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Modification of a Tractor Dynamic Model Considering the Rotatable Front End

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A mathematical model for tractor dynamics was expanded by considering the rotatable tractor front end. The fundamental shortcoming of the simplified model was revealed by the loss of contact of the tire with a rigid horizontal surface in an obstacle–passing case. Further shortcom–ings of the simplified model arise from aspects of the motion and vibration characteristics. The improved model provides a better and more realistic representation of the tire–ground contact condition and is applicable to tractors on lateral slopes. The independent roll motions of the two main tractor parts (the front end and main body) significantly reduce the motions of the tractor and thus increase its stability. Furthermore, the effects of the forward tractor speed and obstacle height were studied for a tractor on a 10° lateral slope. By analyzing the motion amplitude and tire–ground contact condition, the tractor speed and obstacle height parameters associated with danger and risk were evaluated. The results suggest the greater capability of the improved model to predict tractor dynamic response in Phase I overturn.

Key words: Tractor Overturn, Dynamic Response, Rotatable Front End

INTRODUCTION

While the use of tractors has become widely popularized with high–tech support in recent decades, overturn, also known as rollover, is still a concern because of its high mortality rate. To protect operators, remarkable efforts have been devoted to addressing the problem of tractor overturns in recent decades. Despite the importance of the introduction of the rollover protective structures (ROPS), known for their effectiveness in passive protection of drivers, and the ad–vantages from computer–aided analyzing tools (Zhu *et al.*, 2014), further research into vehicle overturning mechanisms is needed to give us a better understanding of tractor dynamics under static, quasi–static, and dynamic conditions (Ahmadi, 2011). The statics, kinematics, and kinet–ics of specific types of vehicle have been analyzed and predicted for specific cases (Guzzomi, 2012; Smith *et al.*, 1974). Among these, the analysis of kinetic parameters is considerably more complex, especially when dynamic conditions are considered, which is more realistic for a moving tractor.

Two–dimensional (2D) models with two degrees of freedom (DOF) are usually used to analyze tractor tipping behavior (Koch *et al.*, 1970; Homori *et al.*, 2003). To analyze a more gener–al case, a three–dimensional (3D) model with up to six DOF for the main body of the tractor is necessary to be studied. As noted by Abubakar *et al.* (2010), more than 75% of tractor overturns are to the side, indicating that tractors are more susceptible to overturning on uneven lateral slopes. Some of the models presented in previous studies are based on assumptions such as ne–glect of the spring–damper effect of pneumatic tires (Ahmadi, 2011, 2013), which may result in the tractor being represented as an oversimplified rigid body system. Similarly, it would be preferable not to model tire deformation as continuing after the tire loses contact with the ground (Pershing and Yoerger, 1969) because a probable misjudgment of the moment when the normal force becomes zero may arise. As a result, the onset of overturn may be estimated in–correctly. In investigations of tractor dynamics, especially vibration characteristics, tractor rota–tions are commonly restricted to be small enough to permit linearizing of the model (Homori *et al.*, 2003; Takeda *et al.*, 2010a, 2010b). Furthermore, pioneering studies of motion analysis for single front–wheeled tricycle tractors (Larson *et al.*, 1976) would have broader implications if the currently popular tractors with wide front ends were the focus of concern. As the above dis–cussion suggests, a modified tractor model with a wide front end that permits large rotations while considering tire deformation, tire–ground contact conditions can yield more precise re–sults. While such considerations have been involved in the models developed by Li *et al.* (2014a, 2014b), further extension should take into consideration the pivoting of the tractor front end.

To understand the limitations of the assumptions mentioned above in modelling a dynamic tractor system and the advantages that an improved model considering a rotatable front end could provide, this study was con-

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ducted to analyze tractor dynamics in response to a distur–bance caused by a road obstacle.

MATERIALS AND METHODS

To study dynamic tractor behaviors in a 3D situation, a half–sine bump was modelled under the right–side tires of the tractor. This bump is able to excite tractor pitch, roll, and vertical displacement with respect to the ground. As the rationality of the concept of an effective tire radius has been proven by Takeda *et al.* (2010b), the dimensional model of the tire–obstacle contact geometry is used in this study in modelling the obstacle–surmounting passage. Furthermore, for the purposes of an in–depth phased analysis, we partition the passing–over process into four periods that have been described in vibration–related studies on automobiles (Yamamoto and Shi–mada, 1957; Shimada, 1961a, 1961b, 1962). Periods 1 and 3 represent excited vibrations for a tractor, while the other two periods are damping–free periods. The dimensions and physical properties of the tractor considered are those described in the work of Takeda *et al.* (2010b). The values of other parameters not measured in their work were assumed.

This study considers the dynamic model presented by Li *et al.* (2014a, 2014b) as the base model and improves it by replacing the previous fixed tractor front end by a rotatable type. Considering that most tractors have a front end that can rotate (approximately 10°) about the pivot pin in the tractor's longitudinal direction, an improved model with one rotational DOF for the tractor front assembly should address the limitation of the simplified model. The tractor can then be modelled as two parts—the anterior part (the front end) and the posterior part (the main tractor body, i.e., the rest of the tractor apart from the front end). Moreover, given the preva–lence of sideways overturning incidents, as noted previously, it would be more meaningful to consider a lateral incline as the base surface.

As both of the two main tractor parts exhibit independent rolling motions, the moment when either uphill tire loses contact with the ground can be defined as the onset of Phase I overturn (Guzzomi, 2012). It should be noted again that the new model with a rotatable front end is intended for prediction of tractor dynamics when the independent swing angle does not meet its limit value. In addition, the predictions obtained are considered reliable before sideslip occurs.

A computerized implementation of the Runge–Kutta method in the Visual Basic Application language within Microsoft Excel is used to perform simulations. Noting that the road profile under the left–side tires are set to zero in our case, the simplified and modified models are compared for the assumed values of the bump height and tractor speed.

RESULTS AND DISCUSSION

Limitations of the previous model

Given an obstacle height of 0.08 m on a horizontal surface and a certain speed, the simulation results shown in Fig. 1 reveal the substantial shortcomings of the fixed–front–end tractor model. Fig. 1 (a) shows the variation in the normal force applied to each tire at a speed of 0.8 m s^{-1}. The left front tire leaves the ground for a long period during period 1. Likewise, both the right front tire and left rear tire are off the ground during period 3. As Fig. 1 (b) shows, as the travel speed increases, more intense fluctuations of the forces appear, along with one more tire losing contact with the ground during period 1. These changes in the responses can be attributed to the obsta–cle height and the tractor speed. A tire loses contact with the ground when the magnitude of its vertical displacement due to lifting by the obstacle equals the magnitude of the tire deformation due to compression. Furthermore, higher speeds reduce the stabilizing time that a tire that loses support needs and causes larger fluctuations in force with higher peaks.

As noted in the Introduction, irrationality may arise if the tire–ground contact condition is not considered in the modelling. Fig. 1 (c), which illustrates a free–contact model, suggests the difference compared to Fig. 1 (a). In spite of the relative smoothness apparent in Fig. 1 (c), the tendencies of the supporting forces for the two different conditions are obviously different. Be–cause the

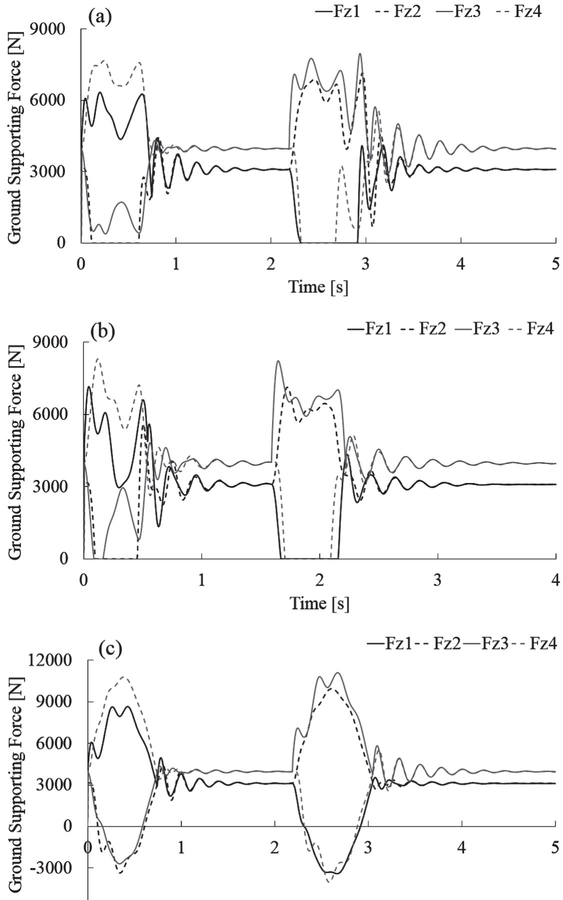

Fig. 1. Variations in the ground supporting forces of the tires: (a) at a tractor speed of 0.8 m s^{-1}; (b) at a tractor speed of 1.1 m s^{-1}; (c) at a tractor speed of 0.8 m s^{-1}, without considering the tire–ground contact condition.

normal force acting on one tire cannot be negative in practice, the corresponding trac–tor model derived on the basis of these forces is flawed in its applicability.

With the necessity of checking the tire–ground contact condition having been proved, it should be noted that in a typical situation, the fundamental limitation of the simplified model is the impossibility of all of the tires being coplanar in the ground plane during an obsta–cle–surmounting passage. However, if the obstacle is wide enough to cause pitching only, this model is still adequate to predict tractor behavior in a 2D case, as shown in the investigation of tractor vibration characteristics performed by Takeda *et al*. (2010a, 2010b).

Comparison of the two models

There is no doubt that in vehicle dynamics, ground reaction forces are the predominant influences on vehicle behavior. Given the limitations of the simplified model, the predictions obtained using the simplified model, illustrated in Fig. 1 (a), were presumed to be significantly different from those obtained using the improved model. Hence, for comparison, the lateral slope in the new model was replaced by a horizontal surface. The obstacle height and the tractor speed were set at 0.08 m and 0.8 m s^{-1}, respectively. The values that were not reported by Takeda *et al*. (2010b) were assumed or determined from geometric calculations in this study.

The vehicle attitudes obtained from the two models are shown in Fig. 2. On the whole, the motions of the COG predicted by the new tractor model are smaller than those predicted by the simplified model in some ways. Specifically, the maximum vertical displacements predicted by the simplified model for periods 1 and 3 are 36.9% and 35.6% larger, respectively, than those predicted by the new model. Note that the two independent tractor rolling motions considered in the new model result in improved tire–ground contact conditions. The maximum pitch angle in period 1 predicted by the new model is noticeably smaller than that predicted by the simplified model. Furthermore, because the posterior part of the tractor accounts for most of the tractor's mass and size, its influence on vehicle response is greater than

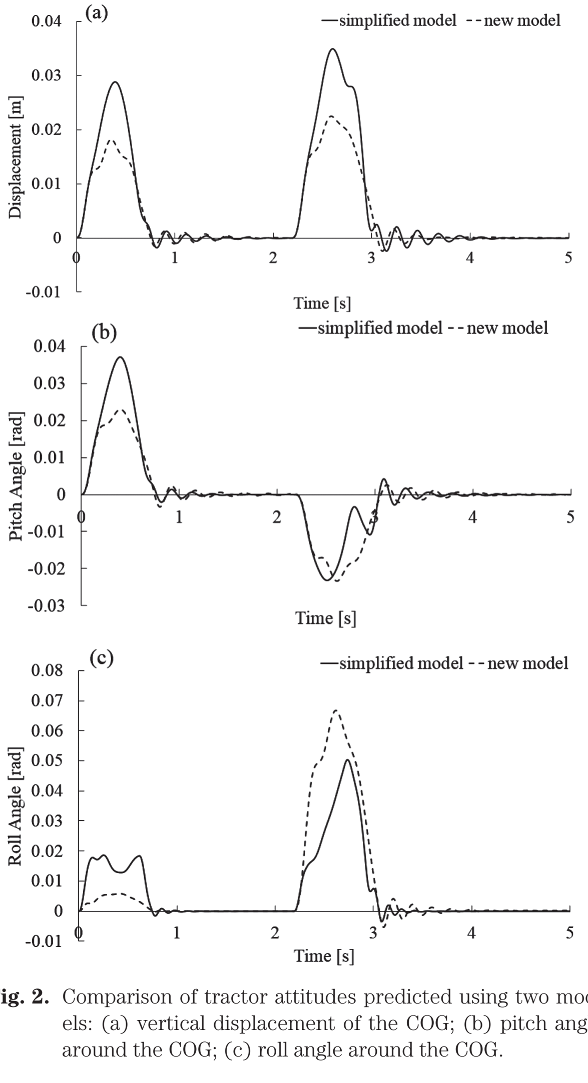

Fig. 2. Comparison of tractor attitudes predicted using two models: (a) vertical displacement of the COG; (b) pitch angle around the COG; (c) roll angle around the COG.

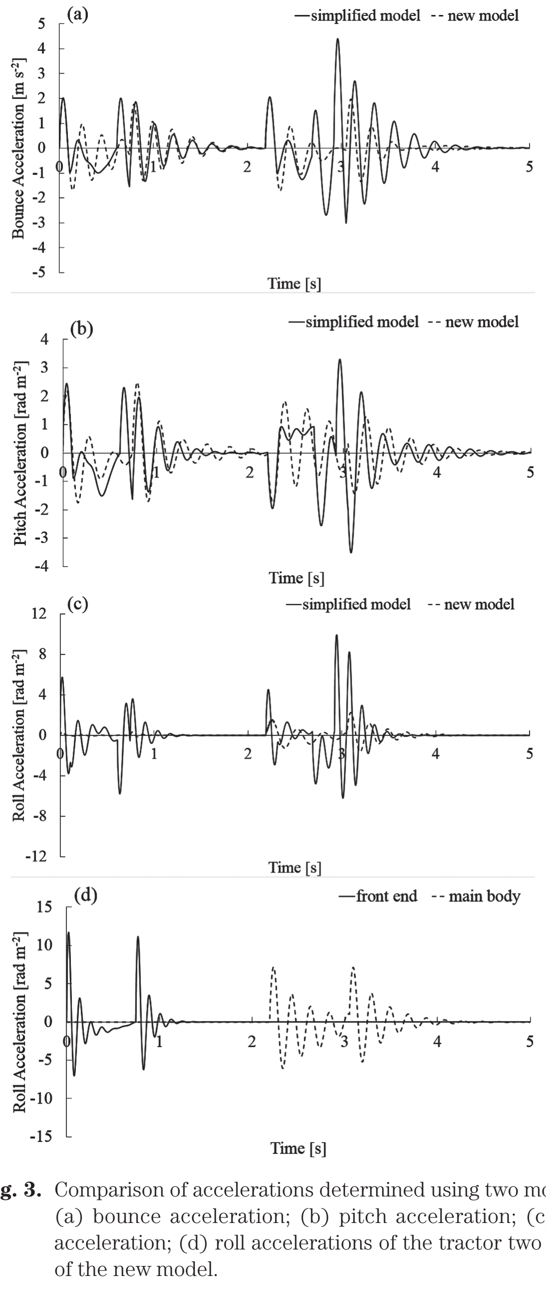

Fig. 3. Comparison of accelerations determined using two models: (a) bounce acceleration; (b) pitch acceleration; (c) roll acceleration; (d) roll accelerations of the tractor two parts of the new model.

that of the anterior part. There–fore, for period 1, the front end of the tractor has less influence on the vehicle roll angle than the main body of the tractor.

The tractor vibration characteristic for each motion is described by the corresponding acceleration, shown in Fig. 3. For tractor bounce and pitch, the acceleration amplitudes predicted by the two models remain almost the same during periods 1 and 2. In period 4, however, the acceleration amplitude is much smaller if the tractor front end is modelled as rotatable. Similarly, for tractor roll, a large disparity in the predicted values is observed. According to the new model, tractor roll acceleration can be nearly ignored for periods 1 and 2. Moreover, according to the new model, the maximum acceleration throughout periods 3 and 4 is approximately one fifth of that determined according to the old model. Such improvements owe to the independent rolling motions of the tractor two parts. In Fig. 3 (d), it is found that the roll accelerations of the front end and the main body of the tractor arise only in their respective obstacle–surmounting related periods. Considering the considerable difference in weight percentage of the tractor two parts, the rolling motion of the front end tends to be large and susceptible to obstacles but has fairly minor influence on the entire vehicle, while that of the tractor main body owns a reverse trend.

The values obtained for the normal forces, tractor attitudes, and accelerations provide insights into the advantages of the new tractor model over the simplified model. Note that the new model was developed to reflect the fact that most tractors are equipped with rotatable front ends. Therefore, a typical tractor used in practice is safer and more stable than is suggested by a theo–retical model based on the assumption of a fixed front end.

Prediction of the improved model on lateral slope

The forward speed and obstacle height were identified as the factors influencing tractor be–havior considering the improved model in this study. For the purposes of comparison, a tractor moving at a speed of 0.5 m s^{-1} on a 10° lateral slope over an obstacle with a height of 0.08 m was defined as the reference case.

The influence of speed on tractor response is shown in Fig. 4. As the speed increases, both the vertical and lateral displacements of the COG increase and exhibit more sharply fluctuating peaks, especially in period 3. Figs. 4 (b) and (c) show that the later displacement and roll angle of the COG follow the same trends. In addition, non–zero values appear after the damped free vibrations in periods 2 and 4. In fact, the gaps reflect the initial tractor attitudes due to the lateral slope. That is, the initial roll angle and lateral displacement for steady–state conditions are cal–culated. Although the magnitude of the pitch angle does not increase significantly with increas–ing speed, a potential danger is indicated by Fig. 4 (e). At a speed of 0.8 m s^{-1}, the uphill rear tire loses contact once in period 3, indicating an occurrence of onset of Phase I overturn, while the same tire loses contact four times in period 3 when the speed is 1.1 m s^{-1}. Therefore, 0.8 m s^{-1} is defined as a dangerous speed and 1.1 m s^{-1} as a considerably unstable and risky speed.

Fig. 5 shows the predicted tractor responses for obstacle heights of 0.08 m, 0.14 m, and 0.2 m. For all of the tractor motions, the amplitudes in periods 1 and 3 increase nearly in proportion to the increase in the obstacle height. Notably, drops near the peaks in period 3 occur in tractor lateral translation and roll motions for a 0.2 m obstacle height, implying instability of the vehi–cle. Furthermore, from Fig. 5 (e), we know that the uphill rear tire loses contact with the ground twice and thrice for obstacle heights of 0.14 m and 0.2 m, respectively.

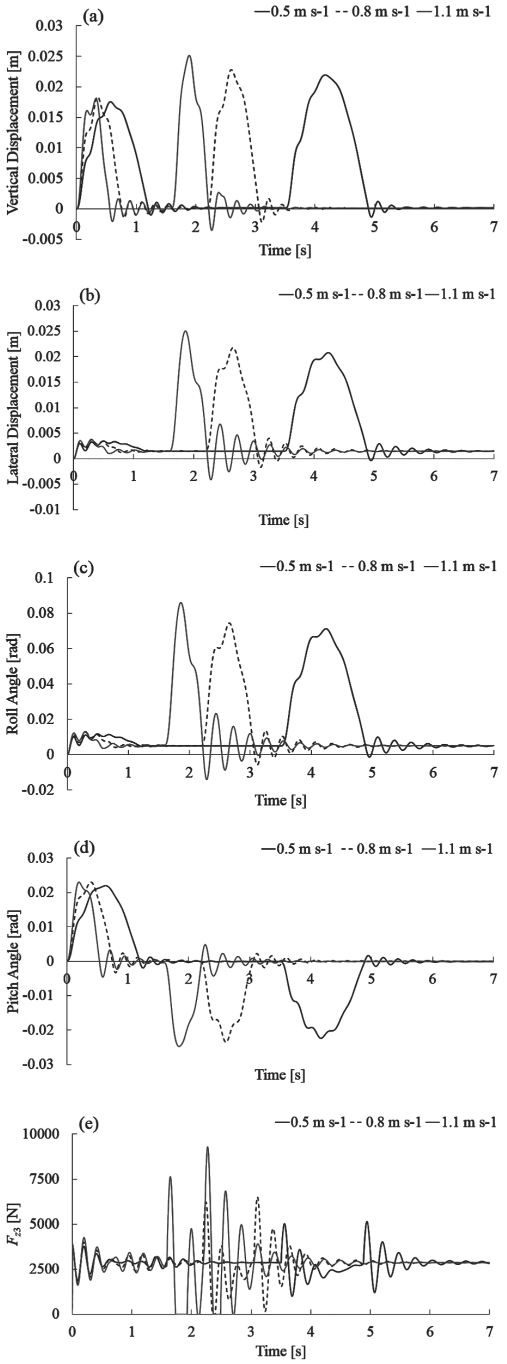

Fig. 4. Effect of speed on tractor response: (a) vertical displacement of the COG; (b) lateral displacement of the COG; (c) roll angle around the COG; (d) pitch angle around the COG; (e) ground supporting force on the uphill rear tire.

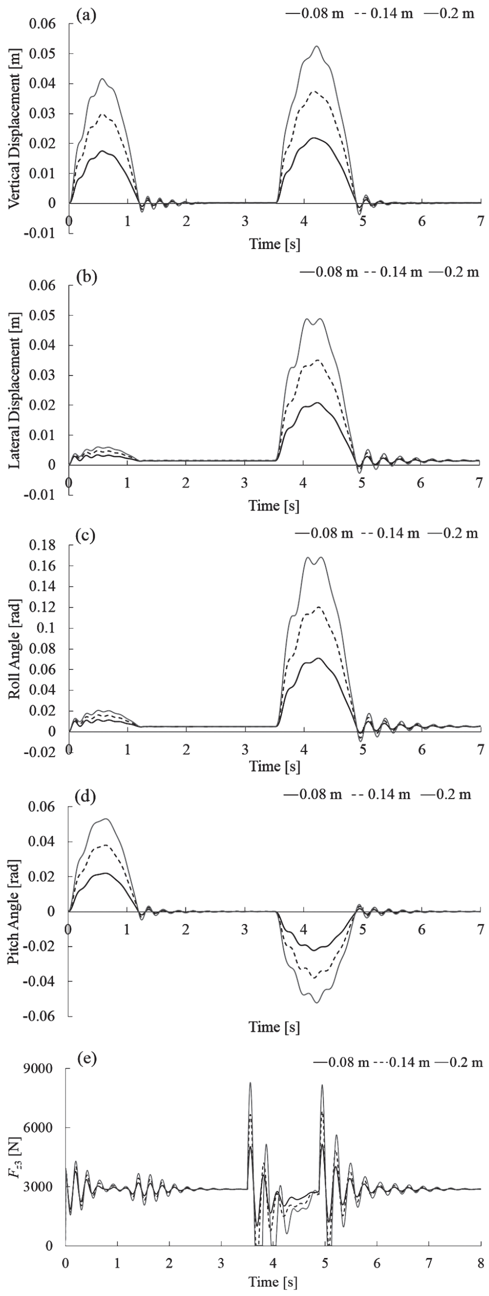

Fig. 5. Effect of obstacle height on tractor response: (a) vertical displacement of the COG; (b) lateral displacement of the COG; (c) roll angle around the COG; (d) pitch angle around the COG; (e) ground supporting force on the uphill rear tire.

Similarly, these two cases are judged as dangerous and risky, respectively.

In the cases discussed above, neither the front nor rear resultant friction force reaches the corresponding maximum static value, suggesting no occurrence of tractor sideslip.

CONCLUSIONS

An improved tractor model with a rotatable front end was considered replacing the fixed type in this study. Although the model of a tractor with a fixed front end takes into consideration the tire spring–damper effect and large rotations in a nonlinear manner, this simplified model has obvious shortcomings.

Compared with the improved model, the simplified fixed–front–end tractor model predicts larger motions, as well as more intense vibrations in period 4. In particular, the roll acceleration is predicted to be much lower if the tractor front end is modelled as rotatable, because of the two independent roll motions considered. It is also apparent that the new model provides better con–tact between the tractor tires and the ground and therefore predicts significantly safer operation.

Taking into consideration a relative roll angle between the two main tractor parts that is within the limited pivot angle, the new model was used to assess the effects of forward tractor speed and obstacle height on tractor response. A lateral slope of 10°, a speed of 0.5 m s^{-1}, and an obstacle height of 0.08 m were considered for the reference case, and variations in tractor response were observed with increasing forward speed and obstacle height. Basing on the increase in motion amplitude and the tire–ground contact conditions, a speed of 0.8 m s^{-1} and an obstacle height of 0.14 m were identified as dangerous conditions, while a speed of 1.1 m s^{-1} and an obstacle height of 0.2 m were identified as unstable and risky conditions. It should be noted that the uphill rear tire is more sensitive to obstacles and thus is more likely to lose contact with the ground.

The results of this study indicate that the new model provides a better way to analyze tractor motions and force variations. However, the new model is only applicable to tractor dynamics within Phase I overturn. Further efforts should be made to model full–stage response so that a quantitative lateral overturn criterion can be developed.

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