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PID Control of Backward Motion of Tractor-Trailer

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Abstract— Tractor-trailers are very useful in raising the transportation efficiency of objects. The objective of this paper is to effectively model the backward motion of the nonlinear tractor-trailer system and apply controller. The three stages of control of the backward motion of tractor-trailer system – (a) controlling the hitch angle, (b) tracking the movement of the system along a lateral distance with a hitch angle, and (c) tracking the movement of the system along a backward distance with zero hitch angle towards the parking point - will be analyzed. The state equation of the backward motion of the tractor-trailer system is then obtained by combining the kinematic, steering and velocity models. The stability analysis of the different stages of control of the backward motion of the tractor-trailer system is done using Lyapunov's technique and the gain is then evaluated using Routh-Hurwitz criteria. The state space model of the first/second/third stages are obtained as a third-order/ third-order/fourth-order/system respectively. The performance of LQR controller on the three stages of the system has also been examined.

Index Terms—Tractor-Trailer; Hitch angle;Lyapunov's technique; Routh-Hurwitz criteria; Jack-knife phenomenon; Kinematic modeling; Dynamic modeling: LQR/LQT Control

I INTRODUCTION

The backward motion of the articulated vehicles consisting of a tractor and a trailer naturally exhibits unstable behavior because of the difficulty in smooth and coordinated movement of trailer and the tractor. The joint between the tractor and the trailer has no actuator, so the trailer can be controlled only by maneuvering the tractor. Such manoeuvres often require counterintuitive inputs, making them error-prone, cumbersome, frustrating or worse, dangerous, especially for uninitiated drivers. This paper proposes a suitable model for the backing up of an articulated system. Fig (1) shows an articulated vehicle studied in [1]. If the driver of the tractor-trailer makes a mistake in the steering input, the joint angle often expands to π/2rad. This is called the "jack-knife phenomenon" [1]. When a jack-knife holonomic constraint. Comparisons of different control strategies are also presented.When a jack-knife phenomenon happens, the tractor-trailer becomes uncontrollable and the backward motion stops. This paper proposes a suitable model that makes a tractor-trailer to track a specified trajectory in backward motion during parking by independently driving the non holonomic constraint.

Fig (1): Tractor – trailer system with carlike tractor and two wheeled trailer [1]

A lot of related work has been done for the stabilization problem of non-holonomic systems [2-4]. Model for timevarying or discontinuous feedback control approaches are studied in [2-4]. Murray and Sastry suggested a model for converting non holonomic systems into chained form [5]. Based on the transformed chained form system, a model for time-varying control law for the stabilization problem with sinusoidal inputs is presented [6, 7]. In [8], a set of globally stabilizing non-stationary smooth feedback control laws is derived for a front-wheel drive car-like robot by a

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different modeling approach. With higher order sliding mode, Zhang proposed a system model for discontinuous feedback control with finite-time convergence for the stabilization problem of a tractor-trailer mobile robot [9].Another discontinuous approach is fuzzy logic system. In [10], a fuzzy selection system for the parallel parking problem of a tractor-trailer mobile robot is constructed. The objective of this paper is to effectively model the backward motion of the nonlinear tractor-trailer system in three stages. In the first stage, the hitch angle is controlled. In the second stage, the system hasto move a lateral distance with a hitch angle. In the third stage, the system has to travel a backward distance with zero hitch angle towards the parking point. The hitch angle is setas zero in the absence of parked vehicle and it is set as a fixed positive value in the presence of parked vehicle.

The backward motion of the tractor-trailer system will be modeled in two ways - kinematic modeling and dynamic modeling. Kinematic modeling is done from the geometrical model of a car-like vehicle and obtained the velocity coordinates along x and y directions and also the expression for the angular velocity of the tractor. Dynamic modeling is done by steering and velocity modeling. The state equation of the backward motion of the tractor-trailer system is then obtained by combining the kinematic, steering and velocity models. The stability analysis of the different stages of control of the backward motion of the tractor-trailer systemis done using Lyapunov‟s technique and the gain is then evaluated using Routh-Hurwitz criteria. The state space model of the first/second/third stages are obtained as a third order system, third order system and fourth order system respectively. In this paper the parking task is divided in to three stages of control of the backward motion of the tractor trailer and also LQR controller. The three stages of control of the backward motion of the tractor-trailer system are (a) controlling the hitch angle, (b) tracking the movement of the system along a lateral distance with a hitch angle, (c) tracking the movement of the system along a backward distance with zero hitch angles towards theparking point.

This paper is organized as follows: Section II gives the system description and modelling and will give a detailed explanation of kinematic and dynamic modeling of the system. Section III explains about the three different stages of control under consideration, Section IV explains about the design of PID controller and will give the simulation results of the three different stages of control under consideration with PID controller. Conclusions are presented in section V.

II SYSTEM DESCRIPTION AND MODELING

Consider the Fig (2) given below. As shown in the given figure a two wheeled tractor is connected to the mid-point of the back axle of a four wheeled tractor.The length between the front and back axle is L which is considered to be 1m. The velocity of the vehicle is kept constant as 1.0 m/s (backward velocity).

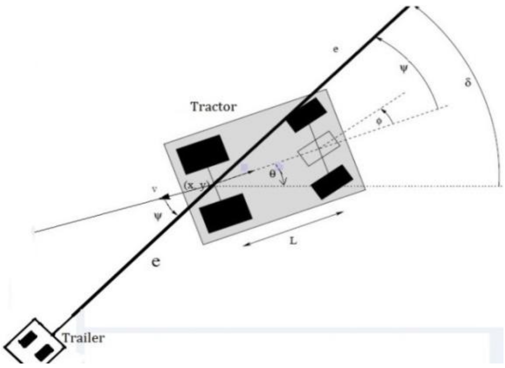


Fig (2): The tractor-trailer system

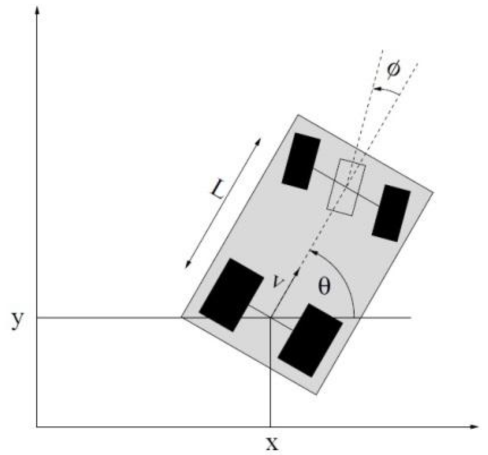
Referring Fig (2) '$\boldsymbol{\Psi}$' is the hitch angle which is to be controlled by the rate of change of steering angle '$\emptyset$'. 'θ'is the angle made by the velocity vector 'v' with X axis, which is called the orientation angle of the tractor. The direction of the orientation angle is opposite to the steering angle and that of the hitch angle is opposite to the orientation angle. The system must move laterally along a distance 'e' with a hitch angle '$\boldsymbol{\Psi}$'which is the second stage of control.

The system is modeled in two ways - Kinematic modelling andDynamicmodeling. Kinematic modeling deals with the geometric modeling of the tractor-trailer system as a car-like vehicle. Dynamic modeling is done by Steering and Velocity modeling. In steering modeling, an approximate model is obtained from the response of vehicle's steering loop to unit step changes in desired steering angle. In velocity modeling, an approximate model is obtained from the response of vehicle's velocity loop to unit step changes in desired velocity.

A Kinematic Modeling

To obtain kinematic modeling consider the figure given below. $\boldsymbol{\Psi}$' is the hitch angle which is to be controlled by the rate of change of steering angle '$\emptyset$'. 'θ'is the angle made by the velocity vector'v'

with X axis , which can be considered as the orientationangleas shown in the Fig(3).



Fig(3): A Geometrical model of a car-like vehicle.

It can be considered as a bicycle model. The Kinematic equation of this vehicle (rear tyres align with the vehicle and front tyres allowed to rotate about vertical axis) is $\dot{x}$ = v$\cos\theta$, $\dot{y}$ = v$\sin\theta$ and $\dot{\theta}$ = $\frac{v\tan\emptyset}{L}$ .Where ‘v’ is the translational velocity, ’$\emptyset$’is the steering angle and θ is the orientation angle.

Dynamic Modeling

Dynamic modeling is done by Steering and Velocity modeling. These models are obtained by considering the models of vehicle’s dynamic response to demands. The advantages of this modeling are as follows:-

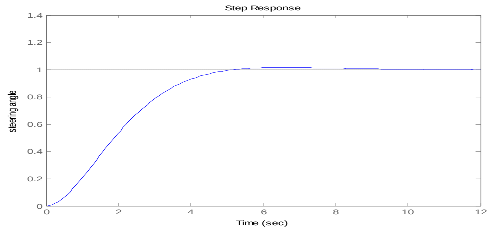
1. It considers the realities on the implementation on vehicles
2. Realistic testing and development of control algorithm in simulation is possible
3. Models are simple, but sufficient to capture general behaviour of the system

B Steering Model

An approximate model is obtained from the response of vehicle’s steering loop to unit step changes in desired steering angle. It is a second order system with transfer function

$$\frac{\emptyset(s)}{\emptyset(s)^*} = \frac{\omega_n^2}{S^2+2\xi\omega_n S+\omega_n^2} \qquad (1)$$

Where $\emptyset(s)^*$ is demand in steering angle, and the damping ratio ξ can be taken as 0.8 and the natural frequency w_n can be taken as 0.8 rad/sec .This second order transfer function can be split into two as $\dot{\rho} = \omega_n{}^2(\emptyset_d{}^* - \emptyset)$ and $\dot{\omega} = \rho - 2\xi\,\omega_n\omega$, where $\omega = \frac{v}{L}\emptyset$



Fig(4):Step response of steering loop [11]

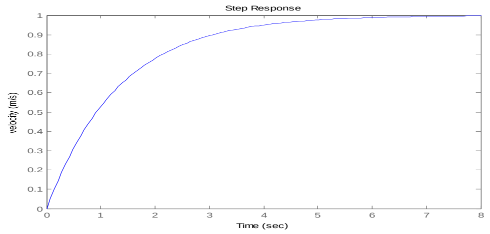
Referring to Fig(4) it is observed that the system is a second order system with damping ratio 0.8 and natural frequency 0.8 rad/sec it has a small overshoot and will settle to steady state within 12 seconds.

B Velocity Model

An approximate velocity model is obtained from the response of vehicle’s velocity loop to unit step changes in desired velocity. It is a first order system with transfer function [11]

$$\frac{V(S)}{V(S)^*} = \frac{K_V}{\tau_V S+1} \qquad (2)$$

Where K_V is the velocity gain constant and it can be taken as 1, and τ_V is the velocity time constant and can be taken as 1.33 sec.The step response of velocity loop is as given in Fig (5).



Fig(5): Step response of velocity loop

Referring to Fig (5) it is observed that the system is a first order system and it will settled to steady state within 7.5 seconds.

C Vehicle State Equations

Considering Fig(3.1),the vehicle state equation is derived from the kinematic, steering and velocity models as $\dot{e} = v\cos\boldsymbol{\Psi}$, $\dot{\Psi} = \frac{v\sin\Psi}{e} + \omega$ and

$$\dot{\delta} = \frac{v\sin\Psi}{e} \qquad (3)$$

These vehicle state equations are obtained by applying the properties of congruent triangles to Fig (3) and considering the direction of the orientation angle as opposite to the steering angle and that of the hitch angle as opposite to the orientation angle.

III STAGES OF CONTROL

There are three different stages of control of backward motion of tractor-trailer system. In first stage, the hitch angle is controlled. In second stage, the system has to move a lateral distance with a hitch angle. In third stage, the system has to travel a backward distance with zero hitch angletowards the parking point. The hitch angle is set as zero in the absence of parked vehicle and it is set as a fixed positive value in the presence of parked vehicle. Stability analysis of the stages of operation of the tractor-trailer system is done using Lyapunov‟s technique. The gain is then evaluated using Routh-Hurwitz criteria. The closed loop response of the system is also analyzed

A First Stage of Control

The first stage of control is to control the hitch angle only. Hitch angle is to be controlled either to zero or to a desired value. When the system move back ward, if there is any already parked vehicle the hitch angle must be maintained at a desired value to avoid collision with it and if there is no vehicle it should be maintained to zero.

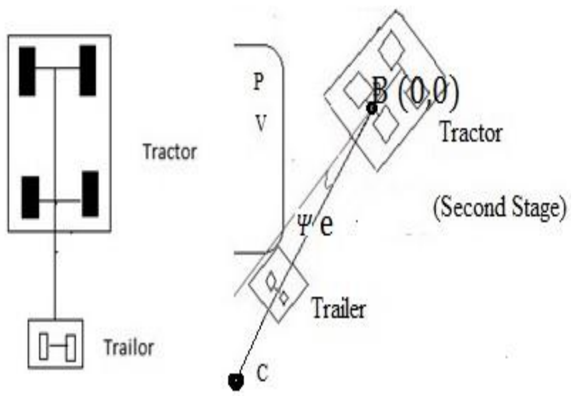


Fig (6): (a)$\Psi$is maintained at Zero

Fig (7):$\Psi$is maintained at a desired value (b)

The first stage of system is as shown in Fig (6) and Fig (6).Fig (7) shows how the hitch angle should be maintained to zero if there is no other parked vehicle and the Fig (7) shows howthe hitch angle should be maintained to a desired value to avoid collision with other parked vehicle.

The block diagram for the first stage of system can be shown in Fig (9). Both the kinematic and dynamic models are adopted to get a system description given as

$$\dot{\Psi} = \omega \ , \quad \dot{\rho} = {\omega_n}^2(\emptyset^* - \emptyset) \ , \quad \omega = \frac{v}{L}\emptyset \ , \quad \dot{\omega} = \rho - 2\xi\, \omega_n \emptyset \ \text{and} \omega^* = -K_1\boldsymbol{\Psi} \qquad (4)$$

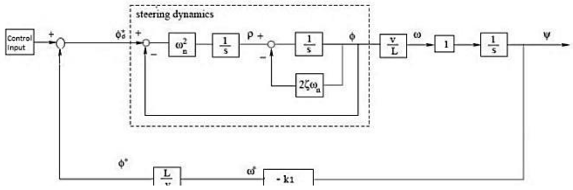


Fig (8): Block diagram of first stage of control [11]

The output is the Hitch angle to be controlled. The hitch angle can be controlled by changing the rate of steering angle $\emptyset$ by appropriate control input. Any controller can be used to control the hitch angle in the forward path or in the feedback path. By Routh-Hurwitz criteria K_1 is obtained as 0.23.The transfer function can be obtained by the reduction of the block diagram thereby the state-space model can be obtained. The transfer function obtained is given by

$$\frac{\Psi(s)}{U(s)} = \frac{.64}{S^3+1.28S^2+.64S+.14} \quad (5)$$

From the transfer function obtained the state space model of the first stage of control can be obtained. The A, B, C, D matrix can be obtained from the transfer function as given below.

$$A = \begin{bmatrix} 0 & 1 & 0 \\ 0 & 0 & 1 \\ -.14 & -.64 & -1.28 \end{bmatrix}, \quad B = \begin{bmatrix} 0 \\ 0 \\ .64 \end{bmatrix},$$

$$C = [1 \quad 0 \quad 0], \quad D = 0, X = \begin{bmatrix} \Psi \\ w \\ \rho \end{bmatrix} \qquad (6)$$

The state variables of the first stage of control are the hitch angle $'\Psi'$ the angular velocity $'\omega'$ and the angular acceleration$'\rho'$.

The closed loop response of the backward motion of the tractor-trailer system for a unit step input is obtained in this section. The hitch angle Vs time is plotted, as shown in Fig (9).

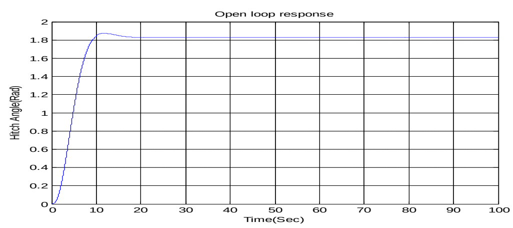


Fig (9): Closed loop stable response of the system

Referring Fig (9) the hitch angle of the system is settled to 1.81rad with a settling time of 19 seconds with zero overshoot and zero steady state error. Also the response is having no delay and the rise time is about 8 seconds. Jack-knife phenomenon occurs at a hitch angle of 1.81 rad. The hitch angle is not increased further because of the limitations in

mechanical linkage between tractor and trailer. As the closed loop system does not settle at the reference value (unity), a controller is required to get the desired performance.

B Second Stage

The second stage of operation of the backward motion of the tractor-trailer system can be shown as given in the Fig (10). The system must move backward through a lateral distance 'e' with a hitch angle Ψ. In this stage two parameters, the hitch angle 'Ψ' and the lateral displacement 'e' should be controlled. Here two modeling are required; Ψ loop modeling for Hitch angle control and 'e' loop modeling for lateral displacement control.

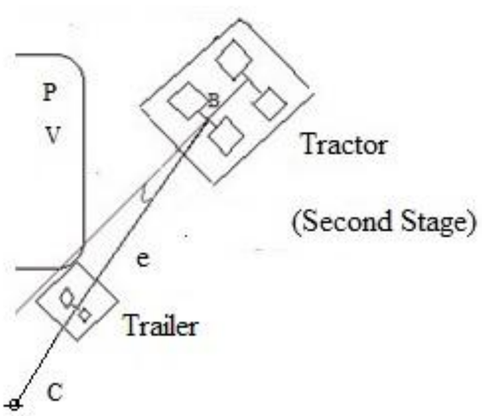


Fig (10): The second stage of control

The block diagram of the second stage of operation of the backward motion of the tractor-trailer system can be shown in Fig (10). Both the kinematic and dynamic models are adopted to get a system description as $\dot{\Psi} = \frac{v\Psi}{e} + \omega$; $\dot{\delta} = \frac{v\Psi}{e}$; $\dot{\rho} = {\omega_n}^2({\emptyset_d}^* - \emptyset)$; $\omega = \frac{v}{L}\emptyset$; $\dot{\omega} = \rho - 2\xi\,\omega_n\omega$; $\omega^* = (k_2 - k_1)\boldsymbol{\Psi}$; $\frac{v}{e} = k_1$ (7)

The output is the hitch angle to be controlled. The hitch angle can be controlled by changing the rate of steering angle $\emptyset$ by appropriate control input. By Routh-Hurwitz criteria K_1 is obtained as 0.1 and K_2 is obtained as 0.5.

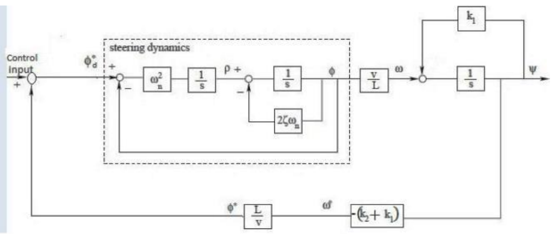


Fig (11): Block diagram of second stage of control [11]

State space model of the system can be obtained from transfer function model and which can be obtained by the reduction of the block diagram, and the transfer function obtained is given by is

$$\frac{\Psi(s)}{U(s)} = \frac{.64}{S^3+1.78S^2+1.28S+.32} \,. \qquad (8)$$

The state matrices of state space model can be obtained from the transfer function as given below.

$$A = \begin{bmatrix} 0 & 1 & 0 \\ 0 & 0 & 1 \\ -.32 & -1.28 & -1.78 \end{bmatrix}, B = \begin{bmatrix} 0 \\ 0 \\ .64 \end{bmatrix}, C = [1 \quad 0 \quad 0]\,, D = [\,0\,]\,, X = \begin{bmatrix} \Psi \\ w \\ \rho \end{bmatrix} \qquad (9)$$

The state variables of the second stage of control are the hitch angle $'\Psi'$ the angular velocity $'\omega'$ and the angular acceleration $'\rho'$

C The e – Loop

The lateral displacement of the system 'e' can be modeled by considering the fact that the lateral displacement 'e' is independent of all other states. It depends only on the input velocity by the linearized equation (3) and we will get $\dot{e}$= v and from equation (12) we will get the velocity as v = $k_1 e$, thus the lateral velocity $\dot{e}$ can be written in terms of change in velocity $\tilde{v}$as $\dot{e} = \tilde{v}$

As shown in the Fig (11), the lateral displacement "e" and the time taken to cover the specified distance can be controlled by changing the velocity input to the tractor-trailer system.

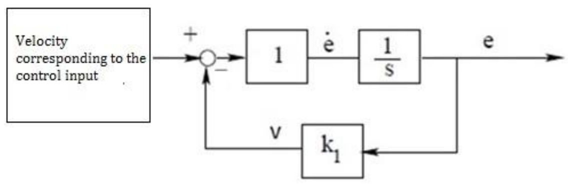


Fig (12): Block diagram of second stage (e-loop) [11]

So the control input from suitably chosen controller should give desired velocity to the system to settle at a desired distance within specified time. By taking k_1=0.1,the transfer function obtained is given by

$$\frac{e(s)}{U(S)} = \frac{1}{(S+.1)} \qquad (10)$$

The closed loop response of the second stage of operation of the backward motion of the tractor-trailer systemisobtained in this section. Hitch angle Vs time, lateral displacement Vs time are shown in Fig (13) and Fig (14) respectively.

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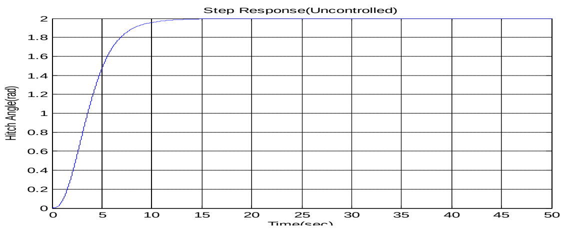

Fig (13): Hitch Angle Vs Time

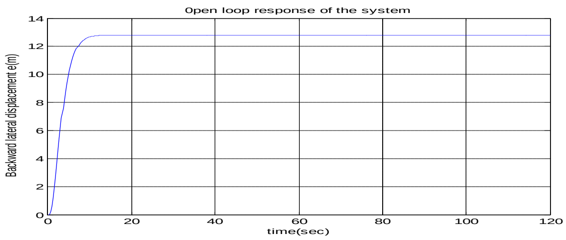

Fig (14): Lateral displacement (e) Vs time

Referring Fig (13) the hitch angle of the system is settled to 2 rad with a settling time of 13 seconds with zero overshoot and zero steady state error. Also the response is having no delay and the rise time is about 8 seconds.Referring Fig (14) the lateral displacementof the system is settled to 13m with a settling time of 15 seconds with zero overshoot and zero steady state error.Also the response is having no delay and the rise time is about 9 seconds.As the closed loop system does not settle at the reference value (unity), a controller is required to get the desired performance.

C Third Stage Of Control

The third stage of the system control is as shown in Fig (15) below;

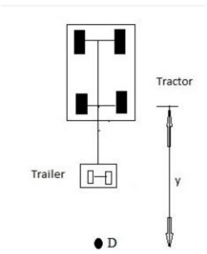

Fig (15):The third stage of control

The third stage of the systemcontrol is as shown in Fig (15). In this stage the system should move backward towards the X axis with zero hitch angle. Here only backward motion is considered and the hitch angle is zero. In this stage zero velocity along X axis. The system moves backward with only the y-component of the velocity. The system must move with an angle of orientation $\theta = 90^0$back ward towards the parking point D. Linearizing the equation (3), where the system have to move along the y-direction and no movement is there along the x-direction .So the orientation angle of the tractor θ can be taken as 90^0 and the obtained linearized equation can be written as

$\dot{x} = 0,\ \dot{y} = \mathrm{v}\,\theta,$

$\dot{\theta} = \omega$ (11)

The block diagram of the third stage of control is as shown in Fig (4.11). The output is the distance travelled backward which is to be controlled. This can be controlled by changing the rate of angular velocity which in turn depends on linear velocity "v" ,by appropriate control input.Fromeqn (3.1) we will get the following equations by linearizing sin θ ,

$\dot{y} = \mathrm{v}\,\theta$, $\dot{\theta} = \omega$, $\dot{\rho} = \omega_n{}^2(\emptyset_d{}^* - \emptyset)$,$\dot{\omega} = \rho - 2\xi\,\omega_n\omega$,$\omega^* = -(k_2\theta + k_1 v^*\, y)$ (12)

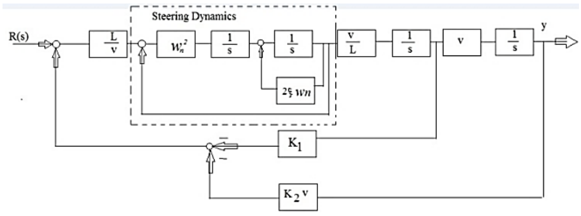

Fig (16): Block diagram for the third stage of control

Any controller can be used to control the position of the tractor trailer by placing the controller in the forward path or in the feedback path. By Routh-Hurwitz criteria K_1 is obtained as 0.1 and K_2 is obtained as 0.5.

State space model can be obtained from the transfer function model and which can be obtained by the reduction of the block diagram given in Fig (16) and the transfer function obtained for the third stage of control is given by

$$\frac{y(s)}{R(s)} = \frac{.32}{S^4+1.28S^3+.64S^2+.32S+.016}. \qquad (13)$$

The A, B, C, D matrix can be obtained from the transfer function in the phase variable form and the state variables are the backward distance to be travelled 'y', the linear velocity along y direction 'θ '(linear velocity along y direction is given by 'v θ',v is taken as 1m/sec),angular velocity 'ω' and angular acceleration 'ρ'.

$$A = \begin{bmatrix} 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \\ -.016 & -.32 & -.64 & -1.28 \end{bmatrix},$$

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$$B = \begin{bmatrix} 0 \\ 0 \\ 0 \\ .32 \end{bmatrix}, C = \begin{bmatrix} 1 & 0 & 0 & 0 \end{bmatrix}, D = [0], X = \begin{bmatrix} y \\ \theta \\ \omega \\ \rho \end{bmatrix} \quad (14)$$

The closed loop responseof the third stage of operation of the tractor-trailer system is obtained in this section. Position Vs Time of the third stage of the system is as shown in the Fig (17).

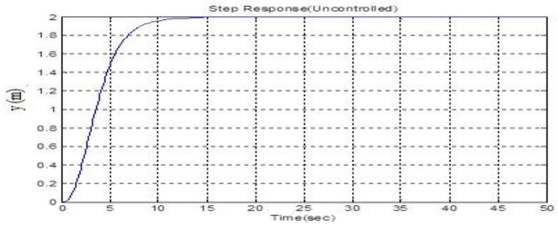

Fig (17): Closed loop response of the stable system

Referring Fig (17) the system is moving backward and stabilized to 2 m with a settling time of 13 seconds with zero overshoot and zero steady state error. Also the response is having no delay and the rise time is about 8 seconds. Though the system is stable, it does not settle at the reference value (unity).

IV PID CONTROLLER

A proportional-integral-derivative controller (PID controller) is a generic control loopfeedback mechanism (controller) widely used in industrial control systems. A PID controller calculates an "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 [4].

The PID controller algorithm involves three separate constant parameters, and is accordingly sometimes called three-term control: the proportional, the integral and derivative values, denoted P,I, and D. Simply put, these values can be interpreted in terms of time: P depends on the present error, I on the accumulation of past errors, and D is a prediction of future errors, based on current rate of change. The weighted sum of these three actions is used to adjust the process via a control element such as the position of a control valve, a damper, or the power supplied to a heating element.

In the absence of knowledge of the underlying process, a PID controller has historically been considered to be the best controller. By tuning the three parameters in the PID controller algorithm, the controller can provide control action designed for specific process requirements. The response of the controller can be described in terms of the responsiveness of the controller to an error, the degree to which the controller overshoots the setpoint, and the degree of system oscillation. Note that the use of the PID algorithm for control does not guarantee optimal control of the system or system stability.

Some applications may require using only one or two actions to provide the appropriate system control. This is achieved by setting the other parameters to zero. 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, since derivative action is sensitive to measurement noise, whereas the absence of an integral term may prevent the system from reaching its target value due to the control action.The PID control scheme isnamed after its three correcting terms, whose sum constitutes the manipulated variable (MV). The proportional, integral, and derivative terms are summed to calculate the output of the PID controller. Defining u(t) as the controller output, the final form of the PID algorithm is:

$$u(t) = MV(t) = K_p\, e(t) + K_d \frac{de(t)}{dt} + K_I \int e(\tau)\, d\tau \quad (15)$$

where

K_p : Proportional gain, a tuning parameter

K_I: Integral gain, a tuning parameter

K_d: Derivative gain, a tuning parameter

e: Error = SP-MV

t: Time or instantaneous time (the present)

τ : Variable of integration; takes on values from time 0 to the present t

The block diagram of PID controller can be shown as shown in Fig (18).

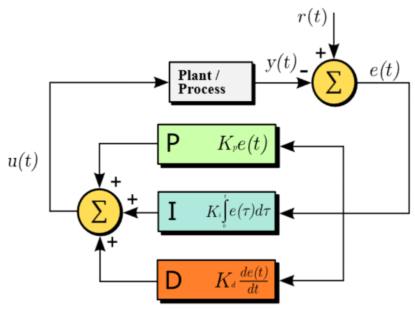

Fig (18): Block diagram of PID Controller

A Proportional Control

The proportional term produces an output value that is proportional to the current error value. The proportional response can be adjusted by multiplying the error by a constant K_p, called the proportional gain constant. The proportional term is given by:

$$P_{out} = K_P e(t) \qquad (16)$$

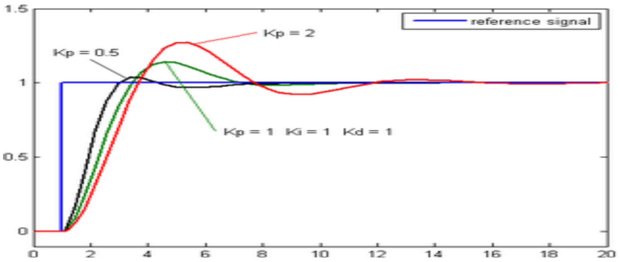

Fig (19): Plot of PV vs time, for three values of K_p (K_i and K_d held constant)

A high proportional gain results in a large change in the output for a given change in the error. If the proportional gain is too high, the system can become unstable. In contrast, a small gain results in a small output response to a large input error, and a less responsive or less sensitive controller. If the proportional gain is too low, the control action may be too small when responding to system disturbances. Tuning theory and industrial practice indicate that the proportional term should contribute the bulk of the output change.

B Integral Control

The contribution from the integral term is proportional to both the magnitude of the error and the duration of the error. The integral in a PID controller is the sum of the instantaneous error over time and gives the accumulated offset that should have been corrected previously. The accumulated error is then multiplied by the integral gain K_I and added to the controller output.

The integral term is given by:

$$+I_{out} = K_I \int e(t)\ dt \qquad (17)$$

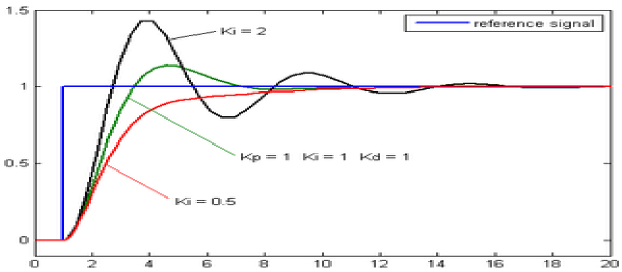

Fig (20):Plot of PV vs time, for three values of K_i (K_p and K_d held constant)

The integral term accelerates the movement of the process towards setpoint and eliminates the residual steady-state error that occurs with a pure proportional controller. However, since the integral term responds to accumulated errors from the past, it can cause the present value to overshoot the set-point value.

C Derivative Control

The derivative of the process error is calculated by determining the slope of the error over time and multiplying this rate of change by the derivative gain K_d. The magnitude of the contribution of the derivative term to the overall control action is termed the derivative gain, K_d.

The derivative term is given by:

$$D_{out} = K_d \frac{de(t)}{dt} \qquad (18)$$

Derivative action predicts system behaviour and thus improves settling time and stability of the system. Derivative action, however, is seldom used in practicebecause of its inherent sensitivity to measurement noise. If this noise is severe enough, the derivative action will be erratic and actually degrade control performance. Large, sudden changes in the measured error (which typically occur when the set point is changed) cause a sudden, large control action stemming from the derivative term, which goes under the name of derivative kick. This problem can be ameliorated to a degree if the measured error is passed through a linear low-pass filter or a nonlinear but simple median filter.

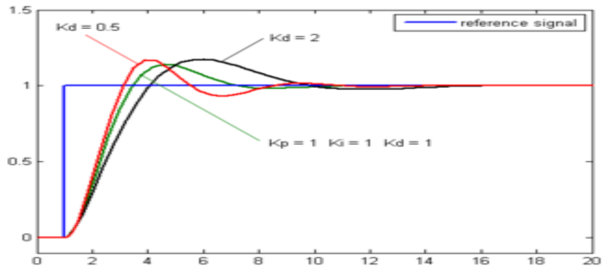

Fig (21) :Plot of PV vs time, for three values of K_d (K_p and K_i held constant)

D Loop Tuning

Tuning a control loop is the adjustment of its control parameters (proportional band/gain, integral gain/reset, derivative gain/rate) to the optimum values for the desired control response. Stability (bounded oscillation) is a basic requirement, but beyond that, different systems have different behaviour, different applications have different

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requirements, and requirements may conflict with one another.

PID tuning is a difficult problem, even though there are only three parameters and in principle is simple to describe, because it must satisfy complex criteria within the limitations of PID control. There are accordingly various methods for loop tuning, and more sophisticated techniques are the subject of patents; this section describes some traditional manual methods for loop tuning.

Designing and tuning a PID controller appears to be conceptually intuitive, but can be hard in practice, if multiple (and often conflicting) objectives such as short transient and high stability are to be achieved. Usually, initial designs need to be adjusted repeatedly through computer simulations until the closed-loop system performs or compromises as desired.

Some processes have a degree of nonlinearity and so parameters that work well at full-load conditions don't work when the process is starting up from no-load; this can be corrected by gain scheduling (using different parameters in different operating regions). PID controllers often provide acceptable control using default tunings, but performance can generally be improved by careful tuning, and performance may be unacceptable with poor tuning.

D Controller Configuration

The PID controller is configured with the system as shown in figure (22).The controller is tuned for a minimum overshoot and settling time according to the requirements in response.

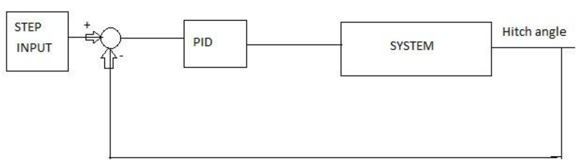

Fig (22): Block diagram of PID configuration

The error between set point and the obtained error is fed to the controller and it will giverequired control action to give a desired response from the system.

AFirst StageOf Control

The control objective of the first stage of operation of the backward motion of the tractor trailer system is to set the hitch angle at a desired reference value. The response of the system using a PID controller for a specified hitch angle (unit step input) in the presence of parked vehicle is shown in Fig (23).

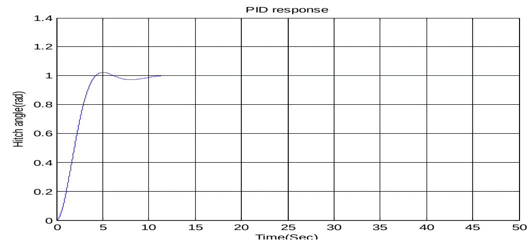

Fig (23): PID response to specified hitch angle of first stage.

The response of the system using a PID controller for a zero hitch angle in the absence of parked vehicle is shown in Fig (24).

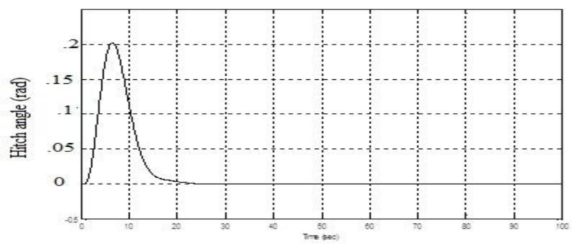

Fig (24): PID response to zero set point of first stage.

Referring Fig (24) the response is having an over shoot of 20% and it settles to zero within 24 seconds and zero steady state error. Also the response is having no delay and the rise time is about 6 seconds.

B PID *Second Stage Of Control*

The control objective of the second stage of operation of the backward motion of the tractor trailer system is to track a specified lateral backward displacement at a specified hitch angle. The response of the system using a PID controller for a specified position (unit step input) for a specified hitch angle (unit step input) is obtained in this section. Hitch angle Vs time and backward motion Vs Time are shown in Fig (25) and Fig (26) respectively.

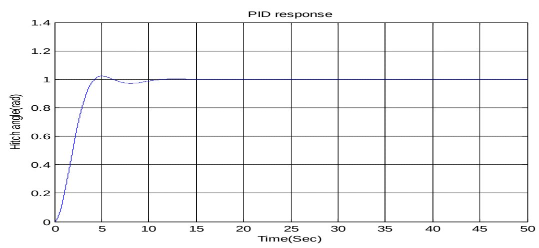

Fig (25): PID response of the second stage – Hitch angle Vs time

Referring Fig (5.10) the response is having an over shoot of 5 % and it settles to the reference value

within 14 seconds and zero steady state error. Also the response is having no time delay and the rise time is about 3.5 seconds.

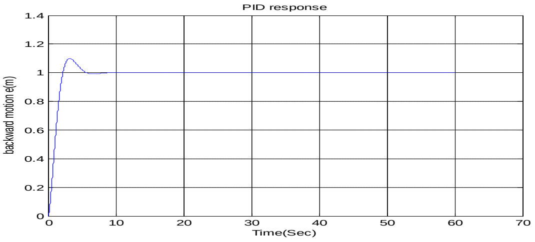

Fig (26): PID response of the second stage Backward motion Vs time

Referring Fig (26)the backward motionthe response is having an over shoot of 10 % and it settles to the reference value within 8 seconds and zero steady state error. Also the response is having no delay and the rise time is about 2.5 seconds.

CThird Stage Of Control

The control objective of the third stage of operation of the backward motion of the tractor trailer system is to track a specified backward displacement at zero hitch angle. The response of the system using a PID controller for a specified position (unit step input) is obtained in this section. Backward motion Vs Time is shown in Fig (27).

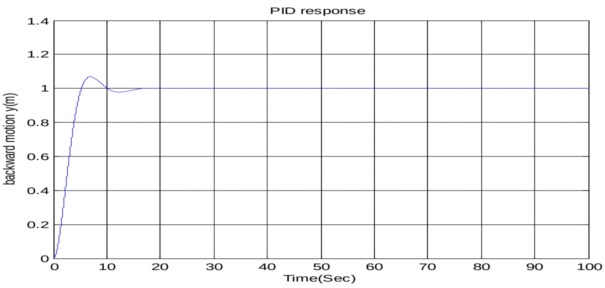

Fig (27): Response of the system under a tuned PID controller

Referring Fig (27) the backward motionthesystem is having an over shoot of 10 % and it settles to the reference value within 16 seconds andwith zero steady state error. Also the response is having no time delay and the rise time is about 2.5 seconds.

DChecking The Robustness Of The Controller

The response of the first stage of the system with disturbance and parameter variation, using a PID controller is analysed in this section for checking the robustness of the controller. The PID response of the first stage under disturbance is shown in Fig (28). The PID response of the first stage under parameter variation of 20% is shown in Fig (28).

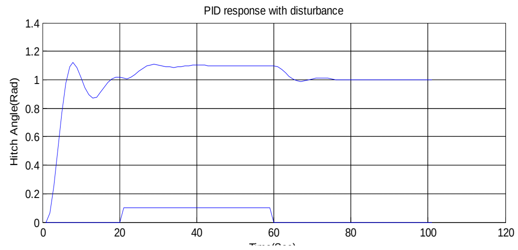

Fig (28):PID controller for first stage with constant disturbance

As inferred from the Fig (28), the response of the first stage goes unstable after 21sec with a constant disturbance of .1units injected after 21 sec. Thus PID controller cannot withstand disturbances occurring in the system.

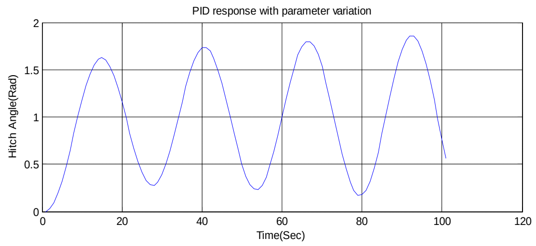

Fig (29) : State feedback controller for first stage with 20 % of parameter variation

As inferred from the Fig (29) , for a 20% of parameter variation, the response of the first stage becomes unbounded under the tuned PID controller. Thus the PID controller is not robust as the response does not meet the required performance specifications with disturbance and parameter variation.

V CONCLUSION

This paper explained the PID control of the three stages of control of backward motion of tractor-trailer system, namely controlling the hitch angle, tracking the movement of the system along a lateral distance with a hitch angle and tracking the movement of the system along a backward distance with zero hitch angle towards the parking point. PID control is analysed with disturbance and parameter variation in the system as well.

The control objective of the first stage of operation of the backward motion of the tractor trailer system is to set the hitch angle at a desired reference value. In the presence of a parked vehicle, the hitch angle initially oscillates for a short time, with an overshoot of 2.5%. Then it settles at the reference value at 12 sec.In the absence of a parked vehicle, the response is having an over shoot of 20% and it settles to zero within 24 seconds. With a

disturbance injected at 20sec, the response of the first stage goes unstable at 20 sec. For a 20% of parameter variation, the response of the first stage oscillates and then comes to a steady state after 80 seconds.

The control objective of the second stage of operation of the backward motion of the tractor trailer system is to track a specified lateral backward displacement at a specified hitch angle. The hitch angle initially oscillates for a short time, with an overshoot of 2.5%. Then it settles at the reference value at 14 sec. The backward motion has an overshoot of 10% and settles at 10 sec.With a disturbance injected at 20sec, the response of the second stage goes unstable at 20 sec. For a 20% of parameter variation, the response of the second stage oscillates and then comes to a steady state after 120 seconds.

The control objective of the third stage of operation of the backward motion of the tractor trailer system is to track a specified backward displacement at zero hitch angles. The backward motion has an overshoot of 10% and settles at 16 sec.

PID controller offers a greater overshoot and higher settling time than the required performance specifications. Alsoit cannot withstand disturbance and parameter variation occurring in the system as given for a 20% parameter variation, the first stage of the system became unbounded under tuned PID controller. Also the response of the first stage goes unstable at 21 sec with a constant disturbance of .1units injected at 21sec. Thus PID controller cannot withstand disturbances occurring in the system and also the parametric variation. Thus the PID controller is not robust as the response does not meet the required performance specifications with parameter variation.. It is not a robust controller. So a better controller is required to control the system effectively, in case of disturbance and parameter variation.

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