

RESEARCH ARTICLE

The road disturbance attenuation for quarter car active suspension system via a new static two-degree-of-freedom design

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ABSTRACT

The main aim of this paper is to attenuate the effects of the road disturbance on the quarter-car active suspension system (ASS) for the passenger comfort by using design. Therefore, a new static disturbance compensator is proposed by using linear matrix inequality method such that the disturbance compensator and feedback controller are simultaneously designed for the disturbances in the linear time-invariant systems, which are measurable or predictable. They have static structure, and the disturbance compensator is designed on the feedforward path. The design is applied against the road disturbance affecting the quarter car ASS. The effectiveness of the design is demonstrated with the simulations.



1. Introduction

Feedforward compensator designs are used for the disturbance rejection or the reference-tracking problem where the external disturbances are measurable or predictable since the designed feedforward element produces an additional control signal according to the measured disturbance values. The disturbance compensator or controller designs to attenuate disturbances are applied to the many chemical and process systems [1-5]. For instance, in [5], a general structure is presented for single-input-single output (SISO) process system. The disturbance compensator/controller designs are used together with a feedback controller [2,3,6-8]. This is why, the feedback controller provides the stability of the system and the disturbance compensator/controller does not affect the stability. In these designs, there are two approaches. The first one: both of these are simultaneously designed as in [1]. The second one: previously the feedback controller is obtained, and then the disturbance compensator/controller is obtained as in [9,10]. In the literature, feedforward designs are proposed by using different approaches for the linear time invariant systems. In view of literature, there are a few studies based on H_∞ approach. The forefront ones among the studies are as in [3,9-12]. In [3], H_∞ dynamic feedforward design is tackled with obtaining the system

inverse. In [9], a dynamic controller is obtained for linear parameter varying (LPV) systems. In [10], a static feedforward controller is proposed for LPV systems while there is a feedback controller. In [11], a reduced order H_∞ controller is designed against the disturbances for active vibration system. In [12], the feedforward designs are obtained with mixed-sensitivity based on inverse of system.

A vehicle suspension system comprises of the springs, damper and linkages that link its wheels to a vehicle. Its essential role is to reduce the vertical acceleration conveyed to the vehicle body. Because, this affects the passenger comfort. The vehicle suspension is generally designed to satisfy three requirements, which are road handling, passenger comfort and load carrying. The suspension system must provide the road handling, load carrying and the passenger comfort, which is provided by an efficient isolation of passengers from the road disturbances. The parameters of a passive suspension consisting of springs and dampers are mostly constant, which are chosen to achieve a specific performance level by considering the road handling, ride comfort and load carrying. Therefore, especially the performances are unchangeable during driving. As for an ASS, it can affect the performances of the road handling and ride comfort by introducing energy by adding an actuator to the system. In view of many road

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inputs or the unevenness of road, the performance of passive suspension is not adequate in contrast to ASS. Thus, the control of ASS is a challenging research topic for the automotive applications in the literature such as [13-15]. In [13], a robust control is made for a quarter-car ASS against the variations on the vehicle body by designing a sliding mode controller with the algebraic estimator for the vehicle body mass. In [14], a fuzzy logic controller is designed to get the desired ride performance under different road profiles corresponding with the quarter-car ASS model. In [15], the control of an electromagnetic ASS with high bandwidth is tackled for a quarter car model. In [16], an adaptive control for vibration rejection is presented in the case of unknown narrow band disturbances in ASS. In [17], the finite-time tracking control with a disturbance compensator is tackled against the external disturbance for ASS. In [18], H_∞ gain-scheduled controller is proposed via convex optimization by using only frequency-domain data. In [19], an output-feedback H_∞ control is proposed for half-vehicle ASS under time-varying input delay. In [20], H_∞ and H_2 optimal control are designed to minimize vehicle vibrations and to improve the comfort of passenger exposed to road disturbances for an ASS model. In [21], a static output-feedback controller is designed for a half car ASS with limited information structure to simultaneously improve the ride comfort and stability. In [22], the H_∞ control is designed via dynamic-output feedback approach for active seat suspension systems. In [23], a mathematical model and H_∞ control are proposed to improve the ride comfort with road handling for an ASS which is subjected to different road profiles. In [24], a feedback controller with feedforward controller is proposed to attenuate vibration for discrete-time ASS.

In this paper, a new simultaneous design of static optimal disturbance compensator and static feedback controller is proposed to minimize the road disturbances on the quarter-car ASS. The proposed design is based on H_∞ control technique via linear matrix inequality. The road handling and passenger comfort are improved by adding extra signal to feedback control input thanks to the disturbance compensator.

2. Problem formulation and disturbance compensator design

In this section, the control problem is formulated. Eq. (1) defines linear time-invariant generalized system G which is generally used H_∞ controller design, where $u(t) \in \mathcal{R}^{n_u}$ is the controller signal, $x(t) \in \mathcal{R}^{n_x}$ are state variables, $z(t) \in \mathcal{R}^{n_z}$ are controlled outputs, $y(t) \in \mathcal{R}^{n_y}$ are the measured outputs, $\omega(t) \in \mathcal{R}^{n_\omega}$ are the disturbance signals.

$$\begin{aligned}\dot{x}(t) &= Ax(t) + B_1\omega(t) + B_2u(t) \\ z(t) &= Cx(t) + D_1\omega(t) + D_2u(t) \\ y(t) &= x(t)\end{aligned}\quad (1)$$

The closed-loop state space model from $\omega(t)$ to $z(t)$ becomes as in Eq. (2) when the controller is realized with $u(t)$ for Eq. (1).

$$\begin{aligned}\dot{x}(t) &= A_{cl}x(t) + B_{cl}\omega(t) \\ z(t) &= C_{cl}x(t) + D_{cl}\omega(t)\end{aligned}\quad (2)$$

Figure 1 shows the block diagram of the ASS coupled with the proposed design, where K_{ff} shows disturbance compensator matrix, K_{fb} shows feedback controller, u_{ff} shows the produced control signal of feedforward path, u_{fb} shows the produced control signal of feedback path, u shows the total applied control signals to the system and ω shows the road disturbance signal to the system. In Section 3, the suspension system is modelled as in Eq. (1).

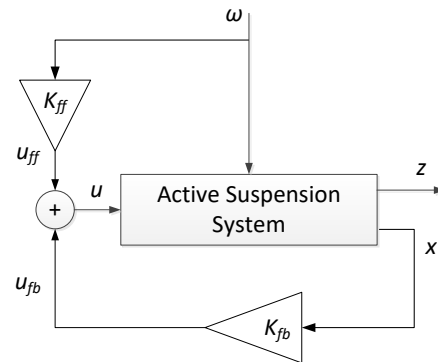


Figure 1. The disturbance compensator and feedback control system

As in Figure 1, the control system contains active suspension model, disturbance compensator and feedback components. The proposed disturbance compensator generates a feedforward signal in addition to feedback signal. Thus, ASS provides better performance during online operation against the road disturbance predictable or measurable.

The proposed feedforward matrix K_{ff} and feedback matrix K_{fb} , which have static structure, are simultaneously designed by using H_∞ control technique based on linear matrix inequality. Accordingly, Lemma 1 is known as bounded real lemma in the literature, and it is commonly used for H_∞ control design.

Lemma 1. Let $G(s) = C_{cl}(sI - A_{cl})^{-1}B_{cl} + D_{cl}$ be transfer matrix of the closed loop system Eq. (2). If and only if $\gamma^2 I - D_{cl}^T D_{cl} > 0$, $\|G(s)\|_\infty < \gamma$ and the following states are equivalent.

a. If there exists a positive symmetric matrix

$P \in \mathcal{R}^{n \times n}$, the following inequality holds.

$$\begin{aligned}A^T P + PA &+ (B^T P + D^T C)^T (\gamma^2 I - D^T D)^{-1} (B^T P + D^T C) \\ &+ C^T C < 0\end{aligned}$$

b. If there exists a positive symmetric matrix

$P \in \Re^{n \times n}$, the linear matrix inequality (3) holds.

$$\begin{bmatrix} A_{cl}^T P + P A_{cl} & P B_{cl} & C_{cl}^T \\ B_{cl}^T P & -\gamma I & D_{cl}^T \\ C_{cl} & D_{cl} & -\gamma I \end{bmatrix} < 0 \quad (3)$$

Proof. See [25].

In the literature, the matrix inequality in the Theorem 1 based on Lemma 1 is commonly used for the static state feedback H_∞ controller.

Theorem 1. If there exist a symmetric positive Q in (4) and a matrix R , there exists a static feedback controller $u(t) = Kx(t)$, which stabilizes the system in Eq. (1). Thus, the optimal H_∞ controller is obtained from $K = RQ^{-1}$.

$$\begin{bmatrix} AQ + QA^T + B_2 R + R^T B_2^T & B_1 & QC^T + R^T D_2^T \\ B_1^T & -\gamma I & D_1^T \\ CQ + D_2 R & D_1 & -\gamma I \end{bmatrix} < 0 \quad (4)$$

Proof. See [25].

Remark 1. As in the theorem, the present form is already appropriate in order to optimize the effects of the disturbances $\omega(t)$ to the outputs $z(t)$ for the system in (1). However, it is known that the feedback controller is here designed by considering potential disturbances. However, in the design of this paper, the proposed disturbance compensator works at minimizing their effects on the outputs according to the online measured or estimated disturbances by including an additional control signal on the feedback controller. In addition, disturbance compensator together is simultaneously designed with feedback controller for the disturbance rejection. Theorem 2 presents the proposed disturbance compensator and feedback controller design of the problem in Figure 1.

Theorem 2: If there exist a symmetric positive Y in (5) and a matrix F , there exists a static feedback controller $u_{fb}(t) = K_{fb}x(t)$, which stabilizes the system in Eq. (1), and a static disturbance compensator $u_{ff}(t) = K_{ff}\omega(t)$, which attenuates the disturbances. In this case, the optimal H_∞ controller and disturbance compensator are computed by (6).

$$\begin{bmatrix} AY + B_2 F + (*)^T & B_1 + B_2 K_F & YC^T + F^T D_2^T \\ B_1^T + K_F^T B_2^T & -\gamma I & D_1^T + K_F^T D_2^T \\ CY + D_2 F & D_1 + D_2 K_F & -\gamma I \end{bmatrix} < 0 \quad (5)$$

$$\begin{aligned} K_{fb} &= FY^{-1} \\ K_{ff} &= K_F \end{aligned} \quad (6)$$

Proof. Considering the control system in Figure 1, the

controller signal is as in Eq. (7).

$$u(t) = \begin{bmatrix} K_{ff} & K_{fb} \end{bmatrix} \begin{bmatrix} \omega(t) \\ x(t) \end{bmatrix} \quad (7)$$

If Eq. (7) is substituted into (1), the closed loop system from $\omega(t)$ to $z(t)$ in Eq. (2) becomes as in Eq. (8).

$$\begin{bmatrix} \dot{x}(t) \\ z(t) \end{bmatrix} = \begin{bmatrix} A + B_2 K_{fb} & B_1 + B_2 K_{ff} \\ C + D_2 K_{fb} & D_1 + D_2 K_{ff} \end{bmatrix} \begin{bmatrix} x(t) \\ \omega(t) \end{bmatrix} \quad (8)$$

The state space matrices are as in (9) according to Eq. (8).

$$\begin{aligned} A_{cl} &= A + B_2 K_{fb}, \quad B_{cl} = B_1 + B_2 K_{ff} \\ C_{cl} &= C + D_2 K_{fb}, \quad D_{cl} = D_1 + D_2 K_{ff} \end{aligned} \quad (9)$$

When the obtained closed loop matrices are substituted into (3) in Lemma 1, the matrix inequality (10) is obtained.

$$\begin{bmatrix} PA + PB_2 K_{fb} + (*)^T & * & * \\ B_1^T P + K_{ff}^T B_2^T P & -\gamma I & * \\ C + D_2 K_{fb} & D_1 + D_2 K_{ff} & -\gamma I \end{bmatrix} < 0 \quad (10)$$

Nevertheless, the inequality is not linear due to the products of unknown matrices. In order to linearize, it is pre- and post-multiplied with the matrix in (11) and its transpose respectively, and so the linear matrix inequality (5) is acquired.

$$\begin{bmatrix} P^{-1} & 0 & 0 \\ 0 & I & 0 \\ 0 & 0 & I \end{bmatrix} \quad (11)$$

The definitions of variables are as in Eq. (12) with regard to the linear matrix inequality (5).

$$Y = P^{-1}, \quad F = K_{fb} Y^{-1}, \quad K_F = K_{ff} \quad (12)$$

3. Quarter car active suspension system model

Figure 2 shows a quarter car ASS model which is one fourth of full vehicle model (one wheel system), where m_1 is one fourth of vehicle body mass (sprung mass), m_2 is suspension mass (unsprung mass - tire and axles), k_1 is suspension spring constant, k_2 is spring constant of wheel and tire, b_1 is suspension damping constant, b_2 is damping constant of wheel and tire, u is generating external force. The tire is modelled as a linear spring having a stiffness constant k_2 . The suspension system consists of a passive spring k_1 and a damper b_1 in parallel with an active control element actuator generating a force u . These passive elements assure a minimum standard of safety and performance, whereas the active one is designed to improve the safety and performance. Hence, the model is a quarter car ASS model where an actuator generating the control force u is included to the passive one in order to improve the

safety and comfort performance for different road disturbances.

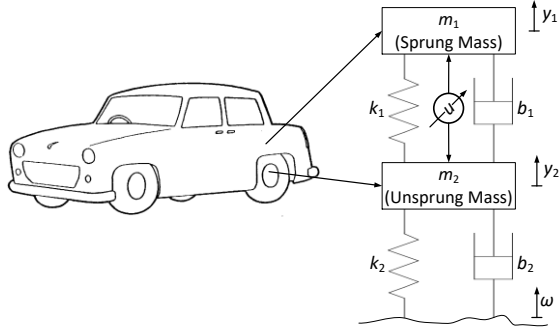


Figure 2. The quarter car ASS

The differential equations are obtained as in (13) by using Newton's law according to Figure 2.

$$\begin{aligned} m_1 \ddot{y}_1 &= u - k_1 (y_1 - y_2) - b_1 (\dot{y}_1 - \dot{y}_2) \\ m_2 \ddot{y}_2 &= -u + k_1 (y_1 - y_2) + b_1 (\dot{y}_1 - \dot{y}_2) \\ &\quad + k_2 (\omega - y_2) + b_2 (\dot{\omega} - \dot{y}_2) \end{aligned} \quad (13)$$

The state space model is needed for the controller design. Accordingly, if the states x are defined as $x_1 = y_1$, $x_2 = \dot{y}_1$, $x_3 = y_1 - y_2$, $x_4 = \dot{y}_1 - \dot{y}_2$ and we define integral action state $x_5 = \int y_1 - y_2$, the state space model is obtained as in (14) where the state space matrices are as in (15) and (16).

$$\begin{aligned} \dot{x}(t) &= Ax(t) + B_1 \omega(t) + B_2 u(t) \\ z(t) &= Cx(t) + D_1 \omega(t) + D_2 u(t) \end{aligned} \quad (14)$$

$$A = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 \\ \frac{-(b_1 b_2)}{(m_1 m_2)} & 0 & \left(\frac{b_1}{m_1}\right) \left[\left(\frac{b_1}{m_1}\right) + \left(\frac{b_1}{m_2}\right) + \left(\frac{b_2}{m_2}\right) \right] - \left(\frac{k_1}{m_1}\right) & -\left(\frac{b_1}{m_1}\right) & 0 \\ \frac{b_2}{m_2} & 0 & -\left(\frac{b_1}{m_1} + \frac{b_1}{m_2} + \frac{b_2}{m_2}\right) & 1 & 0 \\ \frac{k_2}{m_2} & 0 & -\left(\frac{k_1}{m_1} + \frac{k_1}{m_2} + \frac{k_2}{m_2}\right) & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 \end{bmatrix} \quad (15)$$

$$B_1 = \begin{bmatrix} 0 \\ \frac{b_1 b_2}{m_1 m_2} \\ -\frac{b_2}{m_2} \\ -\frac{k_2}{m_2} \\ 0 \end{bmatrix}, B_2 = \begin{bmatrix} 0 \\ \frac{1}{m_1} \\ 0 \\ \frac{1}{m_1} + \frac{1}{m_2} \\ 0 \end{bmatrix}, \quad (16)$$

$$C = \begin{bmatrix} \frac{-(b_1 b_2)}{(m_1 m_2)} & 0 & \left(\frac{b_1}{m_1}\right) \left[\left(\frac{b_1}{m_1}\right) + \left(\frac{b_1}{m_2}\right) + \left(\frac{b_2}{m_2}\right) \right] - \left(\frac{k_1}{m_1}\right) & -\left(\frac{b_1}{m_1}\right) & 0 \\ 1 & 0 & -1 & 0 & 0 \end{bmatrix}$$

$$D_1 = \begin{bmatrix} \frac{b_1 b_2}{m_1 m_2} \\ -1 \end{bmatrix}, D_2 = \begin{bmatrix} \frac{1}{m_1} \\ 0 \end{bmatrix}$$

4. Simulation results

The simulation results are acquired via Matlab with Yalmip parser [26] and Sedumi solver [27]. Figure 3

shows the simulation block diagram for the system. In addition, the ASS parameters are as in Table 1 for the simulation.

Table 1. ASS parameters.

parameter	value	unit
m_1	320	kg
m_2	45	kg
k_1	27000	N/m
k_2	211180	N/m
b_1	935	N.s/m
b_2	20	N.s/m

In this case, the obtained state space matrices are as in (17).

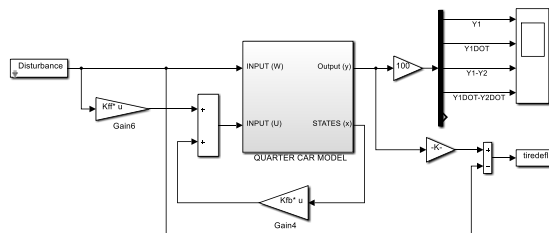
$$A = \begin{bmatrix} 0 & 1 & 0 & 0 & 0 \\ -1.2986 & 0 & -13.829 & -2.9219 & 0 \\ 0.44444 & 0 & -24.144 & 1 & 0 \\ 4692.9 & 0 & -5377.3 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 \end{bmatrix},$$

$$B_1 = \begin{bmatrix} 0 \\ 1.2986 \\ -0.44444 \\ -4692.9 \\ 0 \end{bmatrix}, \quad B_2 = \begin{bmatrix} 0 \\ 0.003125 \\ 0 \\ 0.025347 \\ 0 \end{bmatrix}, \quad (17)$$

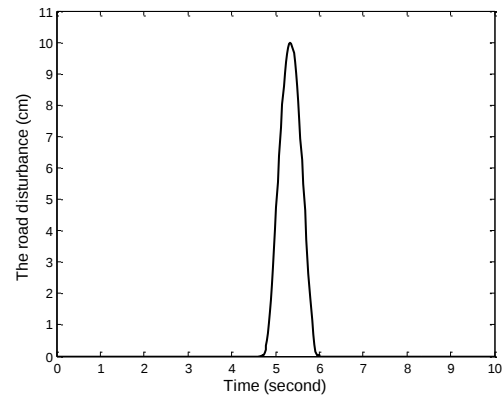
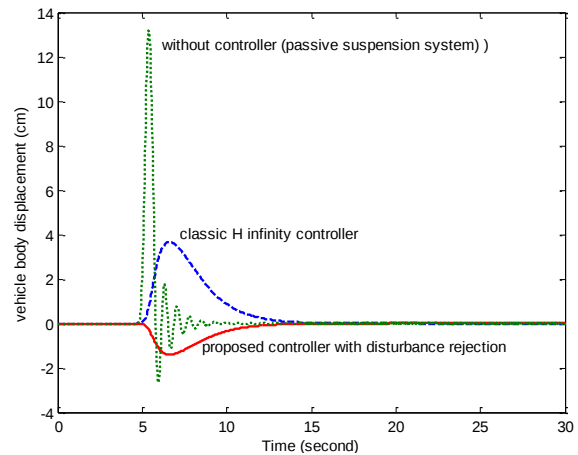
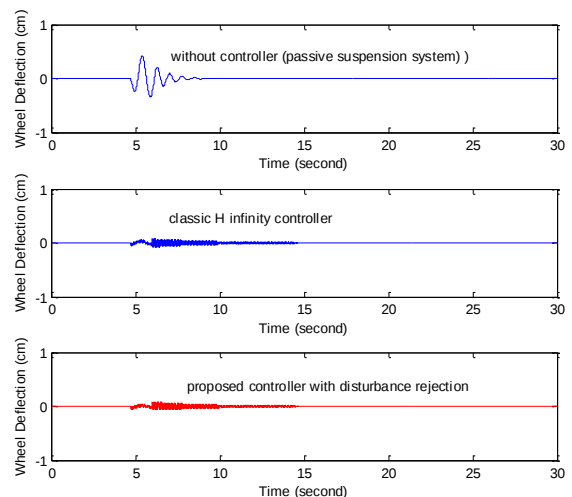
$$C = \begin{bmatrix} 1.2986 & 0 & -13.829 & -2.9219 & 0 \\ 1 & 0 & -1 & 0 & 0 \end{bmatrix},$$

$$D_1 = \begin{bmatrix} 1.2986 \\ -1 \end{bmatrix}, \quad D_2 = \begin{bmatrix} 0.003125 \\ 0 \end{bmatrix}$$

In addition, the obtained classic H_∞ controller matrix is $K=[207.34 \quad -514.74 \quad 4448.6 \quad 933.5 \quad -0.0081947]$ whereas the proposed controller matrix with compensator is $K=[212.26 \quad -508.11 \quad 4450 \quad 933.9 \quad -0.0078739 \quad -574.68]$.

**Figure 3.** The simulation block diagram for the control of quarter-car ASS

The road disturbance is given in Figure 4. According to the simulation, the results are as in Figure 5-8. Figure 5 shows the vehicle body displacements, Figure 6 shows the wheel deflections, Figure 7 shows the car body accelerations, Figure 8 shows the applied control forces.

**Figure 4.** The road disturbance**Figure 5.** The vehicle body displacement**Figure 6.** The wheel deflection

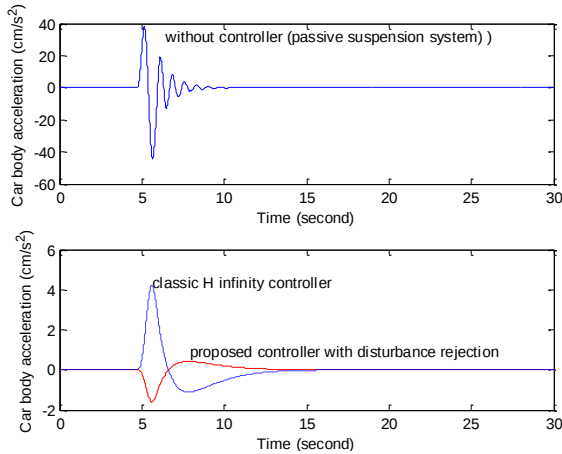


Figure 7. The car body acceleration

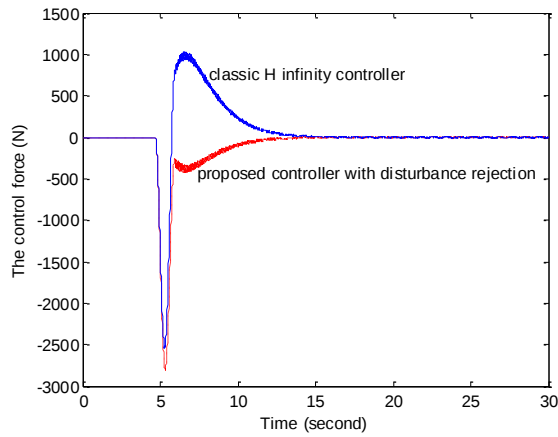


Figure 8. The control force

As shown in Figure 5, the peak vehicle displacement is nearly 3.8 cm when the classic H_∞ controller is applied to the system. However, the value is nearly -1.5 cm for the proposed design and so the proposed design has a better response. In addition, the vehicle displacement negatively changes since the control force generating by adding the feedforward one to the feedback control force is negative as shown in Figure 8. The wheel deflections, which directly affect the road safety, are almost same for both control designs as shown in Figure 6. In view of the vehicle body accelerations, which directly affect the passenger comfort, the proposed design is quite successful as shown in Figure 7. The peak value for classic one is nearly 4.1 whereas the value for the proposed one is nearly -1.5, and so the system becomes more comfortable by the proposed design. The applied total control force of the proposed disturbance compensator controller is as shown in Figure 8 in order to acquire the successes. Hence, the proposed approach has better results than the classic one for the ASS control system in point of the road comfort.

5. Conclusion

This paper shows that a new static disturbance rejection design for the ASS of quarter vehicle model. The disturbance compensator and state feedback controller are simultaneously designed for the disturbance

rejection with respect to passenger comfort. The proposed design is based on optimal H_∞ control method via linear matrix inequality. The simulations have shown that the proposed design has better performance according to classic state feedback H_∞ control against the road disturbance with regard to the passenger comfort. In addition, the design can be used for other dynamic systems, which are multi-input-multi output.

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