Road Vehicle Following System With Adaptive Controller Gain Using Model Reference Adaptive Control Method Lyapunov Approaches

Mohd Razali Sapiee, Aminurrashid Noordin, Mohd Farriz Basar,

Zulhani Rasin, Md Hairul Nizam Talib

Faculty of Electrical Engineering.Email:mohd.razali@utem.edu.my,
aminurrashid@utem.edu.my,mfarriz@utem.edu.my,
zulhani@utem.edu.my, hairulnizam@utem.edu.my
Universiti Teknikal Malaysia Melaka

Abstract—**In order to maintain stability and satisfy operating constraints, the control system on the following vehicle needs information about the motion of preceding vehicle. A one-vehicle look-ahead control strategy is proposed and will be investigated for this operation. A mathematical model for this control strategy is obtained and simulated. This paper describes the process of designing an adaptive controller gain for a road vehicle following system using two Lyapunov approach. One approaches utilizes the perfect modelling while the other does not. This is done through simulations and comparisons and is further discussed to find the effectiveness of the two Lyapunov approaches.**

Keywords—**One-Vehicle Look-Ahead Control; Vehicle Following System; Model Reference Adaptive Control; Lyapunov Approach; Stability Approach**

I. INTRODUCTION

In general, Malaysian drivers tend to follow another vehicle closely. Hence, platoons or convoys appeared to develop rapidly. This normally happens when spacing between vehicles is close and there is no chance for the following vehicle to overtake the preceding vehicle. The vehicle at the back or the following vehicle will have to adjust its speed and spacing with respect to the preceding vehicle. If this is not taken into careful consideration, collision between them may occur.

In order to avoid any collisions between the following vehicle and the preceding vehicle, it is necessary to maintain some safe distances between both vehicles at any speed. So, the following vehicle needs to have information regarding to the speed and the distance between them. In a normal driving, the driver of the following vehicle will estimate the information and adjust his vehicle's speed and position to have a nominal speed and a safe distance between his vehicle and the preceding vehicle.

Nowadays, all the information does not need to be estimated. Sensors are available to measure the speed of the preceding vehicle and the position of the preceding vehicle with respect to the following vehicle. The information from the sensors will be used and processed by the following vehicle controller to produce the required speed and the safe spacing distance.

This is where autonomous control can take place. The controller can automatically ensure safe distance based on the information obtained from the preceding vehicle. The autonomous controller on the following vehicle can activate the vehicle cruise control mode, where the driver does not need to hold the steering nor press the fuel pedal, and automatically apply the brake when necessary in order to ensure the safety of the vehicle.

In developed nations, the autonomous concept leads to the Intelligent Vehicle Highway System (IVHS). As a vehicle enters the highway, his vehicle automatically takes-over the control of the vehicle while following the preceding vehicle. This feature also gives rise to steeringless technology where during the autonomous control in action, the driver does not need to hold the steering wheel. All the driving tasks are taken care by the vehicle intelligent system.

One of the autonomous features is the adaptive type control based on certain control strategy which gives rise to adaptive cruise control (ACC). An ACC controlled vehicle will follow the front vehicle at a safe distance. A Model Reference Adaptive Control (MRAC) can be used in this type of control where the vehicle controller has the ability to adapt to the variation of speed and position of the preceding vehicle.

II. ONE-VEHICLE LOOK-AHEAD CONTROL STRATEGY

When a vehicle follows another vehicle in front of it, a vehicle convoy or a vehicle following system is formed as shown in Fig. 1.
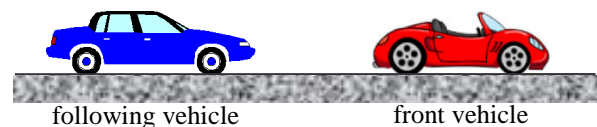

following vehicle   front vehicle

Figure 1.   A Vehicle Following System

The following vehicle can be controlled in such a way that it will maintain either the same speed to that of the immediate preceding vehicle, which is the front vehicle or maintain a safe distance in order to avoid collision between them. This is where the string stability plays an important role by having a string stable vehicle following

system. The system is said to be stable if the range errors decrease as they propagate along the vehicle stream.

In this control strategy, the controlled vehicle only refers the information from the preceding vehicle. So, the control system on the following vehicle needs information about the motion of preceding vehicles. Yanakiev and Kanellakopoulos [1] used a simple spring-mass-damper system to demonstrate the idea of string stability and show the string-stability criterion for constant time-headway and variable time-headway policies. A mathematical modeling for this control strategy is shown in Fig. 2 where i denotes the following vehicle and i-1 denotes the preceding vehicle.
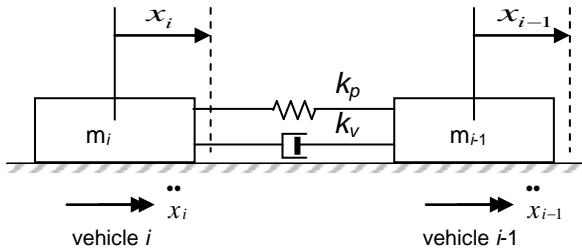

Figure 2. Mathematical Model Of A Vehicle Following System Control Strategy

Performing the mathematical modelling on only the following vehicle i and applying the Newton's Second law results in (1).

$$k_p(x_{i-1} - x_i) + k_v \frac{d}{dt}(x_{i-1} - x_i) = m_i \frac{d^2}{dt^2} x_i \qquad (1)$$

Assuming unit mass for (1) and taking Laplace transform, gives the following transfer function.

$$G(s) = \frac{x_i(s)}{x_{i-1}(s)} = \frac{k_v s + k_p}{s^2 + k_v s + k_p} \qquad (2)$$

The transfer function in (2) depends on the following spacing policy.

The aim of this strategy is to maintain string stability for longitudinal motion within the vehicle following system or the vehicle convoy, particularly between a vehicle with a vehicle or between a vehicle with a following vehicle. This strategy is adopted in order to design a controller by investigating the following two policies.

A. Spacing Policy

Researchers are trying to get close inter-vehicular spacing between vehicles, in order to have an effective following system. Spacing policy is defined as a rule that dictates how the speed of an automatically controlled vehicle must regulate as a function of the following distance. A control system should be designed such that it regulates the vehicle speed according to the designed spacing policy.

There are three basic spacing policies employed by many researchers thus far. Those spacing policies are the fixed spacing policy, the constant time headway policy and the variable time policy. Only the first two will be discussed and investigated in this paper.

1) Fixed Spacing Policy

Under this policy, a fixed inter-vehicular spacing is implemented regardless of the vehicle's speed. A well-known result states that it is impossible to achieve string stability in an autonomous operation when this spacing policy is adopted. This is mainly due to the relative spacing error that does not attenuate as it propagates down the string at all frequencies. Spacing error attenuation will only occur for frequencies above certain level. In addition, keeping the same fixed spacing at different convoy speed would risk the safety and comfort of passengers, especially when the vehicles are closely separated.

Obviously, at higher speed, faster vehicle reaction time is needed in an emergency situation to avoid collision. Nevertheless, the fixed spacing policy can give guaranteed string stability if the front vehicle provides its information on its speed and or position to the rear vehicle in the convoy. This can be done through radio communication or the rear vehicle having sensors to detect the above two parameters. (2) shown before is the transfer function for the fixed spacing policy.

2) Fixed Headway Spacing Policy

This spacing policy keeps a fixed time interval, called time headway or headway, h, between the preceding vehicle and the following vehicle. It is a speed dependent policy where the inter-vehicular spacing will vary according to the preceding vehicle speed. At higher speed, vehicles will be separated in a greater distance but always maintains a fixed time interval between vehicles. Most researchers used this spacing policy in designing controllers to ensure string stability as this policy mimics the behaviour of human drivers. As vehicle speed is increased, a human driver will keep a safe inter-vehicular spacing with the immediate preceding vehicle.

The performance of the fixed headway spacing policy used in autonomous and cooperative vehicles following systems has been studied. It is found that there exists minimum possible fixed headway spacing before the string stability of a convoy collapses which is related to the actual dynamics of the vehicle. The effect of this fixed headway spacing policy is equivalent to the introduction of additional damping in the transfer function, which allows the poles of the transfer function to be moved independently from the zeros of the same transfer function.

With the addition of the fixed headway spacing, (2) then becomes

$$G_1(s) = \frac{x_i(s)}{x_{i-1}(s)} = \frac{k_v s + k_p}{s^2 + (k_v + hk_p)s + k_p} \qquad (3)$$

with the control law developed as

$$u_i = k_p(x_{i-1} - x_i - hv_i) + k_v(v_{i-1} - v_i). \qquad (4)$$

To simplify the control law and at the same time ensure stability, a pole-zero cancellation technique is chosen. This can be achieved by introducing the constraint

$$\frac{k_p}{k_v} = hk_p \qquad (5)$$

Thus,

$$\frac{1}{k_v} = h .\qquad(6)$$

Fig. 3 shows the pole-zero map with pole and zero cancelling each other. Through the pole-zero cancellation technique, (3) is thus reduced to

$$G_2(s) = \frac{x_i(s)}{x_{i-1}(s)} = \frac{k_v}{s + k_v}\qquad(7)$$

which is a first order transfer function with a stable response with time delay. While k_v must always be positive, there exists a pole which is always in the left hand side of the s-plane. Thus, the system is always stable for this spacing policy.
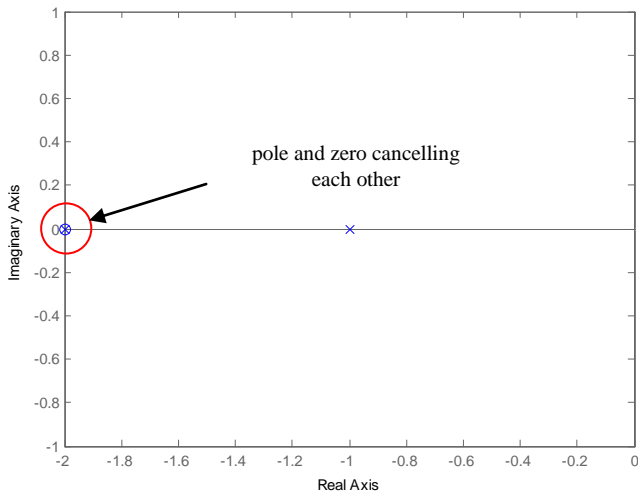

Figure 3.   Pole-Zero Cancellation Map

B. Inclusion Of Vehicle Dynamics

After proving that the fixed time headway policy is suitable to be adopted, a simplified vehicle dynamics model is introduced in order to mimic the actual vehicle internal dynamics. In this case, the external dynamics is not considered. In the simplified model, the internal dynamics is represented as a lag function i.e., the actual vehicle acceleration is obtained after a certain time delay τ. This is given by the relation in (8).

$$\tau\dot{a} + a = u\qquad(8)$$

(3) is modified to include the vehicle dynamics part and this gives a transfer function in (9)

$$G_3(s) = \frac{x_i(s)}{x_{i-1}(s)} = \frac{k_p + sk_v}{\tau s^3 + s^2 + s(k_p h + k_v) + k_p}\qquad(9)$$

Having designed the control strategy and by including the vehicle dynamics, the block diagram of the one-Vehicle Look-Ahead Vehicle Following System Is Shown In Fig. 4.
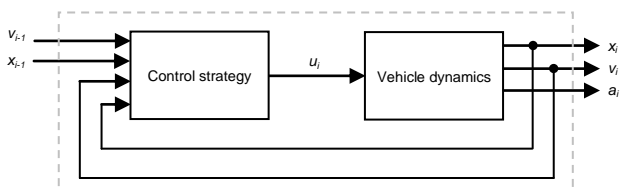

Figure 4.   Block Diagram Consisting Of The Control Strategy And Vehicle Dynamics

If τ Is So Small (As In An Ideal Vehicle) I.E. $\tau \approx 0$, Then $s^3 \to 0$. Hence, The Transfer Function Is Reduced Back To A Second Order Transfer Function,

$$G_4(s) = \frac{x_i(s)}{x_{i-1}(s)} = \frac{k_p + sk_v}{s^2 + s(k_p h + k_v) + k_p} .\qquad(10)$$

(10) Is Then Simulated By Using MATLAB Simulink Giving A Speed Input For H=1s And Kp Values Of 0.5, 1, 2 And 3. Shown In Fig. 5 And Fig. 6 Are The Speed And Acceleration Responses, Respectively. Further Analysis Of Fig. 5 Shows That Kp Value Of 2 Gives The Best Response.
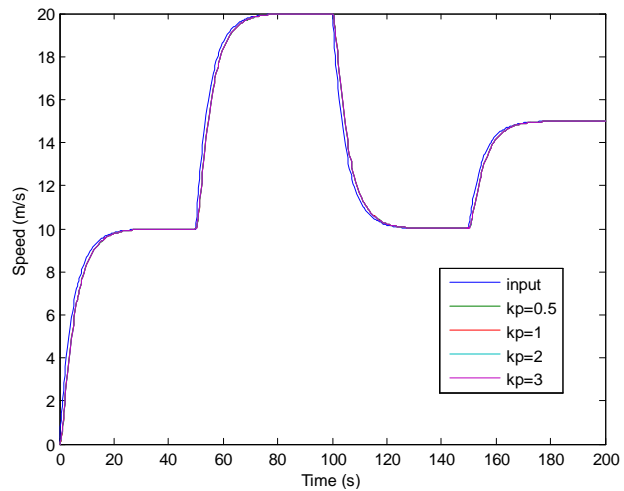

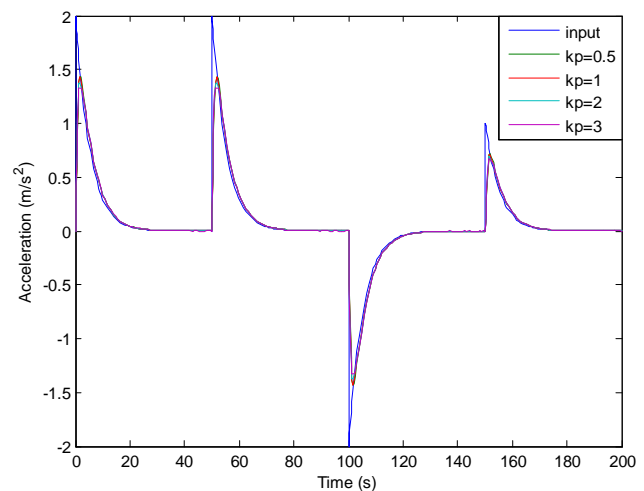Figure 5.   Speed Response Of Fixed Headway With Various *Kp* Values

Figure 6.   Acceleration Response Of Fixed Headway With Various *Kp* Values

III.   MODEL REFERENCE ADAPTIVE CONTROL

An adaptive controller can modify its behaviour in response to changes in the dynamics of a system and the character of any disturbance. It is a controller with adjustable parameter and a mechanism for adjusting the parameter. An adaptive control system consists of two loops, normal feedback loop with plant and controller and

an adaptive parameter mechanism loop. Fig. 7 illustrates the general structure of a Model Reference Adaptive Control (MRAC)system.
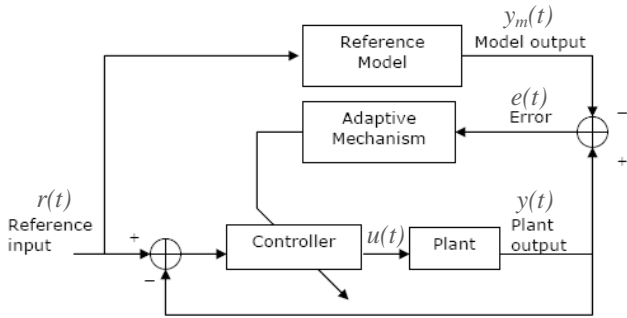

Figure 7.   General Structure Of An MRAC System

The basic MRAC system consists of four main components:

i) Plant to be controlled

ii) Reference model to generate desired closed loop output response

iii) Controller that is time-varying and whose coefficients are adjusted by adaptive mechanism

iv) Adaptive mechanism that uses 'error' (the difference between the plant and the desired model output) to produce controller coefficient

Regardless of the actual process parameters, adaptation in MRAC takes the form of adjustment of some or all of the controller coefficients so as to force the response of the resulting closed-loop control system to that of the reference model. Therefore, the actual parameter values of the controlled system do not really matter.

A. The Stability Approach

The MRAC can be designed such that the globally asymptotic stability of the equilibrium point of the error difference equation is guaranteed. To do this, the Lyapunov Second Approach or stability approach is used. The term stability approach is used throughout this paper. It requires an appropriate Lyapunov function to be chosen, which could be difficult. This approach has stability consideration in mind and is also known as the Lyapunov Approach.

In designing the MRAC controller, we would like the output of the closed-loop system (y) to follow the output of the reference model (y_m). Therefore, we aim to minimise the error ($e=y$-y_m) by designing a controller that has one or more adjustable parameters such that a certain cost function is minimised.

IV. ADAPTIVE GAIN CONTROLLER DESIGN

An adaptive gain controller is to be designed for the one-vehicle look-ahead control strategy with fixed time headway and vehicle dynamics by applying a Model Reference Adaptive Control (MRAC). This section presents a direct adaptive controller design which adapts the unknown vehicle parameter k_p. The advantage of the adaptive approach is that unpredictable changes in the value of k_p can be easily accommodated. Fig. 8 on the other hand shows the one-vehicle look-ahead control strategy based controller with the gain parameter to be

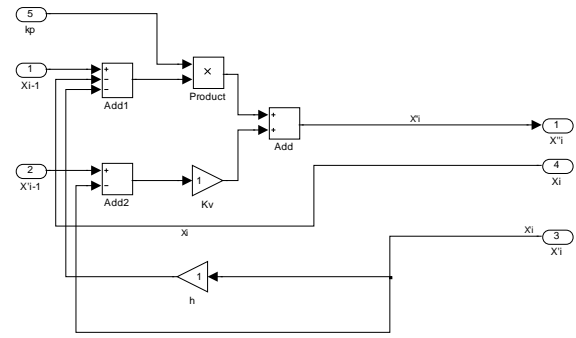adjusted, k_p is the gain adjustment mechanism to be designed in the control strategy.

Figure 8.   A One-Vehicle Look-Ahead Controller With Adjustable Gain

A. The Stability Approach Design With Perfect Modelling

From the analysis of Fig. 6, k_p value of 2 gives the best response. So, it will be used in (10) to give a reference model to be used in designing the adaptive gain controller. The vehicle dynamic has been included in the control law to form the plant.

Plant:

$$y = \frac{s + k_p}{s^2 + (1 + k_p)s + k_p} r \qquad (11)$$

Reference model:

$$y_m = \frac{s + 2}{s^2 + 3s + 2} r \qquad (12)$$

A closed-loop system with a controller has the following parameters:

$r(t)$ = Reference input signal

$u(t)$ = Control signal

$y(t)$ = Plant output

$y_m(t)$ = Reference model output

$e(t)$ = Difference between plant and reference model output

       = $y(t)$ - $y_m(t)$

For perfect model following, the denominator of (11) is assumed to be the same as the denominator of (12). So, in designing an MRAC using stability approach with perfect modelling, the following steps should be followed:

i) Derive a differential equation for error, $e = y - y_m$ (i.e. $\dot{e}, \ddot{e}$ etc.) that contains the adjustable parameter, k_p. From (11) and (12), after replacing, the differential equations become

$$\ddot{y} = -3\dot{y} - 2y + (s + k_p)r \qquad (13)$$

$$\ddot{y}_m = -3\dot{y}_m - 2y_m + (s + 2)r . \qquad (14)$$

Substituting (13) and (14) into $\ddot{e} = \ddot{y} - \ddot{y}_m$, thus

$$\ddot{e} = -3\dot{y} - 2y + (s + k_p)r - [-3\dot{y}_m - 2y_m + (s + 2)r]$$

$$\ddot{e} = -3\dot{e} - 2e + (k_p - 2)r . \qquad (15)$$

Let $k_p - 2 = X$, so that $\dot{X} = \dot{k}_p$.

$$\ddot{e} = -3\dot{e} - 2e + Xr . \qquad (16)$$

ii) Find a suitable Lyapunov function, usually in a quadratic form (to ensure positive definiteness).

The Lyapunov function, $V(\dot{e}, e, X)$, is based on (16).

$$V = \lambda_1\dot{e}^2 + \lambda_2 e^2 + \lambda_3 X^2 \quad , \quad \text{where} \quad \lambda_1, \lambda_2, \lambda_3 > 0 \quad \text{is}$$

positive definite. The derivative of *V* becomes

$$\dot{V} = -6\lambda_1\dot{e}^2 + -4\lambda_1 e\dot{e} + 2\lambda_1\dot{e}Xr + 2\lambda_2 e\dot{e} + 2\lambda_3 X\dot{X} \quad (17)$$

where for stability $\dot{V}$ must be negative i.e. $\dot{V} < 0$.

iii) Derive an adaptation mechanism based on $V(\dot{e}, e, X)$ such that e goes to zero.

$$\dot{X} = -\frac{\lambda_1}{\lambda_3}\dot{e}r \quad (18)$$

Therefore,

$$k_p = -\gamma er \quad (19)$$

The block diagram implementation is given in Fig. 9 where γ is denoted by gamma in the simulation diagram and the red dotted line is the adaptive mechanism. This approach will be known as perfect modeling approach in the paper afterwards.
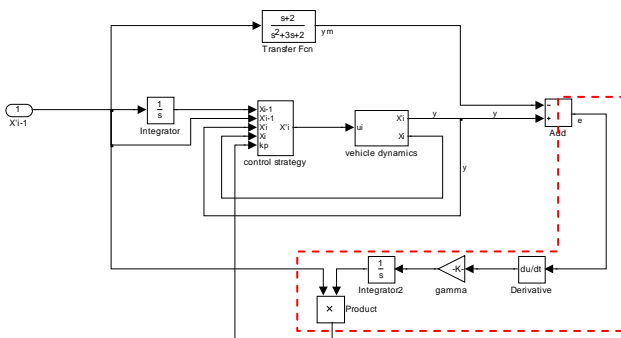

Figure 9. Simulation Diagram For The Stability Approach Adaptive Gain Controller With Perfect Modelling.

The MRAC stability approach adaptive gain controller design with perfect modelling is then simulated again using MATLAB Simulink. Both the output of the system responses (*y* and y_m) are shown in Fig. 10 and Fig. 11. Fig. 10 shows a perfect model following output while Fig. 11 on the other hand shows the acceleration response of *y* where it does not follow a sharp change in input acceleration.
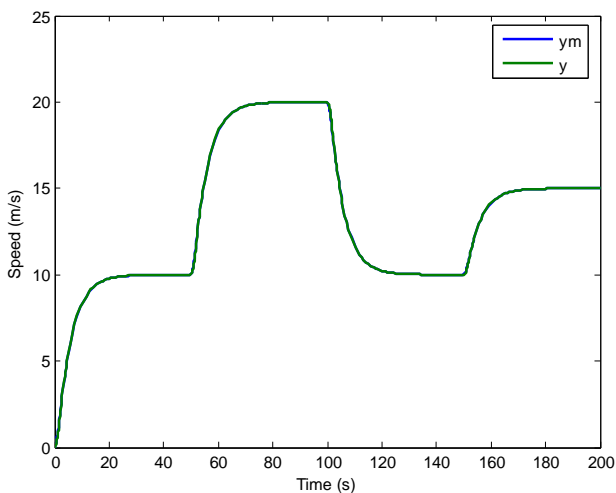

Figure 10. Comparison Of Y And Y_m For Speed From The Stability approach with perfect modelling
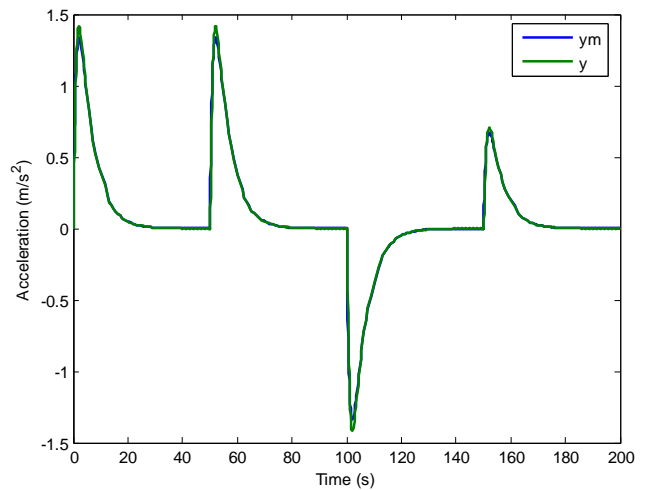

Figure 11. Comparison Of Y And Y_m For Acceleration From The Stability Approach Perfect Modelling

It can be seen that the output response for plant *y* perfectly follows the reference model y_m. It can be said that the system is a perfect model following system.

The adaptive gain controller is again simulated but this time with gamma γ values of 0.1,0.01 0.001 and 0.0001 while the value of *h* is fixed at 1s.

Shown in Fig. 12 and Fig. 13 are the speed and acceleration responses for the adaptive gain controller, respectively. Further analysis of Fig. 13, gamma values of between 0.01 and 0.001 gives the best response by almost fitting the reference model with smooth curve and with jerk of less than 5 m/s^3.
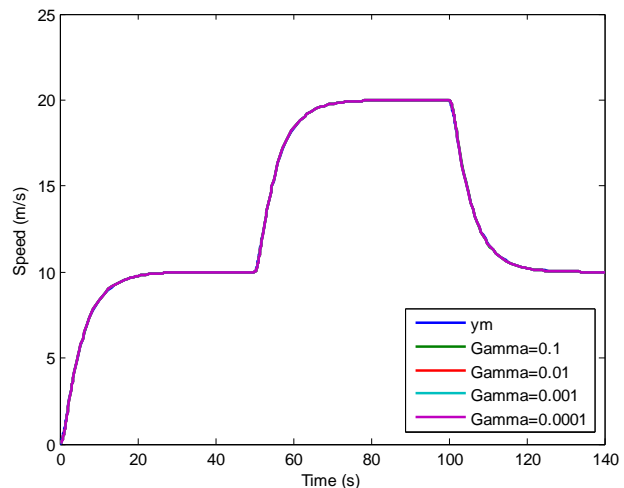

Figure 12. Speed Response With Various Gamma Values For Adaptive Gain Controller
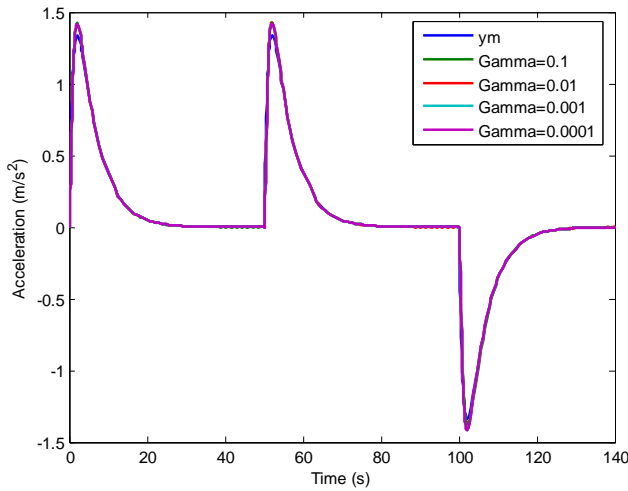
Figure 13. Acceleration Response With Various Gamma Values For Adaptive Gain Controller

B. *The Stability Approach Design With Real Modelling*

For this design, the steps taken in designing the controller are the same as before but with the denominator of (20) remains. So, with the steps repeated, we have followings.

Plant:

$$y = \frac{s + k_p}{s^2 + (1 + k_p)s + k_p} r \qquad (20)$$

Reference Model:

$$y_m = \frac{s + 2}{s^2 + 3s + 2} \cdot r \qquad (21)$$

i) Derive a differential equation for error, $e = y - y_m$ (i.e. $\dot{e}, \ddot{e}$ etc.) that contains the adjustable parameter, k_p. From (20) and (21), after replacing, the differential equations then become

$$\ddot{y} = -(1 + k_p)\dot{y} - k_p y + \dot{r} + k_p r \qquad (22)$$

$$\ddot{y}_m = -3\dot{y}_m - 2y_m + \dot{r} + 2r \qquad (23)$$

Substituting (22) and (23) into $\ddot{e} = \ddot{y} - \ddot{y}_m$, thus

$$\ddot{e} = -3\dot{e} - 2e - (k_p - 2)\dot{y} - (k_p - 2)y + (k_p - 2)r. \qquad (24)$$

Let $k_p - 2 = X$, so that $\dot{X} = \dot{k}_p$. Thus

$$\ddot{e} = -3\dot{e} - 2e - X(\dot{y} + y - r). \qquad (25)$$

ii) Find a suitable Lyapunov function, usually in a quadratic form (to ensure positive definiteness).

The Lyapunov function, $V(\dot{e}, e, X)$, is based on (25). $V = \lambda_1 \dot{e}^2 + \lambda_2 e^2 + \lambda_3 X^2$, where $\lambda_1, \lambda_2, \lambda_3 > 0$ is positive definite. The derivative of *V* becomes $\dot{V} = -6\lambda_1 \dot{e}^2 - (4\lambda_1 - 2\lambda_2)e\dot{e} - (2\lambda_1 \dot{e}(\dot{y} + y - r) - 2\lambda_3 \dot{X})X \qquad (26)$

where for stability $\dot{V}$ must be negative i.e. $\dot{V} < 0$.

iii) Derive an adaptation mechanism based on $V(\dot{e}, e, X)$ such that e goes to zero.

$$\dot{X} = \frac{\lambda_1}{\lambda_3} \dot{e}(\dot{y} + y - r) \qquad (27)$$

Therefore,

$$k_p = \gamma e(y - r) + \gamma e(sy). \qquad (28)$$

The block diagram implementation is given in Fig. 14 where γ is denoted by gamma in the simulation diagram and the red dotted line is the adaptive mechanism. This approach will be known as real modeling approach in the paper afterwards.
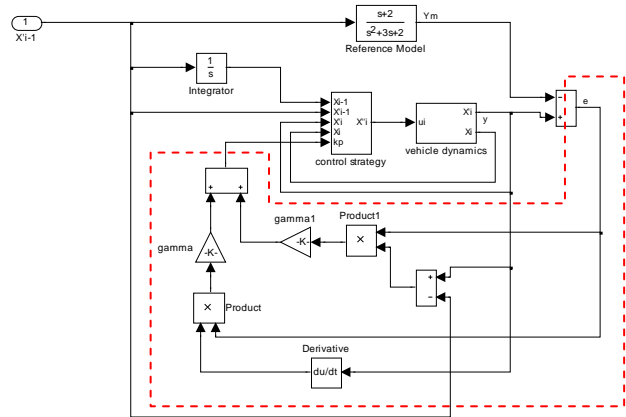

Figure 14. Simulation Diagram For The Stability Approach Adaptive Gain Controller With Real Modelling.

The MRAC stability approach adaptive gain controller design with real modelling is then simulated using MATLAB Simulink. Both the output of the system responses (*y* and *y_m*) are shown in Fig. 15 and Fig. 16. Fig. 15 shows a perfect model following output while Fig. 16 on the other hand shows the acceleration response of *y* where it does not follow a sharp change in input acceleration.
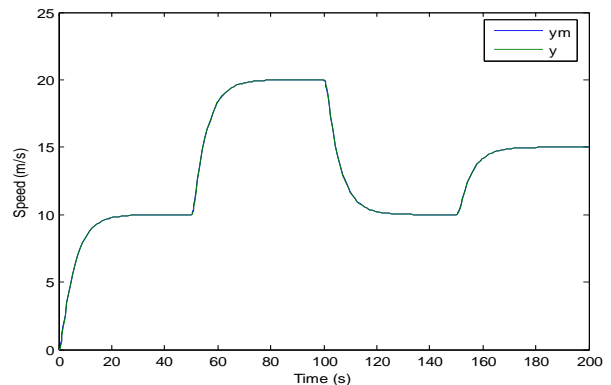

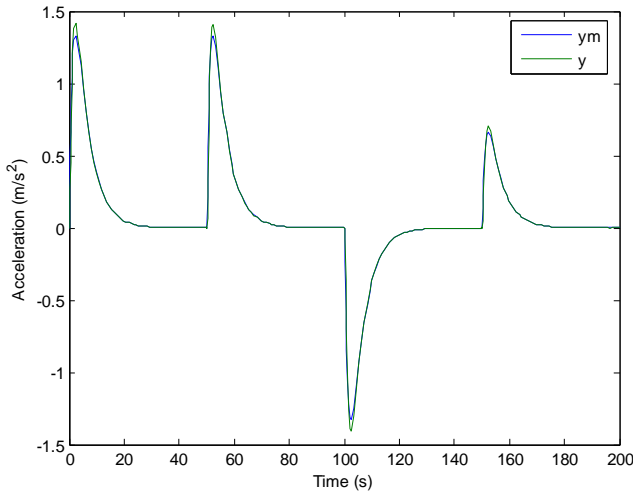Figure 15. Comparison Of Y And Y$_m$ For Speed From The Stability Approach With Real Modelling

Figure 16. Comparison Of Y And Y_m For Acceleration From The Stability Approach With Real Modelling

It can be seen that the output response for plant y perfectly follows the reference model y_m. It can be said that the system is also perfect model following system.

The adaptive gain controller is again simulated but this time with gamma γ values of 0.1,0.01 0.001 and 0.0001 while the value of h is fixed at 1s.

Shown in Fig. 17 and Fig. 18 are the speed and acceleration responses for the adaptive gain controller, respectively. Further analysis of Fig. 18, gamma values of between 0.01 and 0.001 gives the best response by almost fitting the reference model with smooth curve and with jerk of less than 5 m/s^3, same as in the first design.
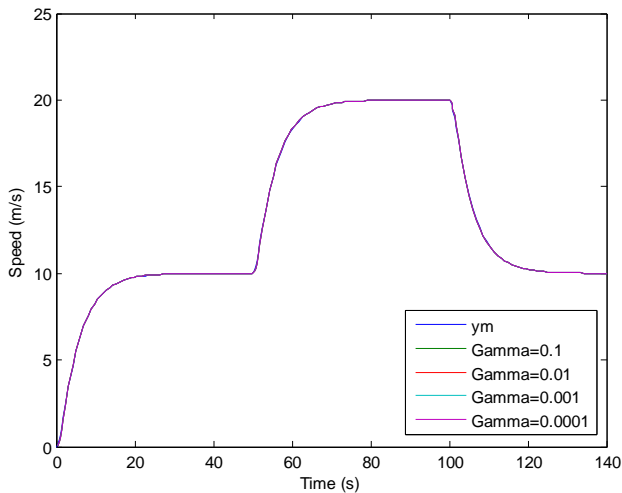

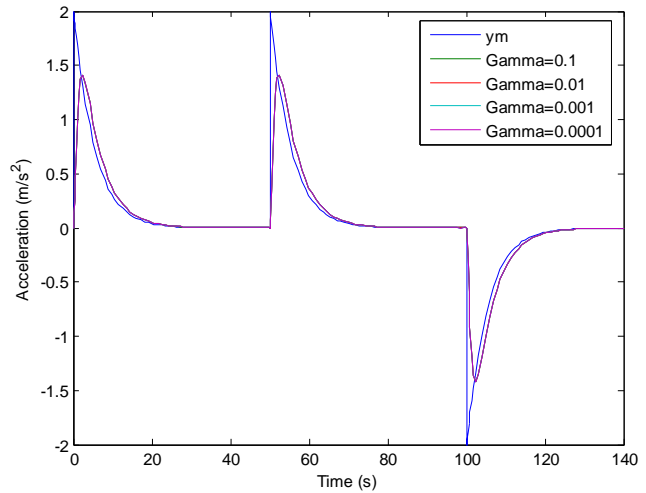Figure 17. Speed Response With Various Gamma Values For Adaptive Gain Controller

Figure 18. Acceleration Response With Various Gamma Values For Adaptive Gain Controller

V. ANALYSIS

The two controller designs from the two approaches are the compared. Both controllers use the gamma value of 0.01 as the value gives the most acceptance jerk responses for both approaches. As both approaches produce perfect model following in terms of the speed response, only the acceleration response is made into comparison, shown in Figure 19.

In Figure 19, no significant difference between the two approaches is observed. However, when the two approaches are compared through their jerk responses as shown in Figure 20, the approach with the real modeling is found to give the best result as it causes the lowest jerk. This is still acceptable as the plots somehow show the real situation when the vehicle internal dynamics is taken into consideration.
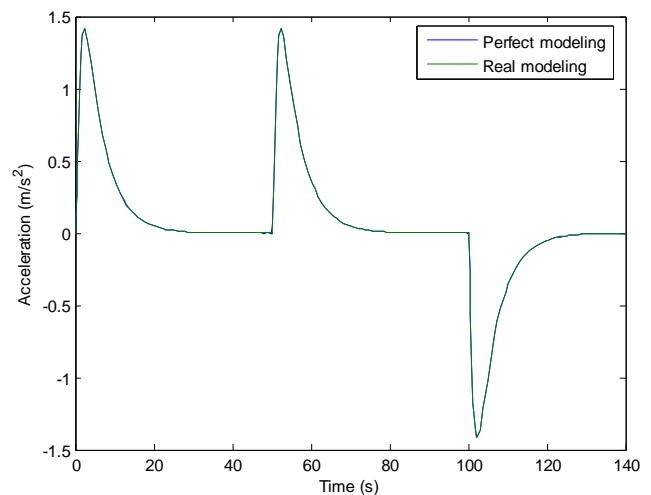

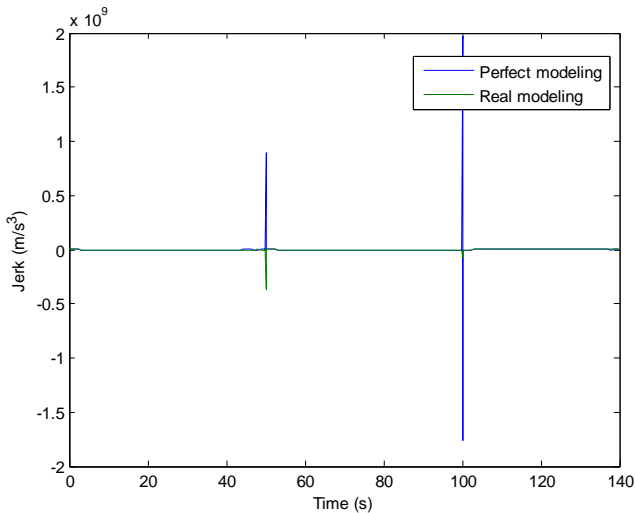Figure 19. Acceleration Responses For The Two Stability Approaches

Figure 20. Jerk Response Comparison For Both Approaches

VI.   RESULTS DISCUSSION

A one-vehicle look-ahead control strategy with fixed headway policy has been adopted in designing a controller to produce an output which can respond immediately to the change in input; in this case, the input is the speed with varying speed conditions. With normal controller, the response does not quite match perfectly with the input. With the introduction of the MRAC adaptive gain controller, the response can be made to follow the input by choosing a suitable reference model. Furthermore, using MRAC adaptive gain controller produces a smooth output as compared to the non-adaptive controller.

From the two MRAC adaptive gain controller stability approaches, the one with the lower jerk response is considered acceptable. So it can be said that in designing the stability approach controller, the actual parameter should be taken into consideration; in this case, the plant transfer function which does not perfectly mimic the reference model.

VII.   CONCLUSION

The adaptive gain controller tuning has been investigated using MRAC concepts through the stability approaches. Simple adaptation law for the controller parameters has been presented assuming that the process under control can be approximated by a second order transfer function. The developed adaptation rules from the two approaches have been applied and simulated. The results obtained show the effectiveness of the technique. The resulting performance could be improved by a better choice of the length of the adaptation period.

The stability approach is used to provide guaranteed nominal stability. However, the stability approach controller can only have very small gain with 1 is the maximum limit. A further limitation of the approach is the assumption of a structure for the nominal system. In this paper, a second order transfer function has been assumed resulting from the assumption of a very small time delay between the command signal and the vehicle dynamics as in ideal vehicle.

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