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Disturbance observer based Takagi-Sugeno fuzzy control for an active seat suspension

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Abstract

In this paper, a disturbance observer based Takagi-Sugeno (TS) fuzzy controller is proposed for an active seat suspension; both simulations and experiments have been performed verifying the performance enhancement and stability of the proposed controller. The controller incorporates closed-loop feedback control using the measured acceleration of the seat and deflection of the suspension; these two variables can be easily measured in practical applications, thus allowing the proposed controller to be robust and adaptable. A disturbance observer that can estimate the disturbance caused by friction, model simplification, and controller output error has also been used to compensate a H¿ state feedback controller. The TS fuzzy control method is applied to enhance the controller's performance by considering the variation of driver's weight during operation. The vibration of a heavy duty vehicle seat is largest in the frequency range between 2 Hz and 4 Hz, in the vertical direction; therefore, it is reasonable to focus on controlling low frequency vibration amplitudes and maintain the seat suspensions passivity at high frequency. Moreover, both the simulation and experimental results show that the active seat suspension with the proposed controller can effectively isolate unwanted vibration amplitudes below 4.5 Hz, when compared with a well-tuned passive seat suspension. The active controller has been further validated under bump and random road tests with both a 55 kg and a 70 kg loads. The bump road test demonstrated the controller has good transient response capabilities. The random road test result has been presented both in the time domain and the frequency domain. When with the above two loads, the controlled seat suspensions root-mean-square (RMS) accelerations were reduced by 45.5% and 49.5%, respectively, compared with a well-tuned passive seat suspension. The proposed active seat suspension controller has great potential and is very practical for application as it can significantly improve heavy duty driver's ride comfort.

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Disturbance observer based Takagi-Sugeno fuzzy control for an active seat suspension

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Abstract

In this paper, a disturbance observer based Takagi-Sugeno (TS) fuzzy controller is proposed for an active seat suspension, and this controller is validated by both simulations and experiments. The proposed controller applies the seat acceleration and seat suspension deflection which can be easily measured in practical application as feedbacks. A disturbance observer that can estimate disturbance caused by friction force, model simplification, and control output error is used to compensate a H_∞ state feedback controller. The TS fuzzy control method is applied to enhance the controller's performance by considering the variation of driver's weight. Because the vertical vibration of heavy duty vehicle seat is highest in the frequency range 2 Hz to 4 Hz, it is reasonable to focus on controlling low frequency vibration and maintain seat suspension's passivity at high frequency (isolating vibration) to release the requirement to the actuator. The simulation and experimental results show that the active seat suspension with the proposed controller can effectively isolate vibration under 4.5 Hz when compared with a well-tuned passive seat suspension. The active controller is further validated under bump and random road tests with 55 Kg and 70 Kg loads, respectively. The bump road test shows the controller has good transient response capability. The random road test result is presented both in time domain and frequency domain. The controlled seat suspension root-mean-square (RMS) acceleration is reduced by 45.5% and 49.5%, respectively, when compared with a well-tuned passive seat suspension. The proposed active seat suspension controller is very practical for application and can greatly improve heavy duty driver's ride comfort.

Keywords: active seat suspension; TS fuzzy control; disturbance observer; acceleration feedback.

1. Introduction

Heavy duty vehicle drivers often suffer from severe vibration caused by rough road or machine tools [1]. Whole body vibration has become a leading factor to deteriorate drivers' ride comfort, fatigue and health. Seat suspension, which is the most direct way to isolate the vibration felt by drivers, is widely applied in heavy duty vehicles to improve drivers' working conditions. For most vehicle seats, the vibration energy is isolated by using passive seat suspension and seat cushions which have limited performance around the resonance frequency of seat suspension and are difficult to optimize [2]. Recently, semi-active seat suspensions have been widely investigated with Electrorheological (ER) and Magnetorheological (MR) materials to improve ride comfort [3, 4]. On the other hand, it is widely believed that active seat suspension is the most efficient way to isolate vibration [5-7]. There are three issues which should be considered when developing an active seat suspension system. First, the active actuator will cost more when it is compared with the semi-active actuator. Apparently, a higher response and higher power output active actuator will have a higher price to achieve better vibration isolation performance. Second, a complicated control algorithm will require a high calculation speed which will increase the hardware cost. Last, the measurable suspension variables are very limited in practical application. Generally, accelerations can be conveniently obtained with cheap sensors, and the suspension relative deflection can also be measured.

There are many reputable control strategies for suspensions, e.g., H_∞ control [8-12], linear quadratic gauss (LQG) [13, 14], adaptive control [15] and fuzzy control [16, 17]. A two-step methodology is proposed to design the static output-feedback controllers for vehicle suspensions in [9]. The finite frequency H_∞ controller for active suspensions has been proposed in [10, 11]. Li et al. [12] present a work about output-feedback-based control strategy for vehicle suspension control with control delay. An adaptive sliding mode control algorithm for nonlinear suspension is proposed in [15]. Du et al. [18]

propose an observer based H_∞ controller with a TS fuzzy model to solve the non-linear problem of a semi-active seat. Maciejewski [19] proposes an active vibration control strategy based on a primary controller and actuator's reverse dynamics. Bououden et al. [20] propose a robust predictive control design for nonlinear active suspension systems via TS fuzzy approach. However, most of the previous researches for vehicle or seat suspension control system do not include the acceleration feedback into controller. Although high frequency noises will be introduced when acceleration signal is measured, with appropriate algorithms, a controller could be designed to be more effective to suppress vibration with acceleration measurement. It is also noted that TS fuzzy approach has been applied in the suspension control; however, very few works consider the friction issue, which is unavoidable in all the practical engineering applications, especially, in the seat suspension, because the friction can greatly affect the seat suspension dynamics.

Disturbance observer has been extensively studied and proven to be able to improve the controller's performance effectively. Deshpande et al. [21] propose a novel nonlinear disturbance compensator for active suspension system. The disturbance observer for sliding model control has been studied in many papers [22, 23]. Kim et al. present a disturbance observer for estimating higher order disturbances [24]. Pan et al. propose tracking control method for nonlinear suspension system with disturbance compensation [25]. Saturation is another important issue in practical applications that needs to be paid attention [26]. In the literature, several methods have been proposed to handle the effects of saturation [27]. Among them, the anti-windup approach is proven to be an effective way to deal with actuator saturation [28], and it is extensively studied in [29, 30].

Some laboratory and field researches supportsthe relationship between low frequency vibration and drivers' fatigue and drowsiness. In this paper, a disturbance observer based TS fuzzy control strategy is proposed for an active seat suspension to improve drivers' comfort. As the vertical vibration of heavy duty vehicle seat is highest in the frequency range 2 Hz to 4 Hz [31], in this paper, the vibration control target is mainly defined in the low frequency range to greatly improve the heavy duty vehicle drivers' ride comfort , and at the same time, to reduce the cost because low rate active actuator can be applied. For the low cost active seat suspension prototype applied in this paper, a control algorithm based on acceleration and suspension deflection, which are measurable in practical application, is developed. In an active seat suspension application, there are several kinds of disturbances, e.g., the friction force, model simplification caused disturbance, and actuator saturation. To deal with these disturbances, in this paper, the disturbance compensation approach is applied with a disturbance observer that estimates all the possible disturbances including the effect caused by actuator saturation. The proposed acceleration measurement based disturbance observer is sensitive to system disturbances, thus it can enhance the controller's response performance. In addition, the TS fuzzy approach is applied to guarantee that the seat suspension has similar performance with different driver weights.

The remainder of the paper is organized as follows: Section 2 presents the proposed disturbance observer based active control; Section 3 discusses the TS fuzzy controller; the simulation and experimental results are presented in Section 4; Finally, Section 5 presents the conclusions of this research.

Notation: I is used to denote the identity matrix of appropriate dimensions. * is used to represent a term that is induced by symmetry.

2. Disturbance observer based active control

2.1 Seat suspension model with disturbance

A seat suspension model is shown in Fig. 1 where m_b is the human body mass; m_s is the seat frame mass; k_c and c_c are the stiffness and damping between human body and seat frame; f_r is the seat suspension friction force; $\bar{u}$ is the saturated active force generated by actuator; z_b and z_s are the displacements of the corresponding masses, respectively, and z_v is the displacement of cabin floor. The dynamic model of the system can be described as:

$$m_s\ddot{z}_s = -k_f(z_s - z_v) + k_c(z_b - z_s) + c_c(\dot{z}_b - \dot{z}_s) - f_r + \bar{u} \tag{1}$$

$$m_b\ddot{z}_b = -k_c(z_c - z_s) - c_c(\dot{z}_c - \dot{z}_s) \tag{2}$$

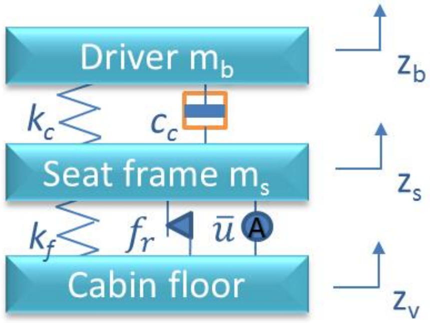

Fig. 1. Seat suspension model

Control algorithms based on the model states measurement or estimation have been extensively explored [32, 33]. However, in practical, the measurable variables are very limited, e.g. accelerations of masses and the relative displacement between seat frame and cabin floor. The complicated multi-states estimation algorithms which always include matrix multiplications and integral of variables require a high calculation speed which will increase the hardware cost. In addition, the active actuator output saturation exists in almost all the real applications [26, 27, 34, 35]. Considering the possible actuator output saturation and other dynamic disturbances, such as the friction in the model (1) and (2), a simple but accurate single degree of freedom (SDOF) seat model is defined as:

$$M\ddot{z}_s = -k_f(z_s - z_v) - F_d + u \tag{3}$$

$$M = m_b + m_s \tag{4}$$

$$F_d = f_r + f_{\Delta m} + \Delta u \tag{5}$$

$$\bar{u} = \text{sat}(u) = \begin{cases} -u_{lim}, & \text{if } u < -u_{lim} \\ u, & \text{if } -u_{lim} < u < u_{lim} \\ u_{lim}, & \text{if } u > u_{lim} \end{cases} \tag{6}$$

where M is the total mass of driver body and seat frame, F_d is the unmeasurable disturbance that includes friction force f_r, model simplification caused disturbance $f_{\Delta m} = m_b(\ddot{z}_b - \ddot{z}_s)$ and control output error $\Delta u = u - \bar{u}$ (caused by actuator output saturation), u is the desired control input, $\bar{u} = \text{sat}(u)$ is the saturated control input defined in Eq. (6), u_{lim} is the control input limit. Because a soft spring is always applied for seat suspension to make the seat comfortable (k_c is always much higher than k_f), in this model, m_b and m_s are taken as a solid mass to simplify the model; the caused

$f_{\Delta m}$ is small in low frequency vibration and will correspondingly increase with vibration frequency. Considering that most of the high frequency vibration has been isolated by vehicle suspension and the low frequency vibration magnitude is hard to be decreased for the stiffness limitation of a passive vehicle suspension, model (3) is practical to be applied for an active seat suspension control.

In this paper, the seat acceleration and seat suspension deflection is measured, and suspension deflection rate is calculated out from seat suspension deflection. Although noise will be introduced when directly differentiating the suspension deflection to obtain its deflection rate, the noise can be easy to deal with by applying a point by point low pass filter. The state variables are chosen as $\mathbf{X} = [z_s - z_v \quad \dot{z}_s - \dot{z}_v]^{\mathrm{T}}$, the vibration disturbance is $d = \ddot{z}_v$, and the measurement variables are $\mathbf{Y_1} = [z_s - z_v \quad \dot{z}_s - \dot{z}_v]^{\mathrm{T}}$ and $Y_2 = \ddot{z}_s$. Thus, combing with Eq. (3) the system model is defined as:

$$\dot{\mathbf{X}} = \mathbf{AX} + \mathbf{B_1}(u - F_d) + \mathbf{B_2}d \tag{7}$$

$$\mathbf{Y_1} = \mathbf{C_1X} \tag{8}$$

$$Y_2 = \mathbf{C_2X} + D_2(u - F_d) \tag{9}$$

where $\mathbf{A} = \begin{bmatrix} 0 & 1 \\ -\frac{k_f}{M} & 0 \end{bmatrix}$, $\mathbf{B_1} = \begin{bmatrix} 0 \\ \frac{1}{M} \end{bmatrix}$, $\mathbf{B_2} = \begin{bmatrix} 0 \\ -1 \end{bmatrix}$, $\mathbf{C_1} = \begin{bmatrix} 1 & 0 \\ 0 & 1 \end{bmatrix}$, $\mathbf{C_2} = \begin{bmatrix} -\frac{k_f}{M} & 0 \end{bmatrix}$, $D_2 = \frac{1}{M}$.

Driver's comfort can be quantified by the seat acceleration, which is the main optimization objective in the controller design process. Therefore, the controlled output is defined as:

$$Z_1 = \mathbf{C_3X} + D_3(u - F_d) \tag{10}$$

where $\mathbf{C_3} = \alpha\mathbf{C_2}$, $D_3 = \alpha D_2$, α is a constant.

2.2 TS fuzzy model

The weight of driver body will change when different drivers are driving. To make the seat suspension have similar performance, the driver's mass variation has been considered into the controller design. The mass variation range is assumed as $[m_{bmin} \quad m_{bmax}]$, where m_{bmin} and

m_{bmax} are the possible minimum and maximum driver's masses, respectively. Therefore, the simplified model mass can be expressed as:

$$\frac{1}{M} = h_1 \frac{1}{m_{bmin} + m_s} + h_2 \frac{1}{m_{bmax} + m_s} = h_1 \frac{1}{M_{min}} + h_2 \frac{1}{M_{max}} \quad (11)$$

where h_1 and h_2 are defined as:

$$h_1 = \frac{1/M - 1/M_{max}}{1/Mmin - 1/M_{max}}, \ h_2 = \frac{1/Mmin - 1/M}{1/Mmin - 1/M_{max}} \quad (12)$$

where $h_i \geq 0,\ i = 1, 2,$ and $\sum_{i=1}^{2} h_i = 1$. The suspension model in Eq. (7) with a variable driver's weight can be expressed as:

$$\dot{\mathbf{X}} = \sum_{i=1}^{2} h_i \, \mathbf{A}_i \mathbf{X} + \sum_{i=1}^{2} h_i \, \mathbf{B}_{1i}(u - F_d) + \mathbf{B}_2 d \quad (13)$$

where matrix $\mathbf{A}_i$ and $\mathbf{B}_{\mathbf{1i}}$, $i = 1, 2$, are obtained by replacing M with M_{min} and M_{max}.

Similarly, the measurement variable (9) and the control output (10) can be expressed as

$$Y_2 = \sum_{i=1}^{2} h_i \, \mathbf{C}_{2i} \mathbf{X} + \sum_{i=1}^{2} h_i \, D_{2i}(u - F_d) \quad (14)$$

$$Z_1 = \sum_{i=1}^{2} h_i \, \mathbf{C}_{3i} \mathbf{X} + \sum_{i=1}^{2} h_i \, D_{3i}(u - F_d) \quad (15)$$

For description simplicity, we define $\mathbf{A}_h = \sum_{i=1}^{2} h_i \, \mathbf{A}_i$, and the same simplification method is applied to other model matrices.

2.3 TS fuzzy disturbance observer

Eq. (3) shows that three forces are exerted on the suspended mass, namely spring force, disturbance force and active force. The friction force f_r is an important component of disturbance force F_d. The active seat suspension prototype applied in this paper has a peak friction of about 80 N which is shown in Appendix. It indicates that F_d has a great influence on the system dynamic, thus F_d needs to be estimated to improve controller's performance. Eq. (3) can be written as:

$$F_d = -M\ddot{z}_s - k_f(z_s - z_v) + u \tag{16}$$

Modifying the estimation by the difference between the estimated value and the actual value is the main idea to design an observer. Thus, a disturbance observer is defined as:

$$\dot{\widehat{F_d}} = -l(F_d - \widehat{F_d}) \tag{17}$$

Combing Eq. (9) and (16), it is rearranged as：

$$\dot{\widehat{F_d}} = l\left(M\ddot{z}_s - \left(-k_f(z_s - z_v) - \widehat{F_d} + u\right)\right) = L[Y_2 - (\mathbf{C_2X} + D_2(u - \widehat{F_d}))] \tag{18}$$

where $L = lM$.

The observer error is defined as:

$$e_f = F_d - \widehat{F_d}. \tag{19}$$

Since, generally, there is no prior information about the derivative of the disturbance F_d, and disturbance varies slowly relative to the observer dynamics, $\dot{F_d} = 0$ is a reasonable assumption. Combining Eq. (9) with (18) yields:

$$\dot{e_f} = \dot{F_d} - \dot{\widehat{F_d}} = LD_2e_f \tag{20}$$

With time t, we can obtain that $e_f(t) = \mathrm{e}^{LD_2(t-t_0)}e_f(t_0)$. Thus, if $\Gamma = LD_2 < 0$, the observer error will be exponentially converged.

The disturbance observer in Eq. (18) is an acceleration measurement based observer. The acceleration is introduced into control algorithm by estimating the differential of the disturbance. The sensitivity of acceleration to vibration can improve algorithm’s dynamic response.

Considering the TS fuzzy model, the TS fuzzy disturbance observer can be defined as:

$$\dot{\widehat{F_d}} = L_h[Y_2 - (\mathbf{C}_{2h}\mathbf{X} + D_{2h}(u - \widehat{F_d}))] \tag{21}$$

The differential of estimation error is:

$$\dot{e_f} = L_h D_{2h} e_f \tag{22}$$

In order to get the same convergence rate for different driver weights, the TS fuzzy observer gains are obtained

$$L_i = \frac{D_{2i}}{\Gamma}, \qquad i = 1,2 \tag{23}$$

where $\Gamma < 0$ is a constant. The convergence rate is inversely proportional to $|\Gamma|$. A smaller $|\Gamma|$ will lead to a more accurate estimation of the disturbance; on the other hand, the chattering problem may be introduced by compensating a fast varying disturbance value. Thus, $|\Gamma|$ needs to be selected according to the practical application.

2.4 TS fuzzy H_∞ controller design with disturbance compensation

Since the disturbance is estimated, a state feedback controller with disturbance compensation is constructed as:

$$u = \mathbf{K}_h \mathbf{X} + \widehat{F_d} \tag{24}$$

where $\mathbf{K}_h$ is the state feedback gain to be designed.

Substitute Eq. (24) into (13), then

$$\dot{\mathbf{X}} = (\mathbf{A}_h + \mathbf{B}_{1h}\mathbf{K}_h)\mathbf{X} - \mathbf{B}_{1h} e_f + \mathbf{B_2} d \tag{25}$$

Combining the disturbance force estimation, a new state space equation is obtained:

$$\dot{\overline{\mathbf{X}}} = \overline{\mathbf{A}}_h \overline{\mathbf{X}} + \overline{\mathbf{B}} d \tag{26}$$

where $\overline{\mathbf{X}} = \begin{bmatrix} \mathbf{X} \\ e_f \end{bmatrix}$, , $\overline{\mathbf{A}}_h = \begin{bmatrix} \mathbf{A}_h + \mathbf{B}_{1h}\mathbf{K}_h & -\mathbf{B}_{1h} \\ 0 & L_h D_{2h} \end{bmatrix}$, $\overline{\mathbf{B}} = \begin{bmatrix} \mathbf{B_2} \\ 0 \end{bmatrix}$,

To achieve good ride comfort and reduce the effect of disturbance's estimation error, the objective output is defined as

$$\mathbf{Z} = \bar{\mathbf{C}}_h \bar{\mathbf{X}} \tag{27}$$

where $\bar{\mathbf{C}}_h = \begin{bmatrix} \mathbf{C}_{3h} + D_{3h}\mathbf{K}_h & -D_{3h} \\ 0 & c_0 \end{bmatrix}$, c_0 is a constant.

The $\mathcal{L}_2$ gain of the system (26) and (27) is defined as:

$$\|T_{Zd}\|_\infty = \sup \frac{\|\mathbf{Z}\|_2}{\|d\|_2} \quad (d \neq 0) \tag{26}$$

where $\|\mathbf{Z}\|_2 = \int_0^\infty \mathbf{Z}^{\mathrm{T}}(t)\mathbf{Z}(t)\mathrm{d}t$ and $\|d\|_2 = \int_0^\infty d^{\mathrm{T}}(t)d(t)\mathrm{d}t$.

It is easy to conclude that if there is a positive definite matrix $\mathbf{P} = \begin{bmatrix} \mathbf{P_1} & * \\ 0 & \mathbf{P_2} \end{bmatrix}$, $\mathbf{P_1} = \mathbf{P_1}^{\mathbf{T}} > \mathbf{0}$, $\mathbf{P_2} = \mathbf{P_2}^{\mathbf{T}} > \mathbf{0}$, such that the following LMI is satisfied [36]:

$$\in = \begin{bmatrix} \bar{\boldsymbol{A}}_h{}^T \boldsymbol{P} + \boldsymbol{P}\bar{\boldsymbol{A}}_h & * & * \\ \bar{\mathbf{B}}^{\mathbf{T}}\mathbf{P} & -\lambda^2\mathbf{I} & * \\ \bar{\boldsymbol{C}}_h & 0 & -\mathbf{I} \end{bmatrix}$$

$$= \begin{bmatrix} (\mathbf{A}_h + \mathbf{B}_{1h}\mathbf{K}_h)^{\mathbf{T}}\mathbf{P_1} + \mathbf{P_1}(\mathbf{A}_h + \mathbf{B}_{1h}\mathbf{K}_h) & * & * & * & * \\ -\mathbf{B}_{1h}{}^{\mathbf{T}}\mathbf{P_1} & D_{2h}{}^{\mathbf{T}} L_h{}^{\mathrm{T}}\mathbf{P_2} + \mathbf{P_2} L_h D_{2h} & * & * & * \\ \mathbf{B_2}{}^{\mathbf{T}}\mathbf{P_1} & 0 & -\lambda^2\mathbf{I} & * & * \\ \boldsymbol{C}_{3h} + D_{3h}\boldsymbol{K}_h & -D_{3h} & 0 & -\mathbf{I} & * \\ 0 & c_0 & 0 & 0 & -\mathbf{I} \end{bmatrix} < 0 \tag{28}$$

then the system (26) is stable with H_∞ performance index $\lambda > 0$.

Pre- and post-multiplying Eq. (28) by diag($\mathbf{P_1}^{-\mathbf{1}}, \mathbf{I}, \mathbf{I}, \mathbf{I}, \mathbf{I}$) and its transpose, respectively. Defining $\mathbf{Q} = \mathbf{P_1}^{-\mathbf{1}}$, $\mathbf{K}_h\mathbf{Q} = \mathbf{R}_h$, and $\mathbf{P_2}$ is set as a constant matrix. The condition $\in < 0$ is equivalent to

$$\sum_{i,j=1}^{2} h_i h_j \ni_{ij} < 0 \tag{29}$$

where

$$
\ni_{ij}= \begin{bmatrix} \mathbf{A}_i\mathbf{Q}+\mathbf{B}_{1i}\mathbf{R}_j+(\mathbf{A}_i\mathbf{Q}+\mathbf{B}_{1i}\mathbf{R}_j)^{\mathrm{T}} & * & * & * & * \\ -\mathbf{B}_{1i}{}^{\mathrm{T}} & (\mathbf{P_2}L_jD_{2i})^{\mathrm{T}}+\mathbf{P_2}L_jD_{2i} & * & * & * \\ \mathbf{B}_2{}^{\mathrm{T}} & 0 & -\lambda^2\mathbf{I} & * & * \\ C_{3i}\mathbf{Q}+\mathbf{D}_{3i}\mathbf{R}_j & -\mathbf{D}_{3i} & 0 & -\mathbf{I} & * \\ 0 & c_0 & 0 & 0 & -\mathbf{I} \end{bmatrix}<0 \quad (30)
$$

Based on parameterized linear matrix inequality (PLMI) techniques [36], the relaxation result can be written as:

$$
\ni_{ii}\leq 0, \qquad i=1,2 \quad (31)
$$

$$
\ni_{ii}+\frac{1}{2}\left(\ni_{ij}+\ni_{ji}\right)<0, \qquad 1\leq i\neq j\leq 2 \quad (32)
$$

After solving Eq. (31) and (32) by linear matrix inequality (LMI) toolbox in Matlab, the controller gain is obtained as $\mathbf{K_i}=\mathbf{R_i}\mathbf{Q^{-1}}$.

The controller implementation procedure is shown in Fig. 2. In the practical scenario, the controller is implemented in discrete-time; h_i only needs to be calculated once when one certain driver is operating the vehicle. In one control sample period, 15 multiplications and 12 additions/subtractions are operated for the controller and observer. This means that the proposed control strategy is easy to be implemented in the practical application. As the applied disturbance compensation strategy considers all the possible disturbances, uncertainties, modelling errors, actuator saturation, etc., the robustness of the proposed method is enhanced.

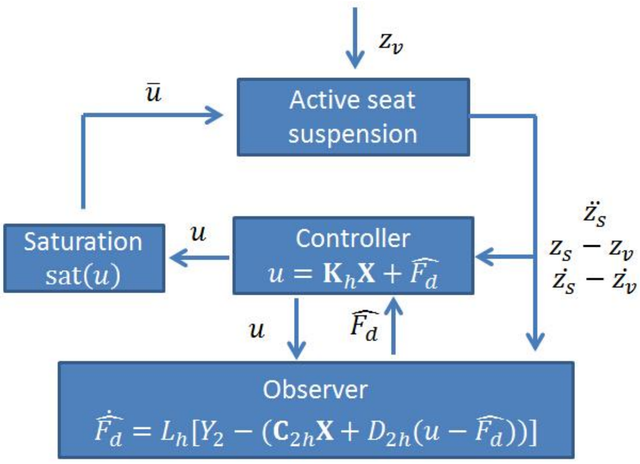

Fig. 2. Controller implementation

Remark 1. To evaluate a new method, it is common to compare it with other available existing methods in dealing with the same problem. We noticed that many references have proposed different control methods for vehicle suspensions [11, 32], but they cannot be directly compared with our proposed method because there are different aspects between seat suspension control and vehicle suspension control. Even though some papers studied about seat suspensions but most of them only considered road input disturbance but did not consider the practical friction issue. On the other hand, from implementation point of view, some control strategies, such as the proposed output feedback control by Li et.al [12], are complicated when compared to the state feedback control using available measurements. Thus to make a fair evaluation for the proposed method, we only compare it with the uncontrolled case in the simulations and passive and uncontrolled cases in the experiments to validate its effectiveness.

3. Control algorithm evaluation

3.1 Active seat suspension prototype

When designing an active set suspension system, the trade-off between its performance and cost needs to be considered. Generally, the higher performance is required, the higher cost actuator is needed. Considering the factor that, the vertical vibration of heavy duty vehicle seat is highest in 2 Hz to 4 Hz and 3 Hz vibration is closely related to drivers' fatigue, if low frequency vibration can be isolated, the heavy duty vehicle drivers' ride comfort can be greatly improved and the drivers' fatigue can be reduced. At the same time, low cost actuator can be applied to the active seat suspension, if the low frequency vibration is mainly focusing on. For the higher frequency vibration, active seat suspension's passive performance can dissipate the vibration energy, and its vibration magnitude is much lower than low frequency vibration.

Heavy duty vehicle drivers always suffer from severer vibration than normal passage vehicle drivers, thus seat suspensions are widely applied to improve ride comfort and protect drivers' health. Fig. 3

shows a well-tuned passive seat suspension (GARPEN GSSC7, left) and an active seat suspension prototype (right) which is applied in this paper to validate the proposed control algorithm. The active seat suspension prototype is modified from the passive seat suspension by removing the damper and installing rotary actuators in the scissor's structure of the passive seat suspension. The rotary actuator is composed with one rotary servo motor and one gear reducer (ratio 20:1). The output torque of the motors can be controlled accurately by their drives.

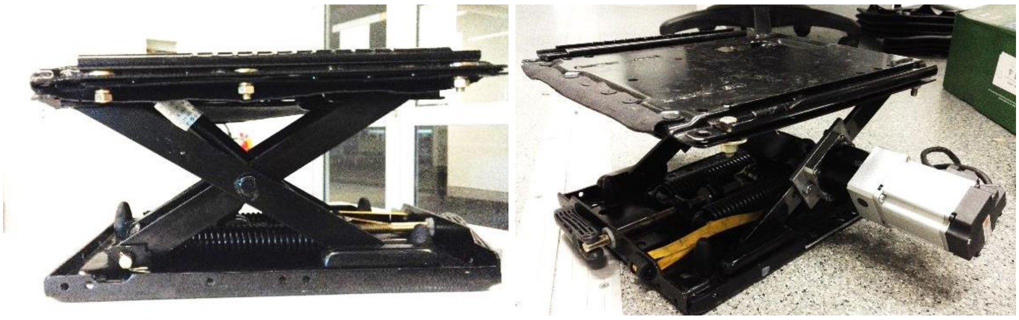

Fig. 3. Conventional passive seat suspension and active seat suspension prototype

For convenient analysis and control, an equivalence equation is derived to transfer the motor torque to a force exerting on the suspended mass by the kinematic model in Fig. 4:

$$T = \frac{u\sqrt{L^2-(h_0+\Delta h)^2}}{2r_g} \tag{33}$$

where T is the torque output by motor, u is the equivalent active force exerting on suspended mass, r_g is the gearbox ratio, L is the scissor link length, h_0 is the initial vertical distance between the upper and base hinges, and Δh is the suspension deflection which can be measured in real-time.

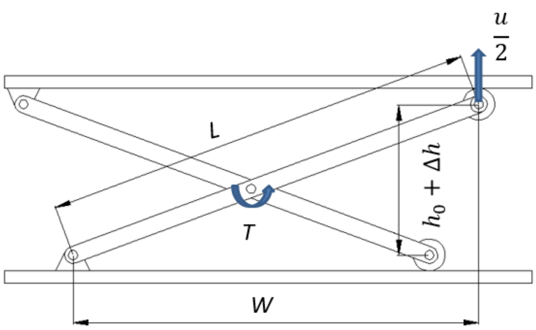

Fig. 4. Torque to equivalent force

In the Appendix, the friction of the active seat suspension prototype is measured and modelled for simulation evaluation.

3.2 Simulation evaluation

In order to evaluate the performance of proposed observer and controller, the simulation is implemented on the 2-DOF system model (1) and (2). The friction model in Appendix is applied with the identified parameters. Table 1 shows the parameters in simulation. The TS fuzzy controller gains are obtained as $K_1 = [4573 \;\; -90]$ and $K_2 = [4560 \;\; -134]$. Two sets of observer parameters are selected, where Γ_1 is chosen for Observer 1 and the disturbance observer gains are $L_{11} = -4566$ and $L_{12} = -3086$; Γ_2 is chosen for Observer 2 and the disturbance observer gains are $L_{21} = -13699$ and $L_{22} = -9259$.

A bump road surface is applied to the simulation model as:

$$z_v(t) = \begin{cases} \frac{a}{2}\left(1 - \cos\left(\frac{2\pi v_0}{l} t\right)\right), & 0 \le t \le \frac{l}{v_0} \\ 0, & t > \frac{l}{v_0} \end{cases} \tag{34}$$

where $a = 0.07$ m and $l = 0.8$ m are the height and length of the bump. The vehicle velocity is set as $v_0 = 1\,\mathrm{m/s}$.

Table 1. Parameters used in the simulation

m_b	55 Kg	α	0.02
m_s	28 Kg	P_2	400
k_c	90000 N/m	λ	1.2
c_c	2500 Ns/m	Γ_1	$-3e^{-6}$
k_f	4600 N/m	Γ_2	$-1e^{-6}$
u_{lim}	250 N	m_{bmin}	45 Kg
m_{bmax}	120 Kg		

The driver body acceleration is shown in Fig. 5 where there is big peak acceleration about 4 $\mathrm{m/s^2}$ with uncontrolled seat suspension, and the driver body accelerations are both decreased when the controller is implemented with either Observer 1 or Observer 2. Although the acceleration magnitudes are very close with the two observers, with Observer 2, the acceleration goes to zero faster. Fig. 6 shows the observers performance. The Observer 2 can track the disturbance better than Observer 1, but it has a bigger peak disturbance because of actuator saturation. The saturated control force is shown in Fig. 7. It needs to be explained here that the friction model is included in the simulation, so when the seat suspension is stable, the static friction may not be zero. The simulation with bump surface indicates that, with a smaller $|\Gamma|$, the controller and observer will have better performance when the actuator output is not saturated; but when the actuator output is saturated, a bigger disturbance value is appeared and this will affect the controller's performance.

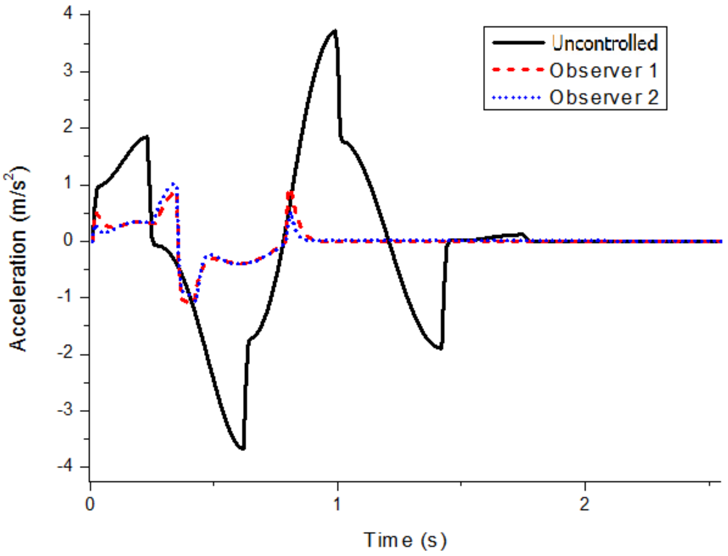

Fig. 5. Acceleration of driver body.

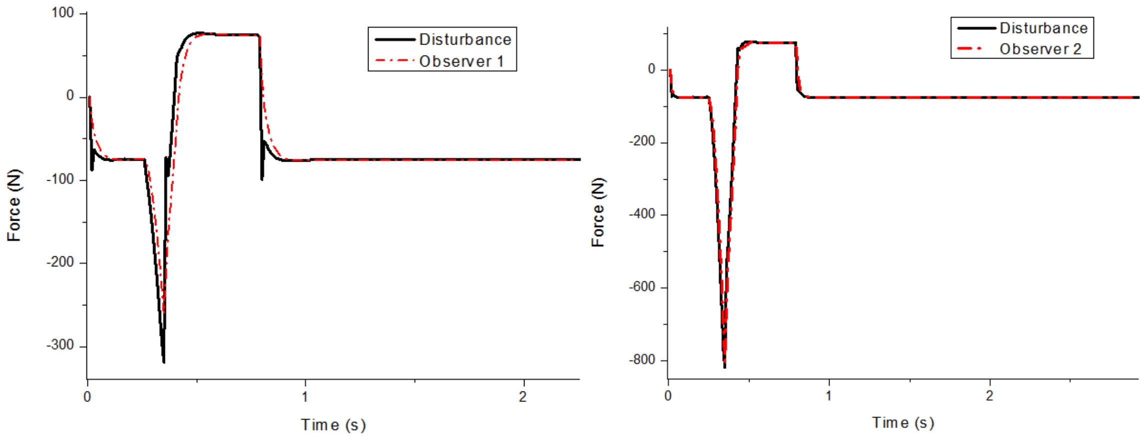

Fig. 6. Observer performance

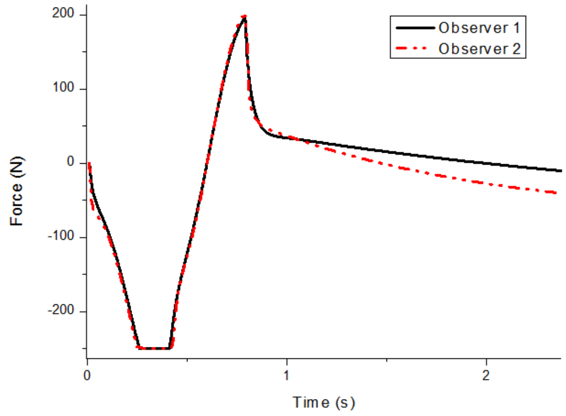

Fig. 7. Saturated control force

Fig. 8 shows the vibration transmissibility of the uncontrolled and controlled active seat suspension with Observer 1. In the low frequency range, the acceleration transmissibility is kept in low magnitude (around 0.4), which means the highest magnitude vibration of heavy duty vehicles can be controlled. Considering that this paper focuses on implementing the algorithm in practical application, more experimental results and analysis are presented in the following sections.

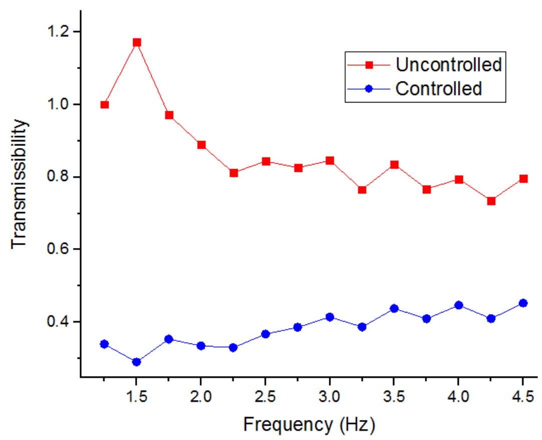

Fig. 8. Simulation acceleration transmissibility of seat suspension

3.3 Experimental setup

The proposed TS fuzzy controller is implemented in the lab. The experimental setup is shown in Fig. 9 where a 6-degrees of freedom (6-DOF) vibration platform is applied to generate vertical excitation and the base of the seat suspension is fixed on this platform. The desired vibration profile can be input into the controller of this 6-DOF vibration platform, and then the upper vibration platform will move based on the vibration commands. The active seat suspension is controlled by a NI CompactRio 9074 with one NI 9205 and one NI 9264 modules. The control frequency is set as 500 Hz. Two accelerometers (ADXL203EB) are utilised to measure the acceleration of vibration platform and seat. The seat suspension deflection is measured by a displacement sensor (Micro Epsilon ILD1302-100).

Three different types of experiments were carried out and the results were compared. The uncontrolled experiment applied the active seat suspension without torque output. The active experiment was done with the proposed TS fuzzy controller. The passive experiment was implemented with the passive seat suspension (GARPEN GSSC7) which is a seat with well-tuned spring and damper for heavy duty vehicles.

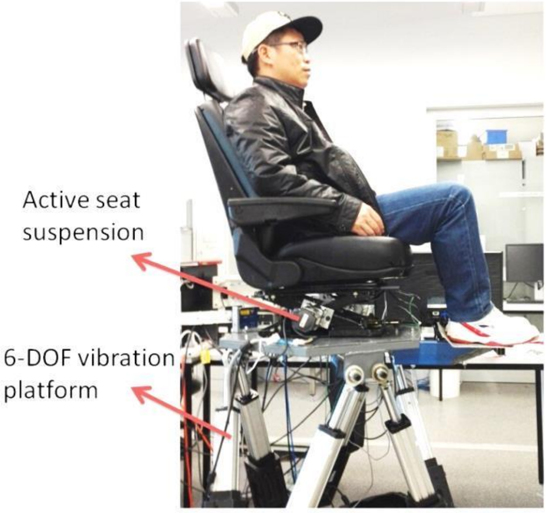

Fig. 9. Experimental setup

3.4 Experimental results

Sinusoidal excitations were applied to the seat suspensions with 55 Kg load. Fig. 10 shows the acceleration transmissibility of three kinds of seat suspensions. The uncontrolled active seat suspension has a better performance in suppressing resonance vibration than the well-tuned passive seat suspension because of its inner friction. When the excitation frequency is increased, both acceleration transmissibility of uncontrolled active seat suspension and passive suspension decreased. In the frequency range 2.5 to 4.5 Hz, this well-tuned passive seat suspension has a better performance than the uncontrolled active seat suspension, but their transmissibility curves are very close. The controlled active seat suspension can successfully isolate the low frequency vibration, which proves the proposed controller's effectiveness. It should be emphasised that when vibration frequency is higher than 4 Hz, the passive seat suspension and uncontrolled active seat suspension have low acceleration transmissibility which indicates that seat suspensions can perform good with their passivity in high frequency vibration. On the other hand, the vertical vibration of heavy duty vehicle seat is highest in the frequency range 2 to 4 Hz, because the vehicle (chassis) suspension system has already isolated most of high frequency vibration from uneven road. This test result proves that it is reasonable to focus on low frequency vibration control for heavy duty vehicle seat suspension.

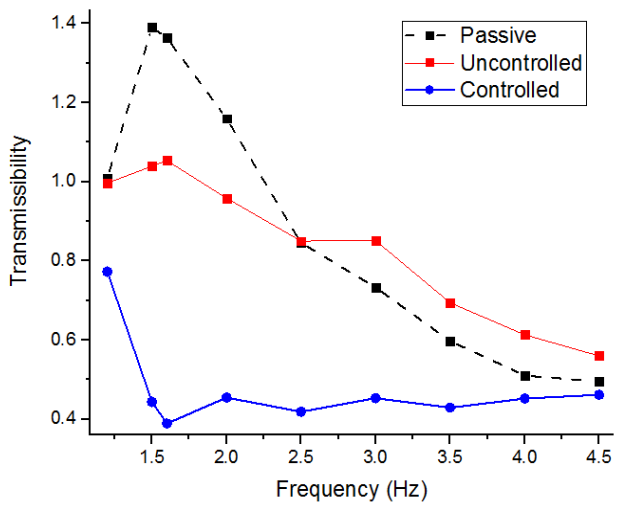

Fig. 10. Experimental acceleration transmissibility to seat

The bump excitation can test the controller's transient response performance. Fig. 11 and 12 show the seat acceleration comparison of the passive seat suspension and controlled active seat suspension with different loads. When 55 Kg mass is loaded on the seat, the peak acceleration magnitude is reduced from 2.02 $\mathrm{m/s^2}$ to 1.32 $\mathrm{m/s^2}$. And when 70 Kg mass is loaded, the peak acceleration magnitude is reduced from 2.33 $\mathrm{m/s^2}$ to 1.03 $\mathrm{m/s^2}$. The results indicated the TS fuzzy controller with disturbance observer can fast response to bump excitation and reduce the vibration peak magnitude with different loads.

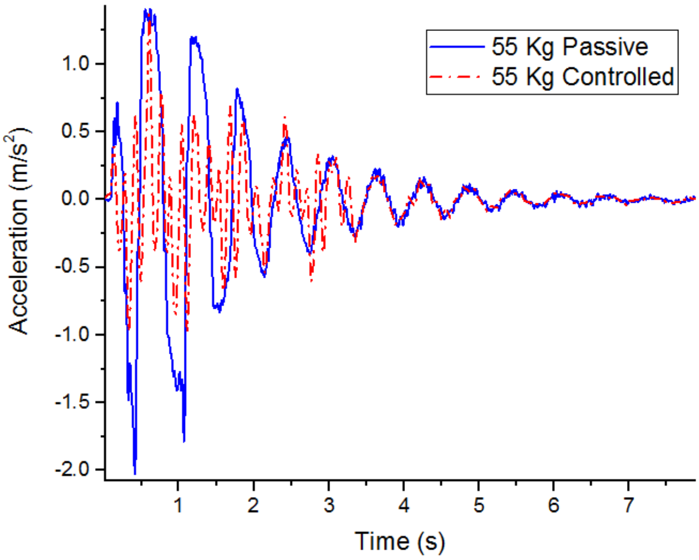

Fig. 11. Bump excitation with 55 Kg load.

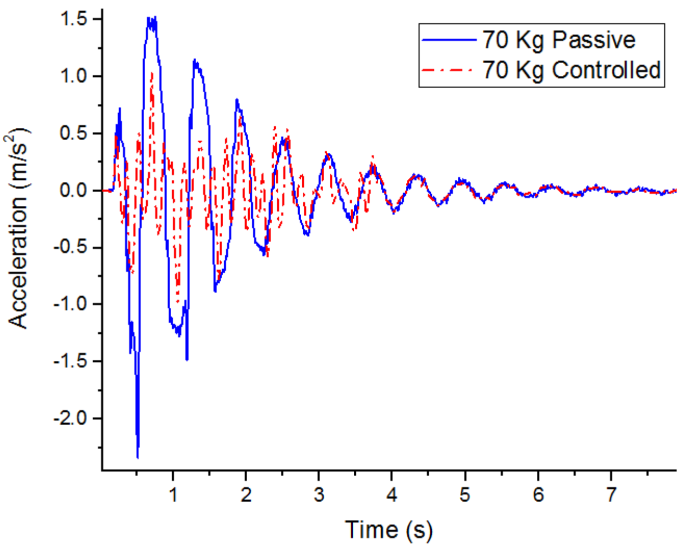

Fig. 12. Bump excitation with 70 Kg load

Random excitation is always applied to evaluate seat suspension's performance in time domain. Fig. 13 and 14 show seats' acceleration with different loads. It can be seen that, for the uncontrolled active seat suspension, it has a big peak vibration magnitude about 6 m/s^2, this is because the inner friction cannot suppress the big magnitude vibration around resonance frequency; the suspension is unstable. The passive seat suspension can suppress this vibration, but has bigger vibration magnitude in other parts. The controlled seat suspension can keep the seat acceleration at low magnitude all the time. Table 2 shows comparison of the Root Mean Square (RMS) acceleration, there are 45.5% and 49.5% reductions between controlled active seat suspension and passive seat suspension with 55 Kg load and 70 Kg load, respectively.

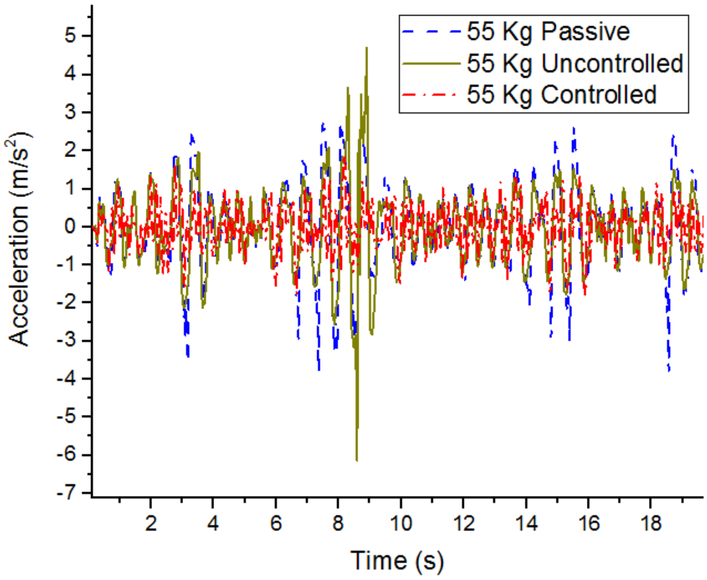

Fig. 13. Random excitation with 50 Kg load

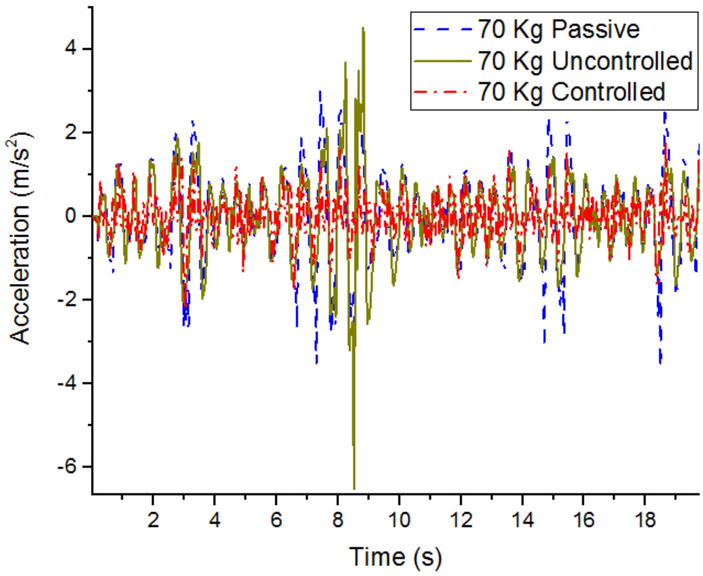

Fig. 14. Random excitation with 70 Kg load

Table 2. RMS acceleration

	Passive (m/s^2)	Uncontrolled (m/s^2)	Controlled (m/s^2)	Reduction (%)
55 Kg	1.061	1.067	0.578	45.5

70 Kg	1.005	1.020	0.508	49.5

For further analysing the random excitation performance, Power Spectral Density (PSD) is shown in Fig. 15 and 16. Both passive seat suspension and uncontrolled active seat suspension have a big PSD value around 2 Hz, and have very low value in higher frequency. The controlled active seat suspension can control the vibration in low magnitude at the whole presented frequency range.

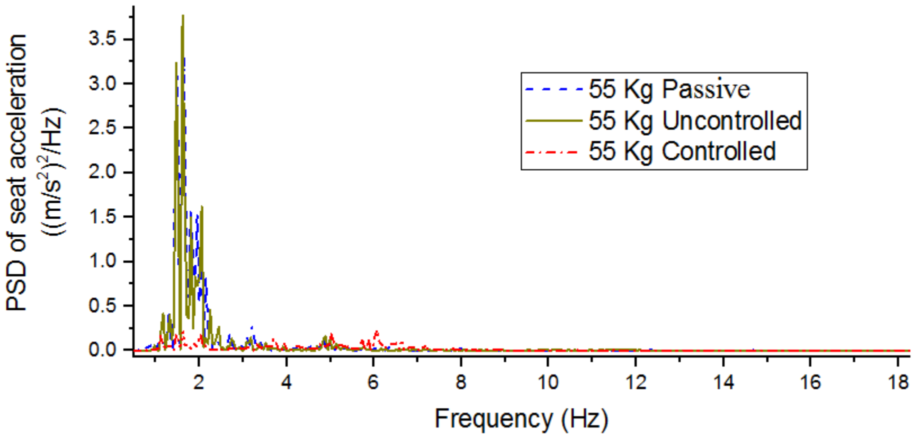

Fig. 15. PSD of vibration with 55 Kg load

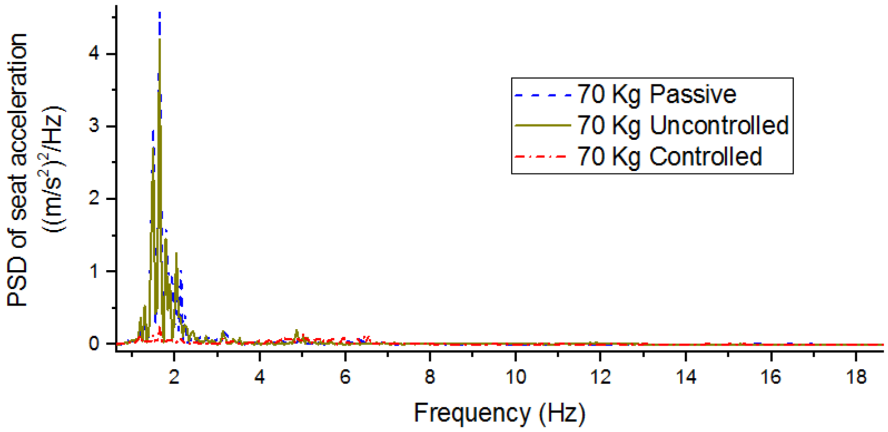

Fig. 16. PSD of vibration with 70 Kg load

4. Conclusion

In this paper, a TS fuzzy controller with a disturbance observer has been designed and validated on an active seat suspension prototype for heavy duty vehicles. The vertical vibration of heavy duty vehicle seat is highest in the low frequency range (2 Hz to 4 Hz). The applied active seat suspension for heavy duty vehicles focus on reducing low frequency vibration which can greatly improve ride comfort and decrease cost at the same time. The designed algorithm introduces seat acceleration into a disturbance observer which can estimate friction force, model simplification caused disturbance and control output error at the same time. The TS fuzzy controller with disturbance compensation applies the seat acceleration and suspension deflection, which can be easy to measure in practical application, as control feedbacks. The friction model of the active seat suspension is identified and applied in simulation to validate the proposed controller. The simulation result shows acceleration transmissibility can be greatly reduced in the low frequency vibration. The experimental results are further conducted, where a well-tuned passive seat suspension is applied to compare with the performance of the active seat suspension system. Loads of 55 Kg and 70 Kg are applied to test the TS fuzzy controller. The bump excitation test shows the active seat suspension has a good transient response performance with both loads. The PSD plot and RMS acceleration are applied to analyse the random excitation test in frequency domain and time domain. The active seat suspension performs much better than the well-tuned passive seat suspension both in time domain and frequency domain. There are 45.5% and 49.5% RMS acceleration reduction with 55 Kg and 70 Kg loads, respectively. The low frequency vibration can be controlled to a very low magnitude by the active seat suspension. The proposed control algorithm can improve driver's ride comfort and reduce fatigue, and it is very practical for real application. In this paper, the mass of the driver body suspended by the seat suspension is assumed to be known and this is applicable to any tested drivers. In practice, the mass of the driver body may have small variation due to, for example, the change of driver's posture. In such a case, the real-time parameter estimation algorithm for the mass of the driver body may be included to improve the performance of the control algorithm in practical application, which is, however, beyond the scope of this paper but reserves a further study. The proposed disturbance compensation control method with disturbance observer can be combined with other controllers, e.g., sliding mode controller. Furthermore, the anti-windup approach applied in this paper can successfully handle the actuator saturation issue, but more theoretical research could be done to get less conservative results in the future,

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Appendix

In order to reduce the requirement of motors, the gear reducers which can amplify the output torque of motors are applied in the active seat suspension prototype. At the same time, the additional friction is introduced into the active seat suspension system. The two gear reducers can work as friction dampers to dissipate vibration energy. For implementing the algorithm in simulations, the friction model parameters are identified in this section. The MTS system with measured force and displacement output was utilised to test the active seat suspension prototype in the laboratory as shown in Fig. 17. The non-energised active seat suspension was loaded with sinusoidal signals of 5 mm and 10 mm at frequency of 0.5 Hz. When the spring force is removed from the measured force, the inner friction is obtained as shown in Fig. 18 which shows that the friction direction will change based on the variation of suspension relative deflection rate, and it will saturate around 80 N.

Fig. 17. MTS test

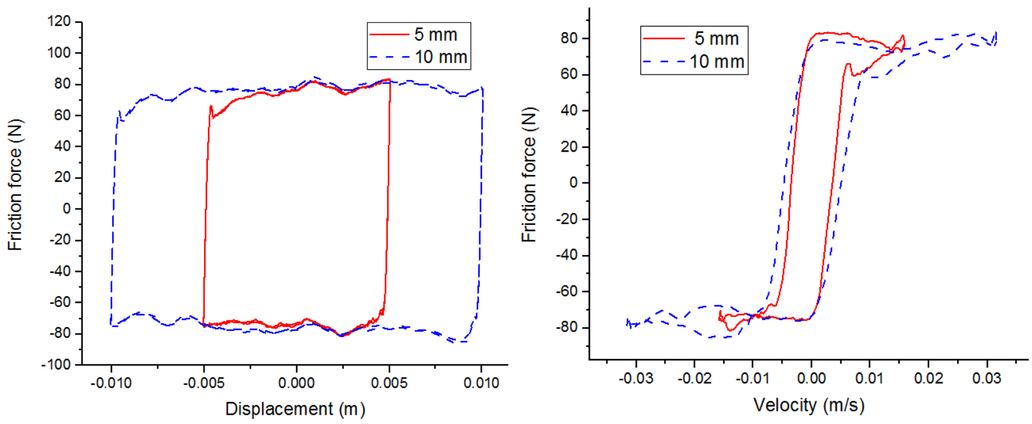

Fig. 18. Displacement-force and velocity-force plot of friction

To date, many models have been proposed to describe the friction. A phenomenological model based on Bouc-Wen hysteresis model is applied in the simulation to validate the proposed control algorithm. The friction can be defined as:

$$f_r = \alpha z_d \tag{35}$$

$$\dot{z_d} = -\gamma_d |v| z_d |z_d| - \beta_d v |z_d|^2 + A_d v \tag{36}$$

where v is the suspension deflection rate. The friction model parameters are identified as $\alpha = 3.0983 \times 10^5$ N/m, $\gamma_d = 2.1617 \times 10^8$ m^{-2}, $\beta_d = -9.8889 \times 10^7$ m^{-2} and $A_d = 6.9321$. The simulated friction is compared with measured friction in Fig. 19 which shows the applied friction model can describe the actual friction in active seat suspension.

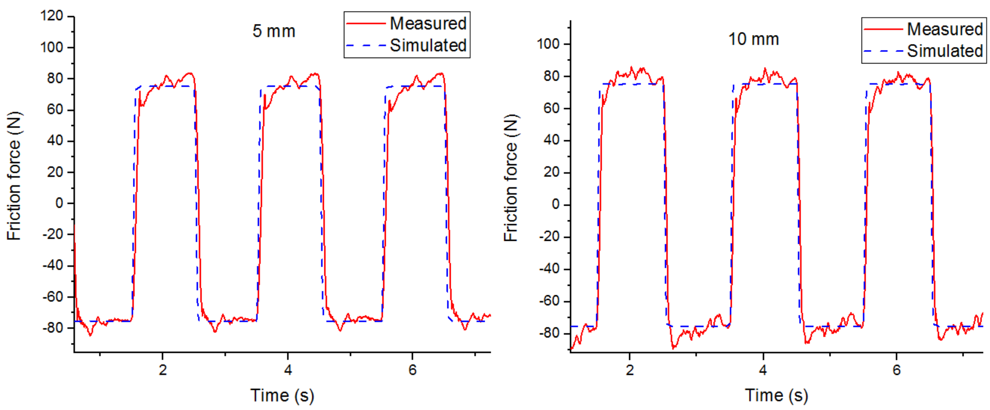

Fig. 19. Friction model identification

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