

VISION BASED AUTOMATIC STEERING CONTROL USING A PID CONTROLLER

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Abstract. Automatic steering control is a vital component of highway automation, currently investigated worldwide in several Intelligent Transportation Systems (ITS) programs. The promise of Intelligent Transportation System lies in the possibility of increasing the capacity of existing highways by safer and more efficient use of available space. This system will include completely “hands-off” driving in which vehicles are fully automatically controlled once they enter the system. In order to achieve the above objective, the Proportional-Integral-Derivative (PID) controller and vision based concept to an automatic steering control system is used to cause the vehicle to track the reference under various conditions. Simulation results show that the proposed control system achieved its objective even though it is less robust in maintaining its performance under various conditions.

Keywords: Automatic steering control, vehicle dynamics, vision system, PID controller

Abstrak. Kawalan stereng automatik adalah satu komponen penting dalam pengautomatan lebuhraya, yang kini diselidik di seluruh dunia di bawah beberapa program *Intelligent Transportation System* (ITS). ITS berpotensi meningkatkan kapasiti lebuhraya yang sedia ada dengan penggunaan yang lebih selamat dan lebih efisien terhadap ruang yang sedia ada. Sistem ini akan terdiri daripada konsep pemanduan “hands-off” secara menyeluruh yang mana kenderaan akan dikawal secara automatik apabila ia memasuki sistem itu. Untuk mencapai objektif di atas, pengawal PID dan konsep dasar penglihatan ke atas sistem kawalan stereng automatik digunakan untuk membolehkan kenderaan menjejak rujukan di dalam pelbagai keadaan. Keputusan simulasi menunjukkan bahawa sistem kawalan yang dicadangkan mencapai objektifnya meskipun ia kurang lasak untuk mengekalkan prestasinya di dalam pelbagai keadaan.

Kata kunci: Kawalan stereng automatik, dinamik kenderaan, sistem penglihatan, kawalan PID

1.0 INTRODUCTION

Automatic steering control is a vital component of highway automation, investigated worldwide in several programs. Overviews of highway automation were given by Bender [1] and Shladover [2]. Automatic steering control of vehicle has been studied since late 1950's [3]. Automatic steering control problem requires addressing two parts: sensing and control. Automatic steering control approaches can be grouped

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into look-ahead and look-down systems. Look-ahead systems replicate human driving behavior by measuring the lateral error ahead of the vehicle. A number of research groups have successfully conducted highway speed experiments with look-ahead systems like machine vision [4] and radar reflective stripes [5]. Another approach is the look-down system which measures the lateral displacement at a location within or in the close vicinity of vehicle boundaries, typically straight down the front bumper. However, despite an impressive amount of literature on control design based on look-down systems [6-9], no highway speed experiments has been reported. This means that look down reference system is limited to low speed [10]. From the point of view of control, several methods have been applied to achieve the purpose of automatic steering control to cause vehicle to track along a reference path with small error automatically [11-17]. A simple Proportional-Integral-Derivative (PID) controller based on the output from a look-down system has been suggested since 1990 [11]. An experiment study on this PID controller was done in 1991 [18]. The researchers concluded that the implementation of a PID with feedforward control action based on look-down approach showed good performance in terms of tracking under nominal conditions. However, the limitations of such a controller were observed as nominal parameters deviated from their nominal values. In 1996, Kosecka [19] demonstrated that with a proportional controller only; by using look-ahead system, the vehicle can reliably track changes in curvature up to 0.004 m^{-1} with velocity up to 28 ms^{-1} . In an attempt to extend the research in this area, this paper proposes a more practical vision dynamics and introduces a PID controller with look-ahead system to vision based automatic steering control systems.

2.0 MATHEMATICAL MODELING

Vehicle model has been studied since late 1950's. Vehicle models with 6 degree of freedoms were developed by Lugner [20], and Peng and Tomizuka [21] to represent the vehicle as realistically as possible. This model is complex and not suitable for controller design. According to Peng and Tomizuka [11], a linear and simplified model which retains only the lateral and yaw motion dynamics has been proven sufficient for studying car steering under nominal conditions, i.e. non-emergency situation.

2.1 Vehicle Model

The simplified two degrees of freedom linear vehicle model is obtained from the complex vehicle model by considering the vehicle is not accelerating along the longitudinal axis and assuming that there are no external forces acting on the vehicle. The linear vehicle model can be expressed as follows [11]:

2.2 Vision Dynamics

The model described by Equation (1) is independent of the road reference. To describe the vehicle relative to the road, the vision dynamics of the system is modeled using visual information and road geometry. Figure 1 shows a vehicle tracking a desired path with road curvature K_L , where $K_L = 1/R_L$. R_L is a desired radius of road with instant centre. In this paper, the vehicle is assumed to be equipped with a set of vision system. The vision system is used to estimate the offset from the centerline y_L and angle between the road tangent and heading of the vehicle ε_L at some look – ahead distance L where measurement is taken. The values of y_L and ε_L are used to estimate lateral error y_v at some look – ahead distance L .

From Figure 1, it can be seen that:

$$y_L = y_v + y_o \quad (2)$$

where y_L is the actual lateral error at L and y_o is the offset because of the road curvature. Referring to Figure 1 and assuming h_ϕ is small, the lateral error y_v can be expressed as:

$$y_v = y_{cg} + Lh_\phi \quad (3)$$

Also from Figure 1, the rate of y_{cg} is

$$\dot{y}_{cg} = V \sin(\beta - h_\phi) \approx V(\beta - h_\phi) = V\beta - Vh_\phi \quad (4)$$

If a set of angles β , ϕ , and δ_f are also assumed to be small, then the following variables can be defined,

$$V_x = V \cos(\beta) = V \quad (5)$$

$$V_y = V \sin(\beta) = V\beta \quad (6)$$

Substituting Equations (5) and (6) into (4),

$$\dot{y}_{cg} = V_y - V_x h_\phi \quad (7)$$

Then, the yaw angle ϕ of the vehicle body with respect to the arbitrary reference direction is

$$\phi = \phi_d - h_\phi \quad (8)$$

where ϕ_d is the desired yaw angle imposed by the road curvature. The yaw rate relative to the inertial coordinate system is given by:

$$\dot{\phi} = \dot{\phi}_d - \dot{h}_\phi \quad (9)$$

where $\dot{\phi}_d = VK_L = V_x K_L$ is a constant value since road curvature K_L and longitudinal velocity V_x are assumed to be constant.

Differentiating Equation (3) and substituting Equations (7) and (9) into it, the lateral velocity at the look-ahead distance L is given by:

$$\dot{y}_v = \dot{y}_{cg} + L\dot{\phi} = V_x h_\phi + LV_x K_L - L\dot{\phi} \quad (10)$$

Then, the vision dynamics which describe the motion of vehicle relative to the road can be represented as:

$$\dot{y}_v = V_y - V_x h_\phi + LV_x K_L - L\dot{\phi} \quad (11)$$

$$\dot{h}_\phi = V_x K_L - \dot{\phi} \quad (12)$$

2.3 Complete Model

The complete model consists of the vehicle model of Equation (1) and the vision dynamics of Equations (11) and (12) which can be represented as:

$$\dot{x} = Ax + Bu + Ew \quad (13)$$

$$y = Cx$$

where $x = [V_y \quad \dot{\phi} \quad y_v \quad h_\phi]^T$, $u = \delta_f$, $w = K_L$ and $y = [y_v \quad h_\phi]^T$;

$$A = \begin{bmatrix} -\frac{C_f + C_r}{mV_x} & -V_x + \frac{C_r l_r - C_f l_f}{mV_x} & 0 & 0 \\ -\frac{l_f C_f - l_r C_r}{I_Z V_x} & -\frac{l_f^2 C_f + l_r^2 C_r}{I_Z V_x} & 0 & 0 \\ 1 & -L & 0 & -V_x \\ 0 & -1 & 0 & 0 \end{bmatrix}; \quad B = \begin{bmatrix} \frac{C_f}{m} \\ l_f C_f \\ I_Z \\ 0 \\ 0 \end{bmatrix}; \quad E = \begin{bmatrix} 0 \\ 0 \\ LV_x \\ V_x \end{bmatrix}$$

$$C = \begin{bmatrix} 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix}.$$

The curvature K_L enters the model as an exogenous disturbance signal.

3.0 OPEN LOOP SYSTEM ANALYSIS

In this paper, we derive the controller based on a single-input single output (SISO) system. This can be done by realizing the complete system of Equation (13) and taking δ_f as the input to the system and y_v as the output from the system. From Figure 2, the open-loop transfer function with $K_L = 0$ is

$$V(s) = \frac{y_v(s)}{\delta_f(s)} = \frac{N(s)}{P(s)}; \quad (14)$$

where

$$\begin{aligned}
 N(\mathbf{s}) &= \mathbf{s}^2 V_x^2 C_f I_Z + \mathbf{s} V_x C_r C_f (l_f l_r + l_r^2) + C_r C_f V_x^2 (l_f + l_r) \\
 &\quad + L (\mathbf{s}_2 V_x^2 C_f l_f m + \mathbf{s} V_x C_r C_f (l_f + l_r)); \\
 P(\mathbf{s}) &= \mathbf{s}^2 (\mathbf{s}^2 m V_x^2 I_Z + \mathbf{s} V_x (I_Z (C_f + C_r) + m (C_f l_f^2 + C_r l_r^2))) \\
 &\quad + m V_x (C_r l_r - C_f l_f) + C_f C_r (l_f + l_r)^2.
 \end{aligned}$$

The open-loop transfer function with $\delta_f = 0$ is

$$D(s) = \frac{y_v(s)}{K_L(s)} = -\frac{V_x^2}{s^2} \quad (15)$$

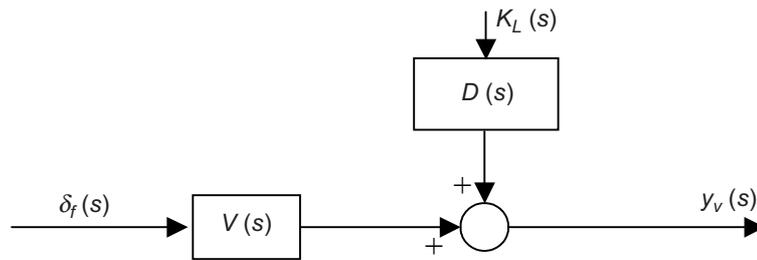


Figure 2 Block diagram for open loop automatic steering control system

The core of the open-loop system analysis lies in understanding the behaviors of the vehicle under various conditions. Notice that the transfer function $V(s)$ from Equation (14) has a pair of poles fixed at the origin, and another two poles and two zeros which characterise the behavior of the vehicle. The poles of $V(s)$ is also a function of longitudinal velocity, V_x and the zeros of $V(s)$ is a function of both longitudinal velocity V_x and the look-ahead distance L . The open-loop analysis of Equation (14) is done using root-locus technique. The parameters for the vehicle are given in Table 1.

Table 1 Parameters of vehicle

Total mass of the vehicle, m	1590 kg
Total inertia of vehicle around the CG, I_Z	2920 kg m ²
Distance of the front axle from the CG, l_f	1.22 m
Distance of the rear axle from the CG, l_r	1.62 m
Cornering stiffness of the front tires, C_f	2×60000 N/rad
Cornering stiffness of the rear tires, C_r	2×60000 N/rad

The root loci of the transfer function $V(s)$ with various vehicle parameters are presented in Figures 3 to 6. From Figure 3, notice that as the longitudinal velocity V_x increases from 5 to 30 m/s, the poles and zeros of the system move towards the right half plane. As a conclusion, the damping ratio, ζ of the system is decreasing as the longitudinal velocity V_x is increasing. The system becomes less stable and exhibit significant oscillatory behavior for higher longitudinal velocity V_x . Additionally, the response of the system become slower as the longitudinal velocity V_x is increasing.

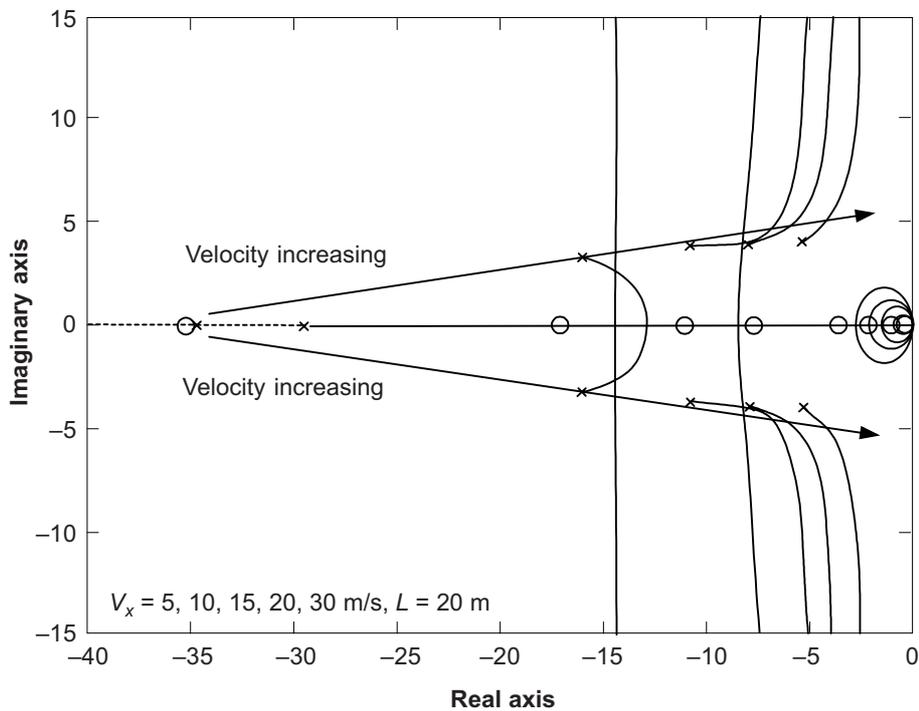


Figure 3 Root locus for $V(s)$ with $V_x = 5, 10, 15, 20,$ and 30 m/s with $L = 20$ m

From Figure 4, notice that as the look-ahead distance L increases, the zeros of the transfer function $V(s)$ move closer to the real axis which will improve the damping of the system. It can be seen that the location of poles is unaffected by the look-ahead distance L . As a conclusion, increasing the look-ahead distance L can provide sufficiently good damping to the system by increasing the damping ratio and reducing percentage overshoot. Therefore, the selection of a proper look-ahead distance is important for stability and performance of the system.

Vehicle's load changes as the number of passengers vary. From Figure 5, notice that as the vehicle's load is increasing from 1590 to 2090 kg, the system tends to overshoot and oscillate significantly.

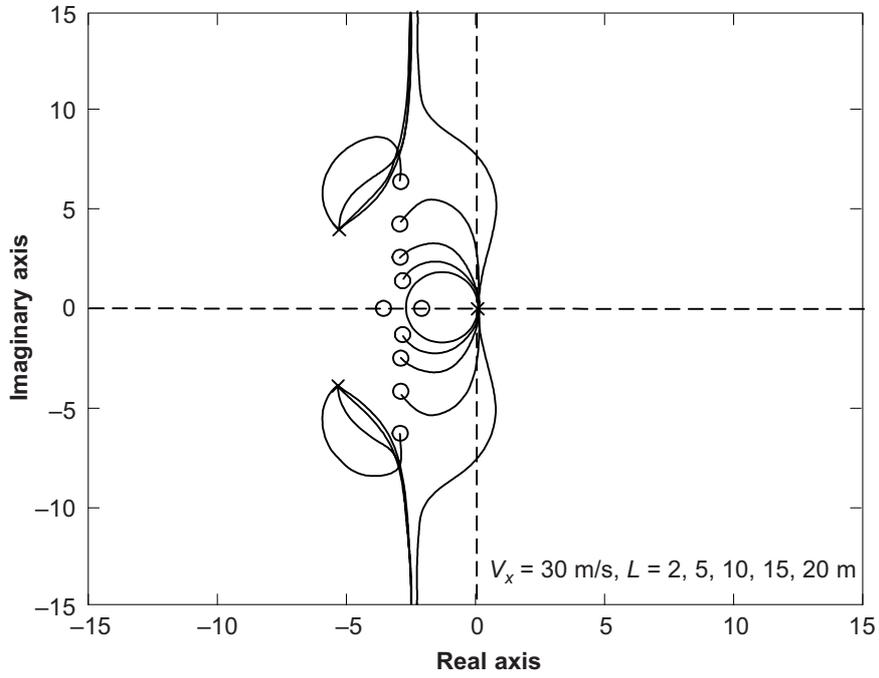


Figure 4 Root locus for $V(s)$ with $L = 2, 5, 10, 15, 20 \text{ m}$ at $V_x = 30 \text{ m/s}$

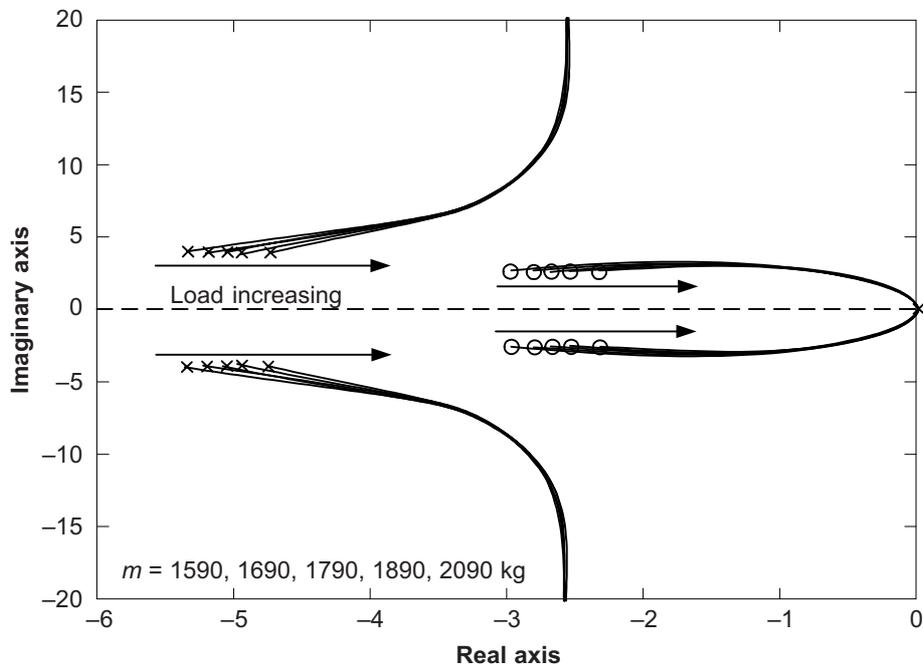


Figure 5 Root locus of $V(s)$ with $L = 10 \text{ m}$ at $V_x = 30 \text{ m/s}$ under various loads

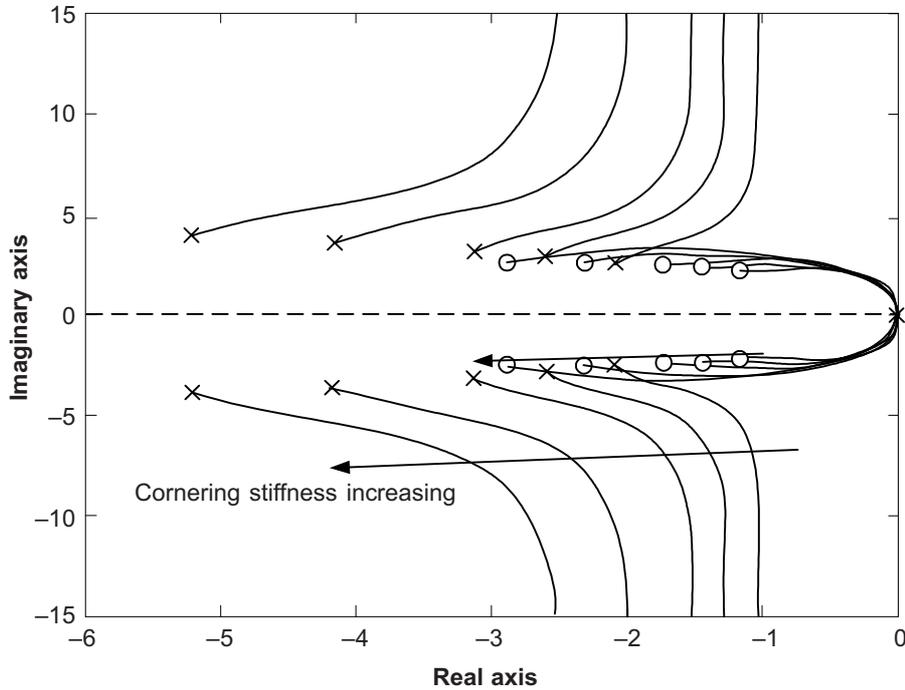


Figure 6 Root locus of $V(s)$ with $L = 10$ m at $V_x = 30$ m/s under various cornering stiffness

From Figure 6, notice that as the cornering stiffness of the vehicle is decreasing, the damping ratio of the system decreases as well. As a conclusion, the system tends to overshoot and oscillate more significantly at a lower cornering stiffness and higher vehicle load conditions. Therefore, robustness is another important design requirement for an automatic steering control system.

4.0 CONTROLLER DESIGN

From the open-loop vehicle model of Equation (13) and disturbance model of Equation (14), the closed-loop system is subjected to one input (steering angle) and another uncontrollable disturbance (road curvature). Subsequently, the automatic steering control problem is to regulate the lateral error y_v at some look-ahead distance L equal to zero for any road curvature under various conditions. The selection of a proper look-ahead distance L is crucial for stability and performance of the system and it is made based on root locus analysis as has been shown in the previous section. The output y_v of the system can achieve zero steady state error with a step disturbance A/s^3 , where $A = -V_x^2 K_L$ by introducing a pure integral into the system. A PID controller is designed by adding one zero and pole as the ideal integral compensator. Meanwhile, another additional zero is added as the ideal derivative

compensator. Using the root locus technique, controller gains and proper location of the compensator poles and zeros can be specified and fine tuning in order to achieve lower-order approximation by zero-pole cancellation. Figure 7 shows the pole-zero plot for the closed-loop system with a proper tuned PID controller. From Figure 7, notice that three pairs of the pole and zero cancelled each other out. Only one pole on real axis is left alone and dominates the response of the closed-loop system.

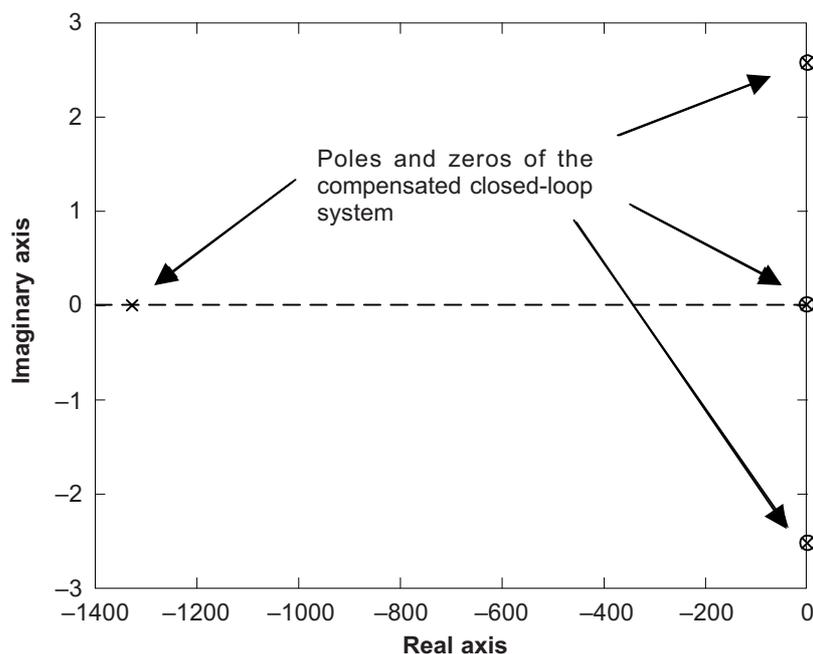


Figure 7 Pole-zero plot for closed-loop system with PID controller

5.0 SIMULATION

The new proposed automatic steering control has been tested in simulation using Equations (13) and (14) and the proposed PID controller. Two different analyses were performed: performance and robustness analysis. The simulation block diagram is shown in Figure 8.

The profile of the road curvature used in the simulation is shown in Figure 9. The road profile begins with a straight road segment to analyse the performance of the system to track a straight road. Then, there is a curvature from $K_L = 0$ to -0.002 m^{-1} to analyze the performance of the system to track a curve road under nominal conditions. Following that is a curvature from the left with a maximum

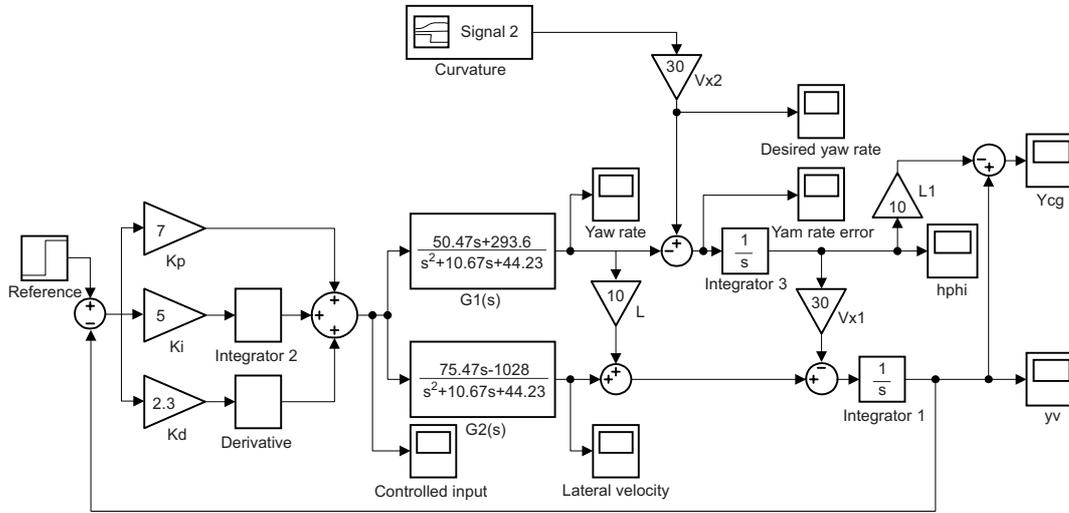


Figure 8 SIMULINK model used in simulation for performance and robustness analysis

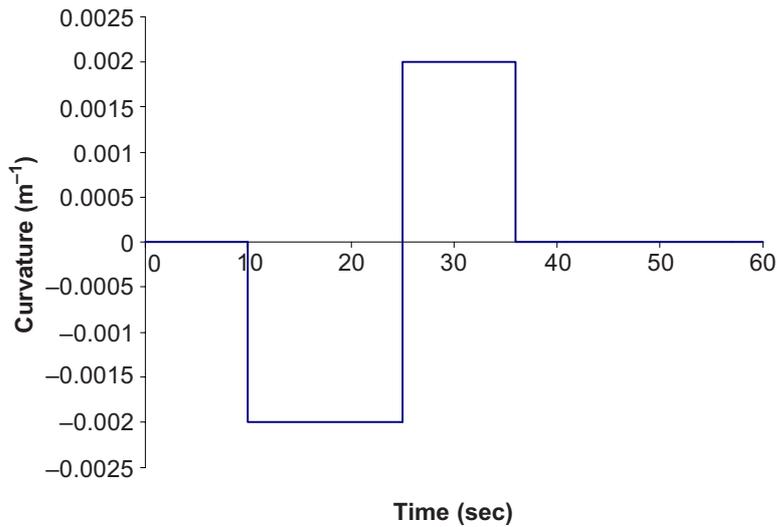


Figure 9 Profile of the road curvature

curvature of -0.002 m^{-1} to the right with a maximum curvature of 0.002 m^{-1} without intermediate straight segments to analyze the performance of the system to track a curve under extreme condition. Finally, the curvature is reduced to 0 m^{-1} to analyse the performance of system to track a straight road when exiting from a curve segment.

5.1 Performance Analysis

The performance is analyzed by simulating the closed-loop system with longitudinal velocity $V_x = 15, 20, 30 \text{ ms}^{-1}$ and two different values of look-ahead distance L for comparison. From Figures 10 to 14, it is found that the controller is able to compensate the yaw rate error of the system and make the lateral error of the vehicle remains null for various road curvature under various conditions. However, the response of the vehicle tends to overshoot as the longitudinal velocity V_x is increased as shown in Figures 10 to 15. Additionally, the system with a smaller look-ahead distance L , the lateral error at the center of gravity y_{cg} of the vehicle improved (decreased) significantly as shown in Figures 12 and 15. However, for a system with high longitudinal velocity V_x but small look-ahead distance; the response of the system tends to overshoot extremely. This may caused passengers to be uncomfortable. As a conclusion, the look-ahead distance, L should be adjusted according to the longitudinal velocity, V_x of the vehicle in order to get a better performance.

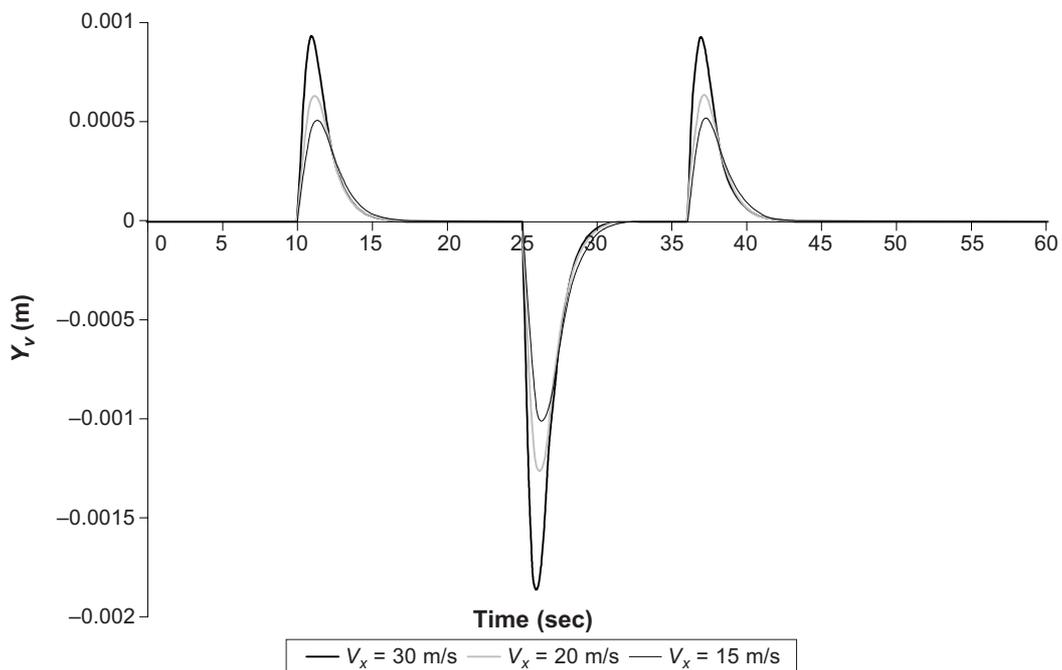


Figure 10 Simulation result of y_v with $L = 10 \text{ m}$

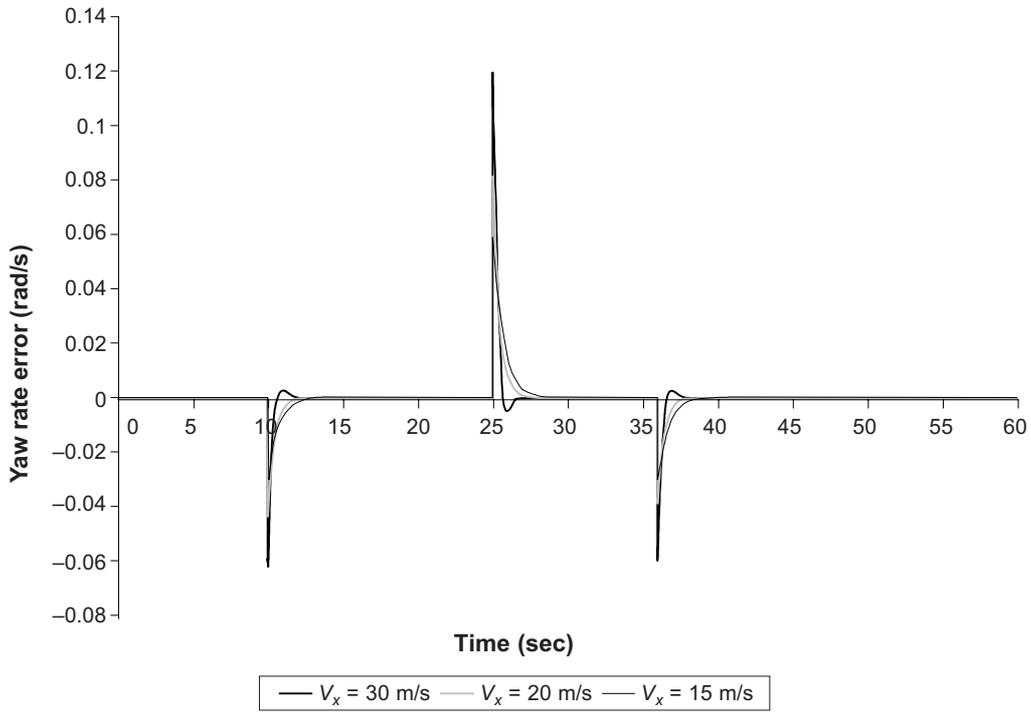


Figure 11 Simulation result of yaw rate error with $L = 10$ m

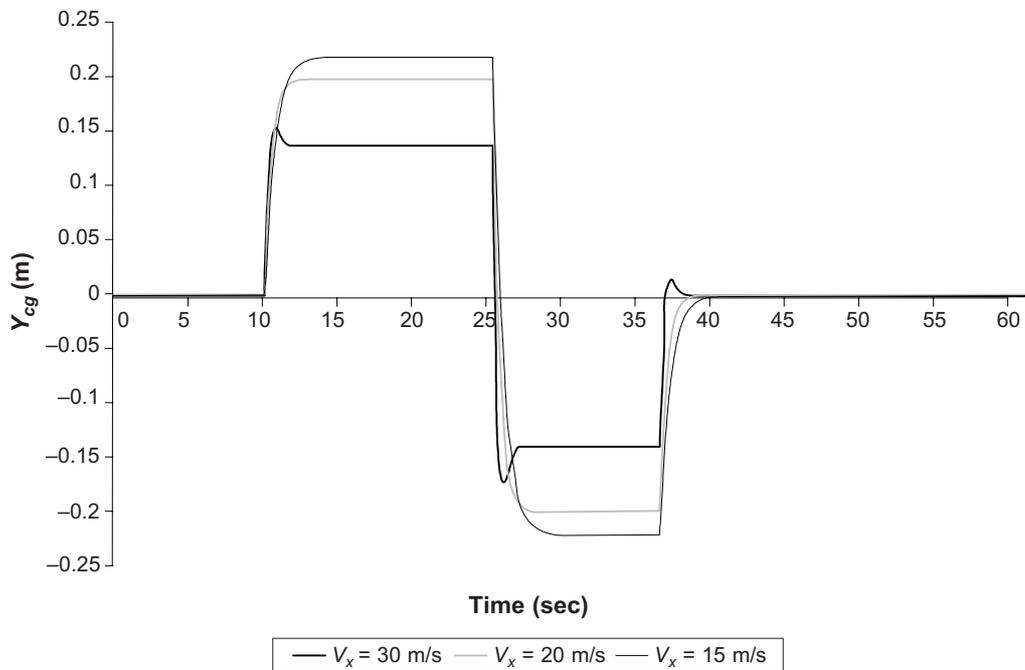


Figure 12 Simulation result of y_{cg} with $L = 10$ m

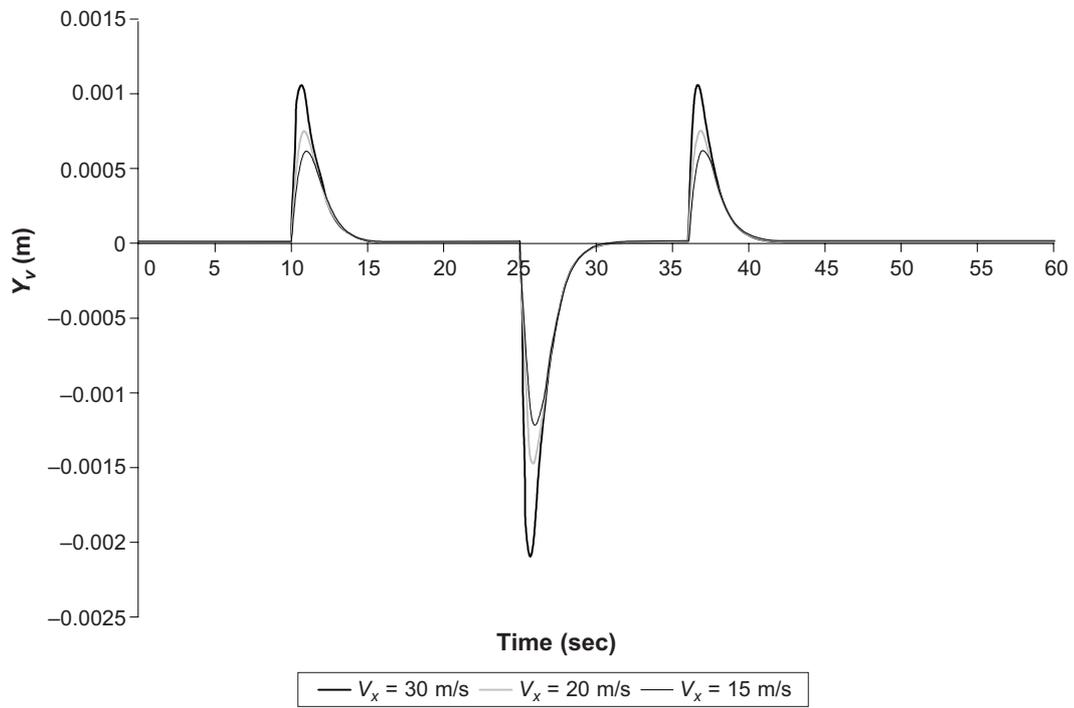


Figure 13 Simulation result of y_v with $L = 5$ m

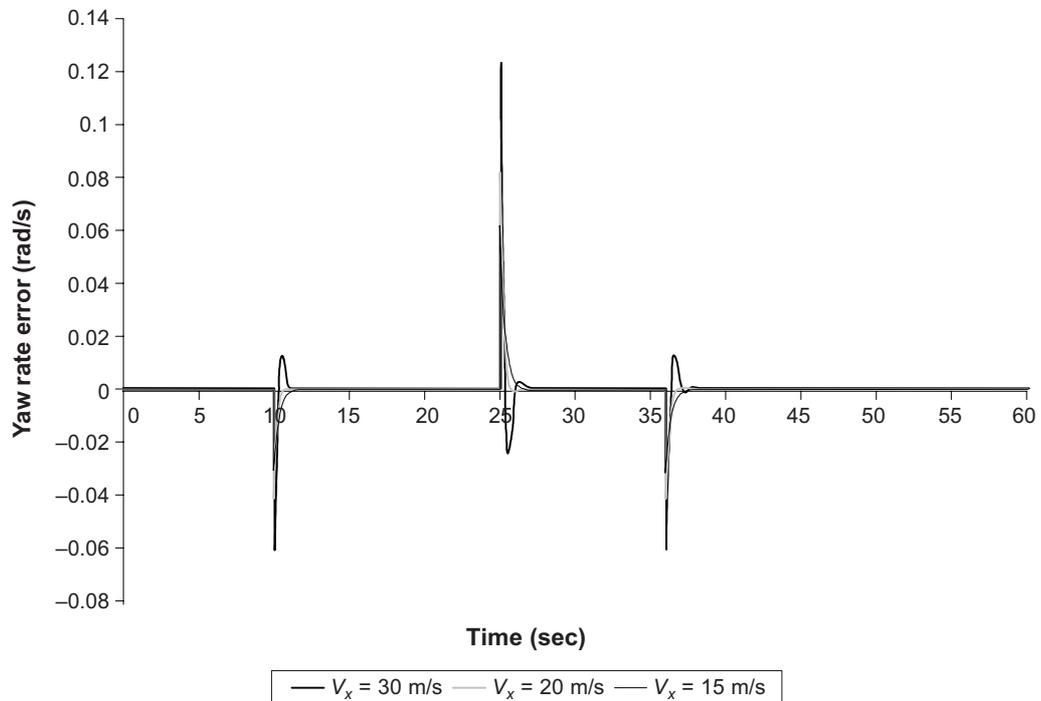


Figure 14 Simulation result of yaw rate error with $L = 5$ m

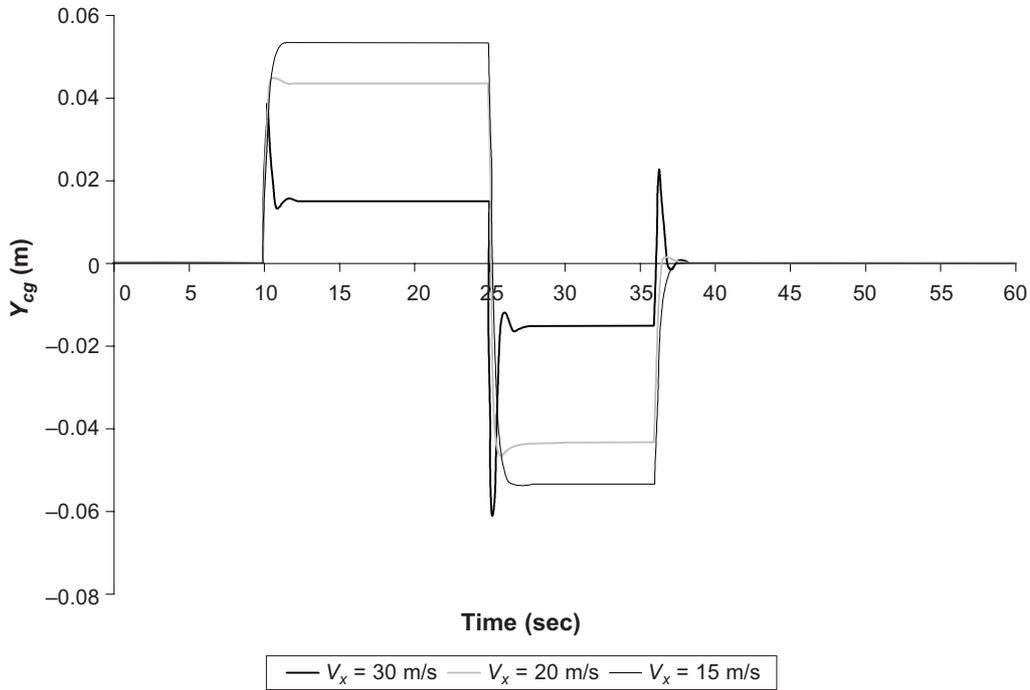


Figure 15 Simulation result of lateral error at center of gravity y_{cg} with $L = 5$ m

5.2 Robustness Analysis

For robustness analysis of the closed-loop system, comparisons are done by varying the mass and cornering stiffness of the vehicle. First, a robustness analysis is performed to analyse the robustness of the control system under various vehicle load, where the vehicle mass $m = 1590, 1790$ and 2090 kg. Secondly, another simulation is performed to analyse the robustness of the control system under different road conditions with varying cornering stiffness as $C = C_r = C_f = 120\,000, 96\,000$ and $60\,000$ N/rad.

Figures 16 and 17 show that the changes in vehicle's mass (load) and cornering stiffness does effect the performance of the automatic steering control system significantly even though all the errors are still within the acceptable range.

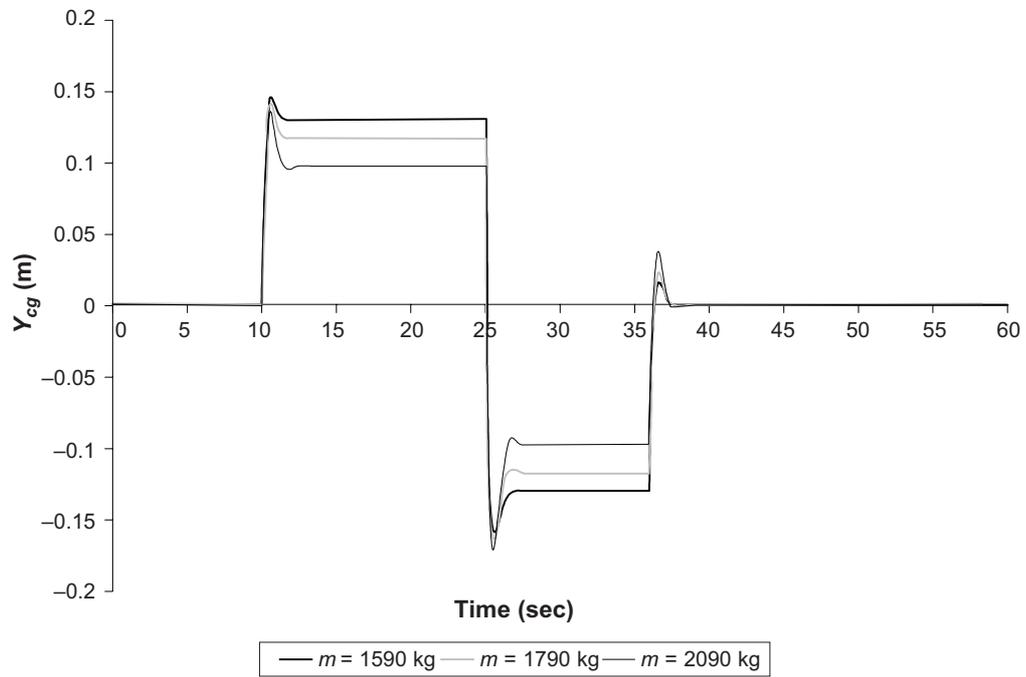


Figure 16 Simulation result for lateral error at center of gravity y_{cg} with $V_x = 30 \text{ ms}^{-1}$ and $L = 10 \text{ m}$ under various vehicle's mass

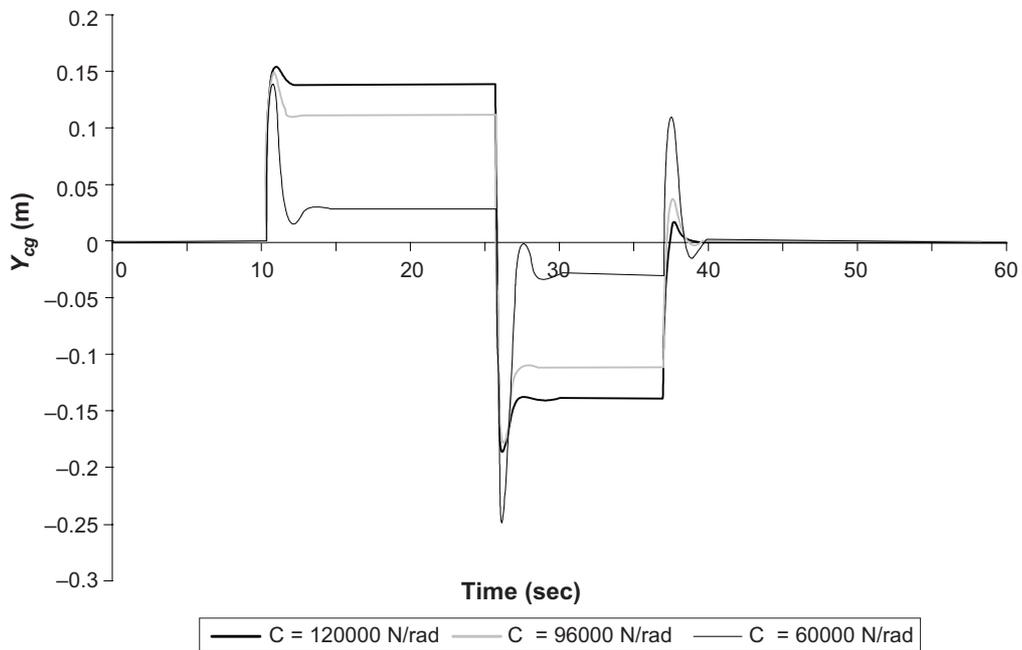


Figure 17 Simulation result for lateral error at center of gravity y_{cg} with $V_x = 30 \text{ ms}^{-1}$ and $L = 10 \text{ m}$ under various road conditions

6.0 CONCLUSIONS

The new proposed automatic steering control based on the look-ahead reference system where the visual information is extracted from image sequences with known camera geometry is employed in the feedback loop of the control system. The proposed PID controller is able to reject the effect of the road curvature disturbance and cause the vehicle to track the reference. However, the vehicle tends to overshoot as the longitudinal velocity of the vehicle is increased. Furthermore, the look-ahead distance, L should also be adjusted according to the longitudinal velocity for a better performance. In general, the primary objective of this research has been achieved where the proposed control system is able to track the centre of a lane automatically with small error under various conditions even though it is less robust.

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