International Journal of Applied Engineering Research
ISSN 0973-4562 Volume 10, Number 7 (2015) pp. 16505-16520

http://www.ripublication.com

Investigations on Vehicle Rollover Prevention Using GA Tuned PID Controller

Binda.M.B.
Research Scholar, Dept of EEE
Sathyabama University., Chennai
bindamb@gmail.com
Dr.M.Rajaram
Vice Chancellor, Anna University
Chennai
rajaramgct@rediffmail.com

Abstract

In this paper, we present a novel methodology for preventing vehicle rollover model and control strategies which is suitable for preventing the untripped rollovers that was to be caused in vehicle. For vehicles that are deemed to be susceptible to wheel-lift off, various control strategies are implemented by earlier researchers. In this work, we propose a method for rollover prevention based on GA tuned PID controller which does not require such accurate contact information. The validity of the proposed methodology is proved, and it is used to realize rollover prevention in the direction of the roll. The primary assumption in this implementation is that the vehicle is equipped with a conventional PID controller system.

Keywords: Rollover, GA Tuned PID Controller, PID Controller

Nomenclature

Parameters	Definition
v_x	Longitudinal velocity (body-fixed frame)
ω_r	Yaw rate (angular rate about vertical axis)
m	Vehicle mass
I_{zz}	Inertia about the vertical axis
a	front-axle-to-CG distance
b	rear-axle-to-CG distance
L	Track of vehicle
t	Width of vehicle

β	Slip angle of the vehicle body
k_1	Front cornering stiffness
k_2	Rear cornering stiffness
δ_f	Front steering angle

Introduction

The increased popularity of vehicles with high center of gravity (center of gravity is hereafter denoted CG),such as SUVs(Sport Utility Vehicles),makes the prevention of rollovers an important safety issue. Rollover occurs when a vehicle flips over, and there are two types: tripped and untripped. Tripped rollover is the most common type, and occurs when the vehicle has started to skid, hits an obstacle, and finally flips over. Untripped rollover is induced by the driver, either during extreme maneuvers, or in panic situations.

The main advantage of our proposed approach is: purely based on the safety Concerns and Understanding the Physics behind Rollover that are occurred in SUV vehicles.

The aim for this work is to find a control system that can prevent untripped rollovers, with minimum trajectory deviation. Vehicle control is an active research field, both in the academic world and in the car industry, and much effort is spent on finding better and better solutions.

This paper deals with standardized SAE vehicle axis system & three-degree-of-freedom vehicle model. GA tuned PID controller system is presented and also explains the need of this controller. The results concerning the aim in this paper is to suggest a solution to the problem of an initial investigation into models and control strategies suitable to prevent vehicle rollover due to untripped driving maneuvers.

Related Works

Rollover prevention is a topical area of research in the automotive industry (see, for example, the rollover section at http://www.safercar.gov/ for a good introduction to the problem) and several studies have recently been published. Relevant publications include that of Palkovics et al. [8], where they proposed the ROP (Roll-Over Prevention) system for use in commercial trucks making use of the wheel slip difference on the two sides of the axles to estimate the tire lift-off prior to rollover.

Wielenga [9] suggested the ARB (Anti Roll Braking) system utilizing braking of the individual front wheel outside the turn or the full front axle instead of the full braking action. The suggested control system is based on lateral acceleration thresholds and/or tire lift-off sensors in the form of simple contact switches.

Chen et al. [10] suggested using an estimated TTR (Time To Rollover) metric as an early indicator for the rollover threat. When TTR is less than a certain preset threshold value for the particular vehicle under interest, they utilized differential braking to prevent rollover. A number of metrics based on geometric principles have been developed for stability measurement. Researchers in mobile robotics have recognized that the location of the vehicle centre of gravity (c.g.) relative to the wheel–terrain contact points is critical to vehicle stability. The stability polygon (or support pattern) is defined as the convex hull of the polygon formed by wheel–terrain contact points projected onto a horizontal plane [1]. An early geo-metric measure defined stable vehicle configurations as those where the horizontal projection of the vehicle c.g. lies within this polygon [2].

A stability margin was then defined based on the shortest distance from the projected c.g. to a side of the polygon. Improvements to this measure were proposed [3, 4]. However, this metric ignores the effects of changes in c.g. height and the destabilizing influence of vehicle dynamic effects. It should be noted that a common geometric stability metric used in vehicle design is the static stability factor, which is computed as the ratio of vehicle width to c.g. height [5].

To help address this public safety issue, impending vehicle rollover needs to be detected and mitigated by activating chassis control actuators based on the vehicle rollover conditions [6].Various mathematical representations of the human driver have models been developed by earlier researchers for the prediction and avoidance of rollover [7]. In [13] several methods for detecting vehicle rollover were tested. The purpose of the tests was to find a method for detecting impending vehicle rollover that will activate active chassis subsystems when needed and minimize unnecessary activations of the chassis control systems. First, estimates obtained from the use of single sensors were examined. A lateral acceleration sensor, roll rate sensor, and suspension relative position sensor were all used to obtain estimates of the vehicle roll angle.

Ackermann et al. and Odenthal et al. [11], [12] proposed a robust active steering controller, as well as a combination of active steering and emergency braking controllers. They utilized a continuous-time active steering controller based on roll rate measurement. They also suggested the use of a static Load Transfer Ratio (LTRs) which is based on lateral acceleration measurement; this was utilized as a criterion to activate the emergency steering and braking controllers.

Rollover Model

All numerical representations follow the standard SAE right-handed sign convention utilizing the simplified three-freedom vehicle model as depicted in Figure 1.

In Figure 1(a), x-y plane, F_{y1} and F_{y2} are the lateral forces over the front and rear tires respectively .The sideslip angle β and the roll angle ϕ are assumed to be negligible. The total forces and torques over the whole vehicle are given by

$$\begin{aligned} \sum F_y &= F_{y1}\cos\delta + F_{y2} = ma_y \\ \sum M_z &= aF_{y1}\cos\delta - bF_{y2} = I_z\dot{\omega}_r \end{aligned} \tag{1}$$

Here a_y is the lateral acceleration,$\dot{\omega}_r$ is the yaw acceleration, and I_z is the yaw moment of the vehicle inertia.

In addition, the lateral forces $F_{y1} = k_1\alpha_1$,$F_{y2} = k_2\alpha_2$,where k_1 and k_2 are the cornering stiffness of front and rear tires, respectively, α_1 and α_2 are the slip angles of front and rear tires, $\alpha_1 = (\beta+a\omega_r/v_x)-\delta$, $\alpha_2 = (\beta-b\omega_r/v_x)$, v_x is the longitudinal velocity, and ω_r is the yaw rate. Consequently, the following equation is derived:

$$a_y = \frac{(k_1+k_2)\beta}{m} + \frac{(ak_1-bk_2)\omega_r}{mv_x} - \frac{k_1\delta}{m}$$
$$\dot{\omega}_r = \frac{(ak_1-bk_2)\beta}{I_z} + \frac{(a^2k_1+b^2k_2)\omega_r}{I_z v_x} - \frac{ak_1\delta}{I_z} \quad (2)$$

In Figure 1(b), y-z plane, the total forces and torques over the whole vehicle are given by

$$\sum F_z = F_{n1} + F_{n2} - mg = 0$$
$$\sum M_x = mgh\sin\phi + ma_y(h_0+\Delta h - h) +$$
$$F_{n2}(\frac{T}{2}+\Delta T) - F_{n1}(\frac{T}{2}-\Delta T) = I_x\ddot{\phi} \quad (3)$$

Where $\ddot{\phi}$ is the body roll acceleration, h_0 is the height of the vehicle's center of gravity (CG) standing above the ground level, h is the distance between the vehicle CG and the assumed roll axis, T is the width of the vehicle track, and Δh and ΔT are the deformations of suspensions and tires. F_{n1} and F_{n2} are normal forces over the left and right wheels, and I_x is the roll moment of vehicle inertia.

The restoring force F_a shown in Figure 1(c) is related with the roll movement, $F_a = (k_\Phi\ \phi + c_\Phi \dot{\phi})/h$, where k_Φ and c_Φ are the total roll stiffness and roll damping of suspensions and $\dot{\phi}$ is the roll rate.

The total torques over the body with respect to the roll axis are given by

$$\sum M_{xs} = m_s a_y h + m_s g h \sin\phi - F_a h = I_{xs}\ddot{\phi} \quad (4)$$

where m_s is the sprung mass that represents the total vehicle mass m excluding the suspension and tire and I_{xs} is the roll moment for the inertia of the sprung mass.

Compared with the sprung mass, the unsprung mass is negligible. Assume that $m_s = m$ and $I_{xs} = I_x$, and establish (5) as follows:

$$\ddot{\phi} = \frac{mh(a_y + g\sin\phi)}{I_x} - \frac{k_\phi \phi}{I_x} - \frac{c_\phi \dot{\phi}}{I_x},$$
$$F_{n1} - F_{n2} = \frac{2ma_y(h_0+\Delta h - h)}{T} + \frac{2mg\Delta T}{T}$$
$$-\frac{2(k_\phi\phi + c_\phi\dot{\phi})}{T}, \quad (5)$$
$$F_{n1} + F_{n2} = mg,$$
$$LTR = (\frac{F_{n1}-F_{n2}}{F_{n1}+F_{n2}})$$

where LTR stands for the load transfer ratio, an important criteria to evaluate the vehicle status.

Since $a_y = \dot{v}_y + v_x\omega_r$, $\beta = v_y / v_x$, after introducing the state vector $x = [v_y \quad \omega_r \quad \dot{\phi} \quad \phi]^T$, the motions of this model can be described by

$$\dot{x} = Ax + B\delta \quad (6)$$

Where

$$A=\begin{bmatrix} \frac{k_1+k_2}{mv_x} & \frac{ak_1-bk_2}{mv_x}-v_x & 0 & 0 \\ \frac{ak_1-bk_2}{I_z v_x} & \frac{a^2k_1+b^2k_2}{I_z v_x} & 0 & 0 \\ \frac{k_1+k_2}{I_x v_x} & \frac{ak_1-bk_2}{I_x v_x} & \frac{-c_\phi}{I_x} & \frac{mgh-k_\phi}{I_x} \\ 0 & 0 & 1 & 0 \end{bmatrix} \tag{7}$$

$$B=\begin{bmatrix} \frac{-k_1}{m} & \frac{-ak_1}{I_z} & \frac{-k_1}{I_z} & 0 \end{bmatrix}^T$$

Rollover Prevention Method By Mitigation Control

The running vehicle intends to overshoot in its response to a suddenly applied lateral acceleration. Therefore, the inner wheels may have lifted off during the steering when the applied acceleration is lower than the nominal permit value that is in compliance with the steady-state laws .

From (5), we can see

$$\ddot{\phi}=\frac{mh(a_y+g\sin\phi)}{I_x}-\frac{k_\phi\phi}{I_x}-\frac{c_\phi\dot{\phi}}{I_x} \tag{8}$$

where the roll angle ϕ is small enough to get $\sin\phi \approx \phi$, and after applying Laplace law to the aforementioned differential equation, we established the following new equation:

$$\frac{\phi(s)}{a_y(s)}=\frac{(k_\phi-mgh)/I_x}{s^2+(c_\phi/I_x)s+(k_\phi-mgh)/I_x}\cdot\frac{mh/I_x}{(k_\phi-mgh)/I_x} \tag{9}$$

where the angular frequency is

$$\omega_n=\left[(k_\phi-mgh)/I_x\right]^{1/2} \tag{10}$$

and the damping ratio is

$$\xi=\frac{c_\phi}{2\left[(k_\phi-mgh)I_x\right]^{1/2}} \tag{11}$$

When the input a_y is set to be the unit step function, the overshoot is

$$M_P=\exp\left(-\frac{\xi\pi}{\sqrt{(1-\xi^2)}}\right) \tag{12}$$

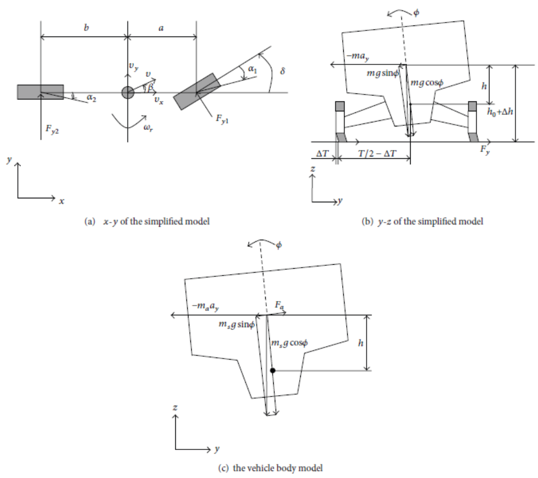

Figure 1: Simplified vehicle rollover model

At this point, we found that vehicle dynamics is determined by the first part of (9), whereas the second part defines the vehicle steady state. These findings form the base of the mitigation control referred to in the rest of this section.

A specific brand of midsize SUV was used as the test vehicle in this study, of which the relevant parameters are shown in Appendix.

With the rollover model and vehicle parameters, we achieved the damping ratio of the original vehicle, ξ = 0.71, and the overshoot value of the roll angle at 38.18% as shown in Figure 8. Thus, the damping ratio needs to be raised in order to reduce the overshoot value. In other words, we can improve the restoring force F_a by increasing the roll stiffness k_{Φ} or the roll damping c_{Φ} . However, it needs to bear in mind that the roll stiffness k_{Φ} and the roll damping c_{Φ} must be increased simultaneously. Otherwise, the damping ratio ξ may be decreased.

With the rollover model and vehicle parameters, we improve the damping ratio of the vehicle, ξ = 0.95, and the overshoot value of the roll angle at 0.71% as shown in Figure 9.

Rollover Prevention Method By PID Controller & Genetic Algorithm Tuned PID Controller

a) PID Controller

A PID is the most commonly used feedback controller. A PID controller calculates error value as the difference between a measured process variable and a desired set point. The controller attempts to minimize the error by adjusting the process control inputs. The PID controller calculation (algorithm) involves three term control: the proportional, integral and derivative values denoted by P, I and D. Where P depends on the present error, I on the accumulation of past errors, and D is prediction of future errors.

A PID controller will be called a PI, PD, P or I controller in the absence of the respective control actions. PI controllers are fairly common, eliminates the steady state error. A PID controller has proportional, integral and derivative terms that can be represented in transfer function form as

$$K(s)=K_p+K_i/s+K_d s \tag{13}$$

K_p: proportional gain, a tuning parameter,
K_i: Integral gain, a tuning parameter,
K_d: Derivative gain, a tuning parameter

By tuning these PID controller gains, the controller can provide control action designed for specific process requirements. The proportional term drives a change to the output that is proportional to the current error. This proportional term is concerned with the current state of the process variable. The integral term (K_i) is proportional to both the magnitude of the error and the duration of the error. It (when added to the proportional term) accelerates the movement of the process towards the set point and often eliminates the residual steady-state error that may occur with a proportional only controller.

Optimizing Of PID Controller

For the system under study, Zieger-Nichols tuning algorithm based on critical gain K_{er} and critical period P_{er} will be used. Here, the integral time T_i will be set to infinity and the derivative time T_d to zero. This is used to get the initial PID setting of the system [15].

In this method, only the proportional control action will be used. The K_p will be increase to a critical value K_{er} at which the system output will exhibit sustained oscillations. In this method, if the system output does not exhibit the sustained oscillations hence this method cannot be used.

Designing PID Parameters

From the response given below in Figure 2, the system under study is indeed oscillatory and hence the Z-N tuning rule based on critical gain K_{er} and critical period P_{er} can be applied.

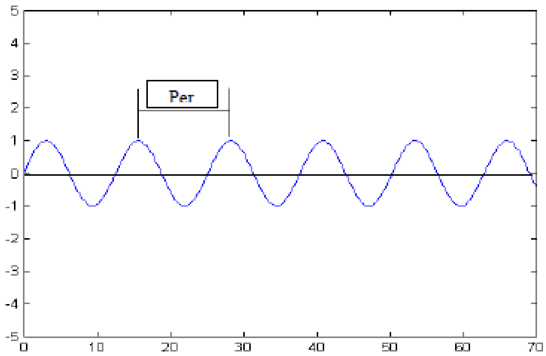

Figure 2: Illustration of Sustained Oscillation with Period P_{er}

The transfer function of the PID controller is

$$G_c (s) = K_p(1+ 1/sT_i + T_d s) \tag{14}$$

The objective is to achieve a unit-step response curve of the designed system that exhibits a maximum overshoot of 25 %. If the maximum overshoot is excessive says about greater than 40%, fine tuning should be done to reduce it to less than 25%.

Since the $T_i = \infty$ and $T_d = 0$, this can be reduced to the transfer function of

$$\frac{C(s)}{R(s)} = \frac{127.33K_p}{s^3 + 16.02s^2 + 127.33s + 127.33K_p} \tag{15}$$

The value of K_p that makes the system marginally stable so that sustained oscillation occurs can be obtained by using the Routh's stability criterion. Since the characteristic equation for the closed-loop system is

$$s^3 + 16.02s^2 + 127.33s + 127.33K_p = 0 \tag{16}$$

From the Routh's Stability Criterion, the value of K_p that makes the system marginally stable can be determined.

The sustained oscillation will occur if K_p =16.02.Hence the critical gain K_{er} is K_{er}=16.02.Thus with K_p set equal to K_{er}, the characteristic equation becomes

$$s^3 + 16.02s^2 + 127.33s + 2039.83 = 0 \tag{17}$$

The frequency of the sustained oscillation can be determined by substituting the s terms with $j\omega$ term. Hence the new equation becomes

$$(j\omega)^3 + 16.02 (j\omega)^2 + 127.33 (j\omega) + 2039.83 = 0 \tag{18}$$

From the above simplification, the sustained oscillation can be reduced to

$\omega^2 = 127.33$

or

$\omega = 11.28$ rad/sec

The period of the sustained oscillation can be calculated as

$P_{er} = 2\pi/11.28 = 0.557$

From Ziegler-Nichols frequency method of the second method ,the table suggested tuning rule according to the formula shown. From these we are able to estimate the parameters of K_p, T_i and T_d .

Table 1: Recommended PID value setting

Type of Controller	K_P	T_i	T_d
P	$0.5K_{er}$	∞	0
PI	$0.45K_{er}$	$(1/1.2)P_{er}$	0
PID	$0.6K_{er}$	$0.5P_{er}$	$0.125P_{er}$

Hence from the above Table 1, the values of the PID parameters K_p, T_i and T_d will be Kp = $0.6K_{er}$ =9.612,T_i = 0.5 X 0.557= 0.2785,T_d = 0.125 X 0.557= 0.0.06925.

b) Genetic Algorithm

Genetic Algorithms are search algorithms based on natural selection and natural genetics. They combine survival of fittest among structures with structured yet randomized information exchange to form a search algorithm. Since GA only requires a way to evaluate the performance of its solution guesses without any prior information, they can be applied generally to nearly any optimization problem. GA is usually extensively modified to suit a particular application. As a result, it is hard to classify a "generic" or "traditional" GA, since there are so many variants. An improvement to the "traditional" GA to provide faster and more efficient searches for GAS that does not rely on average chromosome convergence (i.e. applications which are only interested in the best solution). The "traditional" GA is composed of a fitness function, a selection technique, and crossover and mutation operators which are governed by fixed probabilities. These operations form a genetic loop as shown in Figure 3. Since the probabilities are constant, the average number of local and global searches in each generation is fixed. In this sense, the GA exhibits a fixed convergence rate and therefore will be referred to as the fixed-rate GA.

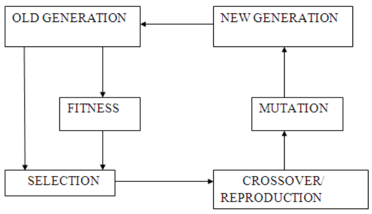

Figure 3: Block Diagram of GA

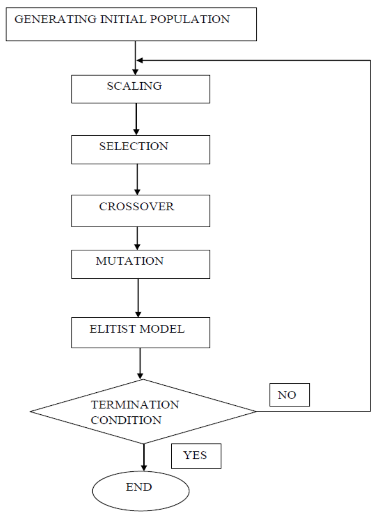

Figure 4: Flow Chart of Genetic Algorithm

A simple GA (Figure 4) consists of five steps:

- Start with a randomly generated population of N chromosomes, where N is the size of population, l – length of chromosome x.
- Calculate the fitness value of function φ(x) of each chromosome x in the population.
- Repeat until N offspring's are created:
- Replace current population with newly created one.
- Go to step 2

Designing Of PID Using Genetic Algorithm

GA can be applied to the tuning of PID controller gains to ensure optimal control performance at nominal operating conditions.

It is good to discuss the differences between Genetics Algorithm against the traditional methods. This will help us understand why GA is more efficient than the latter. Genetic algorithms are substantially different to the more traditional search and optimization techniques. The five main differences are:

1. Genetic algorithms search a population of points in parallel, not from a single point.
2. Genetic algorithms do not require derivative information or other auxiliary knowledge; only the objective function and corresponding fitness levels influence the direction of the search.
3. Genetic algorithms use probabilistic transition rules, not deterministic rules.
4. Genetic algorithms work on an encoding of a parameter set not the parameter set itself (except where real-valued individuals are used).
5. Genetic algorithms may provide a number of potential solutions to a given problem and the choice of the final is left up to the user.

Results of The Implemented Genetic Algorithm PID Controller

In the following section, the results of the implemented Genetic Algorithm PID Controller will be analyzed. The GA designed PID controller is initially initialized with population size of 5 and the response analyzed. It was then initialized with population size of 10, 15, 20 and 25. The response of the GA designed PID will then be analyzed for the smallest overshoot, fastest rise time and the fastest settling time. The best response will then be selected.

Table 2: PID controller gain values

Gain Coefficient	K_p	K_i	K_d
For a population 5	0.8271	0.5514	0.5998
For a population 10	0.703	0.8762	0.9398
For a population 15	0.2436	0.04516	0.953
For a population 20	0.3367	0.7222	0.8433

After giving the parameters as given in Table 3 to GA, PID controllers can be easily tuned and thus system performance can be improved and based on the results, the population size of 5 is selected.

Results of The Implemented Genetic Algorithm PID Controller
In the following section, the results of the implemented Genetic Algorithm tuned PID Controller will be analyzed. The GA designed PID controller is initially initialized with population size of 5 and the response analyzed. It was then initialized with population size of 10, 15, 20 and 25. The response of the GA designed PID will then be analyzed for the smallest overshoot, fastest rise time and the fastest settling time. From the following responses ie by comparing the graphs shown in figure 5 and figure 6, we can observe that GA designed PID will have better response compared to the Z-N tuned PID Method.

Table 3: Parameters of GA

GA property	Value/Method
Population Size	5
Maximum noof generations	4
Performance Index/Fitness Function	Mean Square error
Selection method	Normalized Geometric Selection
Probability of selection	0.05
Crossover Method	Arithmetic Crossover
Number of crossover points	3
Mutation Method	Uniform Mutation
Mutation Probability	0.1

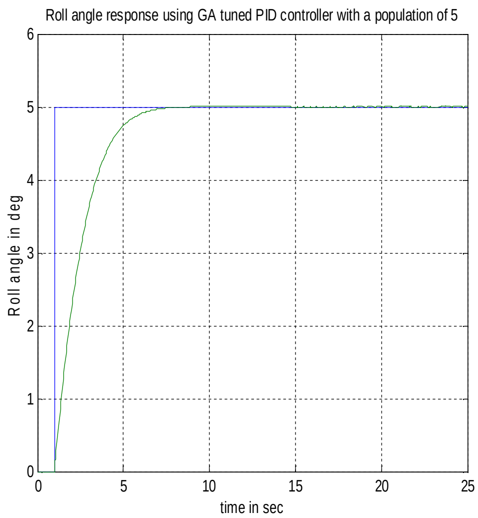

Figure 5: ϕ versus a_y transient response using GA

tuned PID controller with a population of 5

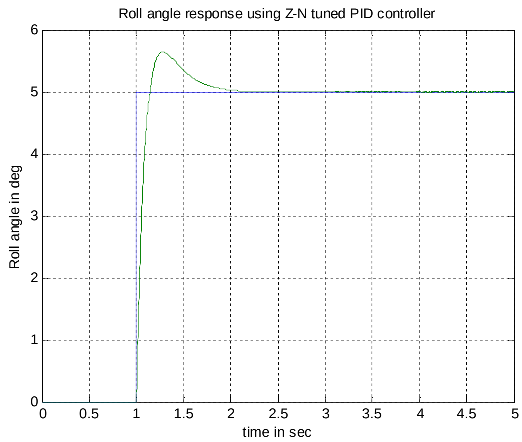

Figure 6: ϕ versus a_y error transient response for Z-N tuned PID controller

Simulation Results

In order to verify the proposed mitigation control, simulation tests are conducted according to NHTSA's Fishhook 1a test description, and the steering maneuver patterns are shown in Figure 10.

Note that the wild fluctuations occur when the hand wheel angle rate changes rapidly, especially within the time ranges of [2s, 3s] and [4s, 6s].

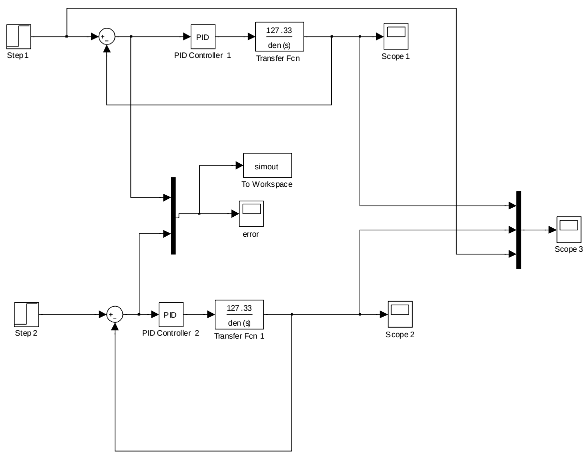

Figure 7: Simulation Diagram for Roll Model

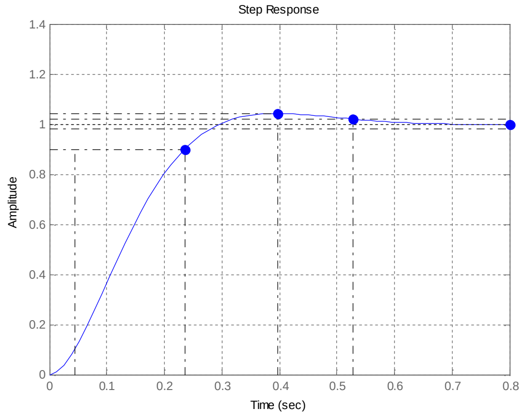

Figure 8: ϕ versus a_y transient response for ξ=0.71

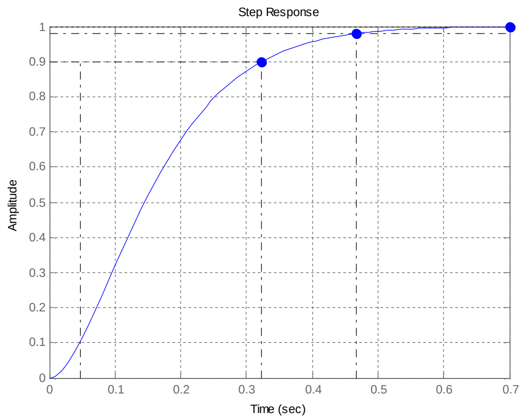

Figure 9: ϕ versus a_y transient response for ξ=0.95

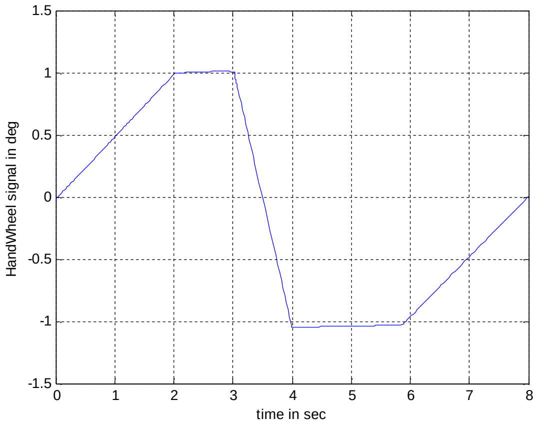

Figure 10: Fishhook 1a maneuver

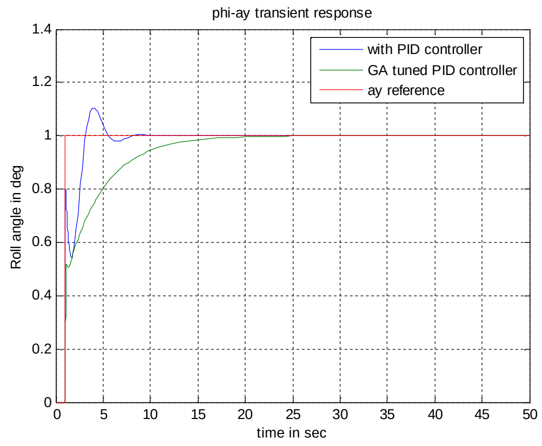

Figure 11: ϕ versus a_y transient response for different controllers

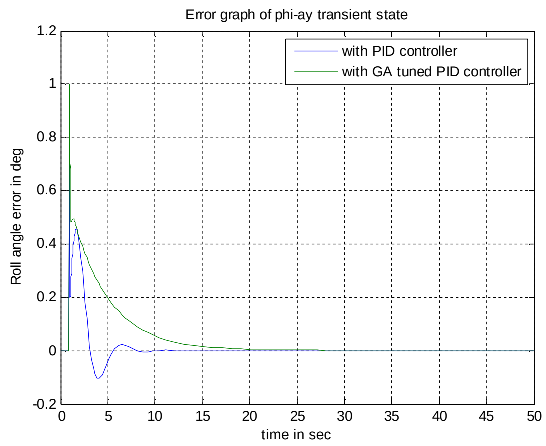

Figure 12: ϕ versus a_y error transient response for different controllers

Table 4: Results of Z-N tuned PID controller & GA tuned PID controller for a 3 DOF vehicle model

Measuring factors	Z-N tuned PID controller	GA tuned PID controller
Rise Time in sec	1.1	5
Maximum Overshoot in %	12.5	NA
Settling Time in sec	2.2	7

Conclusion

The main purpose of this research work is to propose a novel methodology for preventing vehicle wheel-lift prior to sliding. Additionally, multiple control strategies were presented in order to mitigate the vehicle rollover. To investigate the vehicle transient and steady states, an improved rollover model using GA tuned PID controller was established in this study. Simulation results indicated a decreased overshoot of the roll angle and a better confined steady value.

The adjusting method proposed in this paper may help the design of both passive and active suspension controls to increase vehicle stability. We are confident that, with additional studies, the proposed model will be applicable for a real vehicle in the near future.

From the Table 4 ,we can conclude that GA designed PID is much better since it has no overshoot but it suffers in terms of rise time and settling time.

References

[1] R.B. McGhee and A.A. Frank, On the stability properties of quadruped creeping gait, Math.Biosci. 3(2) (1968), pp. 331–351.

[2] R.B. McGhee and G.I. Iswandhi, Adaptive locomotion of a multilegged robot over rough terrain, IEEE Trans. On Syst. Man Cybernet. SMC-9(4) (1979), pp. 176–182.

[3] S. Sugano, Q. Huang, and I. Kato, Stability criteria in controlling mobile robotic systems, IEEE/RSJ International Workshop on Intelligent Robots and Systems, Yokohama, Japan, July 1993, pp. 832–838.

[4] J.K. Davidson and G. Schweitzer, A mechanics based computer algorithm for displaying the margin of static stability in four-legged vehicles, Trans. ASME J. Mech. Design 112 (1990), pp. 480–487.

[5] J.P. Chrstos and D.A. Guenther, The measurement of static rollover metrics, SAE Transactions no. 920582, 1992.

[6] D. W. Shuttlewood, D. A. Crolla, and R. S. Sharp, “Active roll control for passenger cars,” *Vehicle Syst. Dyn.*, vol. 22, no. 5/6, pp. 383–396, 1993.

[7] V. Cherian, R. Shenoy, A. Stothert, J. Shriver, J.Ghidella, and T. D.Gillespie, “Model-Based Besign of a SUV anti-rollover control system,” SAE Paper No. 2008-01-0579, 2008.

[8] Palkovics L., Semsey A` and Gerum E., “Rollover prevention system for commercial vehicles-additional sensorless function of the electronic brake system”, Vehicle System Dynamics, Vol. 4, pp. 285-297, 1999.

[9] Wielenga T.J., “A method for reducing on-road rollovers: anti-rollover braking”, SAE Paper No. 1999-01-0123, 1999.

[10] Chen B. and Peng H., “Differential-brakingbased rollover prevention for sport utility vehicles with humanin- the-loop evaluations”, Vehicle System Dynamics, Vol.36, pp. 359-389, 2001.

[11] Ackermann J. and Odenthal D., “Robust steering control for active rollover avoidance of vehicles with elevated center of gravity”, Proceedings of International Conference on Advances in Vehicle Control and Safety, Amiens, France, 1998.

[12] Odenthal D., B¨unte T. and Ackermann J., “Nonlinear steering and braking control for vehicle rollover avoidance”, Proceedings of European Control Conference, Karlsruhe, Germany, 1999.
[13] Aleksander Hac, T.B., and John Martens, Detection of Vehicle Rollover. SAE,2004(SAE Paper No. 2004-01-1757).
[14] Binda. M.B, Dr. M. Rajaram, “Intelligent Prediction And Prevention Of Vehicle Rollover Using FLQG Regulator”, in Journal of Theoretical and Applied Information Technology,(JATIT),20th November 2014, Volume. 69, No.2, (ISSN: 1992-8645) .
[15] Binda. M. B, Dr. M. Rajaram, “Investigations on vehicle Rollover Prevention using LQG Regulator ”, in Advances in Mathematical Physics, Hindawi Publishing Corporation, Volume 2014, Article ID 308285, 11 pages,(ISSN:1687-9120), December 2014 .
[16] Binda. M. B, Dr. M. Rajaram, “Intelligent Prediction And Prevention Of Vehicle Rollover Using NNLQG Regulator”, in International Journal of Electrical & Electronic Engg &Telecommunications, Volume. 4, No. 1, (ISSN: 2319-2518), January 2015.

Appendix

Parameters	Values
m	1030 Kg
I_{yy}	1705 Kg-m^2
m_s	825 Kg
I_{xx}	375 Kg-m^2
I_{xz}	72 Kg-m^2
I_{zz}	1850 Kg-m^2
a	0.93 m
b	1.56 m
L	1.4 m
h	0.52 m
k_1	-45500 N/rad
k_2	-76650 N/rad
K_φ	53000 N-m/rad
C_φ	6000 N-m-sec/rad^2