Simulation of Active Vibration Suppression Using Internal Motor Sensing

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***Abstract*—Suppression of unnecessary vibration is an important aspect in control system design. Passive, semi active, and active suspension systems are used in vehicles to suppress vibrations and theoretically active suspension systems provide superior performance than the prior two types. This paper proposes a novel method to suppress vibrations using internal motor sensing method. The proposed active vibration suppression system uses a reaction force observer to measure and suppress vibrations acting on the system without using environmental sensors. The motor forcer acceleration and a current sensor measurements are used by the reaction force observer to measure the vibration forces in the system. The proposed system performed system performance, robustness, and applicability is evaluated using quarter car suspension system model. The proposed system is simulated for different conditions to measure the system vibration suppression capabilities. The simulation results provide evidence of robust vibration suppression capabilities and applicability of the controller for real world applications**.

Index Terms—**Active suspension, Active vibration suppression, ,Disturbance observer, Force control , Passive suspension, Reac-tion force observer.**

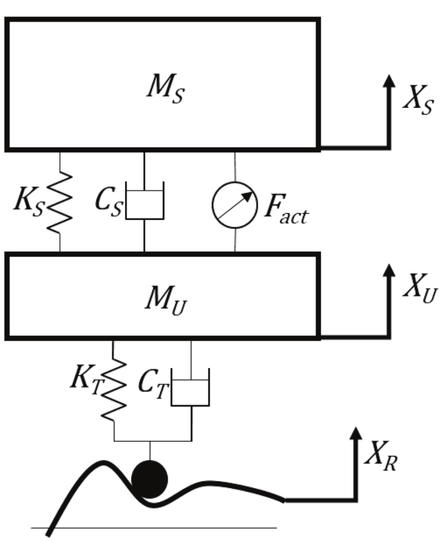

Fig. 1. Quarter car model

I. Introduction

Vibration is a phenomena that occur in real world systems including automobile systems. Vibration causes energy losses, damage to components, and discomfort to humans. Engineers and scientists have designed systems to overcome the problem of vibrations in real world systems with vibration suppression systems. Automobile suspension systems are developed to overcome problem of vibration in vehicles, starting from the development of first shock-absorber by a French cyclist in 1898 [1] that was used in a bicycle. These automobile vibration suppression systems are developed to provide better vehicle control, passenger comfort and better ride quality in automobiles. Humans feel dizziness and sea sickness for vibrations in ranges of 0.5 Hz to 1 Hz and 18 Hz to 20 Hz [2] which could be categorize as uncomfortable vibrations for humans. Suspension systems are connected in between vehicle body and tires suppress vibrations transferring from the road surface to vehicular body through tires and suspension system. In addition the suspension system reduces the vehicle body movement reducing roll and pitch of the body, and providing better ride quality to passengers. Furthermore passive suspension systems, semi-active suspension systems, and active suspension systems were designed to reduce vehicle vibrations, and to improve vehicle handling in automobiles [3].

Passive suspension systems consists of passive elements to suppress vibrations, the conventional passive suspension system is generally built using springs and dampers [4]. The inability to change its properties, cause passive suspension systems to produce uneven vibration suppression in different conditions. Semi-active suspension systems have the ability to control its spring and damper properties. Semi-active suspension system can achieve controllability by changing suspension system properties without introducing energy in to the suspension system, but dissipate or absorb vibration energy from the system by changing the suspension properties [5]. Active vibration suppression systems were developed to overcome problems faced by passive and semi-active suspension systems. Active suspension systems consists of active elements including motors and hydraulic pumps. The active elements have the ability to absorb and introduce energy to suspension system, and to change suspension properties accordingly to

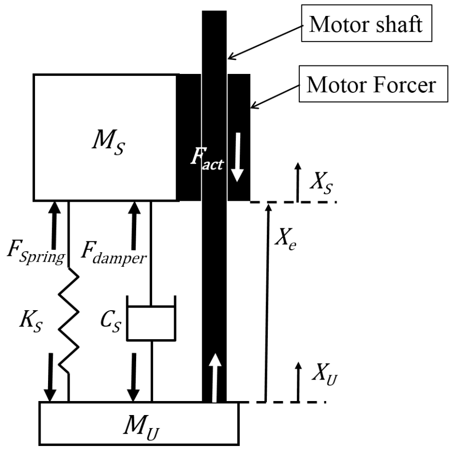

Fig. 2. Proposed suspension system model

TABLE I
SYMBOL DESCRIPTION

Symbol	Variable name
X_R	Road surface displacement (m)
X_U	Unsprung mass displacement (m)
X_S	Sprung mass displacement (m)
$\ddot{X}_S$	Motor forcer acceleration (m/s^2)
M_S	Sprung mass (kg)
M_{Sn}	Nominal sprung mass (kg)
M_U	Unsprung mass (kg)
K_S	Passive Suspension spring coefficient (N/m)
K_T	Tire spring coefficient (N/m)
C_S	Passive Suspension damping coefficient (N/m^2)
C_T	Tire damping coefficient (N/^{2})
F_{act}	Active motor Force (N)
F_{dis}	Disturbance force on motor (N)
F_{ref}	Force reference (N)
F_{RFOB}	RFOB force (N)
K_f	Linear motor force constant (Nm/A)
K_{fn}	Nominal linear motor force constant (Nm/A)
M_m	Linear motor mass (kg)
M_{mn}	Nominal Linear motor mass (kg)
K_P	PID controller proportional gain
K_I	PID controller integral gain
K_D	PID controller differential gain
G_{gis}	RFOB filter gain

provide superior vibration suppression performance. Semi-active suspension systems provide better vibration suppression than passive suspension systems [6], theoretically active suspension systems are far more superior than passive and semi active systems [7].

In semi-active suspension development variable dampers are commonly used to change damping constant in different conditions [8]. Mechanical orifice or smart fluid are used to control the fluid properties in variable dampers [9]. Active suspension systems commonly includes hydraulic or electromagnetic actuators [10]. Electromagnetic active suspension systems have fast responses than hydraulic systems and able to regenerate energy to reduce power consumption of the system [10].

The sensor reliability, performance and cost is main consideration of active suspension systems. The system sensor selection therefore one of the main concerns [11] in design. In addition sensor performance also have grave importance [12]. Failures that could occur due to suspension actuator failure creates safety problems. External environmental sensing is used as an input in vehicle active vibration suppression system controllers [13]. It is necessary to produce internal motor sensing is more reliable than external environmental sensing to provide robust control on sensor failures or where environmental sensing is not possible. The suspension system should be able to provide vibration suppression capabilities even in actuator failures.

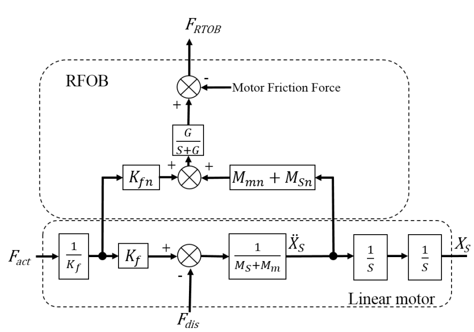

Fig. 3. Proposed linear motor model

This paper proposes a novel method of active vibration suppression using internal motor sensors and use a linear motor as the actuator parallel to the a passive suspension system. The proposed method uses a Reaction Force Observer (RFOB) [14] in the controller which uses motor current and motor forcer acceleration as the measurement. The motor forcer acceleration and motor current is used to measure forces acting on motor forcer. The system reduce vibrations by providing opposite force to the measured force to minimize forces acting on the motor forcer. The proposed controller do not use external environmental sensing for the calculations. The parallel passive suspension system provide vibration suppression even in actuator failures. The proposed method could be especially used in places where environmental sensing is difficult and computational and hardware cost is a problem. This novel method is simulated and results were demonstrated using a Fig.1 quarter car model.

Mathematical model of the active vibration suppression system is presented in section II using a quarter car model. Section III of this paper consists of simulation results of the active vibration suppression system. In addition, simulation parameters are also disclosed in section III. Finally the paper is concluded in section IV.

II. Modelling

The active vibration suppression system is modeled using the quarter car model as shown in Fig.1. The quarter car model is a reduced model to illustrate a vehicle suspension and used in the paper to illustrate vibration suppressor controller applicability in real world systems. The quarter car model consists of an unsprung mass and a sprung mass. Variable symbols and descriptions used in the paper are provided in Table I. The sprung mass is suspended on top of unsprung mass using a passive spring damper system. Linear motor active vibration suppressor is connected parallel to passive suspension elements. Fig. 2 illustrates the physical connection arrangement and force distribution of the system. A frictionless brush less direct current motor is used parallel to passive suspension system as the actuator. The measurements were taken from the initial stable point of the system, under the system weights.

Passive suspension system transfer function (1) between X_R and X_S, has derived using Fig. 1, considering zero F_{act} for passive suspension system.

$$\frac{X_S}{X_R} = \frac{[C_T s + K_T][C_S s + K_S]}{\begin{array}{r}(M_S M_U s^4 + [M_S(C_S + C_T) + M_U C_S]s^3 \\ +[M_S(K_T + K_S) + M_U K_S + C_S C_T]s^2 \\ +[K_S C_T + K_T C_S]S + K_T K_S)\end{array}} \tag{1}$$

After introducing active vibration suppression controller, Sprung mass Newtons equation (2) and unsprung mass Newtons equation (3) could be derived as,

$$M_S \ddot{X_S} = K_S(X_S - X_U) + C_S(\dot{X_S} - \dot{X_U}) - F_{act} \tag{2}$$

$$\begin{aligned} M_U \ddot{X_U} = & - K_S(X_S - X_U) - C_S(\dot{X_S} - \dot{X_U}) + F_{act} \\ & K_T(X_U - X_R) + C_T(\dot{X_U} - \dot{X_R}) \end{aligned} \tag{3}$$

State space equation (4) of active suspension system could be derived using equations (2) and (3) for following state variables.

x_1:- Motor displacement
x_2:- Sprung mass velocity
x_3:- Road to unsprung mass displacement
x_4:- Unsprung mass velocity
Y :-Sprung mass displacement

The state space equations (4), (5) could be used to analyze system response.

$$\begin{aligned} \begin{bmatrix} \dot{x_1} \\ \dot{x_2} \\ \dot{x_3} \\ \dot{x_4} \end{bmatrix} = & \begin{bmatrix} 0 & 1 & 0 & 1 \\ \frac{-K_S}{M_S} & \frac{-C_S}{M_S} & 0 & \frac{C_S}{M_S} \\ 0 & 0 & 0 & 1 \\ \frac{K_S}{M_U} & \frac{C_S}{M_U} & \frac{K_T}{M_U} & \frac{C_S - C_T}{M_U} \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{bmatrix} \\ & + \begin{bmatrix} 1 \\ \frac{1}{M_S} \\ 0 \\ \frac{-1}{M_U} \end{bmatrix} F_{act} + \begin{bmatrix} 0 \\ 0 \\ -1 \\ \frac{C_T}{M_U} \end{bmatrix} \dot{X}_R \end{aligned} \tag{4}$$

$$\begin{bmatrix} Y \end{bmatrix} = \begin{bmatrix} \frac{-K_S}{M_S} & \frac{-C_S}{M_S} & 0 & \frac{C_S}{M_S} \end{bmatrix} \begin{bmatrix} x_1 \\ x_2 \\ x_3 \\ x_4 \end{bmatrix} + \begin{bmatrix} \frac{1}{M_S} \end{bmatrix} F_{act} \tag{5}$$

The controller measures the motor disturbance force,F_{dis} using RFOB. The RFOB force of passive suspension system

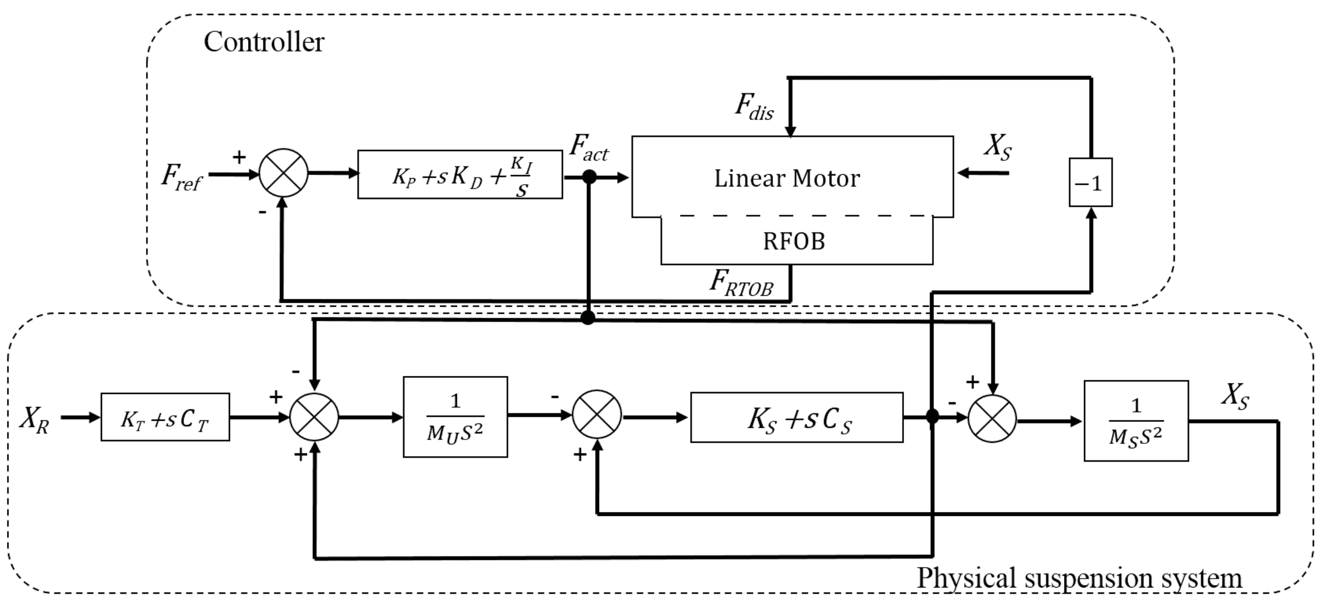

Fig. 4. Proposed Active vibration Suppression system controller

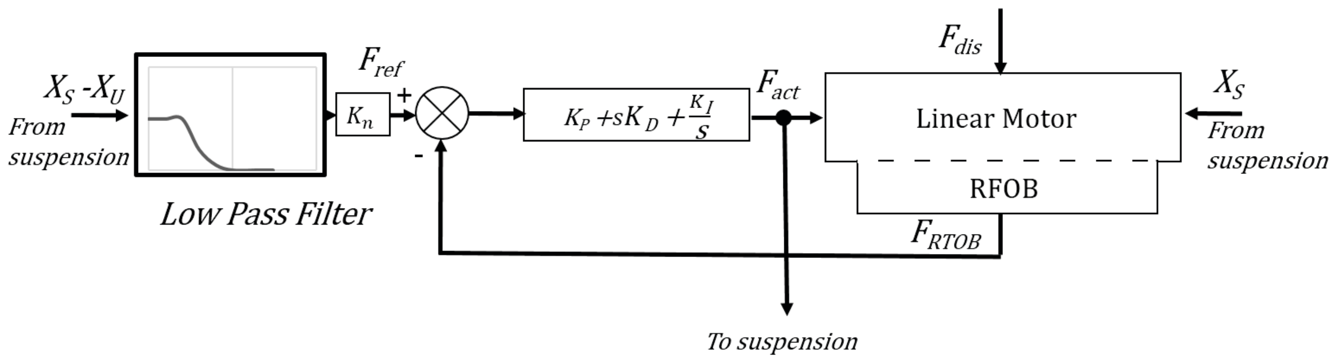

Fig. 5. Suspension deflection control model

could be derived as (6) using motor model in Fig. 3, Where G is the RFOB sampling filter gain that used to reduce sensor sampling errors,

$$F_{RTOB} = \frac{-G}{s+G}(K_S(X_S - X_U) + C_S(\dot{X_S} - \dot{X_U})) \quad (6)$$

In ideal situations where there is no controller latency or sensor errors, we could remove the filter used in RFOB and derive RFOB for ideal situations as (7),

$$F_{RTOB} = -K_S(X_S - X_U) - C_S(\dot{X_S} - \dot{X_U}) \quad (7)$$

The linear motor contains a force controller with a PID controller as shown in Fig.4. The linear motor force $Fact$ could be written as (8) using Fig.3.

$$F_{act} = (F_{ref} - F_{RTOB})(K_P + \frac{K_I}{s} + K_D s) \quad (8)$$

The equations (2), (3) and (8) could be used to derive sprung mass deflection and unsprung mass deflection equations (9) and (10).

$$\begin{aligned} M_S \ddot{X_S} =& K_S(X_S - X_U) + C_S(\dot{X_S} - \dot{X_U}) \\ &+ (F_{ref} - F_{RTOB})(K_P + \frac{K_I}{s} + K_D s) \end{aligned} \quad (9)$$

$$\begin{aligned} M_U \ddot{X_U} =& - K_S(X_S - X_U) - C_S(\dot{X_S} - \dot{X_U}) \\ &- (F_{ref} - F_{RTOB})(K_P + \frac{K_I}{s} + K_D s) \\ &+ K_T(X_U - X_R) + C_T(\dot{X_U} - \dot{X_R}) \end{aligned} \quad (10)$$

In ideal situations the system can achieve zero sprung mass deflection by having a proportional controller with K_p value of one and F_{ref} value of zero. The ideal condition sprung mass Newtons equation (11) could derived using (9) as,

$$\begin{aligned} M_S \ddot{X_S} =& K_S(X_S - X_U) + C_S(\dot{X_S} - \dot{X_U}) \\ &+ (0 - K_S(X_S - X_U) - C_S(\dot{X_S} - \dot{X_U})) \end{aligned} \quad (11)$$

These modeled equation are used to simulate the quarter car model in different conditions. The controller F_{ref} could change dynamically to the change the motor deflection level to limit contraction of the system under actuator forces. In case of a positive F_{ref} value the system would add a force to the motor with the ability to increase the motor deflection, a negative F_{ref} value would add a force to the motor with the ability to decrease the motor deflection. Fig. 5 shows the F_{ref} control mechanism that uses the suspension deflection to direct the suspension deflection to its original length. The F_{ref} could be calculated for Fig. 5 controller as (12) where K_n is the dynamic force constant,

$$F_{ref} = K_n(X_S - X_U) \quad (12)$$

The F_{ref} value with the deflection control requires an addition encoder inside the suspension system to measure suspension system deflection. Zero F_{ref} value and an opposite force to the F_{dis} suppresses passive spring damper effects causes the system to suppress vibrations reducing considerable amount of the sprung mass vibrations.

TABLE II
MODEL SIMULATION PARAMETERS

Parameter	parameter value
M_S	300kg
M_U	60kg
K_S	16000 N/m
K_T	190000 N/m
C_S	1000 N/m^2
C_T	0 N/m^2
K_f	47 N/A
K_{fn}	47 N/A
M_m	1.1kg
M_{sn}	300kg
M_{mn}	1.1kg
K_P	0.99
K_I	0
K_D	0.0000002
G_{gis}	100

III. Simulation and Results

The active vibration suppression system is modeled and simulated in Matlab Simulink simulation environment considering a quarter car system shown in Fig. 1. The controller block diagram that is used in the simulation is shown in Fig. 4. Table II shows the simulation parameters that were used in the active vibration suppression model. A quarter car weights were selection for simulation to produce real world force levels.

The modeled suspension systems frequency response without the active controller is shown in Fig. 6. The active suspension system provides superior vibration suppression properties than in the passive suspension system frequency range from 0.07 Hz. The Fig. 7 shows the active suspension systems frequency response diagram providing evidence to the robust vibration suppression capabilities and high bandwidth of novel proposed controller. The suspension system provides vibration suppression of -5dB even at 0.1Hz. The vibration suppression capabilities of the system improves in higher frequencies.

The active vibration suppression systems pole zero diagram is shown in Fig. 8. There are nine poles and eight zeros in negative side of the imaginary axis of pole zero diagram. The system shows stability for given parameter values. The Fig.7 frequency response diagram and Fig.8 pole zero diagram diagrams shows stability and performance of the proposed vibration suppression system.

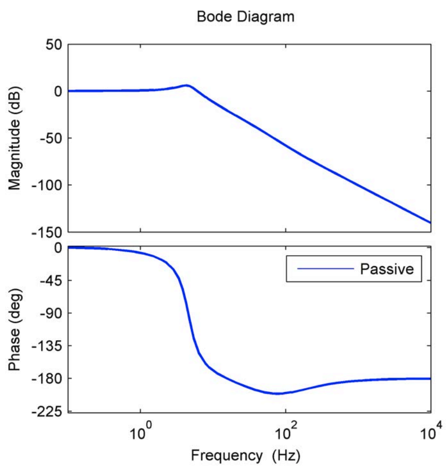

Fig. 6. Frequency response of passive suspension system

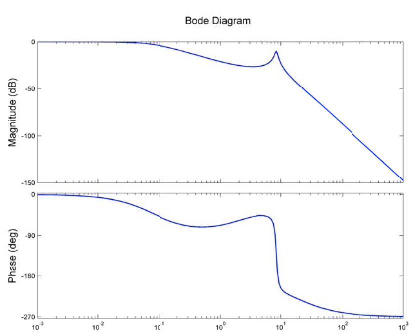

Fig. 7. Frequency response of active suspension system

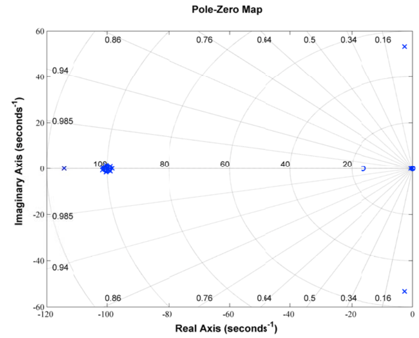

Fig. 8. Pole zero map of active suspension system

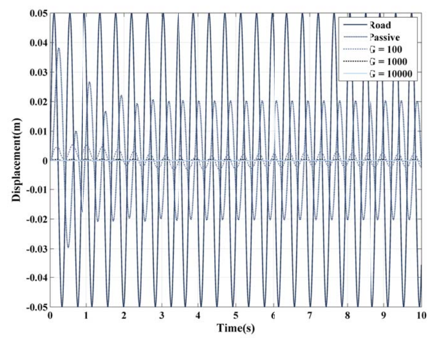

Fig. 9. Active suspension response for different filter gain values

The system RFOB filter gain value depends upon the sampling frequency of the controller. The active suspension system performance therefore varies with the system controller and sensor speed; and the suspension system performance for different gain values shown in Fig.9. The Fig.9 provide evidence that the system vibration suppression improves in higher gain values.

Fig.10 shows the step response of the active suspension system and passive suspension system. Active suspension system has minute vibrations caused by the sample filtration of

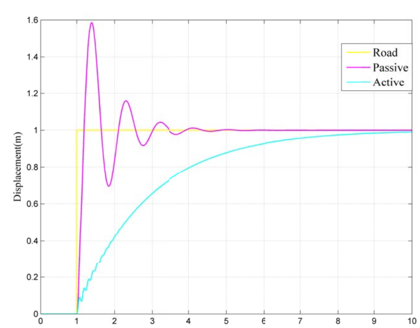

Fig. 10. Step input active suspension response

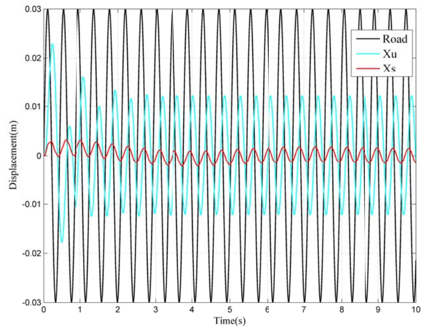

Fig. 11. Sinusoidal input active suspension response

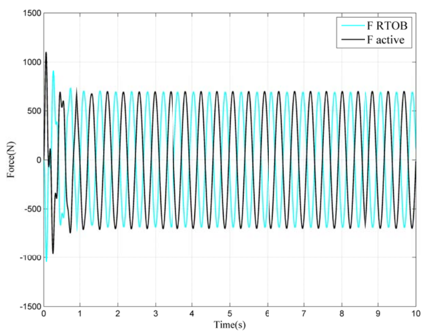

Fig. 12. Active suspension forces for sinusoidal vibration

RFOB, but provide superior system response for step inputs.

The system sprung mass vibration for sinusoidal road surface is shown Fig.11. Fig. 12 shows the active suspension motor force. The system provides good vibration suppression for sinusoidal vibrations and active suspension system motor force is around 800 N in this deflection level which is acceptable. The actuator saturation must be considered when implementing the system.

IV. Conclusion

The proposed active suspension system with internal motor sensing was simulated to provide evidence of vibration suppression capabilities and stability of this novel method. The controller should be designed with proper RFOB sampling rates to achieve desired vibration suppression levels and should consider the controller cost, design cost and the performance of the system implementation. A 100 ms delay is acceptable and prediction mechanisms can be used to improve system performance. The modeled active suspension system only require motor acceleration measurements as sensor inputs. The motor sensing method provides advantages in real world implantation because of the ability to provide vibration suppression with internal sensory data. The system can be implemented for miniature vibration suppressors without high number of sensor installation. The system performance levels and suspension deflection level can be controlled to provide vibration suppression for varied sprung masses using F_{ref} value. The simulated active suspension system with internal motor sensing shows the usability in practical systems.

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