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STATE-FEEDBACK CONTROL APPLIED TO A PENDULAR SUSPENSION

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Abstract: The agriculture sprayer is widely used to control the population of weed, fungus, and insects at crops as a result these populations affects directly the agriculture yield gains. The sprayer booms are responsible to apply the phytosanitary agents to the crops and their length is directly related to the machine efficiency, this fact leads to the seek of wider booms, some up to 50 m wide. But at a wider boom small roll angle variations will generate greater nozzle tips height to crop variations, which has an ideal height to ensure great sprayer distribution and dosage maximizing its efficiency. For this reason, a boom suspension has to ensure a stable boom that will also reduce the impacts on the environment due drift or pesticides overdose and underdose. In this paper, a control of a sprayer boom pendulum suspension using state-feedback is proposed. The control is designed to obtain the desired performance in a closed loop system in terms of transient response and stability. A pole allocation technique was used to define the response time and the settling time. A sprayer chassis model and boom suspension of the passive and state-feedback controlled suspension were created using simulation tools (Simulink/Matlab®), and the road input signals for the simulations were the ISO 5008:2002 rough track at 6 km/h. The performance was measured by comparing the responses between the two boom suspension systems, where the minimum disturbance was the goal. The simulation results showed a significant improvement on the state-feedback control when compared to a passive suspension. In comparison with the passive suspension, the state-feedback control system showed a minimum disturbance, resulting in great boom stability and equipment performance. The state-feedback technique showed to be a more stable and quick response which leads the system to a minimum disturbance.

Keywords: state-feedback control, active suspension, dynamic modeling, sprayer boom suspension

INTRODUCTION

With the increase of grain production demand due to continuous growth of the world population, the use of chemical pesticides have a major impact to improve food production (Carvalho, 2017). The boom motion disturbance resulting of a uneven ground can modify the spraying pattern, resulting on application dosage above and below the ideal specified or even no application of chemical pesticides (Lardoux et al., 2007). This leads to side effects on crop yield, and heavy application of chemical pesticides for controlling disease has seriously damaged the ecological environment (Lechenet et al., 2017).

The angle formed by the boom with crop (θb) in Fig. 1 greatly influences the spraying quality, which is the main reason for underdose and overdose of phytosanitary application. The minimum disturbance of the boom angle is desired and the main contributor to boom disturbance is the vehicle angle (θv) modifications due to ground obstacles. Simulations show that under and over applications caused by boom rolling, vary between 0 and 10 times the desired dose. Even small improvements of the rolling boom stability, significantly reduce spray peaks. (Ramon and Missontten, 1996)

The use of pendulum systems at Sprayer boom suspension is present at most manufactures and these systems uses springs and dampers to control the position and suspension response. The objective of the suspension system is to maintain the boom as parallel as possible to the ground regardless of the vehicle movement.

The objective of this work is to model the boom suspension and to improve its performance by adding a state-feedback control to boom pendulum suspension, resulting in greater boom stability and improving its efficiency.

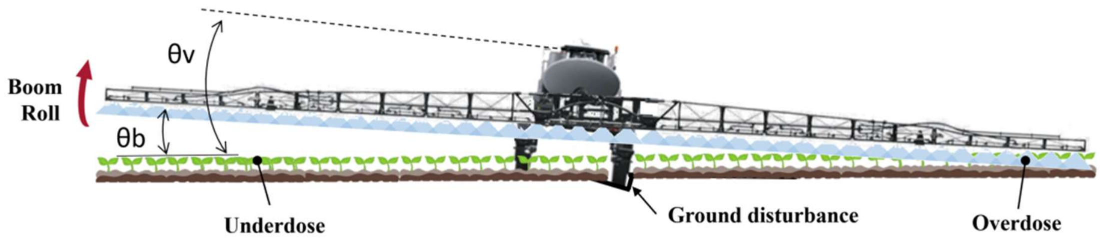

Figure 1. Boom rolling movement influence on application quality (AGCO Corp).

PROBLEM FORMULATION

Track model

The track was modeled using the Standard ISO 5008, 2002 rough track profile at a speed of 6 km/h, the profile is shown in Fig. 2. The track was chosen due to its intense disturbance to both wheels, and it is considered a very severe terrain not often found in real application but very challenging to controllers.

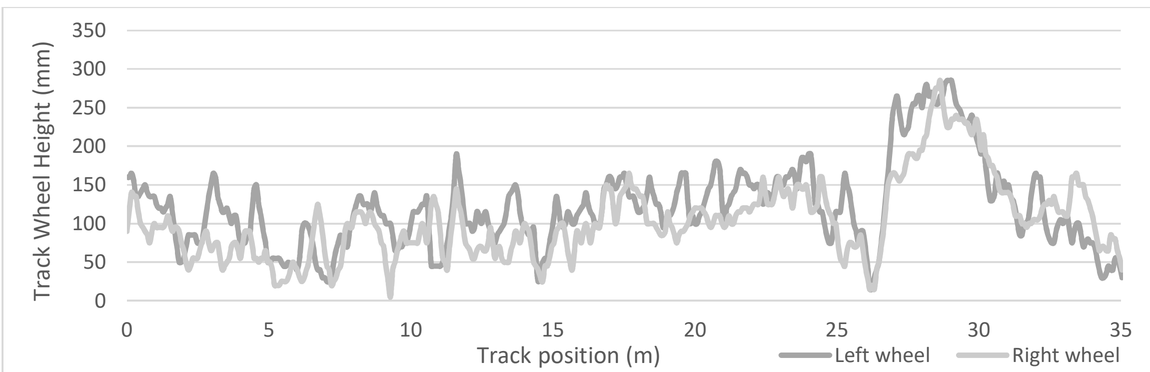

Figure 2. ISO 5008:2002 Rough track profile.

Half vehicle model

Figure 3 illustrates the modeled characteristics of the half-car, M that represents the laden vehicle. The disturbance input W_{LH} and W_{RH} are the road profile of the left-hand and right-hand wheels Z_v represents the position of the sprung of the mass. The dampers are C whereas the springs are K. The left and right suspensions are modeled as spring/damper systems. The vehicle body has roll and height degrees of freedom. They are represented in the model by four states: vertical displacement Z_v, vertical velocity $\dot{Z}_v$, roll angular displacement θ_v, and roll angular velocity $\dot{\theta}_v$. The vehicle properties are shown in table 1, considering the vehicle is full-laden.

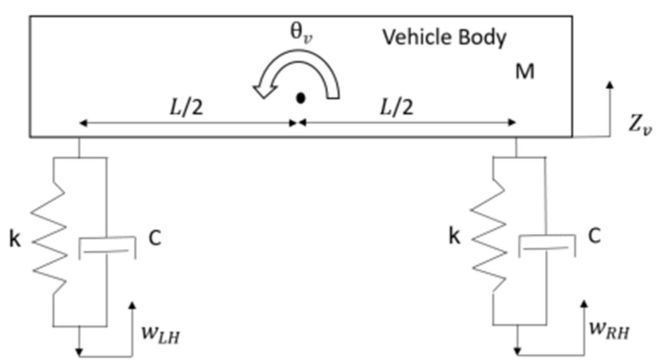

Figure 3. Free-body diagram of the half vehicle model.

Table 1. Vehicle characteristics.

Center of gravity-Vertical height (m)	2.8
L - Track width (m)	3.2
M - Vehicle mass (kg)	10500
I - Mass Moment of Inertia (kg.m²)	84945
K - Suspension spring rate (N/m)	15000
C- Suspension damper rate (N/(m/s))	20000

The mathematical model of the vehicle used a half vehicle model in order to reproduce the vehicle rolling response to feed the boom rolling mathematical model. The open loop model can be verified at Fig. 4 and spring/damper sub-model at Fig. 5.

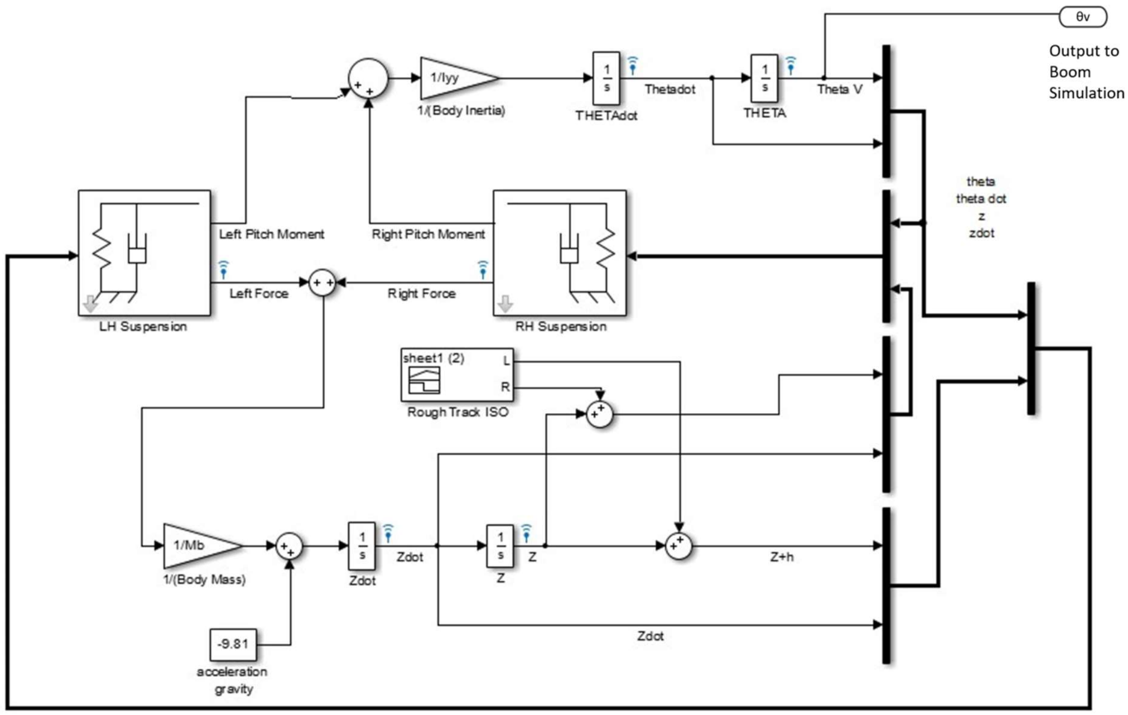

Figure 4. Half vehicle model at Simulink/Matlab®.

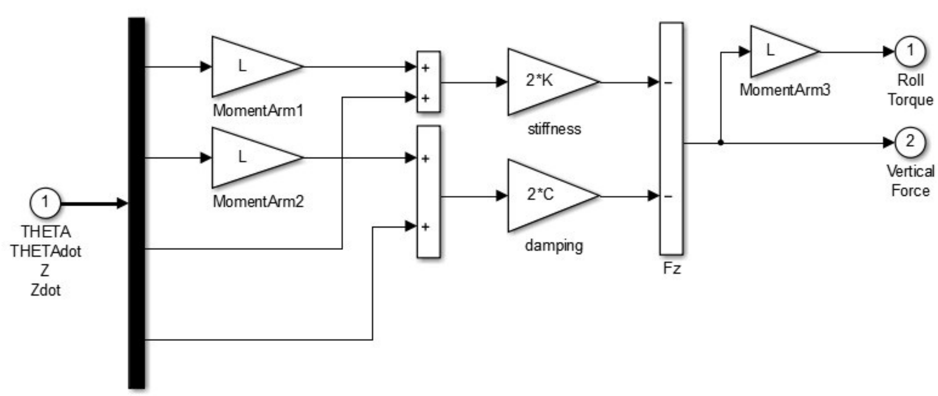

Figure 5. Left hand and Right hand Spring/damper suspension sub-model at Simulink/Matlab®.

Passive boom model

A passive suspension isolates the boom from high frequencies and transmits low frequencies but can only dissipate, temporarily store and later return energy to the system. The boom stabilizing forces are generated in response to local relative motion, velocity and acceleration. (O'Sullivan,1988).

A boom model was created by Fig. 6, considering the vehicle rolling angle (θ_v), and the boom rolling angle (θ_b). Some characteristics distances are fundamentals, for instance the distance from the pendulum rotation (a) and the total length of the boom (L) and its mass (m). The same notations where used at the other models.

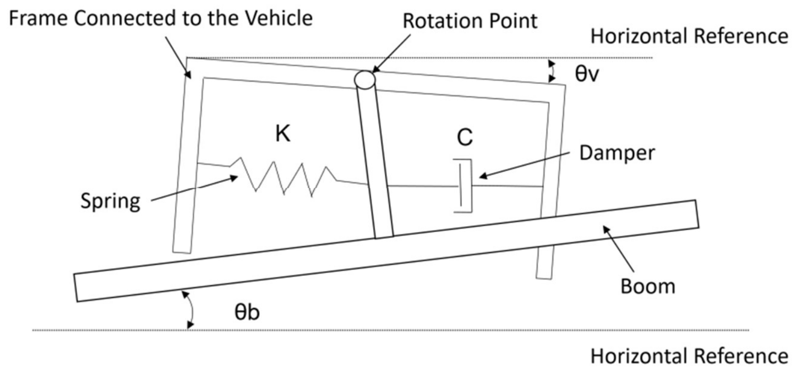

Figure 6. Simplified Model of the Passive system.

The rotational movement of the system is described by Eqs. (1) and (2), and considering the mass of the boom as equally distributed the inertia of the boom is used at Eqs. (3) and (4).

$$J\ddot{\theta} = T \tag{1}$$

$$T = -2k\theta a^2 - mglsin\theta - C\dot{\theta}a \tag{2}$$

$$J = I_{boom} \tag{3}$$

$$J = \frac{1}{12} M * L^2 \tag{4}$$

Since the boom maximum working angle is less than 8°, the approximation of $\sin\theta \cong \theta$ is valid and Eq. 2 can be approximated as linear as resulting in Eq. 5.

$$\frac{1}{12} ML^2\ddot{\theta} = -2ka^2\theta - Ca\dot{\theta} - mgl\theta \tag{5}$$

The model of the passive system showed at Fig. 7 was created using the Simulink/Matlab® software and compare its performance with others controllers.

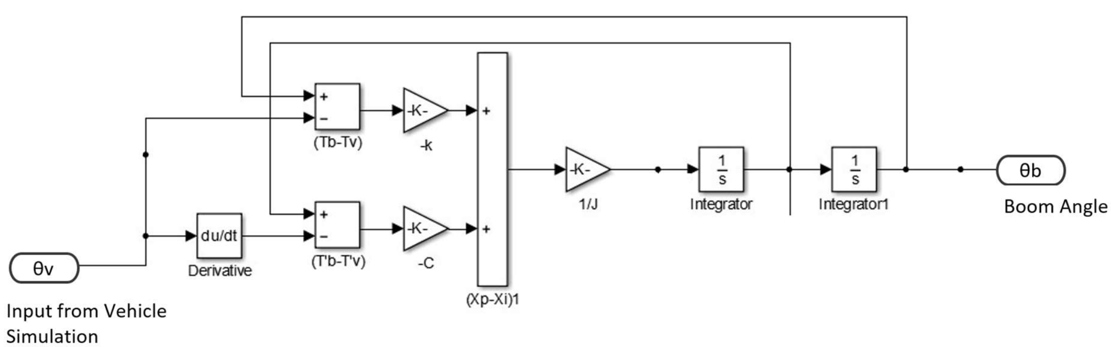

Figure 7. Passive model using Simulink/Matlab®, the input signal is the vehicle-rolling angle and the outputs is boom angle.

State feedback controller design

State Feedback Control, shown at Fig 8, is designed to obtain the desired closed loop performance in terms of characteristics of the transient response and stability. The arbitrary location of eigenvalues in closed loop through state feedback can be achieved if the state equation in open loop is controllable (Yakub, Lee and Mori, 2016). The system in open loop, that is the system plant, is presented by the state equation time invariant and linear (Du, Li and Zhang, 2012). The state-space representation of the controlled system in Fig. 3 can be formalized as following:

$$\dot{x}(t) = Ax(t) + Bu(t) \tag{6}$$

$$\dot{y}(t) = Cx(t) \tag{7}$$

The resulting control law of the state feedback is shown in Eq. 8, where *K* is the constant gain matrix of the state feedback that results in the state equation in a closed loop with the desired performance in order to achieve stability in a closed loop. The controller of state feedback must be designed to control the transient response in a closed loop, such as overshooting and the settling time.

$$u(t) = -Kx(t) + r(t) \tag{8}$$

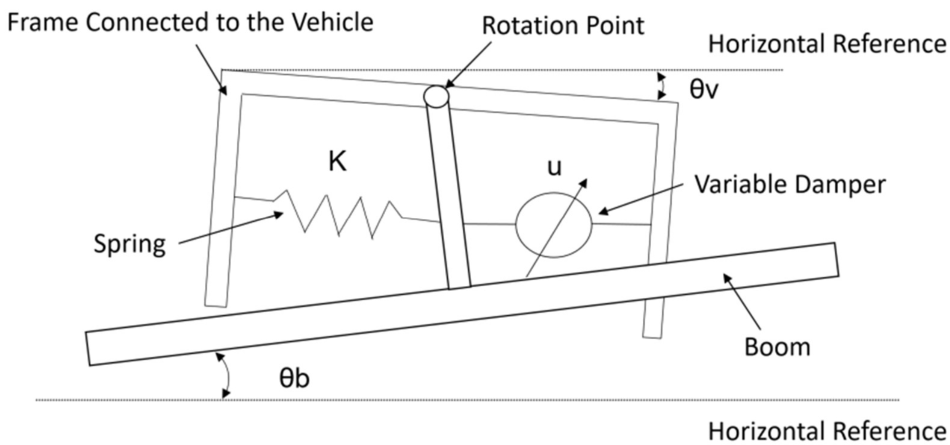

Figure 8. Plant model of the State Feedback Controller

To the controller achieve closed-loop stability, the state-feedback should be designed to control the closed-loop transient response as overshoot and settling time. The desired closed-loop system behavior via eigenvalue selection is known as shaping the dynamic response or pole placement. The poles allocation used the chosen settling time of (1%) at 0.25 s and overshoot of 0.15 resulting the poles are found at Eq. 9.

P1: -18.42 + j 30.50
P2: -18.42 - j 30.50 (9)

In order to improve the system response a second real pole (-28.42) was the chosen, resulting on:

$$P = [-18.42 \quad -28.42] \tag{10}$$

To formulate the system mathematical model the following equations:

$$M\ddot{y} + C\dot{y} + Ky = u \tag{11}$$

$$M\dot{x}_2 + Cx_2 + Kx_1 = u \tag{12}$$

Then

$$\dot{x}_1 = x2 \tag{13}$$

$$\dot{x}_2 = \frac{1}{M}(-Cx_2 - Kx1) + \frac{1}{M}u \tag{14}$$

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \end{bmatrix} = \begin{bmatrix} 0 & 1 \\ \frac{-K}{M} & \frac{-C}{M} \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} + \begin{bmatrix} 0 \\ \frac{1}{M} \end{bmatrix} \tag{15}$$

$$\begin{bmatrix} \dot{x}_1 \\ \dot{x}_2 \end{bmatrix} = \begin{bmatrix} 0 & 1 \\ -3{,}5 & -2{,}5 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} + \begin{bmatrix} 0 \\ 9E-4 \end{bmatrix} u \tag{16}$$

$$y = Cx \tag{17}$$

$$y = \begin{bmatrix} 1 & 0 \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \end{bmatrix} \tag{18}$$

The defined gains are displaced in matrix K:

$$K = \begin{bmatrix} 44{,}44 & -116{,}81 \end{bmatrix} \tag{19}$$

Figure 9 shows the system response in open loop and Fig. 10 shows the system response in a closed loop after the Kr calculation.

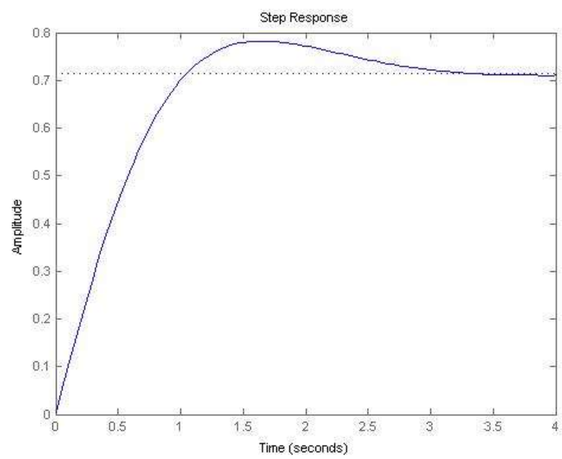

Figure 9. Open loop response

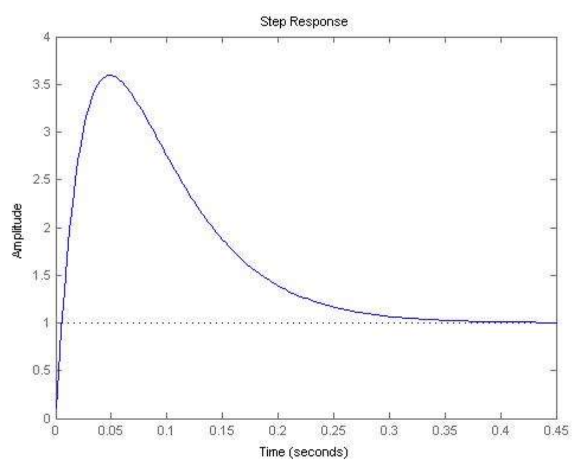

Figure 10. Closed loop response after Kr= 209.32 adjustment.

The closed-loop control system block diagram model is shown in Fig. 11, which was set with the K_1 and K_2 gains and also the K_r gain in the system input.

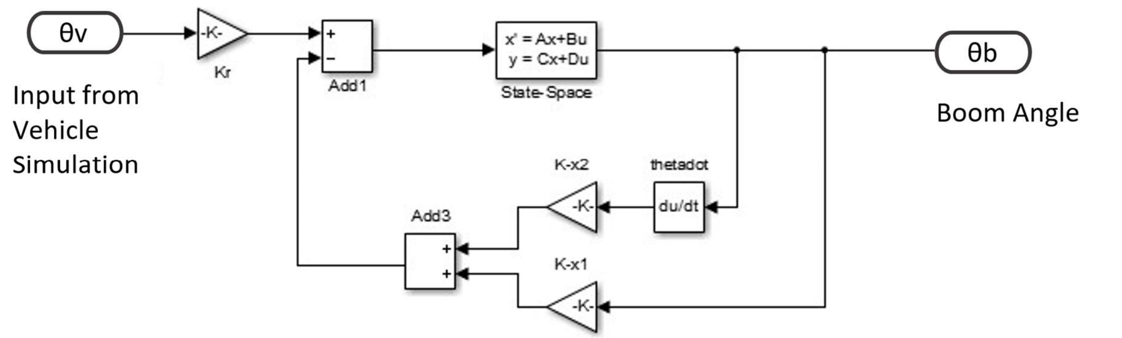

Figure 11. State feedback closed loop block diagram.

RESULTS

The results of the vehicle response to track disturbance shown in Fig. 12 and its angle response show that the track inputs are severe when compared to a normal terrain application. These inputs are necessary to compare the efficiency of the control method.

The results of the simulations of the Passive and state-space or state feedback control θb response compared in Fig. 12.

The performance of the passive suspension shown that it reduces the response of the input from the vehicle, but still following the angle of the vehicle.

The plot shows the stability of the boom in closed-loop compared to the open-loop strategy and its capability to maintain the original angle regardless the vehicle motion.

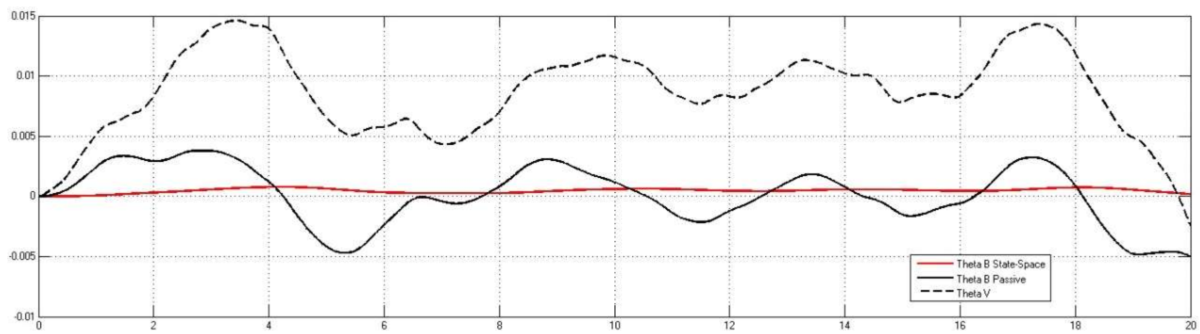

Figure 12. Graph show vehicle response θv (dashed) and comparison between passive system response angle θb (Black) and state feedback control θb (red).

CONCLUSION

This paper addressed modeling aspects of suspension control system and the great performance that can be achieved. The system with state feedback presented a much smaller displacement compared to an open loop suspension, which means better stability of the boom and, with that, a better performance of the equipment. The state feedback system proved to be a more stable and fast-response system, which led the system to have minimal disturbance. In conclusion, the state feedback system is a viable and effective control system for a significant performance improvement.

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