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Mostafa I. Yacoub, Mootaz E. Abo-Elnor, Ahmed M. Ali, Ibrahim A. Elsherif, Mohamed A. El-Latif, Alhossein Mostafa Sharaf

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Mostafa I. Yacoub*

Automotive Engineering Department,
Military Technical College,
P.O. Box 11766, Cairo, Egypt
Email: mostafa.yacoub@mtc.edu.eg

*Corresponding author

Mootaz E. Abo-Elnor

Mechanical Equipment Engineering Department,
Military Technical College,
P.O. Box 11766, Cairo, Egypt
Email: mootaz_aboelnor@mtc.edu.eg

Ahmed M. Ali

Automotive Engineering Department,
Military Technical College,
P.O. Box 11766, Cairo, Egypt
Email: ahmed.m.ali@mtc.edu.eg

Ibrahim A. Elsherif

Mechatronics Engineering Department,
Military Technical College,
P.O. Box 11766, Cairo, Egypt
Email: drielsherif@mtc.edu.eg

Mohamed A. El-Latif

Mechanical Design and Production Department,
Military Technical College,
P.O. Box 11766, Cairo, Egypt
Email: abdelmenim@mtc.edu.eg

Alhossein Mostafa Sharaf

Automotive Engineering Department,
Military Technical College,
P.O. Box 11766, Cairo, Egypt
Email: a.m.sharaf@mtc.edu.eg

Abstract: One of the strategies to protect vehicle platoons against improvised explosive devices (IEDs) is to use a mine-rollers mounted at the front of the vehicle with a telescopic lever and fork ended up with two sets of freely rotating solid tyres. However, rough vehicle handling or cornering instability conditions may be encountered particularly at high speeds. This paper presents a detailed investigation to ensure sufficient handling and stability of mine rollers-equipped vehicle. In the design phase both the kinematic and dynamic analyses are illustrated. The mathematical analysis of the dynamic stability problem is validated by in-field measurements of real mine rollers-equipped heavy vehicle. Results show that vehicle stability could be achieved using pre-manufacturing mathematical analysis illustrated in this paper. Furthermore, the analysis is extended to address the effect of some mine-rollers design parameters, like increasing setup length and weight which would certainly enhance the protection of the mobile platoon.

Keywords: mine-rollers; heavy vehicle stability; handling performance; conveying security; constant radius test; double-lane change test; tyre cornering stiffness; tyre sinkage; steering stress analysis.

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Biographical notes: Mostafa I. Yacoub received his PhD in Mechanical Engineering from University of Ottawa, Canada, in 2013. He received his BSc and MSc in Automotive Engineering from the Military Technical College, Cairo, Egypt in 1998 and 2005, respectively. He joined the Military Technical College in 2013 as a Lecturer. He joined University of Waterloo in 2021 as a Visiting Associate Professor. Currently he is affiliated as an Associate Professor in the Automotive Engineering Department in the Military Technical College. His research interests are Vehicle Dynamics and Control, Advanced Driver Assistance Systems (ADAS), Unmanned Ground Vehicles, Electric Vehicles, and Automotive Embedded Networks.

Mootaz E. Abo-Elnor received his BSc and the MSc in Mechanical Engineering from the Military Technical College, Cairo, Egypt, in 1991 and 1997, respectively and PhD from Strathclyde University, Glasgow UK, in 2002. In 2002, he joined the Department of Mechanical Equipment, Military Technical College, as a Lecturer, and in 2017 he became an Associate Professor. His current research interests include soil-tool interaction, heavy load mechanisms, biomechanics, predictive design and machine design. He is a Fellow of Egyptian Syndicate of Engineers and he is a consultant, material handling equipment/Egyptian Syndicate of Engineers since 2010 up till now.

Ahmed M. Ali received his BSc and MSc, both in Automotive Engineering at Military Technical College, Cairo, Egypt, in 2008 and 2015 respectively. He received his PhD (Dr.-Ing.) degree in Mechanical Engineering in 2019 at the University of Duisburg-Essen, Germany. He is now with the Automotive Engineering Department, Military Technical College, Egypt. His fields of interest include optimal design and control of hybrid and electric powertrains, intelligent transportation system, and fleets electrification.

Ibrahim A. Elsherif received his BSc from Military Technical College, Egypt in 1981, MSc from Cairo University, Egypt in 1990 and PhD from University of Alabama in Huntsville, USA in 1995. His former collage assignments include Head of Mechanical Equipment Department, Head of Mechatronic Department,

Chairman of Mechanical Engineering Branch, and Head of Scientific Council of Mechatronic Department. His main interests are mechatronic systems, power hydraulic, and construction equipment.

Mohamed A. El-Latif graduated from Military Technical College (MTC) in 1983 with a major in Mechanical Engineering. He holds a Master degree in Hydraulic Control from the MTC. He finished his PhD in Mechanical and Aerospace Engineering from the University of Alabama, USA. He assumed many positions during his career including consultant for the head of the Arab Authority of Industry (AAI), a member of the mechanical parts and engineering drawing standardisation committee at the standards and quality authority, a member in the Aviation Research Centre board of directors, a member in the general assembly of Shubra Engineering Industries Company.

Alhossein Mostafa Sharaf received his BSc in Mechanical Engineering from the Military Technical College (MTC) in Cairo in 1993 with grade excellent with honours. In 1998, he achieved his MSc in Heavy Vehicles Dynamics from MTC. In October 2007, he finished his PhD from Loughborough University, UK. From January to July, 2016, he awarded a Post-Doctoral Fellowship in the University of Ontario Institute of Technology, Canada. Currently, he is a Professor of Automotive Engineering in the Military Technical College. He has contributed to several typical research projects in the field of vehicle dynamics. Furthermore, he is concerned about AWD vehicle drivetrain issues comprising modelling, simulation and control.

1 Introduction

Mine-rollers equipped vehicles are widely used in convoying security by military engineers in the battle fields as shown in Figure 1. Although they provide proper protection for soldiers against improvised explosive devices (IEDs), their stability and reliability are questionable. Adding a heavy axle to the front of the vehicle will definitely affect its handling characteristics and roll stability. Also, the reliability of some subsystems' critical components, like steering system, needs investigation before adding such a setup.

In literature, one can find several research papers analysing the handling and roll stability of special purpose heavy-duty vehicles and articulated vehicles, but not for mine-rollers equipped vehicles. Despite that, there are several interesting research topics that contributed indirectly to the present work. Since mine-rollers equipped vehicles in consideration are heavy vehicles, yet feature a special extended articulation, the following analysis explores the state-of-the-art research from a point of view related to handling and roll of heavy vehicles, articulated vehicles and tyre-terrain interaction.

Kang et al. (2016) presented a sensitivity analysis of the suspension components of heavy-duty mining trucks (HDMT) aimed to optimise the design of the composite link (CL) and the candle (CA) independent suspension components of such vehicles to improve steady-state and transient handling performance. Pazooki et al. (2014) presented a three-dimensional model of an articulated frame steer vehicle (ASV) to investigate the ride and handling characteristics of such vehicles. The model was based on the kineto-dynamic properties of the frame steering mechanism while assuming a steady forward

velocity. After validation, the authors presented this model as a tool for investigating and tuning the suspension and steering design of such vehicles.

Figure 1 A typical vehicle equipped with a mine-rollers system (see online version for colours)



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Fancher and Winkler (2007), emphasised directional performance issues for articulated heavy vehicles design. In their review, the authors treated the aspects of evaluation and design of articulated heavy vehicles' critical components that affect the directional performance such as articulation joints, multiple axle suspensions, hitches, trailing units and combination of towing units with trailing units. They proposed an example for future development by placing the instantaneous path curvature at the mid-wheelbase of an articulated heavy vehicle using sliding mode control by manipulating the angles of the rear self-steered axles to achieve a predefined yaw acceleration trajectory. Sorge (2020) presented a nonlinear dynamic model for articulated vehicles. The aim of their study was to represent the large variations of side-slip angles against the large steering angles of the vehicle to help assess the off-tracking behaviour. Although the study was targeting the tractor-semi-trailer combination, the presented kinematic analysis correlating the side-slip angles and the steering angles was an adequate start for the present work. Another nonlinear model for an articulated vehicle with a hydraulic steering system was presented by Lei et al. (2021). The authors presented a theoretical stability analysis using the proposed model. Hamersma and Els (2020), developed a suspension force estimator using Kalman Filter observer for the road profile and suspension displacement. Gong and Yan (2021) proposed and evaluated a heavy vehicle active suspension control strategy using adaptive robust H_∞ controller. In order to address the vehicle roll stability, the rollover threshold should be identified. Although static rollover metrics were used (Chrstos and Guenther, 1992), throughout the long history of rollover research, several methodologies were proposed to address the dynamic rollover threshold including; field testing (Sharaf et al., 2019), lab testing (Sherif et al., 2010) and mathematical modelling (Ikhsan et al., 2021). Since the actual rollover threshold depends on the vehicle and the manoeuvre, it is hard to constitute a single straight forward method to estimate that threshold. Instead, a rollover threshold of 0.4 g lateral acceleration was commonly used

by approval organisations before which the tested vehicle of any size should not overturn (Economic Commission for Europe of the United Nations – United Nations Economic and Social Council, 2001).

For tyre-ground interaction, Romano et al. (2020) presented the unsteady-state brush theory of the tyre modelling. The authors analysed the contact patch adhesion by deriving an analytical solution to the bristle deformation by means of method of characteristics. They theoretically proved that the solution is valid for any shape of contact patches including time-varying ones without symmetries. El-Sayegh et al. (2020) presented and experimentally validated a modelling technique for off-road tyre interaction with gravelly soil. The model was developed using the finite element analysis (FEA) while the soil was modelled using the smoothed-particle hydrodynamics (SPH). The authors presented this model as a tool to estimate the rolling resistance and vertical force in static and dynamic tyre-ground interactions.

In light of presented overview, it can be perceived that the assessment of handling and stability in mine-rollers-articulated vehicles using experimental validation is still an open challenge that requires a particular attention. This paper presents a novel experimental approach to ensure handling stability and manoeuvring boundaries of mine-rollers-articulated vehicles without exceeding allowable stresses on steering mechanisms of the vehicle.

This paper is organised as follows: in Section 2 the mine-rollers setup under investigation is presented. In Section 3 the vehicle static loading is illustrated. Sections 4 and 5 explain the proposed theoretical kinematic and dynamic analyses of integrating the mine-rollers setup to the vehicle. In the latter sections, the theoretical results were obtained and analysed considering the quasi-steady dynamic load transfer and the mine rollers' tyres-terrain interactions. In Section 6, a theoretical investigation of increasing the mine-rollers system mass and length is presented. In Section 7, a theoretical stress analysis of the steering system critical components is presented. The experimental validation is presented in Section 8 including the double lane change (DLC), emergency brake, constant radius tests and the experimental stress analysis of the critical components. Finally, section nine concludes the paper.

2 Mine-rollers system description

The proposed mine-rollers system under investigation consists of a main longitudinal beam, a lateral front beam with two vertical pivots and two mine-rollers sets, Figure 2. The parametric main dimensions of the system are shown in Figure 3. The vehicle that was used in the present work was a *Mine-Resistant Ambush Protected* (MRAP) vehicle, see Figure 1. The fixation location of the main beam in the vehicle chassis was chosen to be in the front towing holes, Figure 4, as this is the strongest location to bear the stresses caused by mine-rollers system operation. The main longitudinal beam was initially proposed to be one rigid structure that was fixed to the front towing holes in a way that it was allowed to swivel vertically around a lateral hinge axis, directly placed in front of the front bumper of the vehicle. The front lateral beam of the set-up was proposed to be fixed to the longitudinal beam using a hinge that allowed it to swivel around a longitudinal axis. Each of the two mine-rollers sets was proposed to yaw about a vertical hinge mounted on the lateral front beam and were separated by a distance of 1.9 m.

Figure 2 Mine-rollers system main components

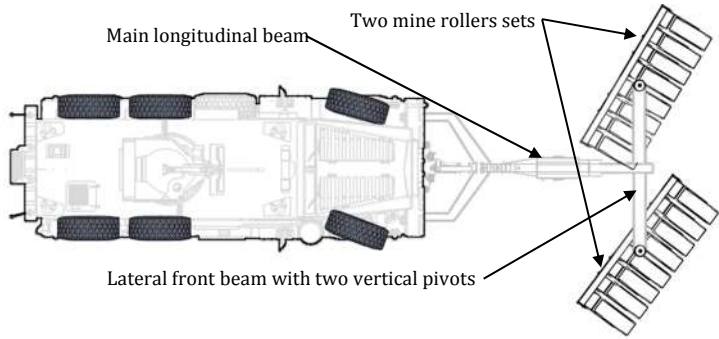


Figure 3 Main dimensions of the mine-rollers system; $l_b = 3.45$ m, $h_m = 1.2$ m and $b_m = 1.9$ m:
(a) side view and (b) top view

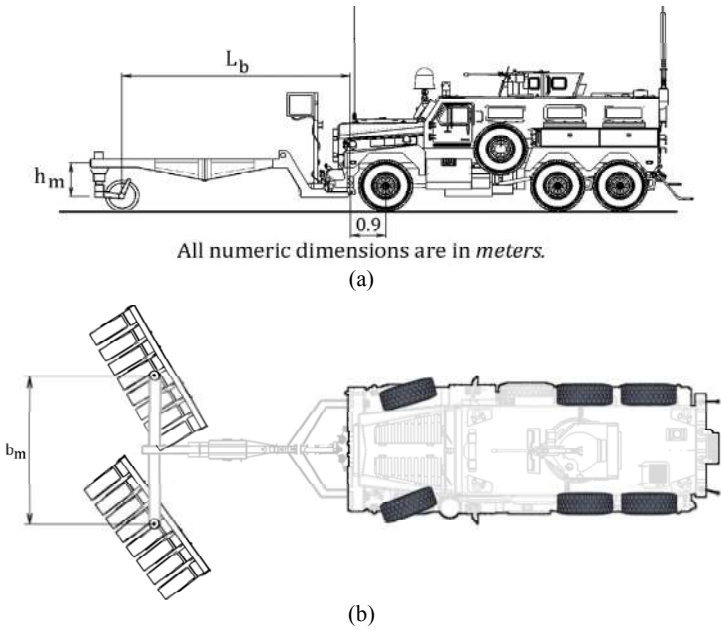
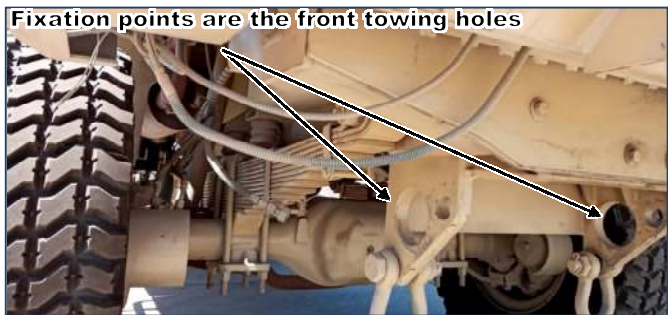


Figure 4 Fixation points of the main beam of the mine-rollers system (see online version for colours)



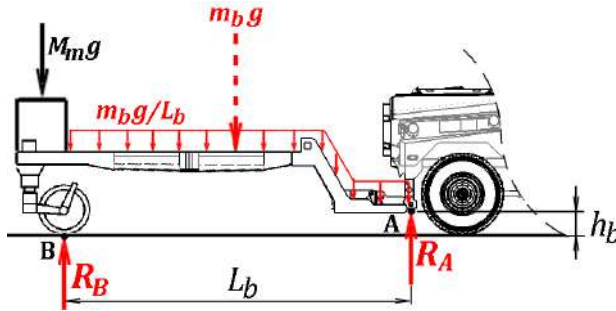
3 Checking of the static loading

In this section, the static loads at the chassis fixation point of the mine-rollers system were checked. In addition, the combined centre of gravity of the vehicle with mine-rollers system was determined.

3.1 Static load on vehicle front towing holes

It was essential to make sure that the vertical static load on the vehicle front towing holes, caused by the mine-rollers assembly, won't cause the vehicle axles loading capacity to be exceeded. To calculate the static load on the front towing holes (R_A), the mine-rollers system was modelled as two main parts; the longitudinal beam and the mine-rollers sets with their load masses. The longitudinal beam was modelled as a rigid homogeneous beam, where its mass (m_b) is distributed equally along its length (L_b) as shown in Figure 5. Also, the total mass of the mine-rollers sets, including their load masses, is denoted by (M_m). Since the system is hinged at the front towing holes and rolled at the mine-rollers sets, the beam was considered a simple supported beam.

Figure 5 Static loading on the vehicle chassis caused by the mine-rollers system (see online version for colours)



The static load on the front towing holes (R_A) and mine-rollers wheels (R_B) are calculated as follows:

$$R_A = \frac{m_b}{2}, \quad R_B = \left(M_m + \frac{m_b}{2} \right) g \quad (1)$$

As shown in equation (1), the beam length won't affect the static loads at the ends of the beam, instead they are affected by the values of the lumped masses of the system parts; beam (m_b) and mine-rollers sets (M_m).

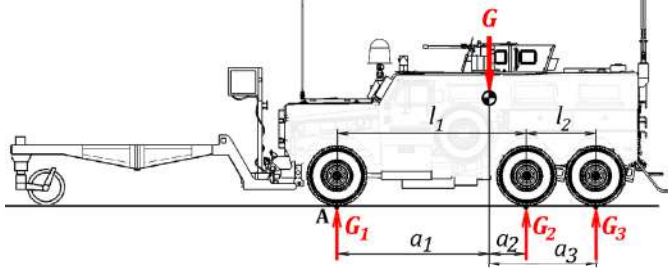
3.2 Location of the combined centre of gravity (C.G)

To find the location of the combined center of gravity of the vehicle with mine-rollers system, the static loads, m_1 , m_2 and m_3 , on each of the individual vehicle axles were used. By finding the sum of pitching moments (M_A) about the front axle (point A in Figure 6) caused by vehicle static loads, following equations determine the longitudinal distances of each of the axles from the CG:

$$a_1 = \frac{m_2 l_1 + m_3 (l_1 + l_2)}{M}, a_2 = l_1 - a_1, a_3 = l_2 + a_2 \quad (2)$$

where M is Gross Vehicle Mass (GVM), l_1 is the distance between axle no. 1 and axle no. 2, l_2 is the distance between axle no. 2 and axle no. 3, a_1, a_2, a_3, a_4 is the distances between vehicle CG and the front middle and rear axles respectively.

Figure 6 Vehicle GVW (G) and the reactions under its axles (G_1, G_2 and G_3) (see online version for colours)



4 Kinematic analysis

The objective of this section is to check the altered kinematic features of the vehicle when equipped with the proposed design of the mine-rollers system. The analysis revises the turning radii of the mine-rollers governed by the original vehicle turning radii.

4.1 Vehicle turning radii

As each of the two mine-rollers sets are allowed to swivel (yaw) freely around its own vertical pivot, its behaviour acts like a castor wheel. The steering (yawing) angles of both mine-rollers sets are constituted by the turning curvature achieved by the original steering system. During cornering about a centre of rotation (O), the input steering angle (δ_f) governs the vehicle CG turning radius (R), rear tandem axle centre turning radius (R_1), inner and outer mine-rollers sets' steering angles (δ_{m_i} and δ_{m_o}) and the mine-rollers sets' turning radii (R_{m_i} and R_{m_o}) as illustrated in Figure 7.

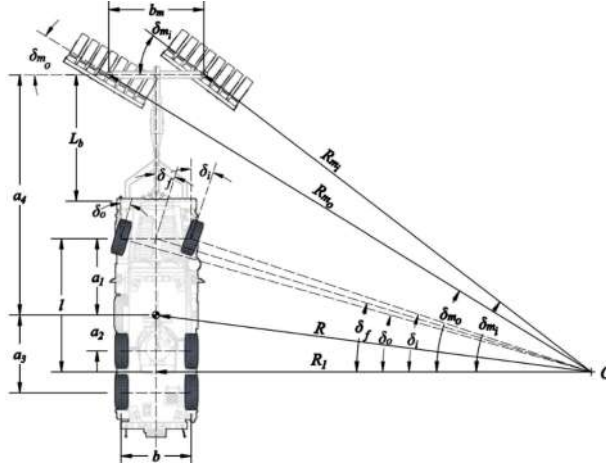
The longitudinal distance from front lateral beam of the mine-rollers and vehicle CG is denoted by a_4 while the vehicle wheel track is denoted by b . For a given six-wheeled vehicle, the turning radius of the rear tandem axle (R_1) was determined from the following equation:

$$R_1 = l \cdot \cot(\delta_f) \quad (3)$$

where δ_f is the steering angle of front axle, R_1 is the turning radius of the combined (tandem) rear axles about point O . l is the vehicle wheelbase which equals $l_1 + l_2/2$. The vehicle turning radius (R) governed by the input steering angle was calculated using the following formula (Jazar, 2017):

$$R = \sqrt{\left(a_2 + \frac{l_2}{2}\right)^2 + R_1^2} \quad (4)$$

Figure 7 Turning radii and steering angles of the vehicle equipped with mine-rollers system



4.2 Mine-rollers steering angles and turning radii

As mentioned earlier, the steering angles of the mine-rollers sets were determined by the vehicle steering angle as they act like castor wheels. Using trigonometry basics, the steering angles ($\delta_{m_i}, \delta_{m_o}$) and turning radii (R_{m_i}, R_{m_o}) of both inner and outer mine-rollers sets were calculated using the following formulae:

$$\delta_{m_i} = \tan^{-1} \frac{a_3 + a_4 - \frac{l_2}{2}}{R_1 - \frac{b_m}{2}}, \quad \delta_{m_o} = \tan^{-1} \frac{a_3 + a_4 - \frac{l_2}{2}}{R_1 + \frac{b_m}{2}} \quad (5)$$

$$R_{m_i} = \sqrt{\left(a_3 + a_4 - \frac{l_2}{2}\right)^2 + \left(R_1 - \frac{b_m}{2}\right)^2}, \quad R_{m_o} = \sqrt{\left(a_3 + a_4 - \frac{l_2}{2}\right)^2 + \left(R_1 + \frac{b_m}{2}\right)^2} \quad (6)$$

Figure 8 shows the change of the turning radii of the rear tandem axle, vehicle CG, inner mine-rollers set and outer mine-rollers set according to the change of the wheel steering input during steady-state cornering. It should be noted that, this figure only shows the range of input steering angle from 3° to 18.3° . On the other hand, Figure 9 shows the relation between the mine-rollers sets steering angles and the vehicle steering angle input during steady state cornering.

As could be found from the above analysis, the inner mine-rollers steering angle reaches 35.7° while the outer mine-rollers steering angle won't exceed 32° . These values could be used to place swivel limiters for the vertical pivots of the front lateral beam. Therefore, this would protect the mechanism from swivelling-over to the supplementary angular positions of the mine-rollers sets. Also, from the kinematic analysis, it was found

that the clearance between the mine-rollers sets must accommodate for the swivel (steering) of both sets within the mentioned angles ranges without collision.

Figure 8 Vehicle turning radii and mine-rollers sets turning radii during steady-state cornering (see online version for colours)

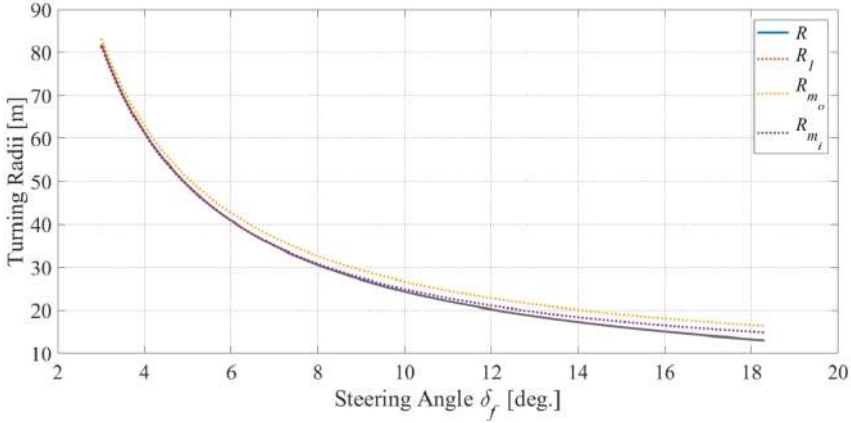
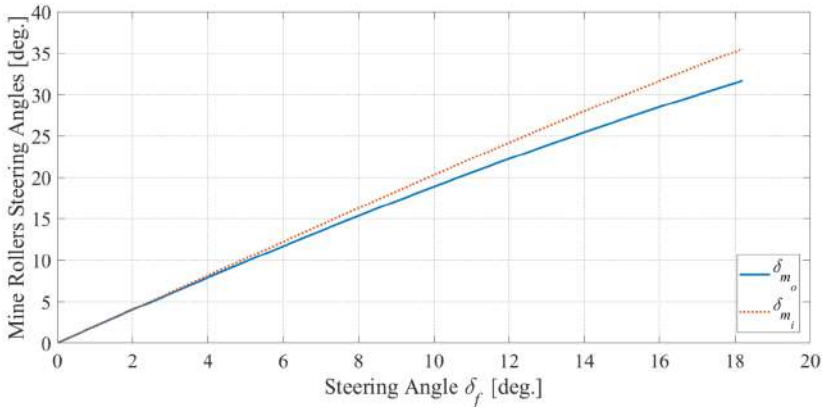


Figure 9 Mine-rollers sets steering angles during steady-state cornering (see online version for colours)



5 Dynamics analysis

To evaluate the performance and strength of the proposed design during steady-state cornering, the forces acting on the mine-rollers wheels need to be primarily determined.

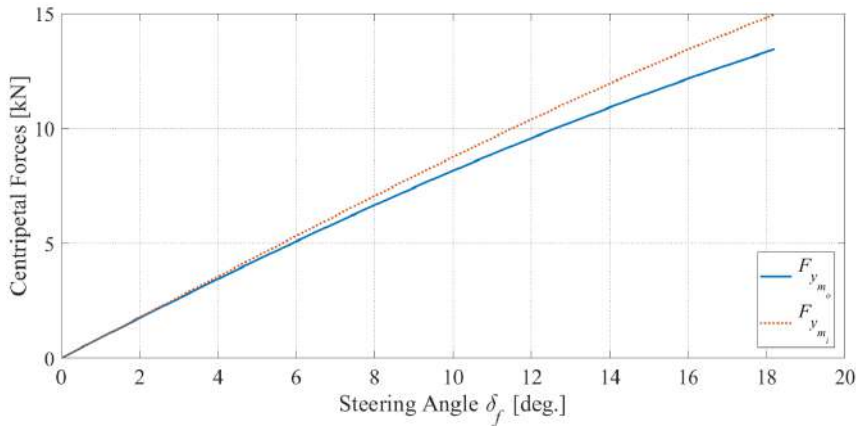
5.1 Steady-state cornering without load transfer

Considering the calculated turning radii of the mine-rollers sets, the centripetal forces of each of the mine-rollers sets can be calculated as follows:

$$F_{y_{m_i}} = \frac{M_m}{2} \frac{V^2}{R_{m_i}}, F_{y_{m_o}} = \frac{M_m}{2} \frac{V^2}{R_{m_o}} \quad (7)$$

where ($F_{y_{m_i}}$) is the centripetal forces at the inner mine-rollers assembly ($F_{y_{m_o}}$) is the centripetal forces at the outer mine-rollers assembly (M_m) is the total mass of the front mine-rollers assembly (V) is the vehicle forward speed. Figure 10 shows the relation between both centripetal forces at the mine-rollers sets and the steering angle input during steady-state cornering. The forward speed was fixed constant at 40 km/h as this is the expected maximum operating speed. As shown in this figure, the maximum centripetal force (10 kN) was reached at the inner mine-rollers set at the maximum steering angle. It's worth mentioning that the inner centripetal force won't reach that value due to the load transfer to the outer set during cornering. On the other hand, the outer set is subject to a higher centripetal force than the value shown (9 kN). In the worst-case of load transfer scenarios, the centripetal force on the outer set won't exceed 18 kN. This value is significantly less than the lateral force at the combined tandem rear axles (138 kN).

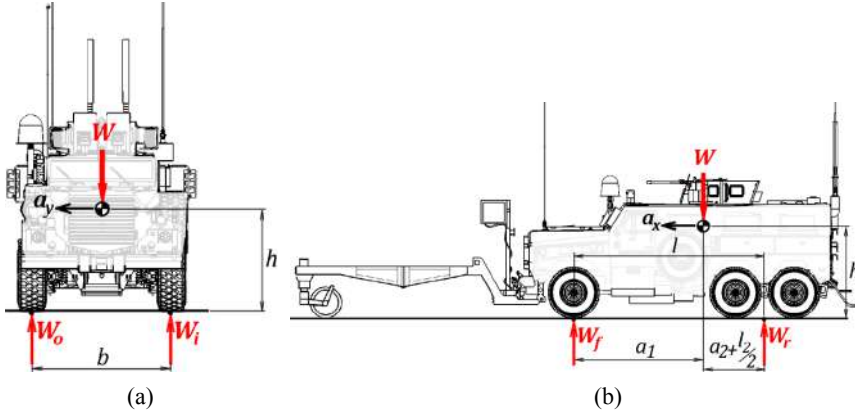
Figure 10 The centripetal forces of the mine-rollers during steady-state cornering (see online version for colours)



5.2 Steady-state cornering considering load transfer

The load transfer due to steady-state cornering at different turning radii was calculated. The analysis was based on the maximum operating speed (40 km/h) in off-road conditions and a changing turning radius down to its minimum value (13 m). Both lateral and longitudinal load transfers were considered. Figure 11 shows the free body diagram of the vehicle in front and side view of the vehicle. Figure 11(a) shows the forces exerted on the vehicle including lateral load transfer caused by lateral acceleration. Figure 11(b) shows the forces exerted on the vehicle including longitudinal load transfer due to longitudinal acceleration.

Figure 11 Load transfer on wheels during cornering caused by lateral acceleration: (a) front view: inner and outer wheels load transfer and (b) side view: front and rear load transfer (see online version for colours)



Since the mine-rollers system is hinged at the vehicle front towing holes and the front beam of the mine-rollers sets is free to roll independently, there is no moment and hence vertical load transfer between the mine-rollers system and the vehicle chassis. Therefore, the mine-rollers system was not considered in calculating the vertical forces on the vehicle wheels. The following equations represent the load on each of the wheels of the vehicle assuming the rear tandem axles were considered as a combined single axle.

$$W_{r_o} = \frac{W}{4} \left(\frac{a_1}{l} + \frac{a_x}{g} \frac{h}{l} + \frac{1}{2} + \frac{a_y}{g} \frac{h}{b} \right), \quad W_{f_o} = \frac{W}{4} \left(\frac{a_2}{l} + \frac{l_2}{2l} - \frac{a_x}{g} \frac{h}{l} + \frac{1}{2} + \frac{a_y}{g} \frac{h}{b} \right) \quad (8)$$

$$W_{r_i} = \frac{W}{4} \left(\frac{a_1}{l} + \frac{a_x}{g} \frac{h}{l} + \frac{1}{2} - \frac{a_y}{g} \frac{h}{b} \right), \quad W_{f_i} = \frac{W}{4} \left(\frac{a_1}{l} + \frac{a_x}{g} \frac{h}{l} + \frac{1}{2} - \frac{a_y}{g} \frac{h}{b} \right) \quad (9)$$

where (W_{r_o}) is the load on outer rear tandem axle tyres (W_{f_o}) is the load on outer front axle tyre (W_{r_i}) is the load on inner rear tandem axle tyres (W_{f_i}) is the load on inner front axle tyre (h) is the vehicle CG height (a_x , a_y) are the vehicle longitudinal and lateral acceleration respectively.

It should be noted that, the lateral acceleration is the component of the centripetal acceleration (a_c) in the direction of the vehicle lateral axis. The longitudinal acceleration is the sum of the longitudinal component of the centripetal acceleration and the longitudinal inertia acceleration caused by the driving/braking of the vehicle (a_d). Figure 12 shows the horizontal kinematics of the vehicle during cornering. Accordingly, the longitudinal and lateral accelerations were calculated as follows:

$$a_x = \frac{WV^2}{gR^2} \left(a_2 + \frac{l_2}{2} \right) + a_d, \quad a_y = \frac{WV^2}{gR^2} (R_l) \quad (10)$$

Figure 13 shows the change of the vehicle longitudinal, lateral and horizontal accelerations with the steering angle input. The horizontal acceleration (a_h) is the vector sum of the longitudinal and lateral accelerations. Figure 14 shows the change of vertical load on each tyre with the steering angle input. As shown in the figure, the maximum

vertical load occurred at the combined outer rear tandem axle tyres (95.5 kN). The obtained vertical loads were used to calculate the side slip angles of the front axle and the combined rear tandem axles.

$$\alpha_r = \frac{(W_{r_o} + W_{r_i})V^2}{4gC_r R}, \quad \alpha_f = \frac{(W_{f_o} + W_{f_i})V^2}{2gC_f R} \quad (11)$$

where (α_r, α_f) are the slip angle of the combined rear tandem axle and the front axle respectively. (C_r, C_f) are the cornering stiffness of each of the rear and front tyres, respectively. The cornering stiffnesses of rear and front tyres are equal as they are all identical.

The cornering characteristics of this specific tyre were modelled and validated earlier by Babulal et al. (2015). Figure 15 shows the reproduced cornering characteristics of the tyre according to the experimentally validated Pacejka model by the authors. Figure 16 shows the depicted cornering stiffness of the tyre according to the slip angle and the tyre normal force. This tyre model was used to predict the quasi-steady cornering performance of the vehicle. Figure 17 shows the change in side slip angles of front and rear axles with the change of steering angle input.

Figure 12 Horizontal kinematics of the vehicle during cornering (see online version for colours)

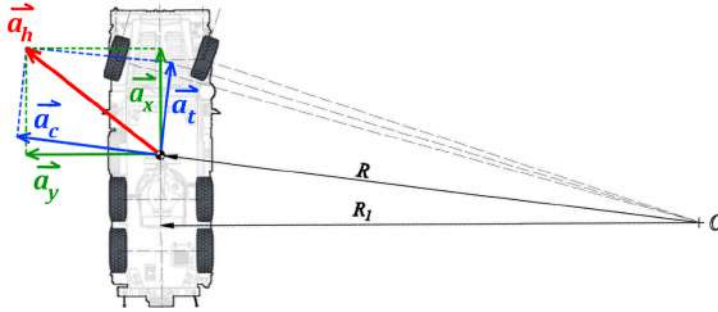


Figure 13 Vehicle accelerations and steering input during steady-state cornering (see online version for colours)

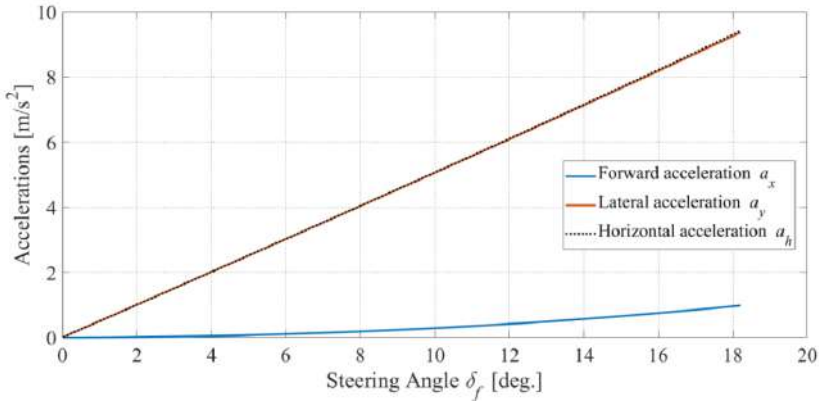


Figure 14 Tyres vertical loads and the steering input during steady-state cornering (see online version for colours)

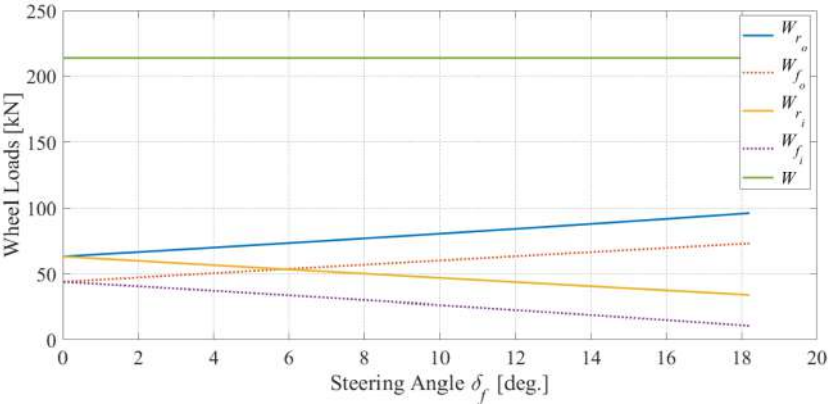
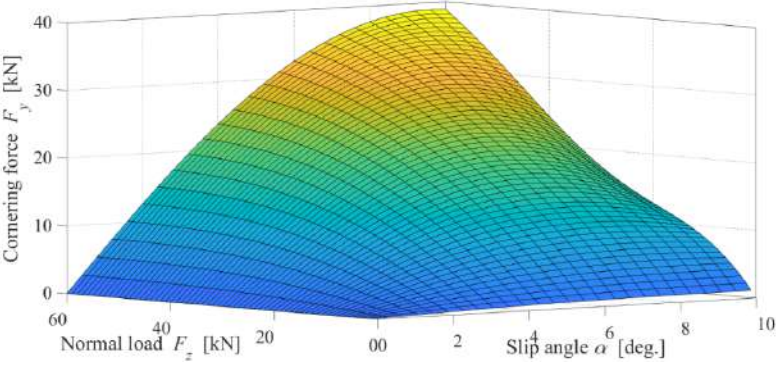


Figure 15 Cornering characteristics of the vehicle tyre (see online version for colours)



Source: Babulal et al. (2015)

Figure 16 Cornering stiffness of the vehicle tyre as depicted from the tyre characteristics surface (see online version for colours)

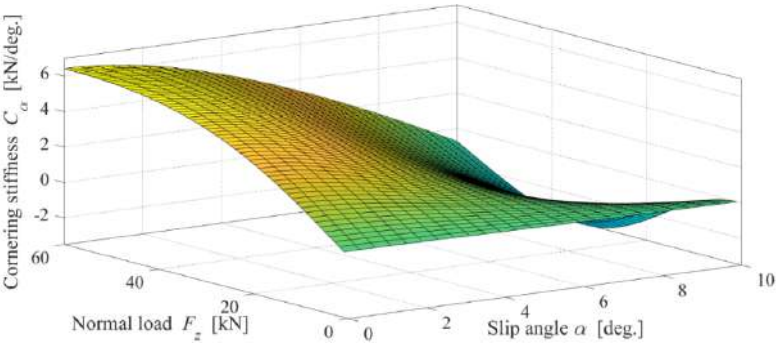
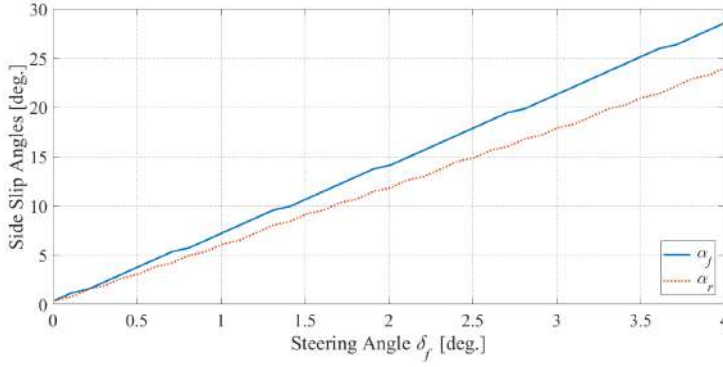


Figure 17 The front/rear axles slip angles during steady-state cornering (see online version for colours)

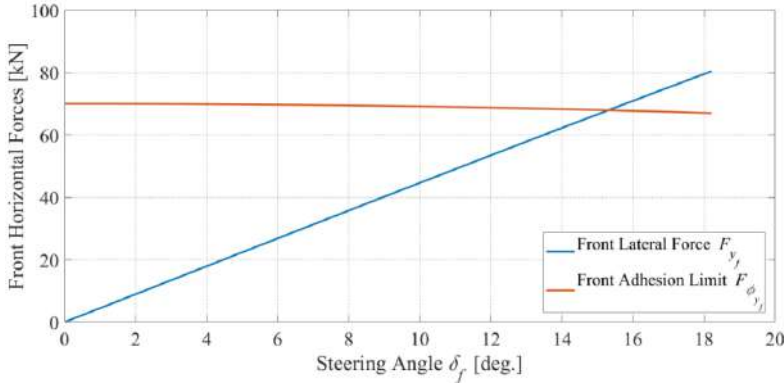


The lateral force (F_{y_f}) and adhesion force (F_{ϕ_f}) at the front axle tyres are calculated as follows:

$$F_{y_f} = \frac{(W_{f_o} + W_{f_i})V^2}{gR}, \quad F_{\phi_f} = (W_{f_o} + W_{f_i}) \cdot \mu \quad (12)$$

where (μ) is the adhesion coefficient between tyre and ground. Figure 18 shows the relation between the front axle cornering force and the input steering angle. The graph also shows the variation of the adhesion limit with the steering angle. The adhesion coefficient between tyre and road was assumed to be 0.8. As shown in figure, the maximum achievable lateral force on the front axle was 67 kN at steering angle 15.5 deg. Beyond this angle, the lower value of the adhesion limit will allow the front axle to laterally skid during cornering. Accordingly, the front axle lateral force value (67 kN) was used in the stress analysis of the steering system as will discussed later.

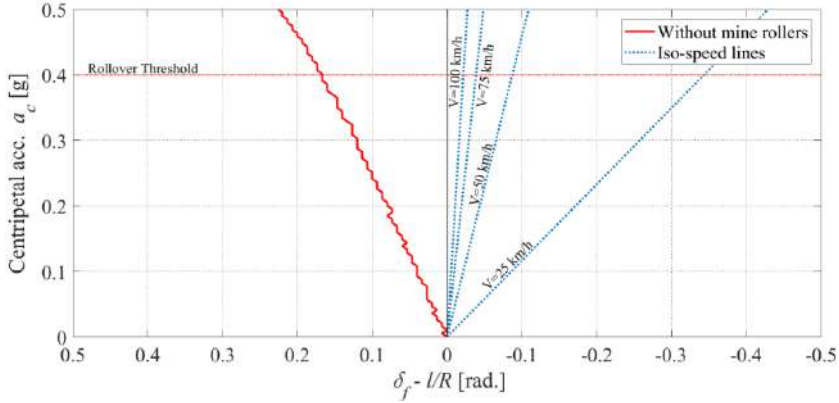
Figure 18 The front axle lateral force/adhesion limit during steady-state cornering (see online version for colours)



The theoretically obtained handling diagram of the vehicle without the mine-rollers setup attached is shown in Figure 19 where the vehicle did not undergo oversteer within the entire range of the steady-state cornering. This means that the vehicle is safe from

suffering oversteer up to and beyond the rollover threshold (0.4 g). Since each of the mine-rollers sets is free to pivot (castor) about its own vertical axis, their wheels won't significantly suffer from side slip. Therefore, the vehicle handling characteristics would not be affected by the mine-rollers wheels side-slip. The vehicle understeer coefficient (K_{us}) was further affected by the added lateral forces due to mine-rollers centripetal forces along with the soil compaction and rolling resistances as follows.

Figure 19 Handling diagram of the vehicle without mine-rollers (see online version for colours)



5.3 Steady-state cornering considering mine rollers-terrain interactions

For locomotion on unprepared, off-road terrain the main mechanisms of energy losses are the wheel's compaction, bulldozing and dragging of soil. On slopes, resistance due to the gravitational component parallel to a slope is an additional impediment to forward motion. The summary of such resistive forces is known as the external motion resistance. Consider the case of rigid wheel which is freely rolling without driving torque (driven-wheel) over a deformable terrain as shown in Figure 20. The formulas of predicting both the motion (compaction) resistance to the wheel (R_c) and the tyre maximum sinkage (z_o) have been predicted by Bekker and given as follows (Wong, 2001):

$$z_o = \left[\frac{3W}{b \cdot (3-n) \cdot (K_C/b + K_\phi) \cdot \sqrt{D}} \right]^{\left(\frac{2}{(2n+1)} \right)} = \left[\frac{3W}{b \cdot (3-n) \cdot K \cdot \sqrt{D}} \right]^{\left(\frac{2}{(2n+1)} \right)} \quad (13)$$

$$R_c = b \cdot \left(\frac{K_C}{b} + K_\phi \right) \cdot \frac{z_o^{n+1}}{n+1} = b \cdot K \cdot \frac{z_o^{n+1}}{n+1} \quad (14)$$

where (W) is the tyre vertical load, (K) is the modulus of plastic soil deformation, (K_C, K_ϕ) are the cohesive and frictional moduli of soil deformation respectively, (n) is the soil sinkage exponent.

Figure 21 shows the relation between the tyre sinkage and the tyre applied weight for different types of off-road terrains. Figure 22 shows the relation between the compaction resistance force and the applied weight for different types of off-road terrains. Since the

compaction resistance force increases with the increase of the applied weight, there was a need to further investigate its effect on the vehicle handling characteristics.

Figure 20 Simplified rigid wheel-soil interaction model (see online version for colours)

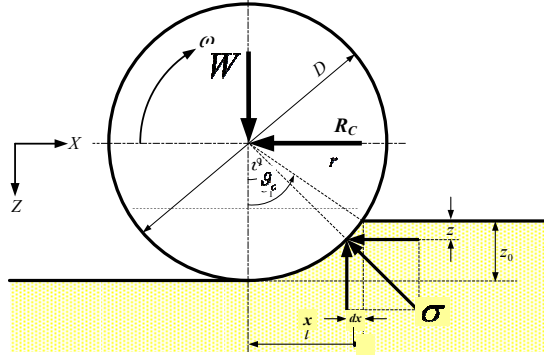


Figure 21 The tyre sinkage for different off-road terrain types (see online version for colours)

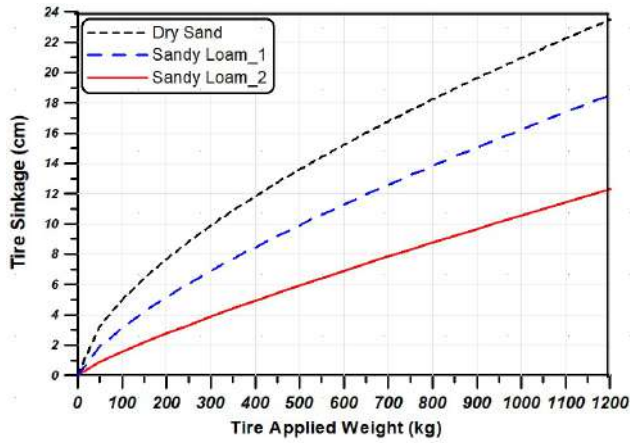
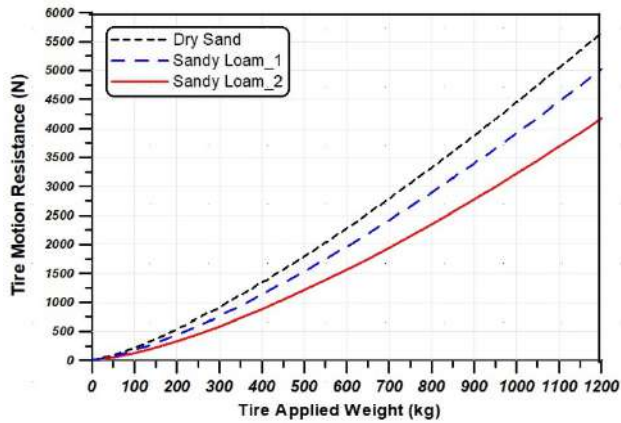


Figure 22 The compaction resistance force for different off-road terrain types (see online version for colours)



The compaction resistance forces are always opposing the motion of the mine rollers' wheels, Figure 23. As shown in figure, the compaction resistance force on each of the mine rollers' sets (R_{c_i} or R_{c_o}) is resolved into two components; one in the longitudinal direction of the vehicle motion ($R_{c_{xi}}, R_{c_{xo}}$) and the other component ($R_{c_{yi}}, R_{c_{yo}}$) in the lateral direction of vehicle motion. The same applies to the rolling resistance of the mine-rollers wheels (not shown in Figure 23). The lateral components of the compaction and rolling resistances were added to the centripetal forces of the mine rollers, calculated earlier, to calculate the total mine-rollers lateral forces as follows:

$$R_{t_y} = R_{c_y} + R_{f_y} + F_{y_{m_i}} + F_{y_{m_o}} \quad (15)$$

where, R_{f_y} is the lateral component of the rolling resistance force that exert on the mine-rollers wheels. To study the effect of the generated lateral forces applied to the mine-rollers setup on the handling characteristics of the vehicle, the free body diagram of the vehicle, with the mine-rollers system installed, was constructed as shown in Figure 24. As shown in figure, the sum of the lateral components of the resistance forces (R_{t_y}) was transferred to the vehicle chassis at the front axle and rear tandem axles by f_1 and f_2 , respectively. The values of f_1 and f_2 are governed by the vehicle longitudinal dimensions and the lateral stiffness of the tyres (K_y). To find the values of f_1 and f_2 , the vehicle chassis (along with the mine-rollers longitudinal beam) was considered a single lateral rigid beam, hence the lateral and angular deflections were solved as a statically determinant beam problem. The magnitudes of the lateral springs' reactions (caused by the lateral tyre stiffness) at the front and rear axles are equivalent to the required forces; f_1 and f_2 , but opposite in direction. The angular deflection of the beam (α) was calculated as follows:

$$\alpha = \frac{3a_4 - a_1 + 2a_2 + l_2}{2K_y \left(2a_1^2 - 2a_2^2 - \frac{1}{2}l_2^2 + 6a_2l_2 + 4a_1a_2 + 2a_1l_2 \right)} R_{t_y} \quad (16)$$

while the lateral deflection (y) of the vehicle chassis was calculated as follows:

$$y = \frac{R_{t_y} - 2K_y(a_1 - 2a_2 - l_2)\alpha}{6K_y} \quad (17)$$

$$f_1 = 2K_y(y + a_1\alpha) \text{ and } f_2 = 4K_y \left(y - \left(a_2 + \frac{l_2}{2} \right) \alpha \right) \quad (18)$$

The lateral stiffness of the tyre (K_y) was obtained from a previously published work by Stallmann et al. (2014). The obtained lateral forces, f_1 and f_2 , were added to the original lateral forces at the front axle and rear tandem axles F_{y_f} and F_{y_r} , respectively. Then, the new front and rear slip angles differences were calculated. The new handling diagram was constructed and compared to that of the vehicle without the mine-rollers setup attached as shown in Figure 25. As shown in figure, adding the effect of mine rollers' wheels resistance forces increased the understeer behaviour of the vehicle.

Figure 23 Compaction resistance forces at the mine-rollers wheels (R_{ci} and R_{co}) (see online version for colours)

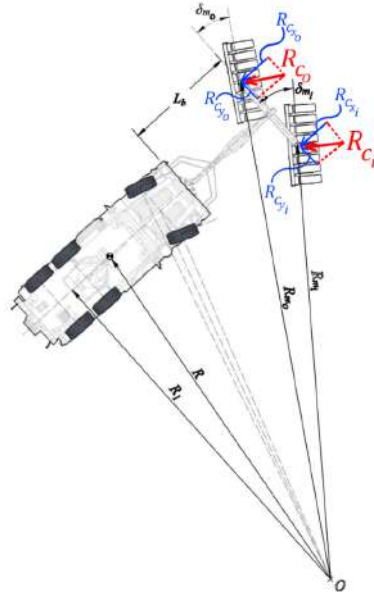


Figure 24 (Left) top view of the vehicle chassis when simplified as a simple beam to study the effect of mine-rollers resistances forces on lateral deflections and forces; additional lateral deflection, y , and additional yaw deflection α . (Right) free body diagram of vehicle – top view (see online version for colours)

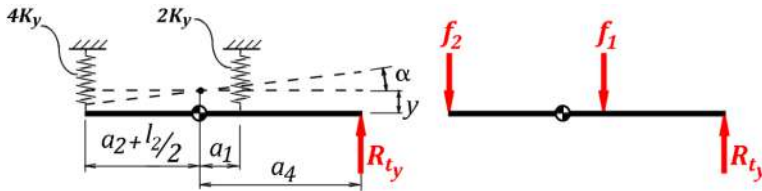
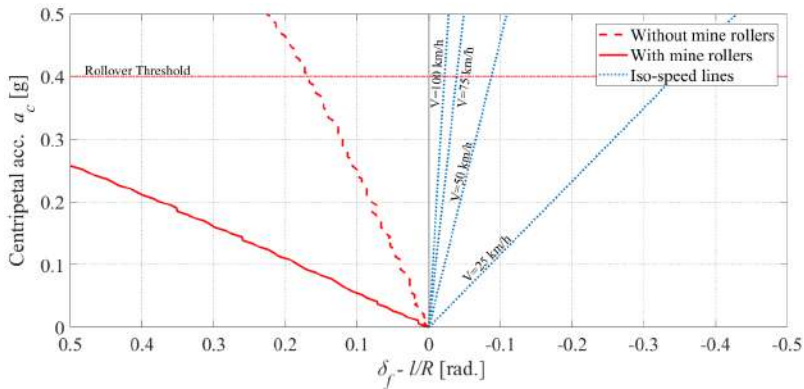


Figure 25 Comparison of handling diagrams of the vehicle with and without the mine-rollers system with 3500 kg mass and 3.45 m longitudinal beam length (effect of resistance forces) (see online version for colours)



6 Effect of increasing the mine-rollers masses and lengths

Increasing the mine-rollers masses will increase the compaction resistance forces in front of the mine-rollers wheels. This will affect the added lateral forces at the front axle and the rear tandem axles. Figure 26 shows the handling diagram of the vehicle when equipped with mine-rollers sets with a mass of 6 tons. As shown in figure, the increased mine-rollers mass increased the understeer coefficient of the vehicle (slope of the handling diagram). The analysis was extended to include the effect of increasing the length of the mine-rollers longitudinal beam. Figure 27 shows the handling diagram of the vehicle when equipped with a mine-rollers setup with a longitudinal beam length of 5m and a mass of 6 tons. Increasing beam length further increased the vehicle understeer coefficient.

Figure 26 Comparison of handling diagrams of the vehicle with and without the mine-rollers system with **6000 kg** mass and **3.45 m** longitudinal beam length (effect of resistance forces) (see online version for colours)

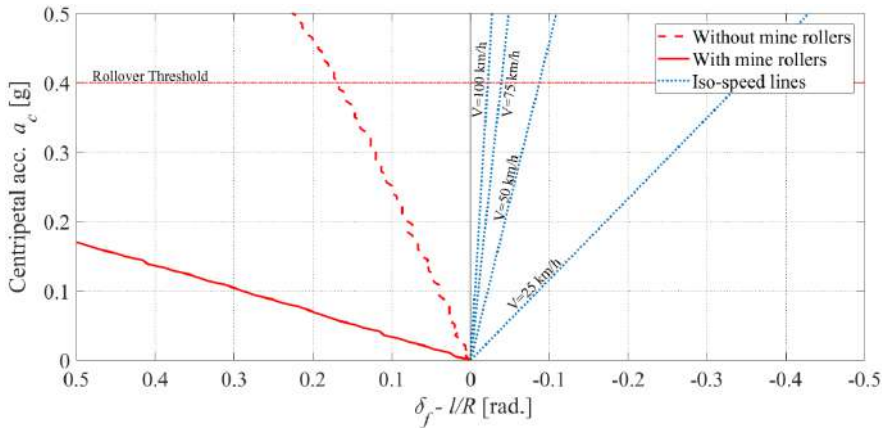
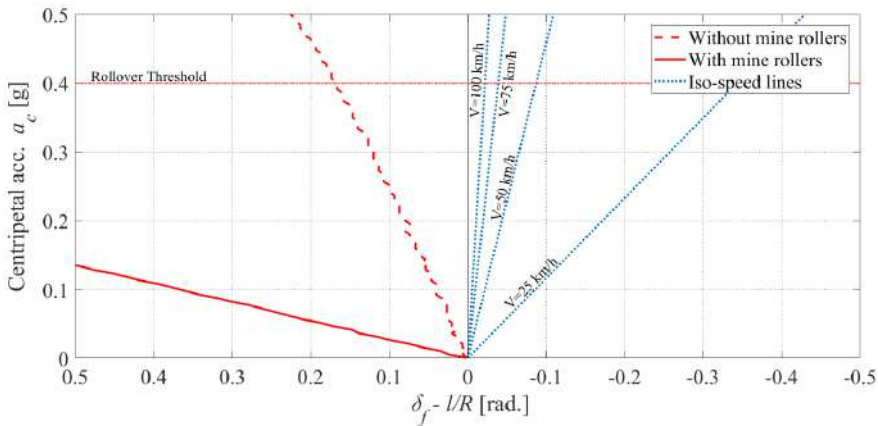


Figure 27 Comparison of handling diagrams of the vehicle with and without the mine-rollers system with **6000 kg** mass and **5 m** longitudinal beam length (effect of resistance forces) (see online version for colours)



7 Stress analysis of steering mechanism

The vehicle steering mechanism is a drag link type that consists of a reduction gearbox, hydraulic power assisting unit, and steering linkages (Pitman arm – drag link – tie rod). Increased lateral forces during cornering causes a combination of shear stresses (on tie rod pins) and normal stresses (on lateral drag link). These lateral forces are subjected to sudden peaks due to several effects including: cornering in off-road terrains, tyres bulldozing forces, and sudden shocks due to terrain irregularities. Therefore, stress analysis on the critical pinned connection of steering mechanism was carried out to ensure structural suitability of the modified vehicle during operations. The following load and boundary conditions had been considered: calculated tyre lateral forces due to cornering – see dynamic analysis section ($F_{yf} = 35$ kN each tyre), dynamic shocks have been assumed to increase lateral forces by the factor of three, and driver steering forces exerted through total multiplication of 350 with an equal magnitude of forces on the Pittman-arm knuckle pinned connection.

The following techniques had been carried out in this stress analysis: spatial measurements and CAD drawings of the steering mechanism, spectrum analysis of the tie rod and pins material to give an insight into the alloy material, simulation of the stresses and respective safety factors on the steering mechanism. The spectrum analysis of tie rod and pins material showed a 97% accordance with the steel alloy no. 4140. The estimated chemical composition of the analysed alloy is shown, comparatively to the 4140 alloy, in Table-1. The after treatment, mechanical properties of the 4140 alloy have been approximately considered similar to 4150 alloy as shown in Table-2.

Table 1 Estimated and standard elements content of tie rod alloy

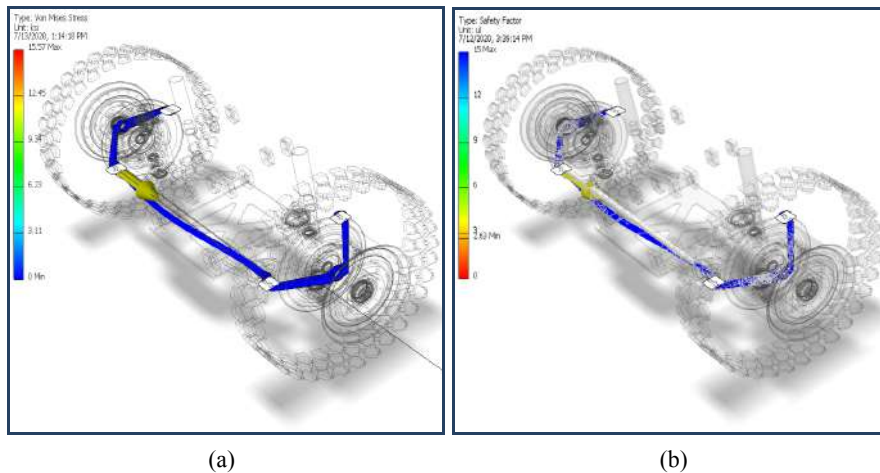
<i>Element</i>	<i>Standard content (%)</i>	<i>Estimated content by ElvaX ProSpector (%)</i>
Chromium, Cr	0.80 – 1.10	1.05
Manganese, Mn	0.75 – 1.0	0.86
Carbon, C	0.380 – 0.430	--
Silicon, Si	0.15 – 0.30	0.88
Molybdenum, Mo	0.15 – 0.25	0.01
Sulphur, S	0.040	0.18
Phosphorus, P	0.035	0.19
Iron, Fe	Balance	96.41

Table 2 Estimated and standard elements content of tie rod alloy

	<i>Alloy 4140</i>	<i>Alloy 4150</i>
Tensile strength (MPa)	655	731
Yield strength (MPa)	415	380
Bulk modulus (typical for steel) (GPa)	140	140
Shear modulus (typical for steel) (GPa)	80	80
Elastic modulus (GPa)	190	190
Hardness, Brinell	197	197

The stress analysis was conducted in Autodesk-Inventor environment considering the above-mentioned stress conditions on the steering mechanism. The dynamic stresses were simulated by tripling and transferring the ground lateral force to the tie rod using the measured mechanism leverage. Also, the boundary conditions were imposed by fixating the left and right wheel hubs. The graphical illustration of the stress analysis is given in Figure 28. As shown in figure, the stresses did not exceed 15.8 ksi (107 MPa) which is less than the yield strength (380 MPa) of the material. Also, the minimum safety factor (**2.63**) obtained at the tie rod ball joint pin indicates that the steering mechanism could withstand the applied cornering loading.

Figure 28 Stress analysis simulation in Autodesk-inventor® environment: (a) stress and (b) safety factor (see online version for colours)



The findings of conducted stress analysis can be summarised as follows:

- 1 Exerting forces on steering linkages (pins) reach up to 100 kN.
- 2 The considered load conditions had been tripled considering extreme driving conditions on off-road and sand terrains.
- 3 No need to further increase the applied stresses on steering mechanism as the tyre forces are bounded by adhesion limits.
- 4 Existing pins of the tie rod were found safe to withstand existing stresses with a minimal safety factor of 2.63.

8 Experimental validation

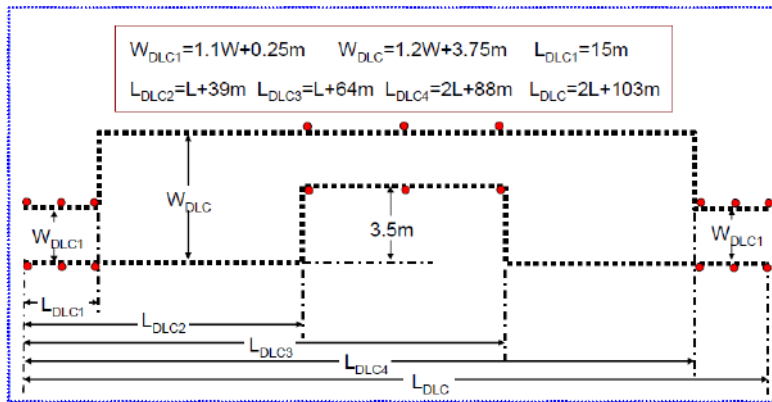
To validate the theoretical analysis, two series of experimental dynamic testing were carried out for the heavy vehicle; one when equipped with the mine-rollers and the other without. A comparison between the results of both cases was conducted. In addition, a correlation between the experimental results and the theoretical results was addressed. The experiments are listed as follows:

- 1 double lane change (DLC) manoeuvre
- 2 constant radius test (CRT)
- 3 emergency brake test.
- 4 strain test for tie rod, Pitman arm and mine-rollers main hinge during driving on severe off-road conditions.

8.1 Double lane change (DLC) manoeuvre

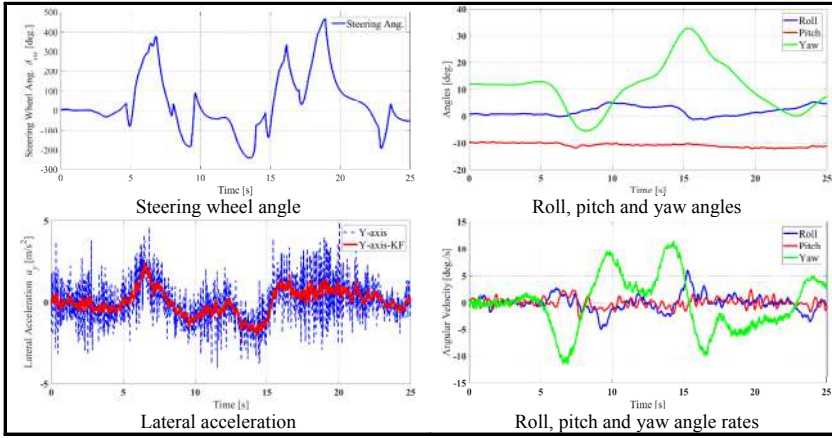
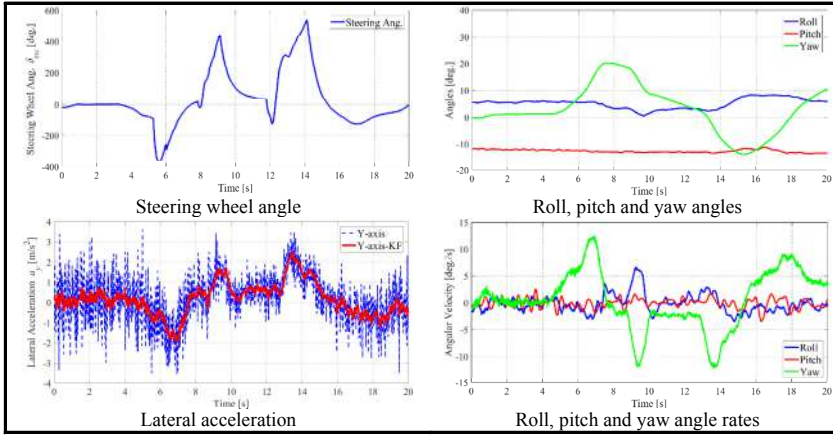
The double lane change (DLC) test was carried out for both cases; with and without the mine-rollers attached. The dimensions of the test track were extracted from the standard test for these types of heavy vehicles (NATO-AVTP-03-160W) (Allied Vehicle Testing Publication, AVTP 03-160 W, 1991) (see Figure 29).

Figure 29 Standard DLC manoeuvre test (NATO-AVTP-03-160W) (see online version for colours)



Source: Allied Vehicle Testing Publication, AVTP 03-160 W (1991)

The test was repeated at several entrance speeds to achieve the maximum safe speed for the vehicle setup. The sensors used to record the results were accelerometers, gyros, a steering wheel angle sensor and the vehicle speed from the vehicle controller area network (CAN) bus. The acceleration signals were filtered using Kalman Filter (KF) to enhance readability. The highest speed recorded for the vehicle with the mine-rollers setup attached was 33 km/h. The results obtained during the DLC test with the mine-rollers attached is shown in Figure 30. As shown in figure, the lateral acceleration achieved was 2.5 m/s^2 (0.225 g). Although the relatively high lateral acceleration achieved compared to this type of vehicles, the vehicle passed the test without touching the track cones or showing any signs of instability as depicted from the roll, yaw and pitch angles and angles' rates figures. For the vehicle testing without the mine rollers, the highest speed recorded was 38 km/h. The results obtained during the DLC test with the mine-rollers attached are shown in Figure 31. As shown in figure, the lateral acceleration achieved was 2.5 m/s^2 (0.225 g). Comparing the roll, yaw and pitch results with those when the mine-rollers were attached, it is found that the mine-rollers did not affect the original vehicle roll and yaw stability.

Figure 30 Experimental results of DLC manoeuvre of the vehicle with mine-rollers (33 km/h)
(see online version for colours)**Figure 31** Experimental results of DLC manoeuvre of the vehicle without mine-rollers (38 km/h)
(see online version for colours)

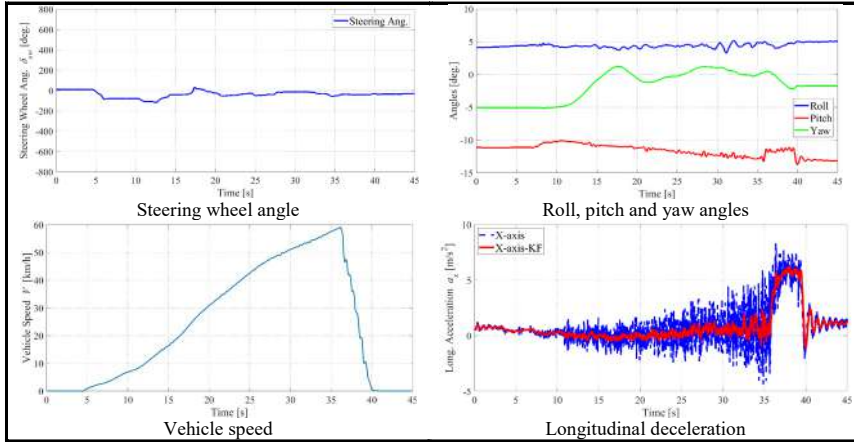
8.2 Emergency brake test

The vehicle was driven to the highest possible safe speed and then the emergency brake was applied. The brake pedal was depressed in full to the complete vehicle stop without readjusting the steering wheel angle. The signals monitored were the vehicle speed, the steering wheel angle, the longitudinal deceleration and the roll/yaw/pitch angles. The signals were evaluated based on the allowance ranges for the brake testing for heavy vehicles (New Zealand Transport Agency, 2007; National Highway Traffic Safety Administration – Department of Transportation, 2009). The safe stopping distance (S) in metres was measured and evaluated based on the recommended range and function of the entrance speed (V) in km/h as follows (New Zealand Transport Agency, 2007):

$$S = 0.15V + \frac{V^2}{130} \quad (19)$$

Figure 32 shows the signals obtained from the emergency brake test for the vehicle equipped with the mine-rollers. The test entrance speed recorded was 60 km/h and the maximum longitudinal deceleration was 6 m/s² (0.61 g).

Figure 32 Experimental results of emergency brake of the vehicle with mine-rollers (60 km/h) (see online version for colours)



As shown from Figure 32, the yaw response at the braking duration did not exceed 3°. This value is less than the recommended range (15°) according to the braking evaluation criteria (Economic Commission for Europe of the United Nations – United Nations Economic and Social Council, 2015, 2016). In addition, the measured stopping distance was 33 m, which is less than the recommended value (36 m) which implies the stability of the vehicle during braking.

8.3 Constant radius test

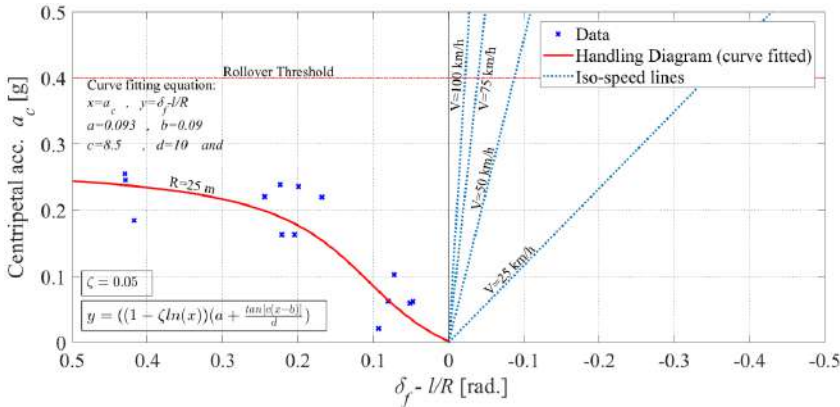
The aim of the constant radius test is to construct the handling diagram experimentally. The constant radius of the test ground was 25 m. The test was repeated at different forward speeds. In each run, while maintaining a steady-state forward speed duration, the same signals were captured as described in the previous section. During the tests, the maximum achievable forward speed at this tuning radius was 42 km/h. The Root Mean Square (RMS) for the signals of interest at the steady-state durations were obtained. Those measured signals were the vehicle centripetal acceleration (a_c), the steering wheel angle (δ_f), and the vehicle longitudinal speed (V). Given the constant turning radius (R) and the vehicle wheelbase (l), the handling diagram was constructed by plotting each individual test point on the graph. In order to depict the curve trend, a nonlinear curve fitting method was applied to the test data, Figure 33. The figure shows blue marks (×) at the points that represent the actual runs of the test. The red line represents the obtained curve fitting equation that best fits the data, which has the following nonlinear form. Where (x) is the handling diagram abscissa ($\delta_f - l/R$), y is its ordinate (a_c), ζ , a , b , c and d are the curve fitting equation coefficients and their values are as follows: $\zeta = 0.05$, $a = 0.093$, $b = 0.09$, $c = 8.5$ and $d = 10$. The graph also shows the iso-speed lines and the

commonly used rollover threshold of 0.4 g centripetal acceleration (Economic Commission for Europe of the United Nations – United Nations Economic, and Social Council, 2001). As shown in figure, the vehicle was in understeer condition in all the range of operation of the vehicle. This implies that the vehicle when equipped with the mine-rollers setup has stable handling characteristics. The constant radius tests were repeated for the same vehicle without the mine-rollers setup. Figure 34 shows the handling diagram of the vehicle in the latter case. The following equation shows the curve-fit relation between the abscissa ($x = \delta_f l/R$) and the ordinate ($y = a_c$):

$$y = (1 - \xi \ln x) \left(a + \frac{\tan[c(x-b)]}{d} \right) \quad (20)$$

where, the equation coefficients are $\xi = -0.025$, $a = 0.135$, $b = 0.29$, $c = 4.2$ and $d = 20$. From the diagram, it is shown that the vehicle was in understeer condition throughout the centripetal acceleration operating range at least beyond the rollover threshold. Figure 35 shows a comparison between the experimentally obtained handling diagrams of the vehicle in both cases, with and without the mine-rollers setup. As shown in figure, adding the mine-rollers setup to the vehicle increased its understeer coefficient. Furthermore, returning to the theoretically obtained handling diagram, Figure 15, it was found that the proposed theoretical analysis presents an adequate prediction for the vehicle handling performance that could help designers to theoretically optimise the mine-rollers setup design which significantly reduces the efforts during prototype testing.

Figure 33 Experimentally obtained handling diagram of the vehicle when equipped with the mine-rollers setup. The curve-fit (red line) shows that the vehicle is in understeer condition throughout the operation range of the vehicle (see online version for colours)



8.4 Stress analysis of critical components

The main purpose of the experimental stress analysis is to investigate the effect of the mine-rollers setup on the critical components of the vehicle. In order to investigate the stresses on those critical parts, five strain gauges were installed as follows: Two strain gauges at the tie rod, one strain gauge at the Pitman arm and two strain gauges at one of

the hinges brackets that support the mine-rollers setup to the vehicle chassis. The test manoeuvres included the following: DLC, constant radius test, emergency braking, driving in on-road environment and driving in off-road environment. All those manoeuvres were repeated twice; with and without the mine-rollers setup attached to the vehicle. Based on the steering system components material and the yield strength, the safe strain limit was calculated and plotted against the measured strain in the graphs. It was found that the maximum strain occurred at the tie rod during the emergency brake test. Figure 36 shows sample strain results obtained during the emergency brake test when the vehicle was equipped with the mine-rollers setup. As seen from the graphs, the maximum strain value at the tie rod during the emergency brake test was 0.07 with a safety factor of 2.8.

Figure 34 Experimentally obtained handling diagram of the vehicle without the mine-rollers setup. The curve-fit (red line) shows that the vehicle is in understeer condition throughout the operation range beyond the rollover threshold (see online version for colours)

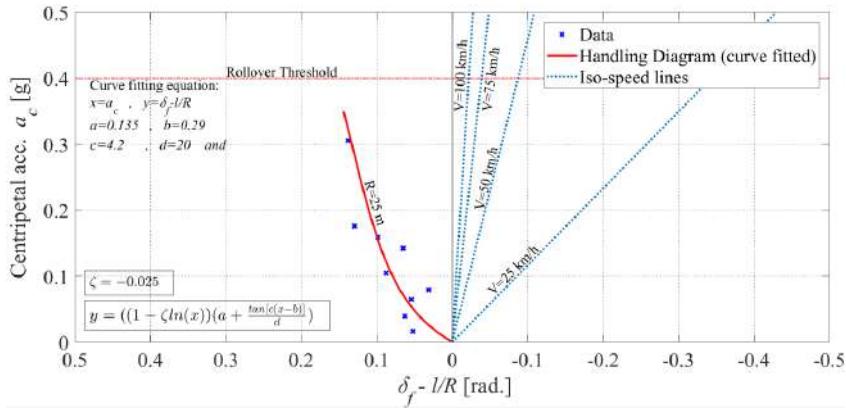


Figure 35 A comparison between the experimentally obtained handling diagrams of the vehicle in both cases; with and without the mine-rollers setup (see online version for colours)

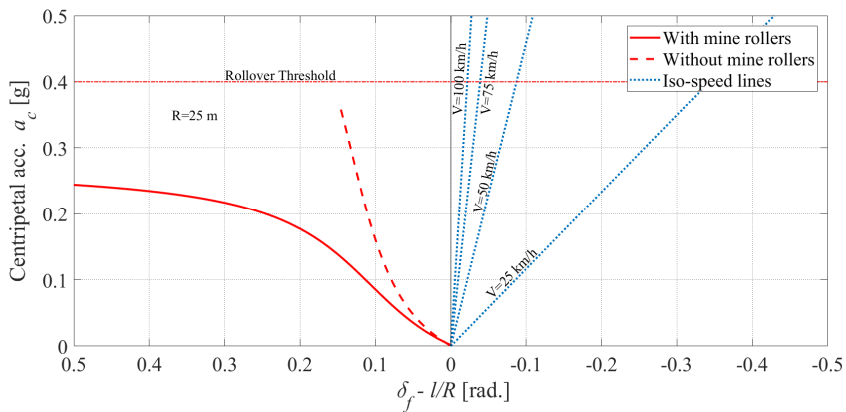
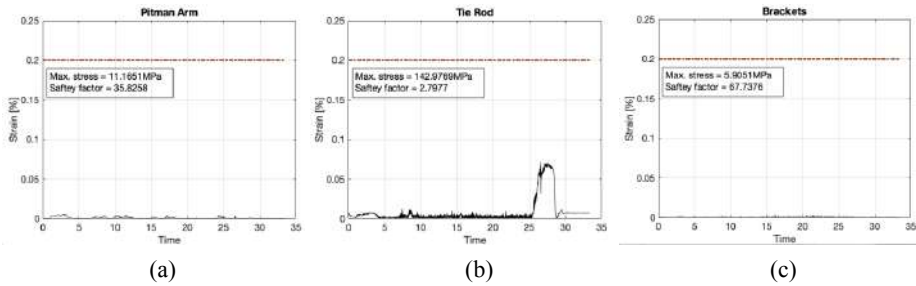
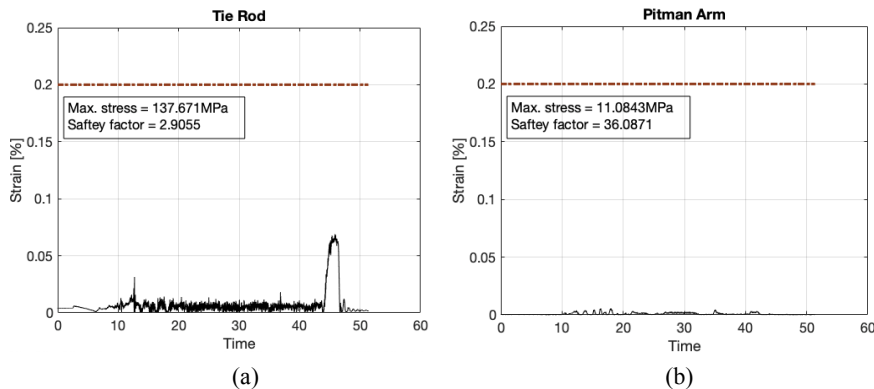


Figure 36 Sample strain results during the emergency brake test with the mine-rollers setup: (a) pitman arm; (b) tie rod and (c) hinged support bracket (see online version for colours)



To further address the significance of adding the mine-rollers setup, the strain results of the all the tests were repeated and recorded for the vehicle without the mine-rollers setup attached. In the latter case, all the recorded strain values were found significantly small for all manoeuvres while the emergency brake test achieved the highest value at the tie rod. Figure 37 shows the results of this test, the strain at the tie rod has a value of 0.069 which gives a safety factor of 2.9. This means that adding the mine-rollers did not significantly increase the stresses on the tie rod during all manoeuvres.

Figure 37 Strain measurements during the emergency brake test without the mine-rollers setup: (a) tie rod and (b) pitman arm (see online version for colours)



9 Conclusions

A theoretical and experimental investigation of the effect of adding a mine-rollers system to a heavy vehicle on its handling and roll stability was presented. The static and the dynamic forces acting on the vehicle during cornering were estimated. Based on the theoretical analysis, the handling diagram was constructed. To investigate the safety of vehicle critical components, a theoretical stress analysis was carried out. The methodology was validated experimentally by implementing DLC manoeuvres, constant radius test manoeuvres and emergency brake tests; both with and without the mine-rollers setup attached. The steering wheel angle, the vehicle body accelerations, the yaw/roll/pitch angles/angles' rates and the vehicle speed were captured during the entire test. From

the DLC manoeuvres and the emergency brake tests, it was found that the roll and braking performance were negligibly affected by the added mine-rollers setup. The experimentally constructed and curve fit handling diagrams showed that adding the mine-rollers setup had increased the understeer coefficient of the vehicle without degrading the handling performance. The stress analysis was validated by measuring the strain at selected critical components (tie rod, Pitman arm and the hinges of the mine-rollers supports) during the mentioned manoeuvres in addition to on-road and off-road driving. The strain values in all manoeuvres at those components were significantly low while it was observed that its highest values occurred at the tie rod during emergency braking. Comparing the strain value with the one obtained for the vehicle without the mine-rollers setup attached showed that the setup did not significantly affect the safety factor of the tie rod. The presented analysis was proved to be an adequate tool to theoretically estimate the handling performance of vehicles equipped with mine-rollers setups in the design phase which reduces a lot of endeavours that may arise in prototyping.

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