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Research Article

Comparison of Quarter Car Model of Active Pneumatic Suspensions using Mass Flow Control for a Small Car

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Abstract

This paper deals with active pneumatic suspension with control strategy based on mass flow control for a commercially manufactured small car. Two different control strategies based on mass flow control are presented. In the first model mass flow rate to the air spring is a function of difference in the velocities of the base and in second model mass flow rate is a sinusoidal function of frequency of excitation from the base. The models are simulated using MATLAB software corresponding to identical suspension parameters. The results of simulation for both models are compared. It is concluded that the active suspension using velocity feedback as the control strategy has lower transmissibility and hence better performance at resonance compared to passive suspension.

Keywords: *Pneumatic suspension, active suspension, air spring, mass flow control, active control.*

1. Introduction

Automobile suspensions have function of improving passenger comfort by isolating vehicle from external disturbances and increase holding ability of the vehicle by providing adequate suspension deflections. Various types of suspension systems have been developed. Pneumatic suspension is one such suspension. One of the most prominent advantages of pneumatic suspension over metallic counterparts is the fact that the main natural frequency of the system can be made independent of sprung mass. Most of the commercially available passenger cars are equipped with passive spring and damper suspension systems. In recent years a few commercially available cars are fitted with active suspension systems.

The natural frequency of active suspensions is low resulting in improved ride, while still maintaining small static deflections. In addition active suspensions have low dynamic deflections under transient excitation and the suspension characteristics are maintained irrespective of loadings. Active suspensions provide greater stability. There is flexibility in the choice of desired dynamic characteristics, particularly between different modes of vehicle body oscillation, which is bounce, pitch and roll (Hall and Gill 1987).

Research in the area of pneumatic suspension has been going on for more than three decades. The complex dynamic stiffness of a damped air spring connected to a tank was obtained (Bacharch 1983). The effect of vibration frequency, vibration amplitude, and volume of auxiliary chamber in air springs with auxiliary chamber was studied (Toyofuku et al.1999). A model was proposed in dimensionless parameters to understand the effect of the parameters, which define the suspension and to select the spring type (Quaglia et al. 2001). A model of the pneumatic suspension which includes a nonlinear fluid dynamics model, for the suspension stiffness, damping factor and transmissibility was developed (Nieto et al. 2008). It is experimentally found that the sizes of the pipe, the tank and the air spring play an important role in determining the suspension's behavior. An airspring with auxiliary chamber connected through an orifice was modeled for calculation of dynamic stiffness of air spring (Zhu et al. 2008).

Despite the widespread use of pneumatic vibration isolation systems, the possibilities of using the pneumatic configuration as an active system have not been sufficiently studied. An active quarter car pneumatic active system where control is exercised by measurement of system states and use of the state information to drive the air pumps to control the pressure in air spring and consequently the spring forces produced was developed (Sharp, et al. 1986). A quarter car model of a pneumatic active car suspension to show the advantages of active control was developed. Control laws were derived using limited state feedback, linear stochastic optimal theory for a quarter car model (Sharp 1988). Active pneumatic spring constructed as flow chamber with changing volume and pressure which is obtained by relative displacement of piston was considered (Palej et al.1993). The relation between dynamic and static stiffness is obtained. Further, the airspring was treated as force actuator and the frequency response function obtained (Ballo, 2001). The

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properties of a quarter car model of an active suspension using air spring as a force actuator with those of a semi-active suspension were compared. The level of compensation of unwanted vibrations by the active vibration control system was compared to that from a semi-active vibration control system (Ballo, 2007). A nonlinear air-spring model based on experimental data considering the force-deflection relationship was developed (Xiao, et al. 2007).A sliding mode controller to control the actuation force of the air spring was designed. The quarter car model and half car model of vehicle suspension, using airspring as an active force actuator for commercial vehicle suspension was designed. The force from this actuator was regulated by controlling the pressure of air supplied to it through an electro pneumatic pressure regulator used to regulate the air to the chamber based on the voltage provided to it (Bhanadri et al. 2010). Using airspring as a force actuator a classic proportional-integral-derivative controller was compared with fuzzy logic controller experimentally (Kothandaraman, et al. 2011). A fuzzy adaptive sliding mode controller for an air spring active suspension system, considering the non-linearity, and pre-load dependence of spring force was designed (Bao, et al. 2012).Controller for an airspring active suspension with an auxiliary tank was developed (Turkkan et.al.2013).

From the very beginning developing controllers for active suspension has interested researchers. (Thompson, 1976), (Karnoop 1983). A vast majority of researchers working in the area of active vibration control in automobiles have focused on developing controllers for active suspension. They have focused on a different control algorithms and done comparative studies of algorithms. (Li et al. 2013)

The great majority of research on active has revolved around some type of adjustable dampers or force actuators that directly impact damping of the suspended mass This work focuses on the use of pneumatic spring alone in active suspension for a small car. The pneumatic spring supports the load due to sprung mass and isolates it from vibration from the road without acting as a force actuator. The control strategy proposed is novel as it is based on mass flow control.

Commercially available small cars are being considered. Since these cars are symmetric a quarter car model or single wheel approximation is feasible. The paper focuses on modeling the system including the mass flow equations and the equations of motion. The response of the sprung mass is modeled for two different cases of mass flow control. In one case the mass flow rate to the air spring is a function of difference in the velocity of base and sprung mass. In another case t mass flow rate to the air spring is a function of sinusoidal excitations from the road. In order to have maximum passenger comfort sprung mass position is maintained at its static equilibrium position.

2. Mathematical Model

2.1 Equation of Motion

A quarter car model of a suspension as shown in Fig.1. is considered where the sprung mass M has only bounce motion x. The system is to be a closed loop control system. Area of the air spring is assumed to change negligibly. The weight exerted by the sprung mass is supported by the air spring.

Hence, $Mg = p_0 A$ (1)

Where the pressure of air under static load is p_0 and A is the area of cross- section.

In dynamic condition the force due to acceleration of the sprung mass

$$M\ddot{x} + Mg = pA \quad (2)$$

$\ddot{x}$ is the acceleration of the sprung mass.

$$M\ddot{x} = \Delta pA \quad (3)$$

Δp is the small change in pressure in air spring due to the acceleration of the sprung mass.

Differentiate Eq. (19) with respect to time

$$M\dddot{x} - \dot{p}A = 0 \quad (4)$$

The mass flow equations are obtained using thermodynamic laws and equations of state. The mass equations are modeled in order to find the rate of pressure change inside the air spring which is treated as the control volume. It is assumed that the air behaves as an ideal gas in the spring. Process is assumed adiabatic. The air spring is assumed to have uniform area of cross-section. The effect of the properties of the airspring material is neglected.

The rate of change of pressure is obtained as

$$\dot{p} = \frac{dp}{dt} = \frac{\gamma}{V}\left[\dot{m}\left(\frac{pV}{m}\right) - p\dot{V}\right] \quad (5)$$

Where,

$\dot{m} = \frac{dm}{dt}$, $\dot{V} = \frac{dV}{dt}$,

m is the mass of air in the airspring.

p is the pressure inside the airspring,

V is volume of air spring,

R is ideal gas constant and

T is the absolute temperature in the airspring. Substituting the relation for rate of change of pressure $\dot{p}$ from Eq. (5) in Eq. (4) we get

$$M\dddot{x} - \frac{\gamma RTA}{V}\dot{m} + \frac{\gamma pA}{V}\dot{V} = 0 \quad (6)$$

Volume of the air spring is

$$V = (x - y)A$$

Where y is the displacement of the base of the air spring and it is assumed to be equal to the excitation from the road.

Therefore,

$$M\dddot{x} + \frac{\gamma pA^2}{V}\dot{x} = \frac{\gamma RTA}{V}\dot{m} + \frac{\gamma pA^2}{V}\dot{y} \quad (7)$$

2.2 Response of the Sprung Mass

The response of the system is obtained by solving Eq. (7). It is a third order linear differential equation. The complete solution of this equation corresponds to the displacement of the sprung mass x. Only the steady state part of response is being considered. The solution is obtained considering two different cases.

2.2.1 Mass flow rate as a function of velocity difference between excitation and sprung mass

The mass flow rate is defined as a function of difference between the velocity of base and sprung mass by

$$G_v = c_1(\dot{x} - \dot{y}) \quad (8)$$

This also acts as an excitation to the system.
The response of the system is defined by

$$y = Ye^{i\omega t} \quad (9)$$

The transmissibility is given by

$$TR = \frac{\gamma(pA^2 + RTc_1)}{\gamma(pA^2 + RTc_1) - MV\omega^2} \quad (10)$$

2.2.2 Mass flow rate as a sinusoidal function

We assume that a sinusoidal excitation is acting on the base of the air spring.

$$y = Y_0 sin\omega t \quad (11)$$

Y_0 is the amplitude of base excitation and ω is the frequency of excitation .

Air flow takes place into and out of the air spring. This acts as an excitation to the system. A phase lag φ is assumed between the base excitation and the mass flow. The phase lag is considered in order to compensate for the inherent time delay in valve controlling the flow. The mass flow rate is given by

$$\dot{m} = Gsin(\omega t - \varphi) \quad (12)$$

G is the amplitude of mass flow rate which is determined later. In order to obtain the steady state response of the sprung mass the two excitations are considered independently. The response corresponding to the excitation due to mass flow to the air spring is lagging the response due to the base excitation by phase angle $\emptyset$. The response to the excitations is superimposed. Therefore the steady state response of the sprung mass is given by

$$x_p = \frac{RTG}{pA}\left[\frac{2\omega^2 MV - \gamma pA^2}{\omega(\gamma pA^2 - \omega^2 MV)}\right]\cos(\omega t - \varphi) + Y_0\omega\left[\frac{\gamma pA^2}{\omega(\gamma pA^2 - \omega^2 MV)}\right]\sin \omega t \quad (13)$$

Transmissibility of the system can be obtained by obtaining the ratio of response of system from Eq. (13) to the excitation from the base.

3. Control Strategy

3.1 Mass flow rate as a function of velocity difference between excitation and sprung mass

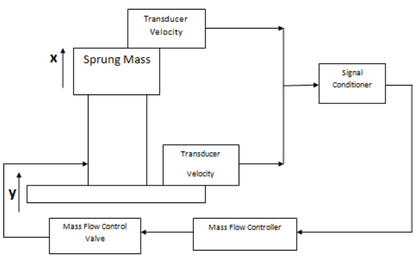

Fig. 1 Control scheme for Case 1

The difference in the velocity of the base and sprung mass is measured using suitable transducer. The signal is sent to a signal conditioner, which in turn sends the signal to a mass flow controller. The mass flow controller in turn controls the control valve so that air can be added or removed from inside the airspring so that zero transmissibility is ensured Fig.2.

The mass flow rate is obtained using Eq.12. The value of the flow constant c_1 is obtained by substituting the amplitude of response corresponding to the excitation transmitted to zero, as zero transmissibility is an ideal scenario, so that

$$c_1 = -\frac{mgA}{RT} \quad (14)$$

3.2 Mass flow rate as a sinusoidal function of frequency of excitation.

In order to achieve active control the amplitude of mass flow rate G and the phase lag $\emptyset$ are controlled. For perfect active vibration control the steady state response of the sprung mass should be zero. The amplitude of the mass flow rate is obtained by considering only the steady response of the system. By equating the response to zero in Eq. 13, we get

$$G = \frac{\gamma\omega Y_0 p_0^2 A^3 \sin \omega t}{RT(2MV_0\omega^2 - \gamma p_0 A)\cos(\omega t - \varphi)} \quad (15)$$

Since the two terms in steady state response vary as a function of $\cos(\omega t - \varphi)$ and $\sin \omega t$. Ideally the response of the sprung mass should be zero displacement. This can be achieved by making the excitation from the road and the mass flow out of phase. This is done by putting $\varphi = \frac{\pi}{2}$ in the active air suspension. Since the velocity vector leads displacement by phase of 90° introducing a negative feedback will ensure the phase difference. Hence Eq. (15) reduces to

$$G = \frac{\gamma(\omega Y_0)(p_0 A)p_0 A^2}{RT(2MV_0\omega^2 - \gamma p_0 A)} \quad (16)$$

In other words

$$G = \frac{\beta(velocity\ of\ base\ excitation)}{(2MV_0\omega^2 - \gamma(static\ load))} \quad (17)$$

Here,

$$\beta = \frac{\gamma(static\ load\ of\ sprungmass)p_0 A^2}{RT}$$

It is obvious that amplitude of mass flow rate is dependent on the excitation frequency. The mass flow rate is controlled in order to control the response of the sprung mass. It is proposed to sense the velocity of the unsprung mass which is assumed to have infinite stiffness. This signal after passing through signal conditioner will go to a mass flow controller. The mass flow controller will adjust the flow corresponding to the signal received so as to ensure zero displacement of the sprung mass as shown in Fig. 3.

The mass flow rate will be defined using Eq. (17).The excitation from the road acts as an excitation force on the base of the vehicle. Transducer can be used to sense the velocity and frequency of the base. The output of transducer after passing through signal conditioner is proposed to be given to a programmable logic controller which would in turn operate the mass control valve. Air is proposed to be drawn into or out of air spring so that the sprung mass remains in static equilibrium condition.

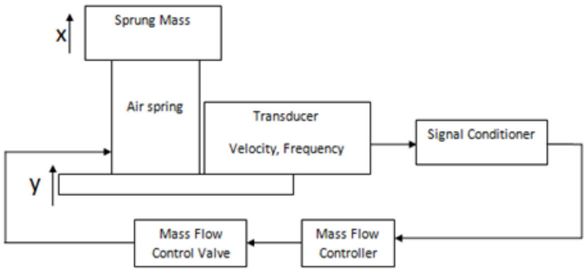

Fig. 1 Control scheme for Case 2

4. Simulation

If the assumptions made in the simulation model are appropriate the results of simulation are close to that of actual experimentation. In the present paper in order to select the suitable control system it was decided to first carry out simulation. The results of the simulation may give guidance to select hardware for the experimental setup, especially the flow control valves. The suspension parameters have been chosen corresponding to sprung mass of commercially available small cars. Simulation is carried out using actual the characters for Firestone 4001 air spring. The steady state response of the sprung mass for an active quarter small car is modeled using MATLAB 7.0.The suspension parameters used are given in the simulation of the response are as in Table1.

Table 1 - Suspension parameters used for simulation.

Sprung Mass	178kg
Amplitude of base Excitation	$5X10^{-3}$m
Spring Height	140mm
Pressure	$7*10^5$ N/ m^2
Area of Cross-section	$25*10^{-4}$ m^2
Volume of Air Spring	0.000325 m^3
Specific Gas Constant	286.9J/Kg k
Temperature	300K

4.1 Passive Suspension

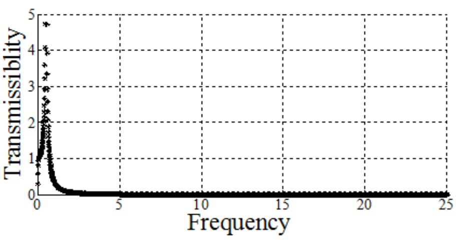

Fig. 2 Response of a passive air suspension

Initially the transmissibility of a passive quarter car suspension system is simulated for the suspension parameters given in Table 1 with excitation frequency ranging from 0 to 10 Hz. Here the mass flow rate is maintained at zero. The natural frequency of the system is obtained as 0.52 Hz. The result of simulation is shown in Fig. 4.

4.2 Active Suspension

4.2.1 Mass flow rate as a function of velocity difference between excitation and sprung mass

The response of the active suspension was simulated using the suspension parameters given in Table 1. The transmissibility of the system for system with mass flow rate as a function of velocity difference of base excitation and sprung mass was obtained using Eq.(10).The value of flow constant $c_1$was obtained using Eq.(14). The effect of change in frequency of excitation on transmissibility of the system was plotted as shown in Fig 5. It is observed that as the frequency of excitation increases the transmissibility decreases. Results of simulation indicate that at a value of frequency of excitation equal to 1Hz the transmissibility of both passive and active suspension with mass flow rate as a function of difference in velocity of base excitation and sprung mass, is identical. The results of simulation indicate that transmissibility is less than 1 even at resonance.

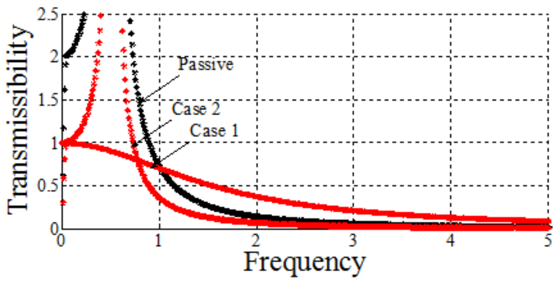

Fig. 3 Comparison of response of active suspension

4.2.2 Mass flow rate as a sinusoidal function of frequency of excitation.

The transmissibility of the active suspension with mass flow rate as a sinusoidal function of frequency of excitation is obtained using Eq.13. The suspension parameters given in Table 1 are used for simulation. The value of mass flow rate is obtained from Eq.15. The effect of resonance is evident as transmissibility is above 1 at resonance. Simulation results also indicate that the transmissibility is very high for a small band of frequency near resonance. However, the value of transmissibility for any value of excitation frequency is lower than that for a passive pneumatic suspension. The transmissibility of the system active suspension with mass flow rate as a function frequency of excitation in general is significantly lower than that of passive suspension with same suspension parameters.

5. Conclusion

It is concluded that the transmissibility of an active suspension is significantly lower at resonance for an active suspension using velocity feed back as demonstrated by active suspension with Mass flow rate as a function of velocity difference between excitation and sprung mass. The variation in transmissibility is very low in this case. The transmissibility is within acceptable limit for any value of excitation frequency. This type of active suspension using mass flow control is quite effective especially at resonance value of excitation. In the other active suspension model major shortcoming is that the mass flow rate is a function of frequency hence is ineffective at and near resonance.

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