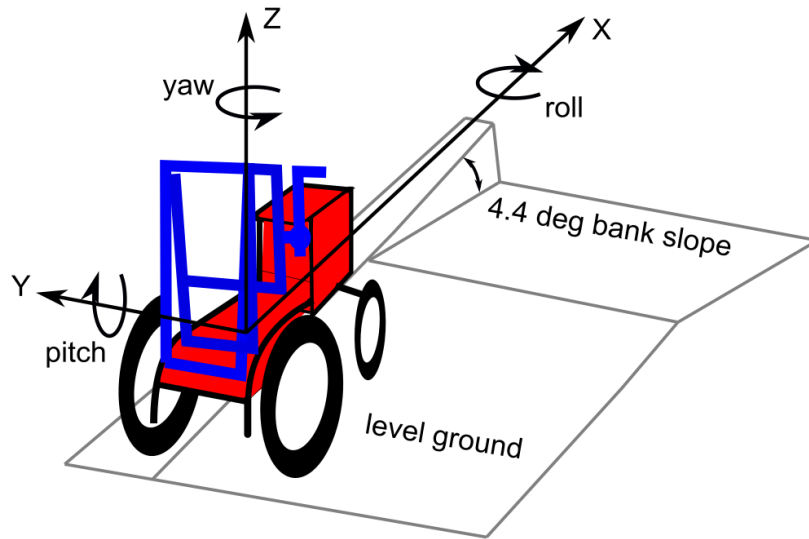


Figure 6.2: Example of tractor bank slope and coordinate.

6.3 Results and discussion

6.3.1 Results of overturning dynamics without traveling speed

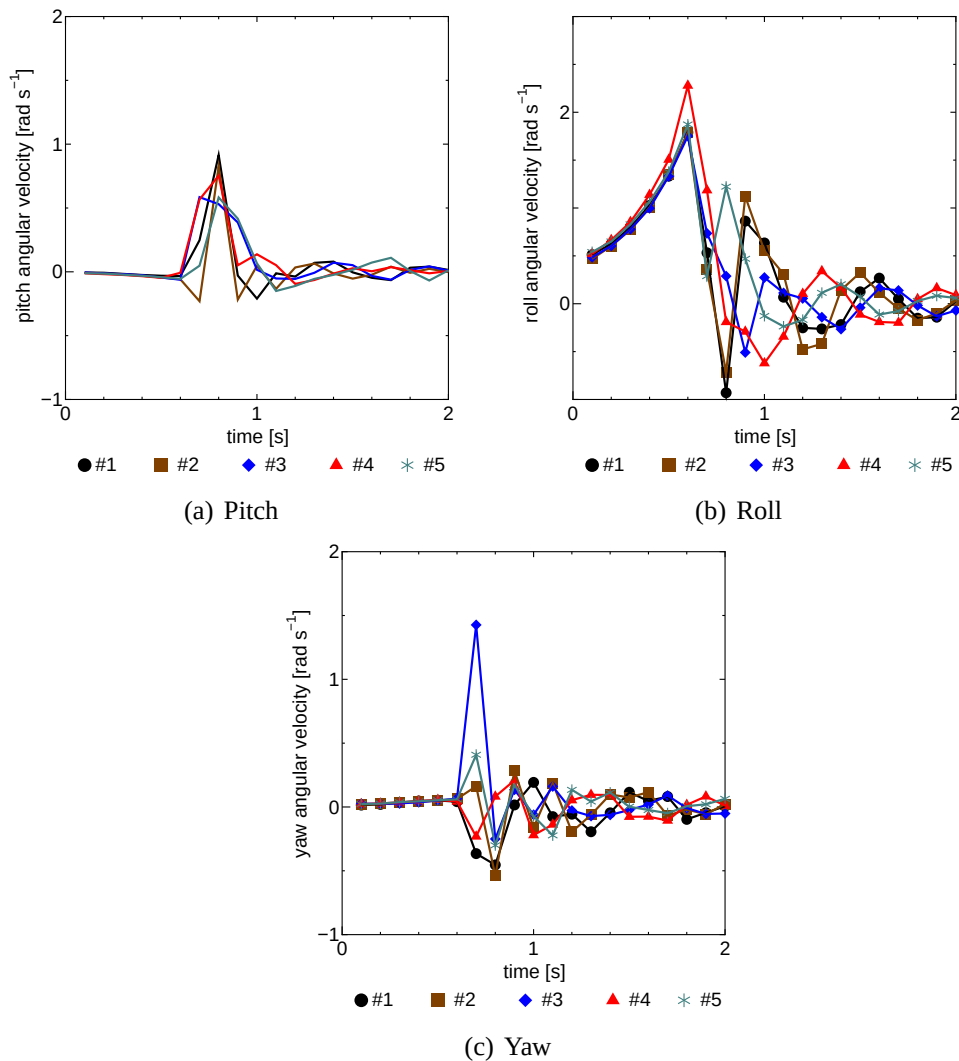


Figure 6.3: Result of angular velocity on left fender with no traveling speed.

Figure 6.3 is the observed angular velocity along three directions for the cases that tractor was laterally towed to overturn toward right side of vehicle. The timing when the tractor overturned by gravity acceleration was regard as the origin. The obvious overturning tendency of the tractor stopped after 2 s. The overturning experiments with no traveling speed were repeated 5 times.

The maximum values in Fig. 6.3(b) at 0.6 s reached a close range to 2 rad s^{-1} when the ROPS was going to impact the level ground. The changes of 5 experiment groups after 0.7 s in Fig. 6.3(b) became different from each other, which might because the main body of the tractor liberated irregularly after the impact of the ROPS.

The dynamics along pitch direction from 0.6 to 0.9 s in Fig. 6.3(a) were similar in all 5 experiment groups, which caused by the fact that the front pivot axle returned to its default pivot angle 0 deg after the impact of the ROPS. The vibrations after 1 s were insignificant.

The results in Fig. 6.3(c) show that the angular velocities along yaw direction might be caused by irregular vibration and be insignificant.

Figure 6.4 is the results of angular velocities along all three directions observed on the right fender. The results along roll direction and pitch on right fender in Fig. 6.4(a) and (b) resembled that on left fender in Fig 6.3(a) and (b). The liberations along yaw direction in Fig 6.4(c) were also irregular and insignificant.

Figure 6.5 portrays the results of rotating angles along roll directions observed on the left and right fender. The results show that the rotating angles which represent the lateral instant posture of the tractor during overturning converge to coherence. Moreover, the curves became smooth after 0.8 s in Fig. 6.5, which also means that the tractor body did not liberated obviously even their angular velocities reported relatively large as shown in Fig. 6.3(b) and Fig 6.4(b).

Figure 6.6 shows the results of lateral acceleration observed on the left and right fender. The accelerations along lateral direction on left fender result in a reversed tendency to that of right fender, because the right fender might contact with the level ground during the overturning of tractor so that the right fender was compressed by both level ground and tractor body.

6.3.2 Results of overturning dynamics with traveling speed

Figure 6.7 is the observed angular velocity along three directions for the cases that tractor was traveling forward at 0.77 m s^{-1} along a bank slope with idling speed. The timing when the tractor overturned by gravity acceleration was regard as the origin. The obvious overturning tendency of the tractor stopped after 2.5 s, which is slightly longer than that of the cases with no traveling speed. The overturning experiments with traveling speed were repeated 3 times.

The curves in Fig. 6.7(b) are similar to the cases with no traveling speed in Fig. 6.3(b) and Fig. 6.4(b). But the curves in Fig. 6.6(a) are different to that in Fig. 6.3(a) and Fig. 6.4(a), because the right front wheel slipped toward the bottom side of the bank slope from 0.6 to 1.1 s. The results along yaw direction as shown in Fig. 6.7(c) are irregular as those in Fig. 6.3(c) and Fig. 6.4(c).

Figure 6.8 portrays the observed angular velocity along three directions for the cases that tractor overturned with traveling speed. The results on right fender in Fig. 6.8 along pitch, roll, and yaw direction resemble that in Fig. 6.7.

Figure 6.9 shows the similarity in rotating angles along roll directions on the left and right fender.

Figure 6.10 portrays the results of lateral acceleration observed on the left and right fender. The accelerations along lateral direction result in the same tendency as the cases with no traveling speed in Fig. 6.6 that the left fender reach the opposite extreme to the right fender.

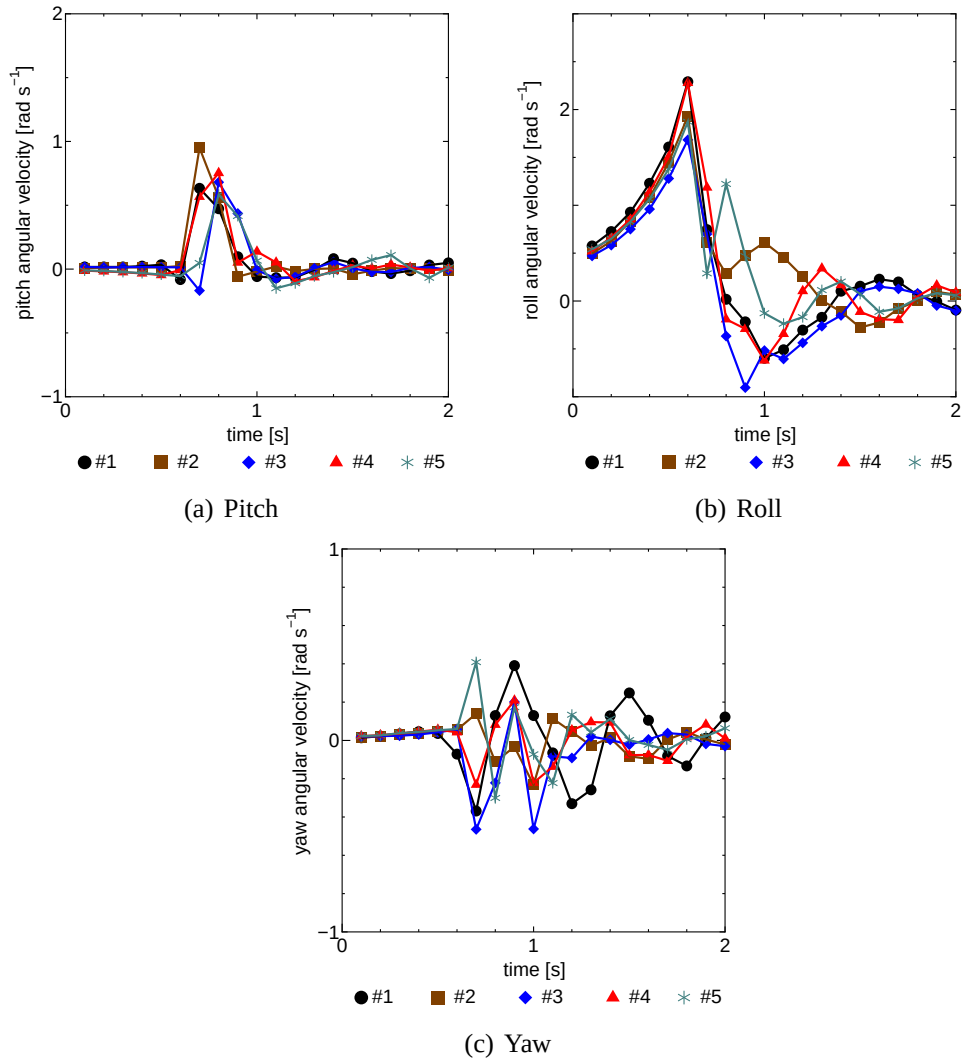


Figure 6.4: Result of angular velocity on right fender with no traveling speed.

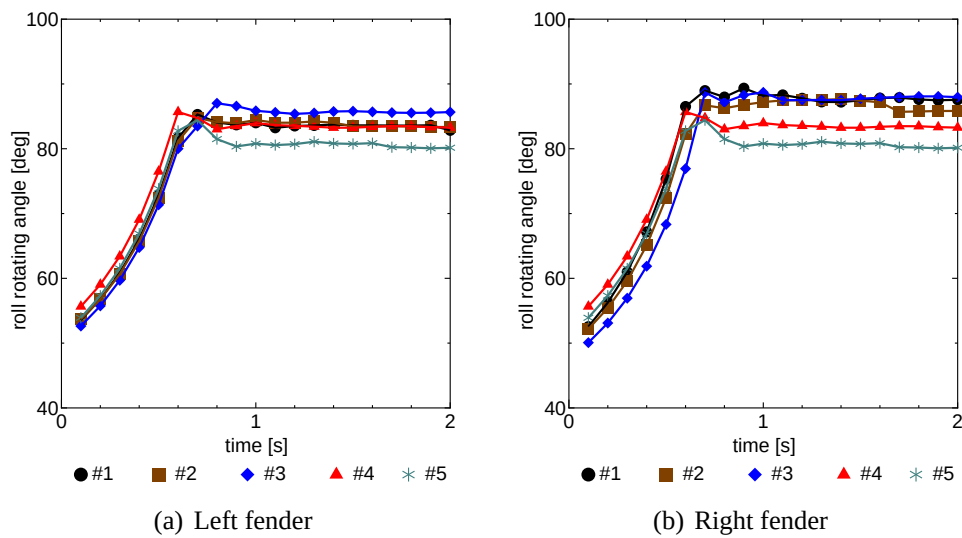


Figure 6.5: Result of rotating angle along roll direction with no traveling speed.

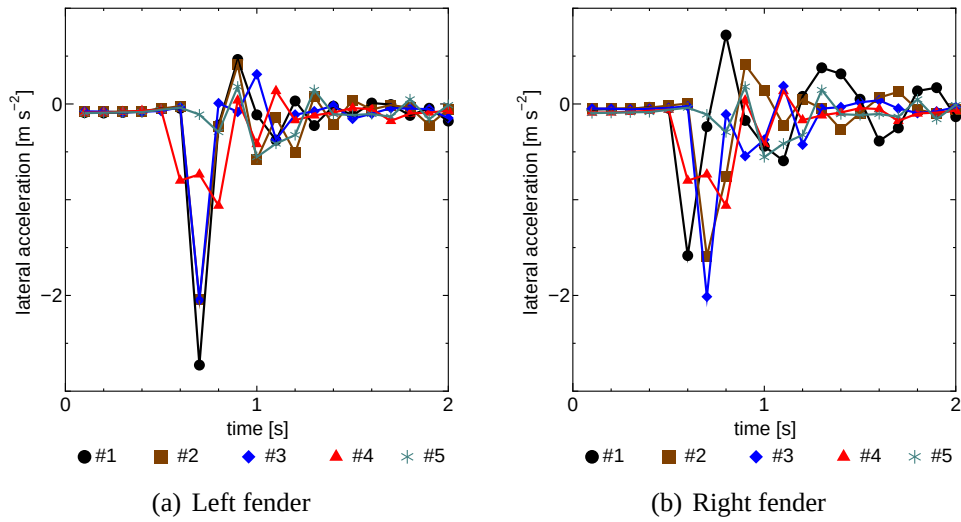


Figure 6.6: Result of lateral acceleration with no traveling speed.

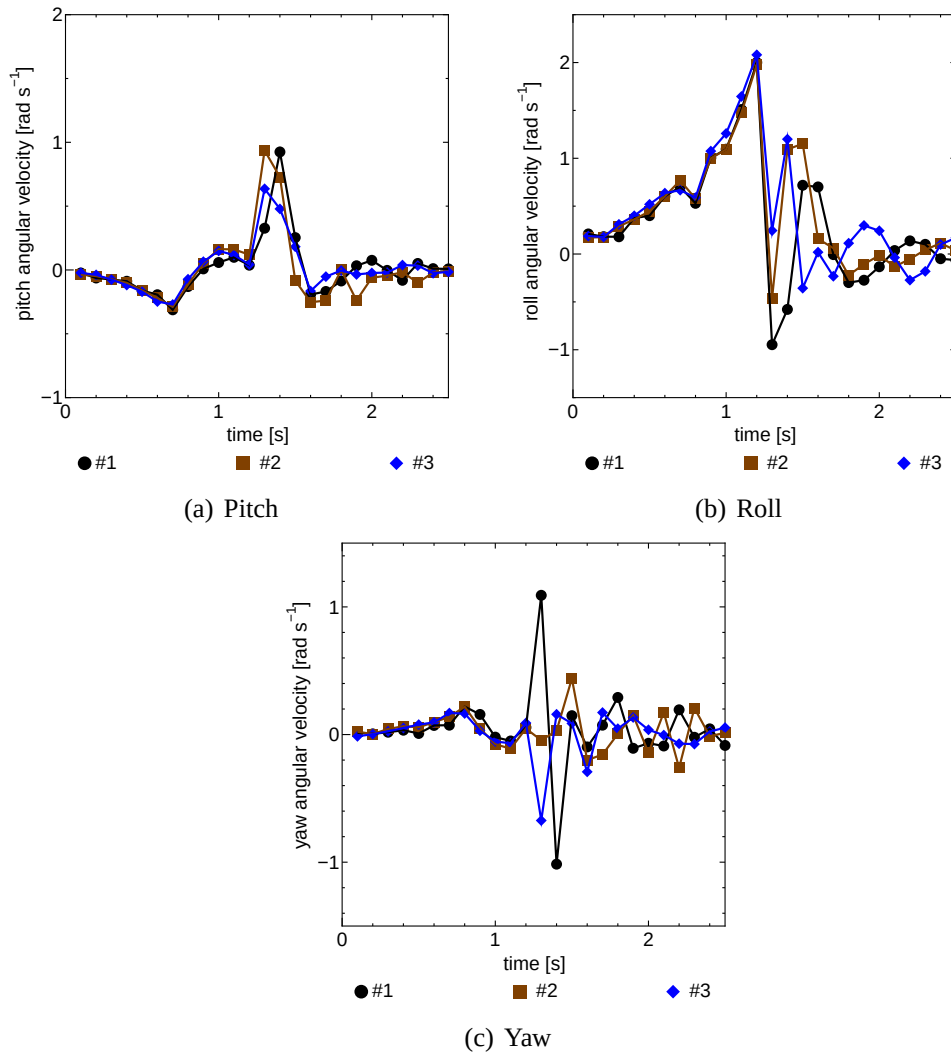


Figure 6.7: Result of angular velocity on left fender with traveling speed.

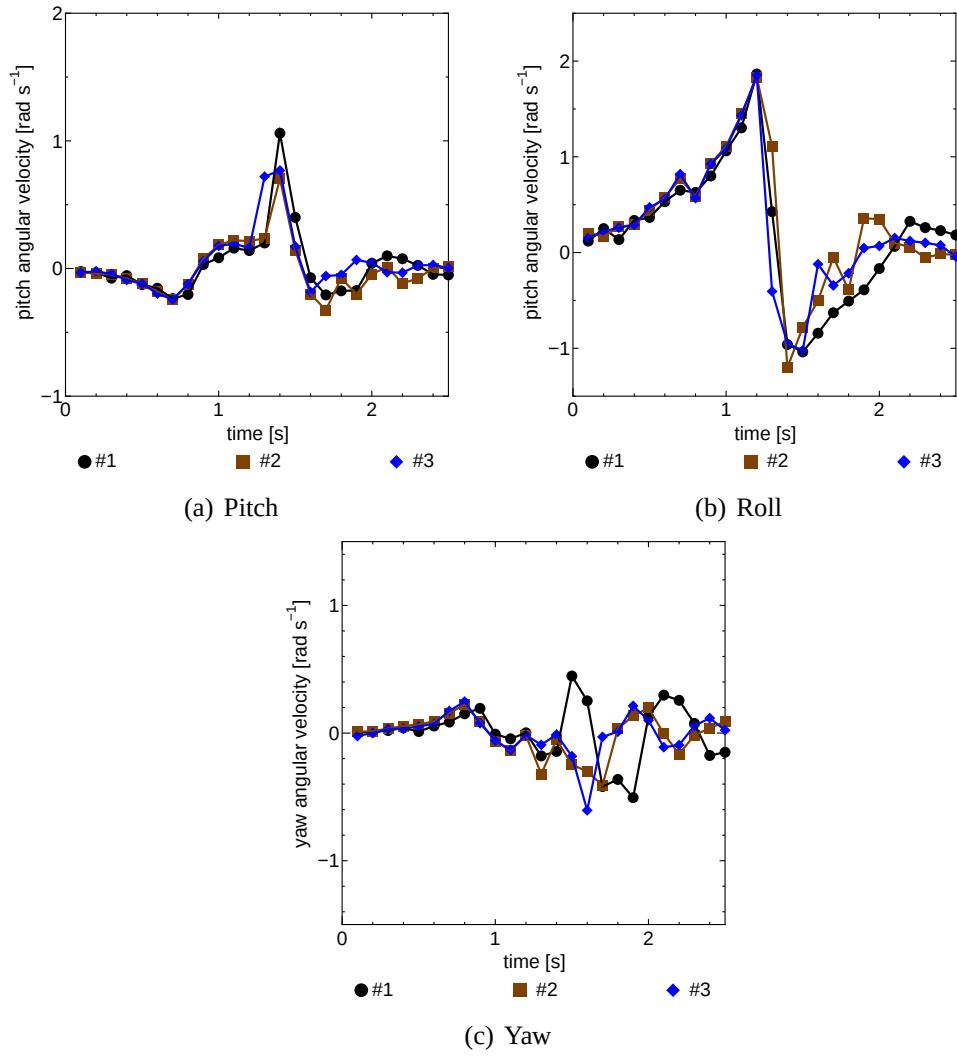


Figure 6.8: Result of angular velocity on right fender with traveling speed.

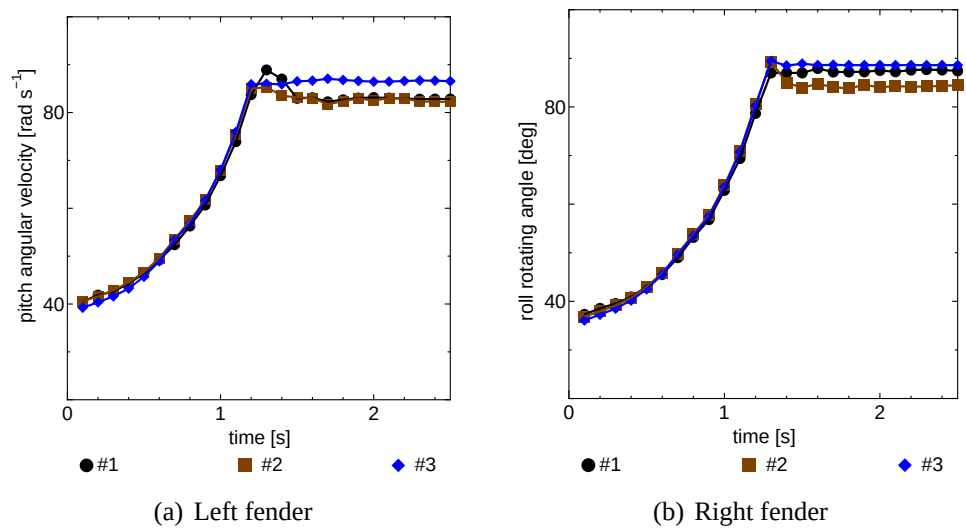


Figure 6.9: Result of rotating angle along roll direction with traveling speed.

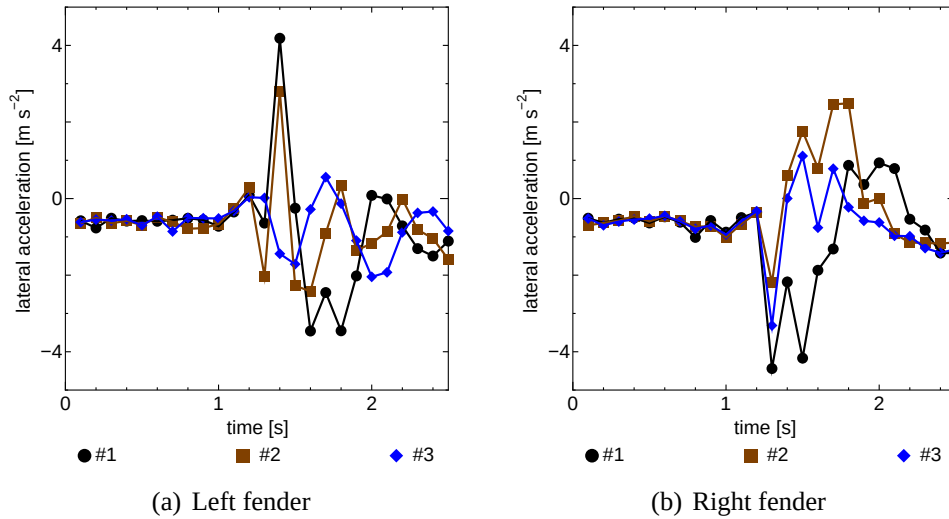


Figure 6.10: Result of lateral acceleration with traveling speed.

Table 6.2: The lateral impact force on ROPS in experiment.

		Maximum impact force [N]	Steady force [N]
No traveling speed	#1	3949	2577
	#2	4165	2489
	#3	5135	cable out
	#4	contact bar stuck into a hole, data not useful	
	#5	3685	2264
With traveling speed	#1	5125	2587
	#2	5125	2783
	#3	4096	2293

6.3.3 Results of lateral impact force

Table 6.2 shows the result of the measured lateral impact force on ROPS including the maximum value during overturning and the steady value after all the motion of tractor stopped. The contact point of ROPS in experiment #4 with no traveling speed happened to rotate into a hole on the level ground which is filled with soft sand and differs the observed data.

The maximum impact forces of the cases with no traveling speed in Table 6.2 result in the same range as the that of the cases with traveling speed. According to those dynamic data specially the angular velocities, the overturning dynamics of the tractor in both cases with and without traveling speed tend to be a stable performance every time. Since there have lots of vibrations along yaw direction, which changes the weight distribution between front right wheel and ROPS, the impact force shared by the front right wheel might cause the differences in maximum impact force.

The steady forces of the cases with and without traveling speed in Table 6.2 converge to coherence because the final postures after the motion of the tractor stopped are much similar. The differences in observed results might be induced by that the pivoted angles of the front axle are irregular during experiments.

6.3.4 Discussion

The observed dynamic data along pitch and roll direction show that the overturning performances with no traveling speed resemble that with traveling speed at 0.77 m s^{-1} . The vibrations in angular velocities along yaw direction portray that the pivot front angle affects on stability of the tractor when the ROPS impacts the level ground, which might also influence the maximum impact force on ROPS. The lateral accelerations on left fender have the opposite extreme value to that on right fender, which means that right fender might contact the level ground during the overturning of the tractor.

Comparing to the previous experiment by Chisholm (1979) which overturned a tractor on a bank slope, the tractor in this experiment was freely overturning before contact the ground and the angular velocity linearly increased to the maximum value as shown Fig. 6.7 and 6.8, but the tractor in Chisholm (1979) was slipping and overturning along the bank slope before the impact of ROPS and the acceleration of angular velocity fluctuated in 4.5 and 4.7. The peak value of angular velocities at all direction in Chisholm (1979) was almost 50 percent larger than the that in our experiment which might because their tractor fall from a 2 meters height bank slope and the motion of tractor was accelerated.

The results of rotating angle at roll direction in our experiment are similar to that in Nichol et al. (2005) which overturn a tractor by using one side wheel hitting a bump of hill even their travelling speed was much faster (1.417 m s^{-1}).

6.4 Conclusion

The experiment of measuring the overturning dynamic of the small-size commercially available tractor and the impact force of ROPS as the primarily lateral impacting point has been conducted. The overturning performances in both the cases without and with 0.77 m s^{-1} traveling speed are similar. The maximum impact force on ROPS to stop the continuous overturning might contribute to the design for rear mounted safety frame.

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Chapter 7

Prediction of Impact Force on a Tractor with ROPS using Physics Engine

7.1 Introduction

In this chapter, whether or not Bullet physics engine can quantify the lateral impact on ROPS when the tractor overturns in the experiments done in Chapter 6 is investigated. The coherence in overturning dynamics which is a precondition for comparing the lateral impact on ROPS, is verified by video identification and statistical methods. The capability of calculating impact force acting on ROPS might contribute to structural analysis for protective efficiency of ROPS against various tractor and slope conditions.

7.2 Method

In the previous experiments, we measured the acceleration data and impact force during overturning of a small-size tractor (1100kg) manufactured by Dongfanghong Industry (Xiao et al., 2018). The tractor could only drive in 2WD.

Since the original tractor did not install ROPS, the newly designed geometry of the ROPS with an extension bar as shown in Fig. 7.1 became wider than any side components of the tractor so that the primarily impacting point of the tractor during sideways overturning could be an edge of ROPS. A force sensor was installed between the main frame of ROPS and the extension bar (see Fig. 7.1). Two gyroscopes were attached for acceleration measurement on the fender of left and right wheels. Measuring frequency for both force and acceleration was 10 Hz.

7.2.1 Tractor model

Dongfanghong SA-250 small-sized tractor is of simple structure and easy to build a simulation model in Bullet. The main components of this tractor are wheels, transmission, engine, bonnet and ROPS, which can be defined by regular shapes. To make our Bullet model as close as the original tractor, all components are replicated by the similar Bullet rigid shapes, such as cylinder shape for wheels and box shape for other components. The critical parameters of the Bullet model kept the same as the measurement of experiment, which are listed in Table 6.1 (Page 71).

The gravity center of the tractor with ROPS is measured as 1.06 m from front axle and 0.64 m height from the ground. The moments of inertia around the gravity center computed by Bullet are



Figure 7.1: Example of overturning experiment with a designed ROPS with force sensor.

678.2 kg m² along pitch direction, 238.4 kg m² along roll direction, and 520.1 kg m² along yaw direction.

7.2.2 Constraint of components

Constraint in Bullet physics engine enables the relative displacement between two connected rigid shapes. Four constraints, which allow the connected shapes rotate along one direction, were used in our tractor model.

The front axle with tyres (shown as green parts in Fig. 7.2(a)) used a constraint which could freely rotate along pitch direction around the front anchor (purple part over front axle in Fig. 7.2(a)). The front anchor was assembled to the main tractor body as a constraint with angular limit along roll direction, which achieved the pivot angle of the front axle.

The designed ROPS as the blue parts in Fig. 7.2(b) connected transmission by a roll direction constraint with feedback function, which output the torque and angular displacement of impact at ROPS. Finally, the rear axle had a constraint which applied torque (30 N m) along pitch direction was given to drive the tractor forward by 2WD at 0.77 m s⁻¹.

Recording the calculated impact on constraint

The source code of this study is saved in **Tank.cpp**, and the part of intercepting the data transport of constraint is also in this file.

- **Frame_con->setJointFeedback(fb);** is the specific command to the constraint **Frame_con** between main tractor body and frame, which orders Bullet physic engine to report its data transport to the memory space created by **btJointFeedback* fb = new btJointFeedback();**
- **void Tank::stepSimulation(float deltaTime)**, this new section in **Tank.cpp** includes the commands to select the data for recording. **FrameHinge2->getRigidBodyA().getInterpolation-**

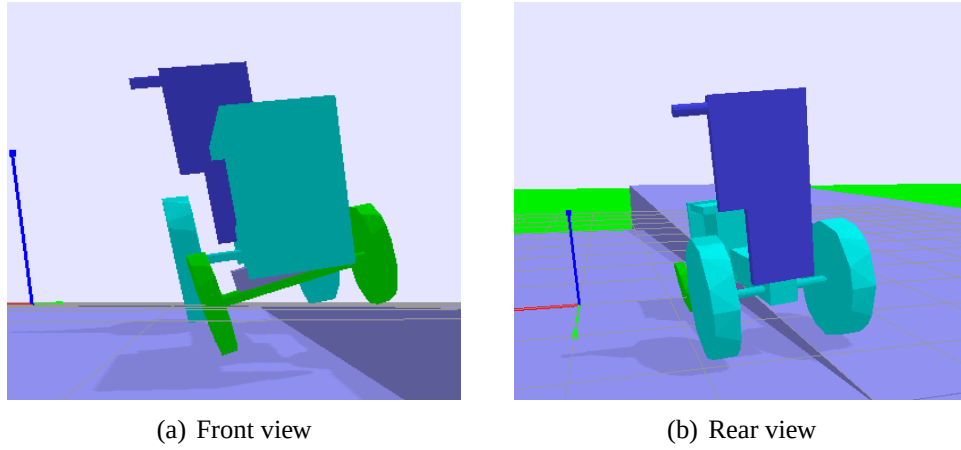


Figure 7.2: Example of tractor model and slope model.

WorldTransform().getRotation().getX(); in this section orders the program to read the data of angular displacement of **BodyA()** from collision pair at **getX()** direction, which can also change for **BodyB()** from collision pair at **getY()**, **getZ()**, and **getW()** direction. The angular displacement of constraint is recorded in “Carrier-ResOutTEST.txt”. Moreover, the command **m_jointFeedback[i]->m_appliedTorqueBodyA.x()** read the data of torque of **BodyA()** from collision pair at **x()** direction, which can also change for **BodyB()** from collision pair at **y()**, and **z()** direction.

7.2.3 Slope condition

The right wheels of the tractor would be travelling on a 4.4 deg slope, and the left wheels would be on the level surface, as shown in Fig. 7.2. When the right side of the tractor became lower and the lateral stability was disturbed, the tractor began rotating sideways. To improve the visual identification of the longitudinal and lateral displacement of tractor model, the 1 m × 1 m scale meshes were imported on the level surface as the gray mesh in Fig. 7.2.

The coefficient of friction of the slope was set at 0.75, and the coefficient of restitution of it was 0.2 in Bullet.

7.2.4 Data processing by Root Mean Square Error

The Root Mean Square Error (RMSE) is used to evaluate the dynamic variations between the simulation result and the experimental result. Firstly, the output data before the overturning is cut that only the period of overturning would be compared. The processed period of cases with no traveling speed is 2 s, and the processed period of cases with traveling speed is 2.5 s, so that the zero variation pairs between simulation and experimental results can be rejected. The RMSE of the processed simulation and experimental data is calculable as

$$RMSE = \sqrt{\frac{\sum_{n=1}^{Total} (data_n^{exp} - data_n^{sim})^2}{Total}}, \quad (7.1)$$

where *Total* is total number of data, which is 20 for the cases with no traveling speed and 25 for the cases with traveling speed; $data_n^{exp}$ is the result of angular velocity or rotating angle at $n/10$ s in

Table 7.1: RMSE result with no traveling speed.

	Pitch [rad s ⁻¹]	Roll [rad s ⁻¹]	Yaw [rad s ⁻¹]	Angle [deg]
#1	0.23	0.40	0.14	6.09
#2	0.21	0.41	0.17	5.99
#3	0.20	0.28	0.37	4.46
#4	0.18	0.35	0.17	8.30

experiment; $data_n^{sim}$ is the result of angular velocity or rotating angle at $n/10$ s in simulation.

The result of RMSE could confirm whether the dynamics of simulation became similar to that of experiment or not. Therefore, the impact force of simulation became comparable to the experimental result.

7.3 Results and discussion

7.3.1 Validation of overturning dynamics without traveling speed

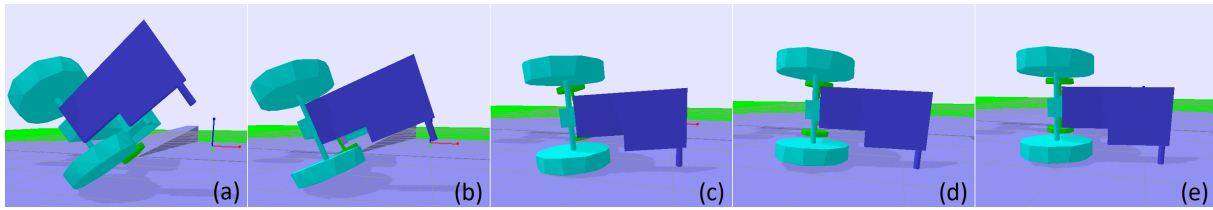


Figure 7.3: Result of overturning dynamics with no traveling speed.

Figure 7.3 shows a series of sideways overturning behaviour of the tractor when the rear wheels are not powered. In Fig. 7.3(a), the tractor sets at an angle to the level surface such that the gravity acceleration can turn over the tractor directly. Then, all postures as (b) during overturning, (c) before impact, (d) doing impact, and (e) after impact are presented in Fig. 7.3.

Because the ROPS is wider than the outside of the wheels, the impact at the bar on the ROPS might become the major contribution to stop the overturning of the tractor.

The simulation result of angular velocities and rotating angle along pitch, roll, and yaw direction can be obtained by Bullet. Since the result of RMSE can portray the difference between simulation and experiment, 20 simulation data of velocities along pitch, roll, or yaw direction and 20 simulation data of rotating angle along the critically roll direction were compared with 20 experimental data for four experimental groups as shown in Table 7.1.

The results of RMSE for roll angular velocity tend to be larger than other angular velocities as summarised in Table 7.1. To investigate the difference between the experimental data and simulated data, the graph of angular velocities and roll rotating angle from the experiment #1 detected at left fender gyroscope and that from simulation are presented in Fig. 7.4.

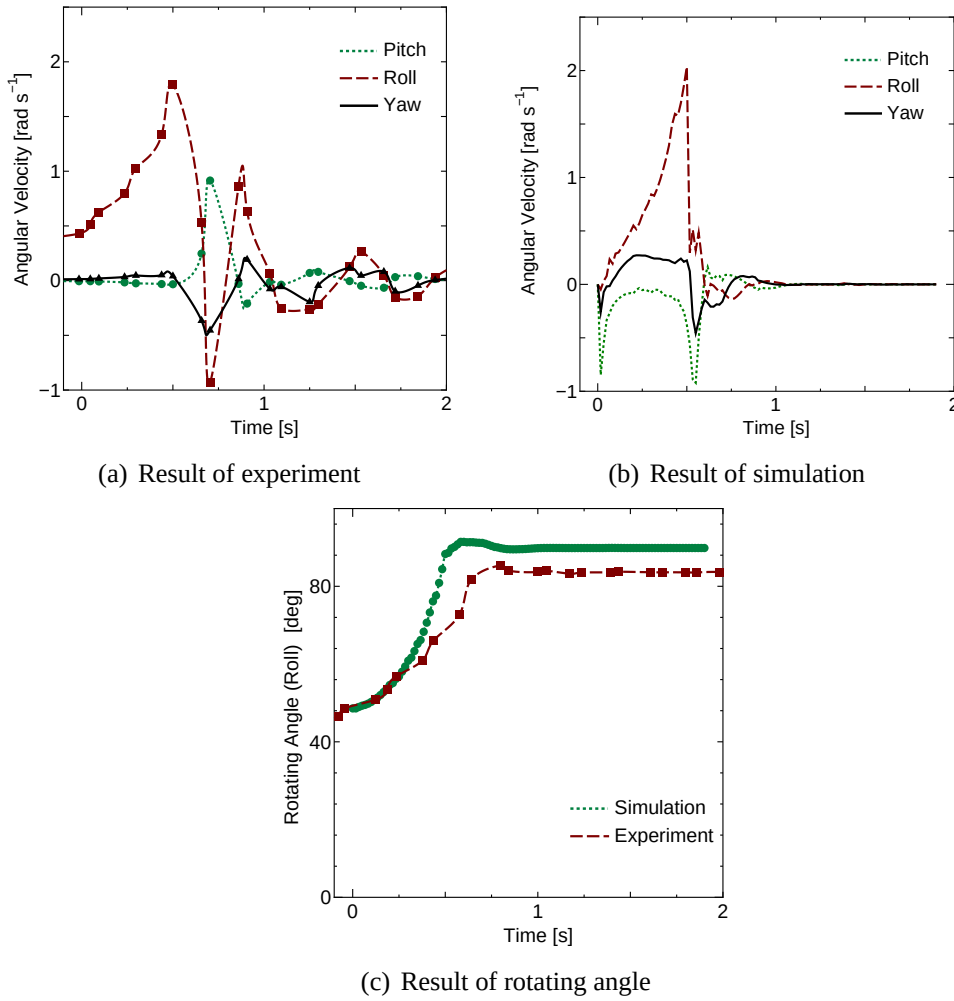


Figure 7.4: Comparison between simulation and experiment #1 with no traveling speed.

The angular velocity along roll direction in Fig. 7.4(a) started from almost 0.5 rad s^{-1} , because the tractor in experiment was laterally towed to rollover. The starting posture for tractor model in Bullet was set at a vertical angle at which the center of gravity of the tractor model had already been the unbalanced side to rollover. After 0.5 s in Fig. 7.4(a), the tractor contacted the horizontal ground surface and the roll velocity did not stop immediately because the left fender attached with the velocity gyroscope was not firmly fixed on the main tractor body and vibrated slightly. But the simulation results output the velocity data of the center of gravity of the tractor since the tractor model was a rigid model. There was no obvious vibration after the contact at 0.5 s between tractor and horizontal surface in Fig. 7.4(b).

The starting angular velocity along pitch direction in Fig. 7.4(b) dropped to almost -1 rad s^{-1} , because the starting posture for simulation model was set at an exact vertical angle as we mentioned above but the front axle was set at the same angle as the main tractor model. The front axle pivoted around the main tractor body to its maximum pivot angle after 0 s, which dropped down the front parts of the main tractor body and resulted in the difference of the beginning of the pitch velocity.

The angular velocity along roll direction ranged from -1 to 2 rad s^{-1} in Fig. 7.4(a) and (b). However, the range of the angular velocity along pitch and yaw direction was three times smaller than that of roll direction, which means that the RMSE result of roll angular velocity is better than the others in Table 7.1.

Table 7.2: RMSE result with no traveling speed.

	Pitch [rad s^{-1}]	Roll [rad s^{-1}]	Yaw [rad s^{-1}]	Angle [deg]
#1	0.37	0.41	0.39	9.32
#2	0.37	0.50	0.21	10.00
#3	0.30	0.47	0.24	8.59

Figure 7.4(c) shows that the tendency of simulation data for roll rotating angle is similar to that of experiment. According to Table 7.1, comparing to the range of rotating angle which is from 48 to 88 deg, the largest RMSE value of experiment #4 (8.30) becomes acceptable.

7.3.2 Validation of overturning dynamics with traveling speed

Figure 7.5 portrays a series of overturning dynamics of the tractor which started from uphill and rotated to impact the slope surface. The forwarding speed of gravity center of the tractor was set at 0.77 m s^{-1} and the power on rear wheels was not cut down until the whole overturning period finished.

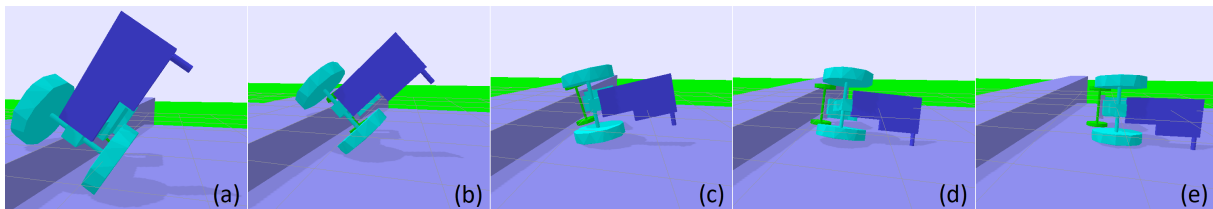
**Figure 7.5:** Result of overturning dynamics with traveling speed.

Table 7.2 is the result of RMSE between experimental data and simulation data. The overturning period of the cases with traveling speed becomes slightly longer than the cases with no traveling speed, so that the total data number for compared pairs from one group between experiment and simulation in Table 7.2 is 25.

The results of RMSE in Table 7.2 are relatively larger than that in Table 7.1, which means that the accuracy of simulation for overturning dynamics with traveling speed is worse than that with no traveling speed.

Figure 7.6 shows the graph of angular velocities and roll rotating angle from the experiment #1 at left fender gyroscope and simulation. Comparing to the result of overturning with no traveling speed in Fig. 7.4, the angular velocities of simulation in Fig. 7.6(b) are much similar to that of experiment in Fig. 7.6(a), which might be caused by the fact that the tractor was driven to rollover and the beginning of overturning in simulation was much similar to that in experiment.

In conclusion, the angular velocities and rotating angle of simulation with traveling speed resembles the experiment #1 result. Although the graph of angular velocities and roll rotating angle in Fig. 7.6 seems to be closer than that in Fig. 7.4, the ranges of angular velocities and roll rotating angle in Fig. 7.6 are almost the same as those in Fig. 7.3. According to the RMSE in Table 7.1 and 7.2, the simulation accuracy of the cases with traveling speed is inferior to that of the cases with no traveling speed.

Consequently, the result of RMSE shows that the difference between simulated dynamics and experimental dynamics is insignificant.

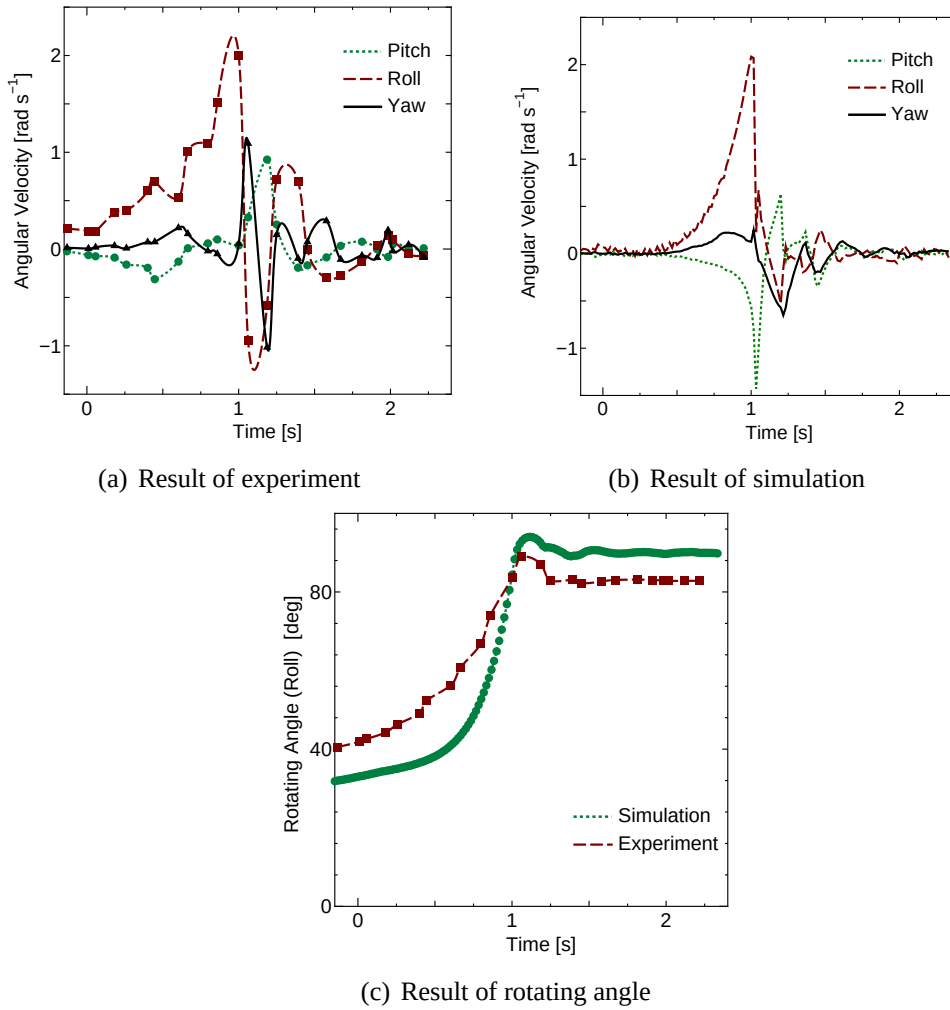


Figure 7.6: Comparison between simulation and experiment #1 with traveling speed.

7.3.3 Result of impact force prediction

As shown in the dynamics data in Fig. 7.4 and 7.6 along with the RMSE result in Table 7.1 and 7.2, the overturning dynamics of tractor in roll direction using Bullet becomes similar to the experimental results. Therefore, the dynamics of ROPS in Bullet model might resemble to that in experiment, which means the impact force on ROPS by Bullet becomes comparable to the experimental result. The maximum impact force on the contacting point of ROPS during overturning of tractor is summarised in Table 7.3. The steady force in Table 7.3 is measured after all the motion of tractor stopped.

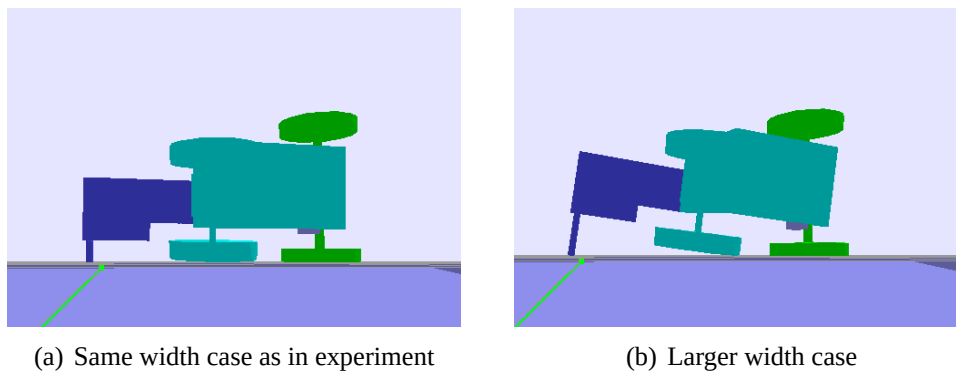
The default speed of the cases with travelling speed in simulation is 0.77 m s^{-1} . To investigate the relationship between minor changes in travelling speed and the influences in impact force, the cases with travelling speed 0.6 m s^{-1} and 0.88 m s^{-1} are also simulated for reference in Table 7.3, and the coherence in overturning dynamics had also been verified by RMSE.

Whichever case of the overturning tractor with or without traveling speed, the results of maximum impact force using Bullet are clearly within the same range as the experimental results. However, there are large differences in the results of steady force between simulation and experiment.

Table 7.3: The lateral impact force on ROPS in experiment.

		Maximum impact force [N]	Steady force [N]
No traveling speed	#1	3949	2577
	#2	4165	2489
	#3	5135	cable out
	#5	3685	2264
	simulation	4028	127
With traveling speed	#1	5125	2587
	#2	5125	2783
	#3	4096	2293
	simulation ^a	4292	804
	simulation ^b	4479	715
	simulation ^c	4851	970

^aTraveling speed is 0.6 m s^{-1} ; ^b Traveling speed is 0.77 m s^{-1} ; ^c Traveling speed is 0.88 m s^{-1}

**Figure 7.7:** Comparison of cases between same width ROPS and wider ROPS in simulation.

To investing further, the effect of width of ROPS is shown in Fig. 7.7. Figure 7.7(a) shows that the tractor was twisted and the engine bonnet became close to the level surface with the width of ROPS set at the same value as simulation and experiment in Fig. 7.3 and 7.5. Therefore, the center part of left rear wheel shared part of body load, which might decrease the load on ROPS and resulted in an extremely small value of steady force as in Table 7.3.

The reason of the twisting of the tractor body is that the constraint between ROPS and tractor body must rotate to measure the torque from the rotating displacement in Bullet. Moreover, this rotation of constraint changed the contact point of left rear wheel from the bottom side to no contact as shown in Fig. 7.7(b). For the case of Fig. 7.7(b), the results of maximum impact force approached 5,331 N and the results of steady force are all close to 2,940 N, which became closer to that of experiment.

7.4 Conclusion

The results of velocities and rotating angle portray that Bullet has sufficient accuracy to simulate the sideways overturning dynamics of a tractor. The measurement capability of Bullet constraint is competent to quantify the maximum impact force on ROPS during overturning. For future plan, the measurement capability of constraint should be improved by minimizing the rotating displacement of constraint.

References

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Chapter 8

Concluding Remarks

In this study, the overturning dynamics of an agricultural vehicle with the relationship between the geometry of ROPS and the continuous overturning were investigated.

In Chapter 2, the developed OECD-based numerical program was capable to predict the effective height and width of a safety frame, or tip-over protective structure (TOPS), for a three-wheeled agricultural carrier (Sun et al., 2017a). The comparison between experiment and simulation results showed that the developed program had qualitatively sufficient accuracy for predicting the TOPS height of a three-wheeled agricultural carrier. However, the developed program could not include precise collision detection and stick or sliding friction.

In Chapter 3, the physics engine (Bullet) could predict the three-dimensional sideways overturning behaviour for an agricultural carrier with tip-over protective structure (TOPS) (Sun et al., 2017b). The comparison of overturning behaviour between experiment and simulation for three-wheeled and four-wheeled agricultural carriers revealed that the collision detection of the physics engine could result in the same overturning process in experiment, which also meant the numerical data calculated by a physics engine could be referential results.

In Chapter 4, the overturning dynamics of agricultural model carrier using a physics engine resulted in the similar dynamics to that of the experiment. (Sun et al., 2018a). The overturning dynamic process has been investigated based on the numerical results of angular velocity, rotating angle, and linear velocity. The changes in the numerical data and the critical height of ROPS with the adjustment of moment of inertia, coefficient of friction and restitution, showed that the physics engine could provide a sufficient physical calculation for our vehicle and slope collision models.

In Chapter 5, the overturning dynamics of agricultural tractors with ROPS on a bank slope and on a uniform slope using Bullet physics engine was completed (Sun et al., 2018b). The results using Root Mean Square Error (RMSE) to measure the accuracy of simulated dynamic data for an overturning farm tractor compared to the previous experiment portrayed that the physics engine could predict the correct overturning dynamics of a tractor over a bank slope. Moreover, the physics engine could also predict a similar result of the critical height of ROPS compared to the OECD program.

In Chapter 6, the overturning experiments of a small-sized tractor with newly designed ROPS were completed with and without travelling speed (Xiao et al., 2018). The observed results of angular velocity and rotating angle during overturning when the tractor with travel velocity of 0.77 m s^{-1} coincide with the cases without travelling speed, which might be the reason that the observed maximum impact force and steady force of the case with travelling speed are close to those without travelling speed.

In Chapter 7, the calculated impact force acting on ROPS of an overturning tractor using a physics engine was calculated (Sun et al., 2018c). The RMSE of angular velocities and rotating angle showed

that the simulation results using the physics engine were in reasonably good agreement with experimental results. The simulation using the physics engine was capable of quantifying the maximum impact force on ROPS during overturning.

For the future development of the collision models of agricultural vehicle and slope using Bullet, the improvement of the rigid-rigid collision models may be recommended. The example in the library “**Soft Body**” → “**Collide**” which demonstrates a tyre conducted by FEM contacting with a rigid ground surface, provides a possibility that some parts of our collision models can be replaced by soft models, such as using soft tyre for tractor model or using soft surface for slope model. Therefore, we attempt to introduce a layer of soft model (FEM) lay on the rigid slope as shown in “**Soft Body**” → “**One**”.

SoftDemo.cpp is the collection of all the examples using FEM soft model. Our case where a rigid three-wheeled agricultural model contacts with a rigid slope with a layer of soft model on its surface can be found in **static void Init_One(SoftDemo* pdemo)**. The deformations of creating a soft body is listed as follows:

- **btSoftBody* psb=btSoftBodyHelpers::CreatePatch(pdemo->m_softBodyWorldInfo, Or₁,Or₂,Or₃,Or₄,Num_X,Num_Y,...;)** means creating a patch model **CreatePatch** named **psb**. **Or₁, Or₂, Or₃, Or₄** mean the coordinates of the fixed endpoints for the patch model, which works to hold the whole patch model. **Num_X and Num_Y** determine the number of FEM element along X and Y direction. There also have other commands in this sentence which we suggest keeping the default setting.
- **psb->getCollisionShape()->setMargin(yoso_width);** the thickness or the width of the FEM element need to be decided by **yoso_width** after the patch has been created. If the set value of the width of FEM element is lower than 0.12 m, the whole soft patch would be a breakdown.
- **btSoftBody::Material* pm=psb->appendMaterial();** format changes into **pm** for the created soft model **psb**.
- **pm->m_kLST= 0.51;** determines the linear stiffness coefficient from 0 to 1. This setting should be coordinated with the predicted impact energy. If the impact model carried a large amount of dynamic energy (high speed or large weight), an incorrect setting of **m_kLST** might also cause breakdown.
- **pm->m_kVST = 1;** determines the area or angular stiffness coefficient from 0 to 1, the change of coefficient did not cause obvious difference in output, the default value 1 is recommended.
- **pm->m_kAST = 1;** determines the volume stiffness coefficient from 0 to 1, the change of this coefficient also had no obvious difference in output, the default value 1 is recommended.
- **psb->generateBendingConstraints(2,pm);** determines the distance of connecting FEM element pairs, the default value 2 means every FEM element is connected another FEM element.

The setting may above enable the each FEM element to be linked by the two endpoints.

According to ZeroGram (2018), there are also some other solutions for the calculation of stress. The following commands can be replaced into the source code after **btSoftBody* psb=btSoftBodyHelpers::CreatePatch(...)**. One of them is that a method applied a reaction force when a stress is applied upon FEM element:

- **psb->setPose(false , true);** provides an original memory of the default setting of the soft model shape, which means a reaction force would be applied to the FEM elements to help them return to the originally defaulted shape.
- **psb->m_cfg.kMT = ???;** determines a coefficient of the reaction rate from 0 to 1. A suitable value adapted to the amount of impact energy can achieve a real bounce feeling and reduce the redundant vibration.

Another method may also be applied a calculation of volume when stress is generated upon an FEM element:

- **psb->setPose(true , false);** activates the calculation of the whole volume of the soft model. The setting of the relationship between stress and reset force, and the relationship between volume and reset force, can be adjust in the following items.
- **psb->m_cfg.kVC = ???;** determines a coefficient of volume conversation which can be a positive number.
- **psb->m_cfg.kPR = ???;** determines a coefficient of pressure which can be any value.

Those three methods above explained the different physics algorithm for holding the shape of the soft body. Moreover, a soft model can also adjust its collision characteristics, for example, the calculation of friction and restitution between the collision pair in our simulation need to be considered. Therefore, the commands **psb->setRestitution(0.35);** and **psb->setRestitution(0.5);** give access to the adjustable condition of the soft surface over our slope model.

Those are our attempt for the improvement of our agricultural and slope models. The final effect of the calculation of friction and restitution still needs to be verified. Because the stress between vehicle model and slope model during contact can press the soft model, where the deformation of soft model (FEM mesh) might have a reaction force not only against the vehicle model but also against the slope model. The reaction force at the normal and tangential direction occurred by the deformation of soft model (FEM mesh) was applied on the vehicle model, might not be a constant rate in every impact. Consequently, the best setting which balance a combination of the coefficient of restitution and friction of rigid slope model, soft surface model, and rigid vehicle model along with the coefficient of stiffness of soft surface model, should be investigated.

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Publication List

Publication of Research Papers

1. Sun, C. R., Nakashima, H., Tsukamoto, S., Minagawa, K., Takahashi, H., Shimizu, H., Miyasaka, J., and Ohdoi, K. 2017. Numerical analysis of sideways overturning of a three-wheeled agricultural carrier with safety frame on a slope. *Engineering in Agriculture, Environment and Food*, 10(4), 249-258.
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