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Lateral Acceleration Potential Field Function Control for Rollover Safety of Multi-wheel Military Vehicle with In-Wheel-Motors

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Abstract: This article proposes an automatic longitudinal deceleration based method for multi-wheel vehicle rollover safety in autonomous mode. The information of lateral acceleration and vehicle roll angle is used to generate the longitudinal acceleration at which the vehicle will remain stable to rollover. The lateral and roll dynamics are coupled with longitudinal dynamics using a potential field function for lateral acceleration. This virtual potential field is developed on g-g diagram which represents vehicle portrait of lateral and longitudinal acceleration on abscissa and ordinate respectively. The motion of vehicle is represented by a point moving on this phase portrait of g-g diagram. TruckSim model of multi-wheel military vehicle with in-wheel motors is used with this algorithm which shows that the vehicle is less susceptible to rollover. The safe longitudinal acceleration is achieved by torque control of in-wheel motors fitted in each wheel. Using this method, the vehicle followed the desired trajectory as higher speeds which are safe. This is particularly useful for vehicle autonomous driving with rollover stability.

Keywords: In-wheel motor, lateral acceleration, multi-axle electric vehicle, potential field function, rollover safety.

1. INTRODUCTION

Multi-axle vehicles are favored for military offroad use and autonomous exploration because of better traction force in the tire contact patch with road and even weight distribution. These vehicle have good mobility even if one or two of the tires are puntured. Traditionally military armoured vehicle are heavy, however there is need for lighter versions for operations at higher speeds in cities without damaging road surfaces. To get benefits of advancements of battery and electric motors ,one class of these vehicles is fitted with in-wheel-motors and electric batteries which has the property of efficient and quite operation. Electric vehicle (EV) with in-wheel-motor not only have obvious advantages from control point of view [1] but also have minimum heat signature due to the absence of any heat engine.

Moreover multiwheeled vehicle with in-wheel-motors have a redundant architecture as there are more actuators than the minimum number required for a general 2D motion on ground. This redundancy offers flexibility for devising the driving control algorithm. This redundancy can be expoited in different ways for example driving the vehicle with minimum energy consumption, better motion performance using torque distribution in real time between left/right and front/rear wheels, better control for having more number of actuators or even for stability such as rollover safety.

Generally multiwheeled architecture offer uniform weight distribution which makes these vehicles more stable however at higher speeds it still show rollover propensity. Vehicle rollover is a dangerous safety related issue for passenger vehicles as well as heavy trucks. NHTSA data shows a large number of fatalities are due to single vehicle road accidents resulting from rollover when a vehicle takes sharp turn at high speed. This motion results into large lateral acceleration at the center of gravity of vehicle. This type of friction induced rollover is called untripped rollover in which the rollover is caused by the inertial forces at center of gravity (CG) and lateral tire forces at contact points with ground. To address this issue of vehicle rollover, impending rollover needs to be detected in timely fashion and mitigated by the action of actuator.

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In this article a novel method is proposed for rollover stability of military multiwheel vehicle with in-wheel motors. First literature review is given here with objective that how the redundancy associated with multiwheeled vehicle have been utilized in previous studies. The problem of rollover detection and mitigation is reported briefly afterwards, followed by the brief explanation of the proposed method.

There have been previous analytical studies on multiwheeled vehicle for improvement of performance utilizing the redundancy of the multiwheel vehicle. Generally multiwheel vehicles exist in different combinations of steerable wheels and non-steerable (skid steering vehicles) and independent/connected driven wheels. In [2, 3] the author used six component force-plate to experimentally measure the tire forces for Kokoon six wheel all-terrain skid steering vehicle for the purpose of improving the steering efficiency. The nonsteerable wheels makes the turning motion complex as compared to turning motion with steerable wheels. On the basis of this study some adjustment to the suspension and mass distribution was proposed that improved the steering and resulted in reduction of propulsion forces of about 20∼30 %. In [4] torque control algorithm is developed using sliding mode control for a six wheeled skid steering vehicle with articulated suspension in which the tire forces optimum distribution cause to minimize the tires work load, slip ratios and realize the driver's commands. In another case sliding mode control is applied to RobuROC6 robot which is also skid steering platform [5] to find out torque commands for each wheel as per desired motion. The sliding mode control was utilized to dealt with the uncertainity associated with tire frictional forces with ground. In another study [6, 7] six wheel drive and six wheel steering (6WD/6WS) system was considered for optimal distribution of tire forces with different weighting factors assigned to each tire in accordance with verticle loading on each tire. The upper lever controller first determine the driving longitudinal force and turning moment in accordance with the driver commands. At lower lever another controller uses the force and moment command to distributes the forces for each wheel optimally using an objective function of an optimization problem. Different objective functions are reported [5, 8–12] in literature for optimum distribution of tire forces for minimizing the driving energy as the sole objective of optimization problem.

There have been few studies for multi-wheeled platform with in-wheel motors for rollover mitigation and other safety related scenario. In [12] the braking force optimal distribution is carried out for rollover mitigation using quadratic programming technique. The redundancy characteristics are utilized for rollover safety here however this problem is addressed at lower level where the effect of torque distribution is utilized to tackle the rollover issue.

Rollover is one of the most fatal accidents that happen during severe turning motion of vehicle. NHTSA data shows a large proporsion of fatal accidents results from rollover. Military vehicles [13] and heavy trucks are also susceptible to rollover. Researchers have determined several metrics [14–17] for rollover detection/warning systems before rollover to happen and also rollover mitigation schemes [14, 15, 18–21] once the rollover is detected. Rollover can be mitigated generally by steering system e.g. active front steering (AFS), differential braking/torque, active suspension system/anti rolling and braking system. In [7] the author proposed a torque control algorithm for a 6WD/6WS vehicle for lateral stability and rollover prevention by keeping the lateral acceleration of vehicle below the limiting value which is determined from vehicle states and road coefficient estimation. G-vectoring control [22–24] is a method to control the longitudinal acceleration in accordance with lateral jerk hence coupling the longitudinal and lateral dynamics of vehicle. This control law uses the information of lateral jerk and is based on expert driver behaviour who can accelerates and decelerates the vheicle while turning in accordance with laterl jerk for improved traction.

There can be different situation where the lateral jerk will not be significant for rollover to occur but having a road bank angle big enough the vehicle may rollover. This means that along with lateral acceleration, the roll angle due to road terrain also needs to be considered for rollover propensity, particularly in autonomous mode of driving.

Based on the above cited literature a method is proposed which has the three properties. First the proposed method constraints the lateral accelertion and roll angles of vehicle by controlling the longitudinal acceleration for rollover safety. Second the driver commands for desired longitudinal acceleration are modified by the controller using the information of lateral acceleration and roll angle which both are very important parameters to detect the rollover propensity of a moving vehicle. Lastly the lateral, roll dynamics are connected with longitudinal dynamics using a potential field smooth function. This method is particularly suitable for high speed motion in variable terrain profile. Driver commands can be received remotely or from a driver sitting in the vehicle.

This paper is organised as following. In next section the vehicle model is explained followed by the driving algorithm. Simulation results are shown in section 4 and some conclusion are drawn which are given at the end.

2. ROLLOVER PREVENTION AND LATERAL ACCELERATION

Rollover prevention or mitigation can be realized by braking the vehicle, generating counter yaw moment by differential braking or traction, roll moment by active suspension system and a steering system e.g., active front steering system. Rollover is caused by large lateral acceleration when the vehicle is turning a sharp turn at hight speed. This causes the lateral load shift from inner wheels to outer wheels and ultimately the inner wheels lift off the road at the moment of rollover. For rollover detection studies different metrics are used such as rollover index which ranges from -1 to 1 as shown in (1) Rollover index represents the difference between vertical tire forces at left and right wheels. As the vertical tire forces cannot be directly measured so rollover index can be estimated using (2). This equation is based on roll dynamics model of vehicle. The same equation can be further simplified [20] to (3) by neglecting the second term as the roll angle remain small until the rollover takes place. The roll angle is neglected in approximate rollover index but it is used in potential field function equation.

$$RI = \frac{F_{zr} - F_{zl}}{F_{zr} + F_{zl}} \qquad -1 < RI < 1, \tag{1}$$

$$RI = \frac{2h}{t.g} a_y \cos\phi + \frac{2h}{t} \sin\phi, \tag{2}$$

$$RI \approx \frac{2h}{t.g} a_y, \tag{3}$$

$$a_{y\_lift\_off} = \left(\frac{t}{2h}\right).g. \tag{4}$$

Equation (4) shows the lateral acceleration at which the wheel lift occurs and inside of vehicle wheels will have no vertical load as it leaves the ground. This limiting value of lateral acceleration will be used as the constraint. It is clear from this equation that rollover is dependent of vehicle geometric parameters along with lateral acceleration. By keeping the lateral acceleration at lower values the danger of rollover can be suppressed.

2.1. Lateral acceleration constraints

The vehicle lateral acceleration can be calculated using (5) which is composed of two terms. First term in this equation is for general motion of vehicle with yaw rate which is moving at certain velocity. The second term is due to the lateral sliding motion of vehicle. Because in this paper only friction induced rollover is considered so the second term can be eliminated.

$$a_y = V_x \cdot \gamma + \ddot{y}, \tag{5}$$

$$a_y \approx V_x \cdot \gamma. \tag{6}$$

The constraint of lateral acceleration on motion of vehicle can be maintained by either controlling the speed or yaw rate of vehicle. Given a value of lateral acceleration and using (6) the plot on vehicle velocity and yaw rate can be plotted as shown in Fig. 1. If the vehicle motion stays below this plot then the value of lateral acceleration will not exceed and lateral stability can be ensured.

2.2. 1DOF roll angle estimator

The vehicle roll angle can be estimated by using the data of roll rate and lateral acceleration data using a one

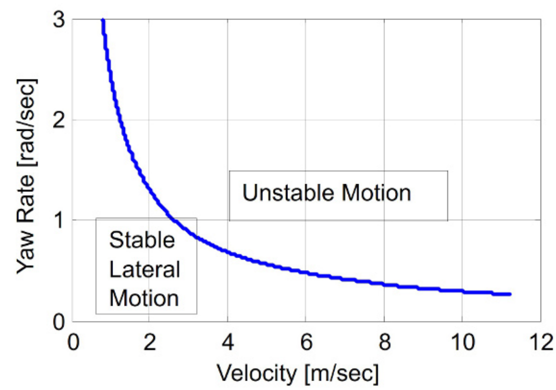

Fig. 1. Lateral acceleration constraints on vehicle velocity-yaw rate.

degree of freedom estimator [19]. This is done by first integrating the roll rate data of sensor and then using a static function of vehicle lateral acceleration. Vehicle roll angle data is used in calculation of lateral acceleration potential field. Vehicle roll dynamics is shown in Eq. (7) using the inertia, damping and spring constant for roll dynamics which is caused by lateral inertial force acting at center of gravity.

$$(I_x + M_s h_x^2)\ddot{\phi} + c_\phi \dot{\phi} + k_\phi \phi = M_s a_{y,m} h_s, \tag{7}$$

where $a_{y,m}$ is the measured lateral acceleration and is given in (8).

$$a_{y,m} = a_y + g \sin\phi. \tag{8}$$

3. LATERAL ACCELERATION POTENTIAL FIELD FUNCTION FOR ROLLOVER STABILITY

The driving controller consists of algorithm where the desired longitudinal tire force and yaw moment is determined on the basis of driver commands. The driver's commands based longitudinal acceleration and desired yaw rate combination may cause the vehicle to rollover by exceeding the limit of latera acceleration for rollover stability. For rollover prevention the driver longitudinal acceleration is modified in order to keep the lateral acceleration and roll angle below their limiting values to avoid rollover while the controller follows the desired yaw rate as per driver commands. The limiting value of lateral acceleration at which the wheel lift-off can be used as a constraint for actual lateral acceleration. The potential field function method is used to determine the longitudinal acceleration from vehicle lateral acceleration and roll angle states.

subsectionPotential Field Function Based Deceleration

Originally this method is used in mobile robotics for obstacle avoidance where the distance from the obstacle is measured and if its within predefined radius range then

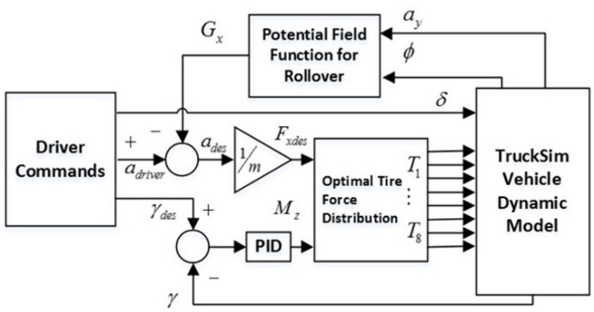

Fig. 2. Driving control algorithm.

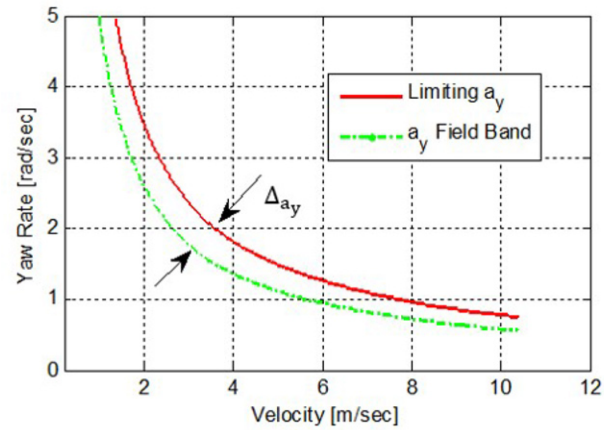

Fig. 3. Lateral acceleration field band on velocity-yaw rate.

a virtual repulsive force is generated that modify the mobile planned path. Using the same analogy the vehicle is decelerated on the basis of difference between the actual and limiting lateral acceleration at which the rollover is likely to happen. The limiting lateral acceleration is treated like an obstacle and the system will use torque control to avoid this value of lateral acceleration by slowing down the vehicle. This deceleration will be generated if the difference between the actual lateral acceleration and limiting acceleration is less than a predefined band gap as shown in (9). Within this band this algorithm will generate more deceleration as the actual lateral acceleration goes more close to the limiting value of lateral acceleration and the driver's command of desired acceleration will be modified. In analogy to an object which is subjected to magnetic field in effective range will feel more repulsive/attractive force as it goes deep inside magnetic field. The driving control algorithm is shown in Fig. 2. The potential field function utilize the lateral acceleration and roll angle and generates deceleration signal G_x. This modifies the driver command of desired longitudinal acceleration. The desired longitudinal force and yaw moment is converted into torque commands by minimizing an objective function. Due to the redundancy of multiwheeled platform the forces can be distributed in several ways. The optimization problem reduces the tire load and help prevent the tire force saturation. The torque signals are calculated by driving algorithm are applied to the TruckSim model and vehicle states are fed back to the algorithm.

Using the same approach in effective range this algorithm will generate more deceleration as the actual lateral acceleration goes more close to the limiting value of lateral acceleration and the driver's command of desired acceleration will be modified. The limiting value and band width of this lateral acceleration potential field is shown in Fig. 3.

Using this concept the algorithm will follow the driver intended motion for smaller values of lateral acceleration but as the value of lateral acceleration and roll angle goes high and enters the lateral acceleration fields the controller will start modifying the driver's commands. This is mathematically represented in (9). This equation generates decelerating force from a repulsive force function between the actual and limiting lateral acceleration.

The *signum* function takes into account the signs of lateral acceleration and roll angle. Their combine effect can be additive or can cancle the effect of each other if they are of opposite signs. The consideration of roll angle along with lateral acceleration results in different safe speeds at roads that have different roll angles for a given motion. This means that a vehicle can move faster with some lateral acceleration on a curved road if the bank angle is negative as compared to zero road bank angle or positive bank angle where the same speed will cause rollover to occur. The η is a tuning scaler that can be tuned during simulations. Equation (9) will results in zero deceleration if the vehicle actual lateral acceleration is less than Δ_{ay} otherwise the deceleration is based on the difference between the actual lateral acceleration and the limiting value for which the rollover is predicted to occure. The selection of the limiting value used in this study is chosen as a constant value on the basis of several extream simulation. In most of the literature the threshold values of roll angle and lateral acceleration is used.

This method can be integreated with another intelligent algorithm that determines this limiting vlaues in real time with some adaptation on the basis of vehicle states. Simulations confirm these results of stable motion on different road bank angles with different speeds.

$$G_x = \begin{cases} -\mathrm{sgn}(a_y,\phi).\frac{1}{2}.\eta\left(\frac{1}{(a_y-a_{y\_\lim})}-\frac{1}{\Delta_{ay}}\right), & \\ \qquad (a_y-a_{y\_\lim})<\Delta_{a_y}, & \\ 0, \qquad (a_y-a_{y\_\lim})>\Delta_{a_y} & \end{cases} \tag{9}$$

$$a_{des} = \begin{cases} a\_\_{driver}-G_x, & (a_y-a_{y\_\lim})<\Delta_{a_y} \\ a_{driver}, & (a_y-a_{y\_\lim})>\Delta_{a_y}. \end{cases} \tag{10}$$

Using the lateral acceleration potential field the combination of safe longitudinal acceleration and desired yaw rate is generated to ensure the rollover stability. The total driving force and yaw moment can be calculate from these two values. The multi wheeled platform offer flexibility in distributing the tire forces in many ways due to its redundant architecture. The tire forces optimal distribution is used to minimize the summation of normalized forces on all tires to minimize the chance of tire saturation of any wheel using the tire friction circle concept. This is explained in following section.

3.1. Optimal tire force distribution method

The redundant architecture of an eight wheel drive vehicle with in-wheel-motor provides the oppurtunity that it can be driven based on an optimal law. The objective function as given in (11) was minimized and the lateral and longitudinal forces to satisfy the desired longitudinal force F_{xdes} and yaw moment M_{zdes} were calculated. The objective function is defined as the sum of the squared normalized forces on each tire from the concept of the friction circle. This minimization reduces the tire force loads and ultimately minimized the energy for a given motion [9–11]. F_{xi}, F_{yi} and F_{zi} are tire longitudinal ,lateral and verticle forces respectively.

$$J = \sum_{i=1}^{8} \mu_i^2 = \sum_{i=1}^{8} \frac{F_{xi}^2 + F_{yi}^2}{F_{zi}^2}. \tag{11}$$

4. VEHICLE DYNAMIC MODEL

4.1. Chassis dynamics model

The chassis dynamic model used in this study consists of 3 rotational degree of freedom i.e., roll, pitch, yaw and 3 translational degree of freedom in 3D space, 8 independent suspension one at each wheel, 8 wheel dynamic models and tire models.

The wheels on front two axles are steerable and all 8 wheels are drivable with in-wheel motors. The vehicle platform is modeled using TruckSim software. The control algorithm is developed in Simulink® and is run in close loop with this vehicle model. Vehicle parameter used in simulations are given in Table.1. This platform is fitted with in-wheel motors and the modeling of wheel and motor is explained next.

4.2. In-wheel motor dynamic model

In-Wheel-Motor have fast response time which is modeled as a first order system with a delay time τ as given in (12). Time delay of the system is based on the inductance L_m and resistance R_m of the coil in the motor.

$$\begin{aligned} T_m &= \frac{1}{1+\tau s} T_{m\_com}, \\ T_m &= \frac{1}{1+(L_m/R_m)\, s} T_{m\_com}. \end{aligned} \tag{12}$$

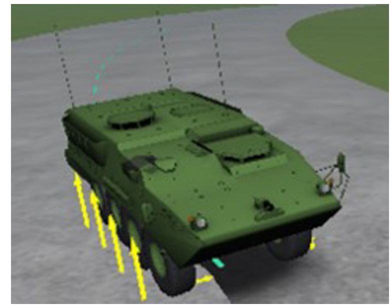

Fig. 4. TruckSim model of 8 wheel vehicle.

Table 1. The vehicle parameters used in simulation.

No	Parameters		Specification
1	Mass m		10000 kg
2	Tread L		4 m
3	Moment of Inertia	Izz Ixx Iyy	7,000 kg-m2 40,000 kg-m2 60,000 kg-m2
4	Wheel Base l		1.9 m
5	Wheel Radius r		0.6 m
6	Gear Ratio of Motor with Wheel		17

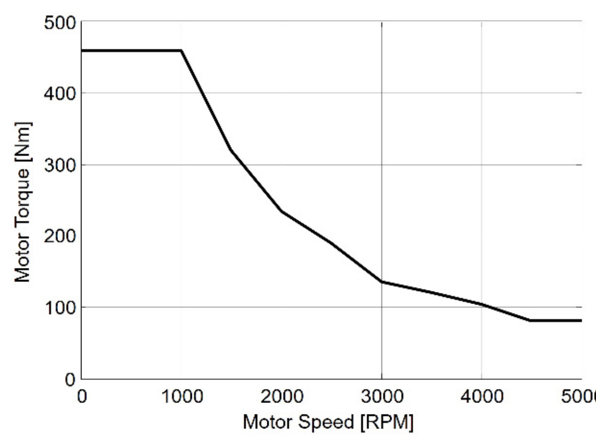

Fig. 5. Motor torque-speed performance curve.

The output torque T_m of the motor model is determined by the input value T_{m_com} which is the wheel torque command. The performance characteristic is shown in Fig. 5 which shows how the torque generated by electric motor drops at the speed of motor increases. The lookup data representing this curve is used in simulation for all the eight driven wheels of vehicle. This is used in simulation for the purpose the system more realistically as it behaves in real world.

5. SIMULATION

The response of this control algorithm is evaluated via simulation. The algorithm was developed in Simulink

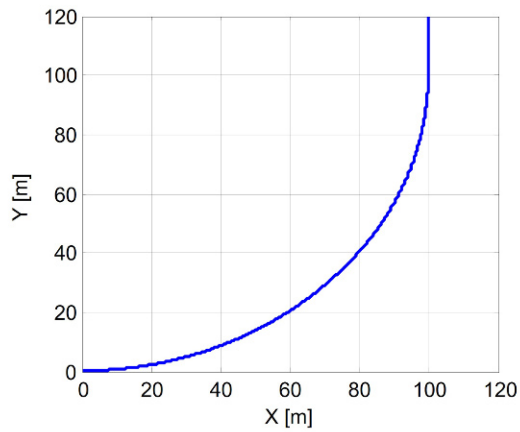

Fig. 6. Constant radius 100 m turn trajectory.

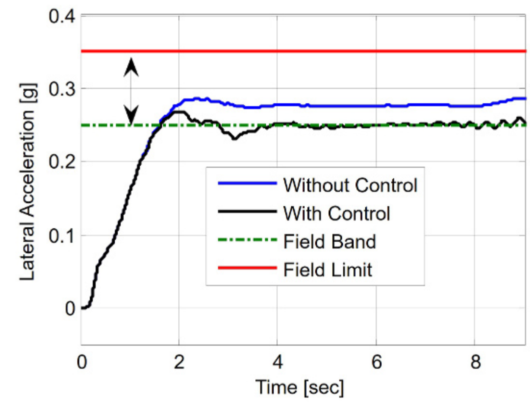

Fig. 7. Lateral acceleration for constant radius turn.

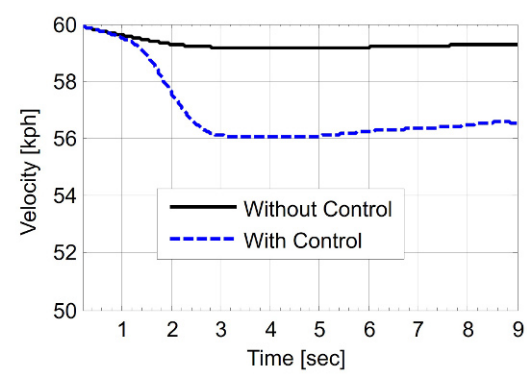

Fig. 8. Velocity during constant radius turn.

Matlab and integrated with vehicle model of TruckSim. The performance was evaluated by switching the potential field function block to On/Off during different maneuvers for checking its effects on rollover prevention.

5.1. Constant radius turn

The constant radius 100 m turn maneuver is performed to test the algorithm in steady manner. The vehicle is moved to follow the trajectory shown in Fig. 6 at initial velocity of 60 kph. In the simulation the rollover control using the potential field function is activated and limiting lateral acceleration of 0.35g is used in the controller which act as a obstacle barrier. In this study the constant predefined values of limiting lateral acceleration is used in simulation on the basis of rollover index as given in (5). This 0.35g of lateral acceleration is suitable value for 60 kph speed while following this trajectory. The controller takes into account the difference between the actual lateral acceleration and limiting value and uses torque control to appply braking force and bring the lateral acceleration down for rollover safety. The controller will be activated if the lateral cceleration is higher than 0.25g and less than 0.35g.

The velocity, yaw rate and lateral acceleration are compared for the two cases of with-control and without-control on these graphs to evaluate the performance of this controller concept. The target vehicle speed is 60 kph so the controller attains this speed but as the lateral acceleration reaches the band gap of 0.25g$\sim$ 0.35g the controller appply braking forces using torque control as a result the lateral acceleration is maintained below 0.35g as shown in Fig. 7.

In Fig. 9 the controller action is compared with another simulation without any control action. The controller action switches between acceleration and deceleration but keeps the lateral acceleration below 0.25g. The g-g diagram portrait for this maneuver is shown in Fig. 9 which shows how the lateral acceleration is maintained by controller while decelerating the vehicle. The controller action suppress the lateral acceleration and develop deceleration as shown in Fig. 9.

5.2. Constant radius turn at 60 kph with different road bank angles

The potential field function control uses the lateral acceleration and vehicle roll angle in the feedback loop and control the longitudinal acceleration of the vehicle accordingly. The consideration of roll angle along with lateral acceleration is focused in this simulation. The same trajectory of Fig. 6 is followed by the vehicle at 60 kph for three different kind of road bank angles with corresponding cross elevation slop of -10% (outward slope), 0% (flat road) and $+10\%$ (inward slope) in lateral direction. The purpose of this simulation is to test the controller performance on variable terrain in a steady way. In this conditions the controller results in different vehicle velocities for each case applying different amount of braking force from torque control. The velocities are compared in Fig. 10. The longitudinal and lateral acceleration are shown in Fig. 11 and Fig. 12 respectively.

The g-g diagram for this maneuver is shown in Fig. 13 in which the lateral acceleration is maintained in safe

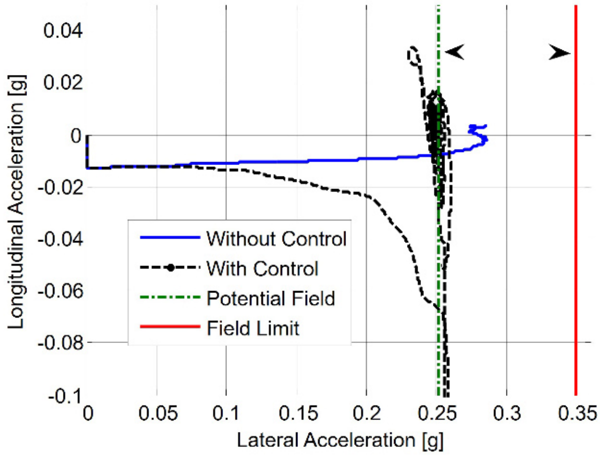

Fig. 9. g-g diagram for vehicle motion.

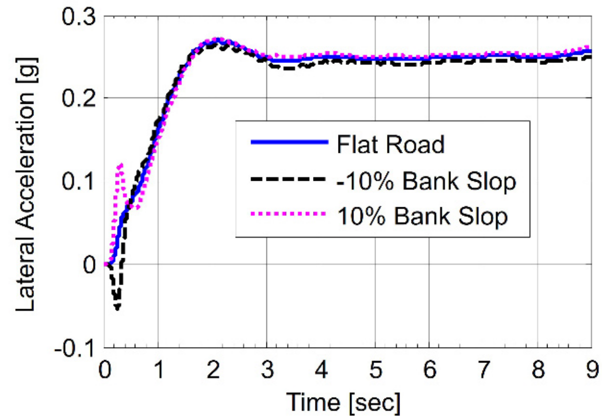

Fig. 12. Lateral acceleration for different road bank angles.

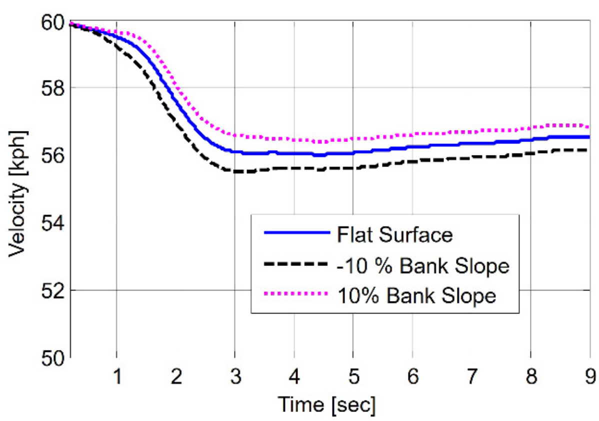

Fig. 10. Velocity profile on different road band angles.

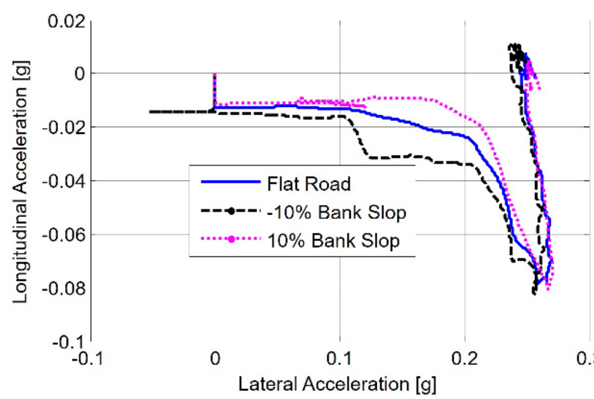

Fig. 13. g-g diagram for constant radius turn with different road bank angles.

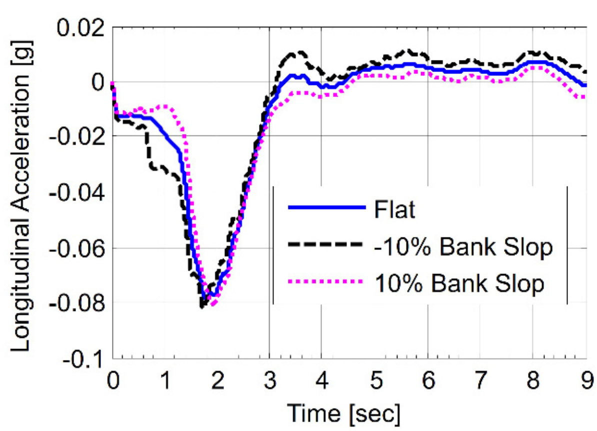

Fig. 11. Longitudinal acceleration for different road bank angles.

bounds while the vehicle is able to move at different longitudinal acceleration. Different road bank angles results in different longitudinal velocities for different road bank angles as shown in Fig. 10. This kind of behavior is required specifically when the vehicle is driving in autonnomous mode of operation to take into account no only the lateral acceleration but also road bank angles. The more the road bank angle is the slower will be the vehicle moving.

5.3. Double lane change at 60 kph

Double lane change maneuver is also performed at 60 kph for which the trajectory is shown in Fig. 14. The vehicle velocity is maintained at 60 kph in close loop manner. The vehicle velocity slightly lowers as the vehicle steers from one lane to another and the controller accelerates the vehicle to compensate this by accelerating the vehicle.

To check the effect of potential field function the controller is switched On and Off. For this maneuver the velocity and longitudinal acceleration are plotted in Fig. 15 and Fig. 16 respectively. As the potential field function is switched on the controller decelerates the vehicle to bring the lateral acceleration below the predefined value.

The lateral acceleration is ploted against the longitudi-

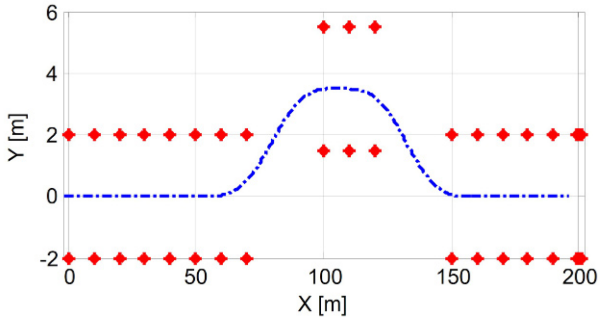

Fig. 14. Double lane change trajectory.

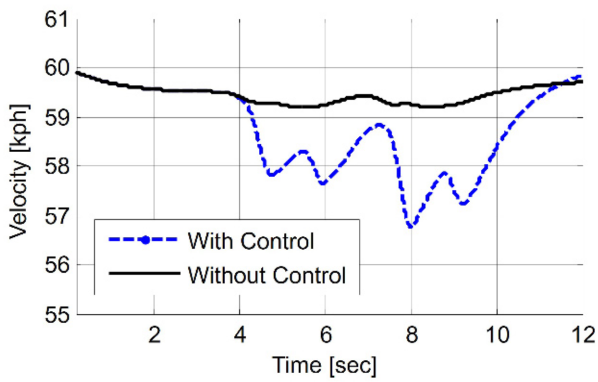

Fig. 15. Vehicle velocity for double lane change maneuver at 60 kph.

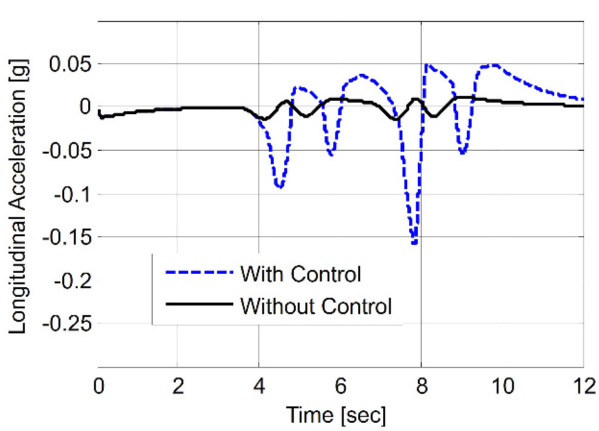

Fig. 16. Longitudinal acceleration for double lane change maneuver at 60 kph.

nal acceleration on a g-g diagram in Fig. 17. The affect of potential field function controller is very obvisous in this plot. The first plot in Fig. 17 is for vehicle acceleration and deceleration without any control. The second plot for this controller is implemented and the longitudinal acceleration is controlled in accordance with lateral acceleration. This action helps prevent tire force saturation in one direction using the friction circle theory concept. Sec-

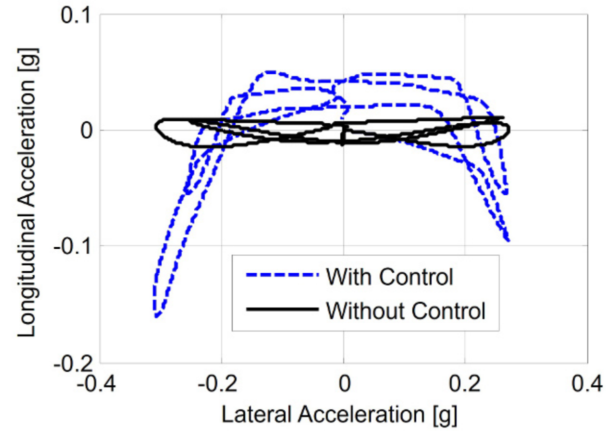

Fig. 17. g-g diagram for double lane change maneuver at 60 kph.

only using this approach the lateral acceleration can be kept under control by controlling the longitudinal acceleration using torque control. In other words by imposing the restriction on lateral acceleration and vehicle roll angles the workspace of vehicle states is contraints by bound that will ensure safety and stability for rollover motion.

6. CONCLUSION

In this study a new method of rollover stability improvement at autonomous mode is proposed which is based on potential field function control. This is particularly suitable for autonomous mode of driving as the controller takes into account the lateral acceleration and road bank angles to calculate the safe higher speeds of driving. The uppper bound on lateral acceleration is taken as an obstacle and potential field function is devised that calculates the safe acceleration that will keep the vehicle safe from rollover. In case of excessive lateal acceleration the controller decelerates the vehicle using torque control method. The controller first determine the forces and moments for desired motion from driver's commands and distributes the tire forces optimally to lower the load of each tire and prevent saturation of tire forces. The potential field function operates in closed loop by monitoring the lateral acceleration and vehicle roll angle and modifies the required longitudinal acceleration for safety. Several simulation is carried out to check the performance of this control concept. It is shown that the designer can use any value of lateral acceleration as a limiting value and the controller action will follow the scheme by maintaing this limit. More over along with the lateral acceleration the controller also takes into account the roll angles of vehicle for detecting the rollover propensity. This controller have the capability of following the exact desired trajectory. Thus this controller achives the rollover safety by limiting the workspace of vehicle.

APPENDIX A

T_m	Output torque of motor
T_{m_com}	Wheel torque command
L_m	Motor coil inductance
R_m	Motor coil resistance
K_t	Motor torque constant
s	Laplase operator
δ	Steering wheel angle
ϕ	Roll angle
T_i	Driving torque of each wheel ($i = 1 \sim 8$)
F_{xi}	Longitudinal force of each wheel ($i = 1 \sim 8$)
F_{yi}	Lateral force of each wheel ($i = 1 \sim 8$)
F_{zi}	Vertical force of each wheel ($i = 1 \sim 8$)
F_r, F_l	Force of right and left side wheels
F_{xdes}	Desired longitudinal force
M_{zdes}	Desired yaw moment
γ	Actual yaw rate from TruckSim
γ_{des}	Desired yaw rate
V_{x}	Longitudinal vehicle velocity
V_{a}	Actual vehicle velocity from TruckSim
K_P, K_I, K_D	PID controller gains
a_x, a_y	Longitudinal and lateral acceleration
$a_{y\_lift\_off}$	Lateral acceleration at which wheel lift off the ground
$a_{y\_\lim}$	Limiting value of lateral acceleration
a_{driver}	Driver's commanded longitudinal acceleration
a_{des}	Desired longitudinal acceleration
G_x	Acceleration by potential field function
m	Vehicle mass
μ	Friction coefficient
J	Cost function
r	Tire radius
RI	Rollover index
t	Tread
I_{xx}, I_{yy}, I_{zz}	Moment of inertia about x, y and z axis
h	Sprung mass height
g	Gravity

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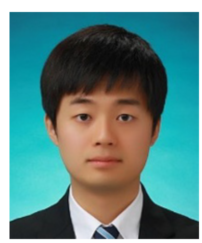

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