Design of Robust Compensating Controller for Lateral Motion of the Vehicle

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Received on:13/1/2013 & Accepted on:4/7/2013

ABSTRAC

In this paper, robust compensating controller is proposed to control the lateral velocity and yaw rate of the vehicle during turning .This controller uses the errors of the lateral velocity and yaw rate as input and the front steering angle and differential brake force as output.

A vehicle model of dynamic lateral motion is heavily influenced by vehicle parameters such as vehicle speed, vehicle mass and road-tire interaction. These vehicle parameters vary during operation; therefore variations of parameters are taken into considering in controller design. The compensators of the controller become function to variations of vehicle parameters to improve the dynamic responses of lateral velocity and yaw rate. The simulation results of the proposed controller with lag brake and front steering models give acceptable responses through reaching to the desired conditions with short transient period and steady – state error is equal to zero for different cases.

Keywords: Robust Controller, Vehicle Dynamics

تصميم مسيطر تعويضي نشيط للحركة الجانبية لمركبة

الخلاصة

في هذا البحث تم تقديم مسيطرمن النـوع التعويضـي (compensating controller) للسيطرة على الحركة الجانبية والدورانية للمركبة اثناء انعطافها. هذا المسيطر يستخدم الخطأ في السرعة الجانبية للمركبة والخطأ في معدل الدوران كادخال له ،اما اخراجه فيكون زاوية توجيه العجلات الامامية و قوة الفرملة الفرقية على عجلات اليسار واليمين في المركبة.

ان ديناميكية الحركة الجانبية والدورانية تتاثر بشكل كبير بتغير عوامل (parameters) المركبة اثناء حركتها مثل سرعة المركبة وحالة العجلات مع نوعية الطريق، لذلك تم جعل معوضات (compensators) المسيطر دالة لعوامل المركبة من اجل تحسين الاستجابة الديناميكية للسرعة الجانبية والدورانية.

https://doi.org/10.30684/etj.31.17A.5
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المحاكـاة لهـذا المسـيطر عنـد تغيـر عوامـل المركبـة أظهـرت اسـتجابة جيـدة للسـرعة الجانبيـة والدورانية من خلال الوصل إلى الحالة التصميمية المطلوبـة وبـزمن مـرور قصـير وخطـا المسـتقر يساوي صفر عند مختلف الظروف .

INTRODUCTION

The automatic control systems in vehicles become very important today and concerning the researchers and vehicle companies. The increasing in the vehicle performance, reducing the driver's efforts, and giving more safety for passengers and the people that use the road. All these achieved by the existence of different automatic control systems in the vehicles.

The automatic control in vehicle can be classified to three fundamental types: 1-longitudinal control, which keeps the interval between vehicles by controlling vehicle speed [1], 2-lateral control, which maintains the lateral position of the vehicle in lane by controlling steering angels or by differential brake [2], 3-rollover control, which prevent vehicle from rollover by controlling on the suspension system [3].

In lateral position control which will focus during this research, the vehicle must follow specified curved path during turning. To achieve this, the lateral velocity must be zero and the required yaw rate must be achieved. The independent control of lateral and yaw motion require at least one additional control input which is independent on the front steering angle. A four wheel steering system whose control inputs are front and rear steering angle is one of the solutions [4]. Also there are other solutions such as control on the transferred engine torque to each wheels [5] and control the distribution brake force on each wheels ,this way is consider best control means and very popular way used today in the vehicles.

In recent years, there is several control algorithms applied to vehicle lateral position. Among them, optimal control [6,7,8],adaptive control[9], fuzzy control [10,11] and neural network control[12] are the most primary control methods. The linear controllers are still using in all practical applications in different fields, and researchers tried to increase the robustness of theses controllers by made its gains adaptive with variations of work condition of the systems.

Practically, a vehicle model of lateral motion dynamics is heavily influenced by vehicle parameters such as vehicle speed, vehicle mass and road-tire interaction. These vehicle parameters vary during operation. Therefore the variations of parameters are taken into considering into presented controller design. The compensators of the suggested controller are made as function of vehicle parameters to improve the lateral response for vehicle and make the controller more robust.

In this paper, the differential brake force on the right and lift wheels are used as additional controller actuator with front steering angle to control on the lateral velocity and yaw rate.

VEHICLE MATHEMATICAL MODEL

The vehicle model proposed by Tom et al. [6] is used in this paper. The equations of lateral motion are formed using figure (1) they are written in state variable from as:

$$\dot{x} = Ax + Bu \qquad\qquad y = cx \qquad\qquad \ldots (1)$$

Where:

$$A = \begin{bmatrix} -\dfrac{C_f + C_r}{mu} & -\dfrac{C_f a - C_r b}{mu} - u \\ -\dfrac{C_f a - C_r b}{Iu} & -\dfrac{C_f a^2 + C_r b^2}{Iu} \end{bmatrix}, \qquad x = \begin{bmatrix} v_y \\ r \end{bmatrix},$$

$$B = \begin{bmatrix} \dfrac{C_f}{m} & 0 \\ \dfrac{aC_f}{I} & \dfrac{d}{I} \end{bmatrix}, \quad u = \begin{bmatrix} \delta_f \\ F_{db} \end{bmatrix}, \quad c = \begin{bmatrix} 1 & 0 \\ 0 & 1 \end{bmatrix}$$

The term ($F_{db} = (F_{x1} + F_{x3}) - (F_{x2} + F_{x4})$) represent the differential brake forces between lift and right side of the vehicle for small δ_f ($\cos\delta_f = 1$). The state-variables and the parameters are defined in list of symbols.

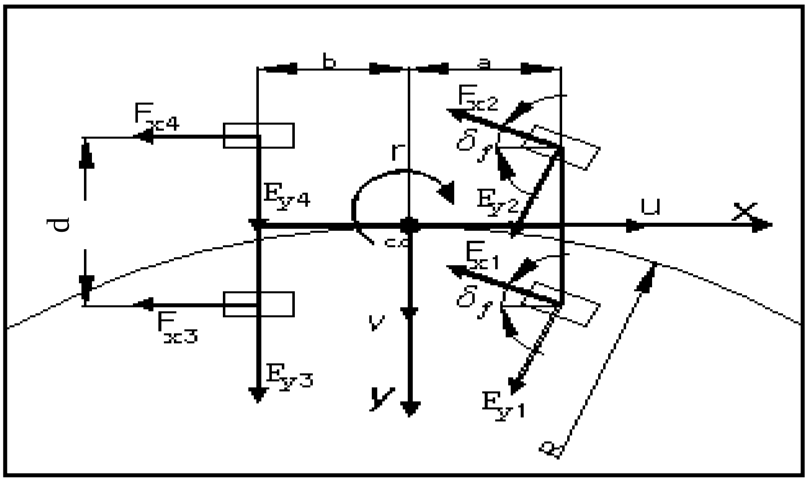

Figure (1) Vehicle model.

It is clear from Equation (1), the variables that must be controlled are lateral velocity and yaw rate. The lateral velocity must become minimum as possible as to prevent the vehicle from slipping out the path and the yaw rate must equal to the desired value that make the vehicle achieve turn in correct path.

At certain vehicle velocity (u) there is a minimum radius (R) of curved path which the vehicle can turn it easily without slipping or rollover. This depend on height of vehicle center of gravity (h) and base wheel (d). Therefore the maximum value of yaw rate (r_{max}) will be limited as shown in Equations (2 ,3)

$$R_{\min} = u^2/(gd/h) \quad \text{.... (2)}$$

$$r_{\max} = (gd/h)/u \quad \text{... (3)}$$

The Equation (1) shows the inputs of the system (δ_f , F_{db})which will be used as controller actuators.

CONTROLLER DESIGN

To achieve the accurate control on the lateral motion of the vehicle, this requires existence of controller has ability to give fast required responses without steady state error. So the responses must have very short period for both transient response and rise time without over shoot. The suggested compensating controller, that its structure shown in Figure (2) can specify the specification of required dynamic response as the designer wants.

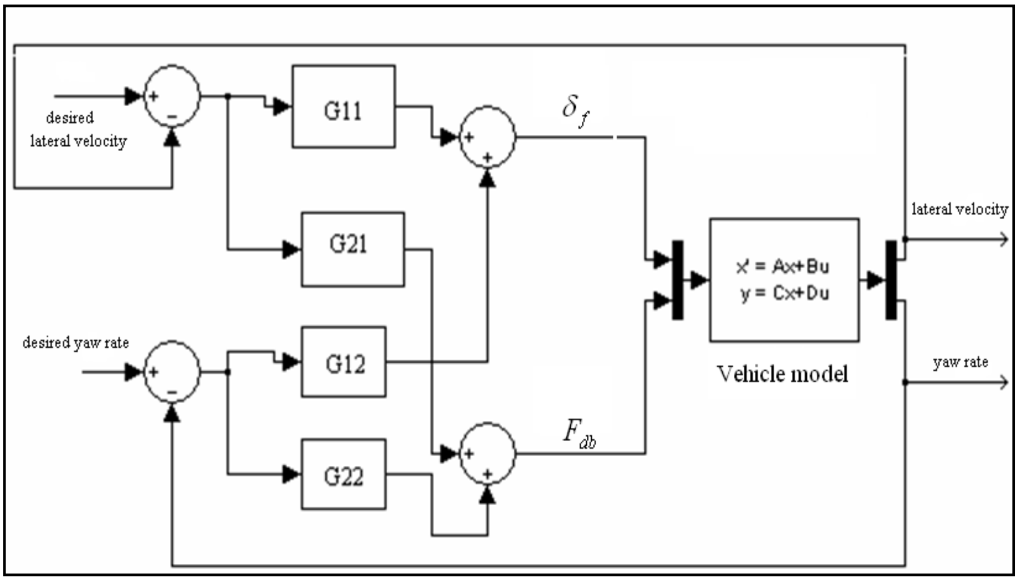

Figure (2) Block diagram for compensator

The procedure of the controller methodology of the compensators G_{c11} , G_{c12}, G_{c21} , G_{c22} as follows:

1- Convert the equation of motion (1) to transfer function form [13]

$$G_p(s) = C(sI - A)^{-1}B \quad \text{... (4)}$$

2- Specify the required responses (G) for lateral velocity and yaw rate for fast response of stable system as follows:

$$(G = \begin{bmatrix} \dfrac{1}{0.1s+1} & 0 \\ 0 & \dfrac{1}{0.1s+1} \end{bmatrix}) \qquad \ldots (5)$$

3- Find the values of compensators (G_c)

$$(G_c = G_p^{-1} G_o) \qquad \ldots (6)$$

Where:

$$G_o = G(I-G)^{-1}$$

For the nominal values of vehicle parameters the compensators are This controller becomes unrobust (insensitive) when the parameters of the vehicle are changing.

$$\left.\begin{aligned} G_{c11} &= \frac{1/605s + 2/121}{0.11s+1} \times \frac{11s+100}{s} \qquad G_{c12} = \frac{7/44(s+10)}{0.1s^2+s} \\ Gc_{21} &= \frac{1/11(-4000/3s - 15000)}{0.11s+1} \times \frac{11s+100}{s} \\ Gc_{22} &= \frac{2000s + 27500/3}{s} \times \frac{s+10}{0.1s+1} \end{aligned}\right\} \qquad \ldots (7)$$

So in this paper, this controller is developed to be more robust by making its compensators as function of the vehicle parameters (mass, velocity, cornering coefficient for front and rear tires). The cornering coefficient is function of other parameters like air pressure in tire, normal load the conditions between the road and tire [14].

So the compensators take the form below

$$\left.\begin{aligned} G_{c11} &= \frac{(s+10)(c_f + c_r + mus)}{(c_f us(s/10+1))} \\ G_{c12} &= \frac{(s+10)(mu^2 + c_f - (3c_r)/2)}{(c_f us(s/10+1))} \\ Gc_{21} &= \frac{-((s+10)(-(4c_f^2) + (10c_f c_r) + (4c_r^2) + (4muc_r s)))}{3(c_f us(s/10+1))} \\ Gc_{22} &= \frac{(s+10)((4c_f^2)/3 + (5c_f c_r)/3 + (4Ic_f us)/3 + 2c_r^2 - (4mc_r u^2)/3)}{(c_f us(s/10+1))} \end{aligned}\right\} \qquad \ldots (8)$$

To give the design of the controller more reliability, dynamic lag term and saturation limit are introduced to represent the brake and steering models. Brake system is modeled as a first order lag with time constant ($\tau_b = 0.1$)[6] such that

$$(\dot{F}_b = -\frac{F_b}{\tau_b} + \frac{F_{b\,comanf}}{\tau_b}) \qquad \ldots (9)$$

The model can be interpreted as the dynamic lag between a brake force command ($F_{b\ command}$) and the resulting brake force (F_b) .Saturation limit for brake force is 3600 N in order to prevent the tire from slipping .
Also the steering is modeled as a first order lag with time constant (τ =*0.25*) [15] such that

$$(\dot{\delta}_f = \frac{1}{\tau_{st}}\delta_f + \frac{1}{\tau_{st}}\delta_{f,comand}) \qquad \ldots (10)$$

Saturation limit for steering angle $\delta_f \leq 40^o (0.7rad)$.

SIMULATION

The performance of the robust controller with the compensators that represented in equation (8) is evaluated using closed loop response for the vehicle model equation (1) including the variations of the vehicle parameters with brake model equation (9) and steering model equation (10) as shown in Figure (3).

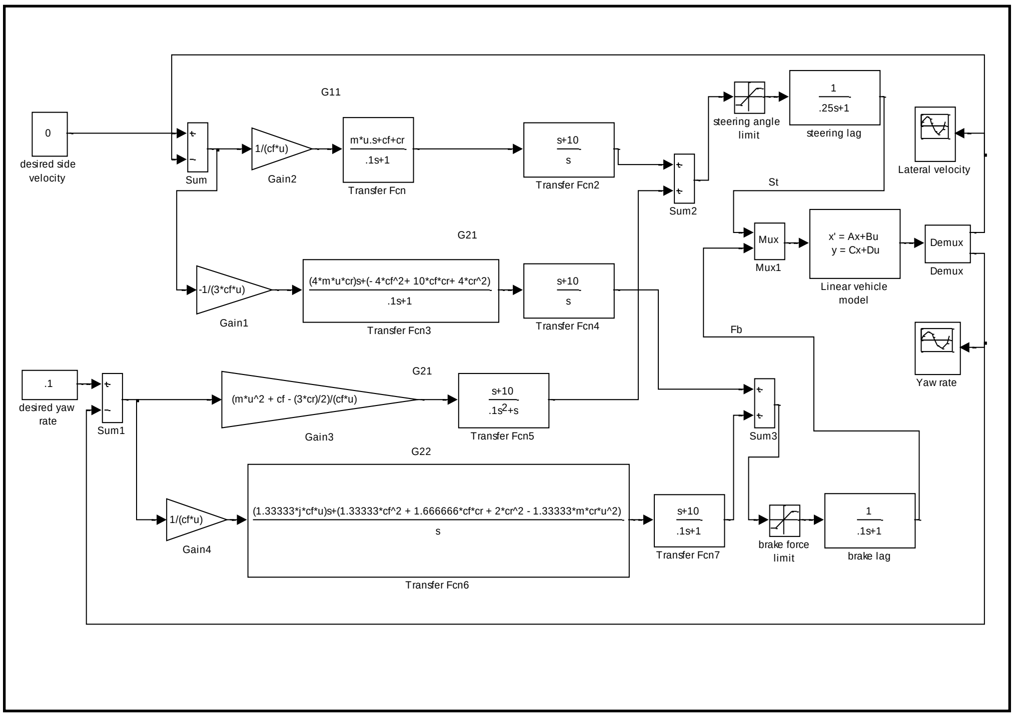

Figure (3) Robust controller

The desired lateral position must be zero to prevent the vehicle from slipping out the correct path and the desired yaw rate is chosen so not exceed the maximum value resulting from Equation (3).

Four cases are studied to test the suggested controller with three types of controller structure: robust controller with lag and saturation limit, robust controller only and unrobust controller.

Case-1

The simulation is performed using the nominal parameters of vehicle The lateral velocity and yaw rate responses are shown in Figures (4,5)

Case-2

In this case, the vehicle velocity is equal to 20m/sec and other parameters are nominal . The lateral velocity and yaw rate responses are shown in Figures (6, 7)

Case-3

The vehicle velocity is increased to 30m/sec , mass of 1400 kg and reduce the front cornering coefficient 20% from nominal value. The lateral velocity and yaw rate responses are shown in Figures (8, 9).

Case-4

In this case, the rear cornering coefficient is reduced 20% from nominal value, vehicle velocity 30m/sec and mass of 1400 kg. The lateral velocity and yaw rate responses are shown in Figures (10,11).

The responses of the differential brake and front steering for all cases as shown in Figure (12, 13) respectively. It can be seen that the responses do not exceed that saturation limits for brake force and steering angle. This is an indication the proposed controller is working in logical physical range for good performance of the vehicle.

DISCUSSION

From the dynamic responses for four above cases, it can be found that the suggested robust controller with lag and saturation elements gives the required responses for lateral velocity and yaw rate. The lateral velocity reaches to zero during short time in spite of existence short transient period with small deviation in lateral velocity about desired value.

The desired responses of yaw rate are achieved with rise time not exceed 0.5sec in all cases and with maximum overshoot 7% in case three.

In case-1 the performance of robust unrobust controller with lag is the same because the unrobust controller is built using nominal parameters which used in case-1. The case-3 and case-4 are considered as difficult cases that the vehicle may be suffered instability condition because of reducing the cornering coefficient (this may be happened due to reduce in tire air pressure [15]) lead to reduce the stability of the vehicle to dangerous level.

It can be seen that the suggested controller gives a good performance until in instability region in comparing with unrobust controller. Because the dynamics change due to vehicle velocity, mass, and cornering stiffness variation was addressed in controller design, therefore the proposed controller became more robust.

Also the present controller can give desired values for lateral velocity and yaw rate with short transient responses about 2.5 sec for all cases while the transient responses in works [6,17] are more than 4.5 sec.

CONCLUSIONS

The dynamic behavior of a vehicle is heavily influenced by changing of the vehicle parameters. Therefore; the performance of any controller may become unrobust and unstable. In this paper, the compensators of the suggested controller are made as function of vehicle speed, vehicle mass and road-tire interaction to improve the lateral response for vehicle and make the controller more robust.

The simulation results show that when the proposed controller is engaged with the lateral vehicle model, the yaw rate can follow its desired value and lateral velocity reaches to zero during short transient period.

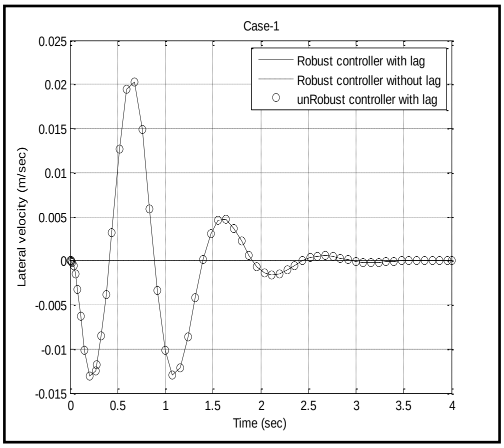

Figure (4) Lateral velocity response for case-1.

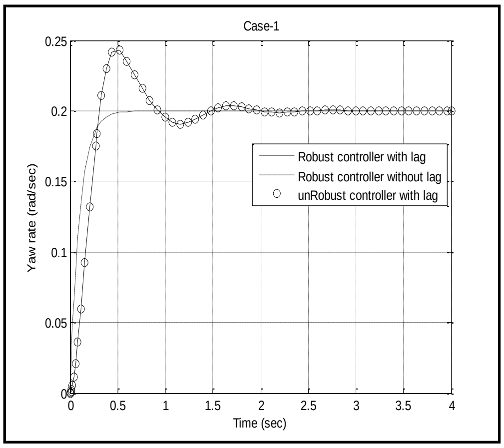

Figure (5) Yaw rate response for case-1.

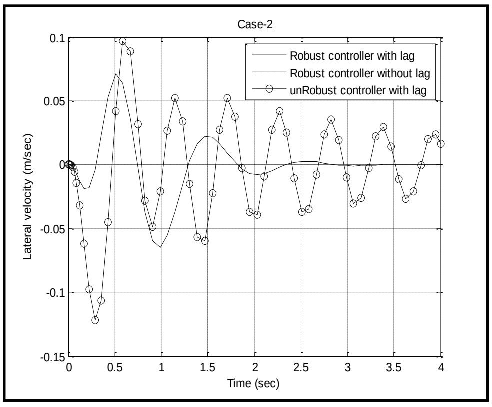

Figure (6) Lateral velocity response for case-2.

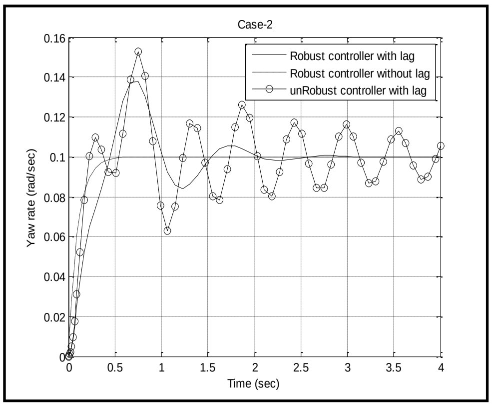

Figure (7) Yaw rate response for case-2.

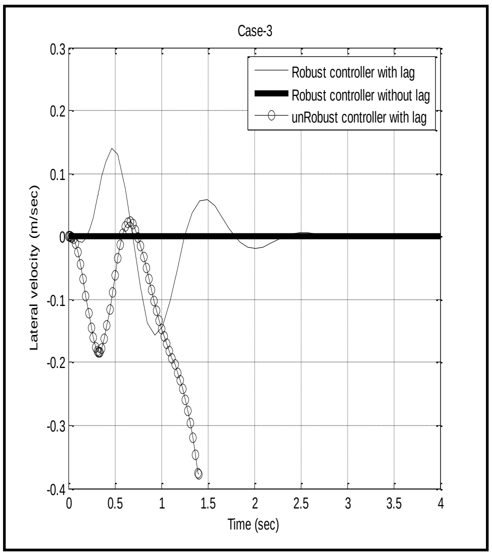

Figure (8) Lateral velocity response for case-3.

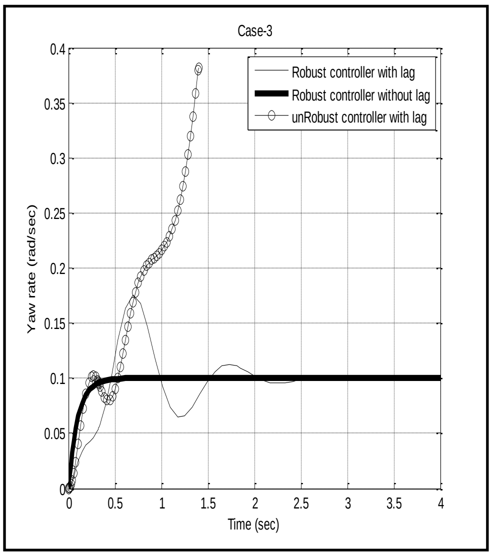

Figure (9) Yaw rate response for case-3.

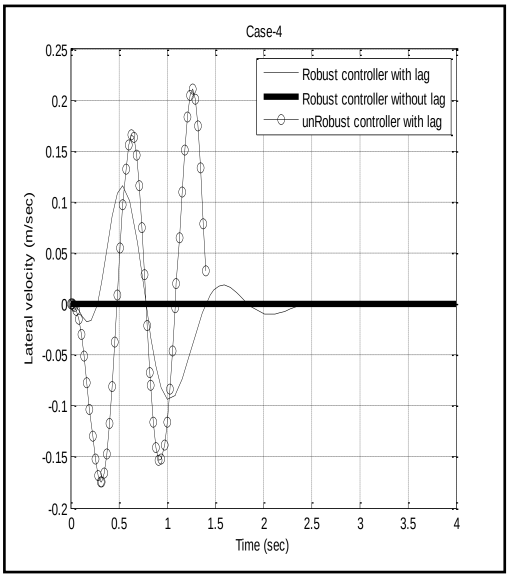

Figure (10) Lateral velocity response for case-4.

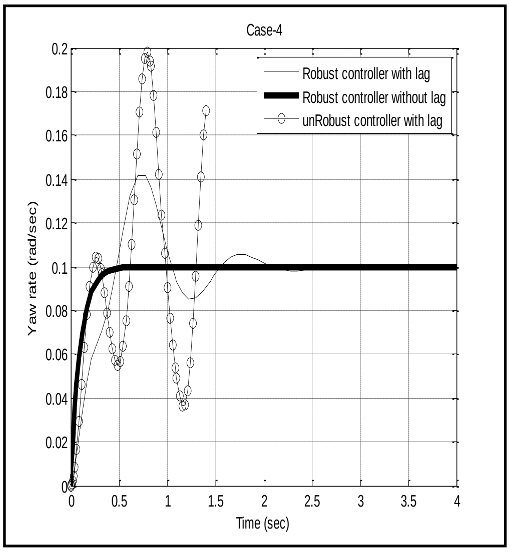

Figure (11) Yaw rate response for case-4.

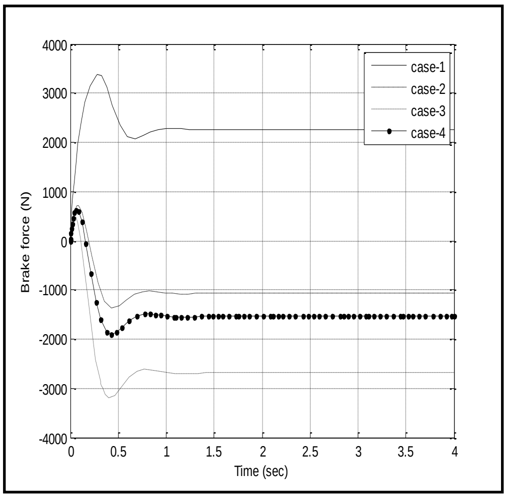

Figure (12) Differential brake response for all cases.

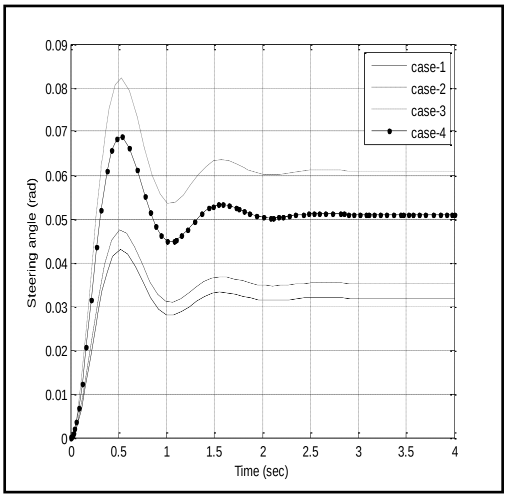

Figure (13) Front steering angle response for all cases.

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LIST OF SYMBOLS.

Symbol	Description	Value	Unit
a	Length from mass center to front axle	1	m
b	Length from mass center to rear axle	1.5	m
C_f	Front cornering coefficient	55000	N/rad
C_r	rear cornering coefficient	45000	N/rad
d	base wheel	1.5	m
F_{xi}	longitudinal tire force for each tire i	-	N
F_{db}	Differential brake force	-	N
h	Height of Center of gravity of Vehicle	0.45	m
I	Mass moment of inertia	1500	Kg m^2
m	Vehicle mass	1000	Kg
r	Yaw rate	-	Rad/sec
R	Radius of curvature	-	m
u	Vehicle velocity	10	m/sec
v	Lateral velocity	-	m/sec
δ_f	Front steering angle	-	rad
τ_b	Brake constant time	0.1	Sec
τ_{st}	steering constant time	0.25	Sec