

PNEUMATIC SUSPENSION SYSTEM WITH INDEPENDENT CONTROL OF DAMPING, STIFFNESS AND RIDE-HEIGHT

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ABSTRACT

In this paper, a novel design for a pneumatic automotive suspension system with independent control of stiffness, damping and ride-height is proposed. Stiffness of an air-spring can be changed by changing the volume of air in the spring. In certain existing adaptive suspensions, air springs are employed to achieve variable stiffness and variable ride height by pumping air into or out of the air springs. But this leads to coupling as the *DP: Amount of air* affects both *FR1: Control stiffness* and *FR2: Control ride-height*. Secondly, stiffness change involves pumping air into the air spring and thereby consumes power and is not instantaneous. In the proposed design, we connect the air spring volume to multiple auxiliary volumes through On-Off valves. Opening and closing of the valves, changes the effective volume of the air spring and thereby changes the stiffness of the air spring. By adequately choosing N unequal auxiliary volumes we can get 2^N stiffness settings. Ride-height control is achieved by using a compressor to pump in more air into the air-spring. The stiffness change is instantaneous, requires no power input, and involves no ride-height change, making this a decoupled design. Thermodynamic modeling of this air spring with auxiliary volumes is presented and it is shown that flow resistance of the valve introduces damping in the suspension system. As a result the suspension has low stiffness at low frequency and high stiffness at high frequency. Performance analysis on the linearized model of the air spring is used to determine the acceptable level of damping and the valve flow-coefficient that achieves it.

Keywords: Axiomatic Design, Automotive Suspension System, Vibration Isolation, Air Spring, Variable Stiffness

1 INTRODUCTION

The motivation for an automotive suspension system with independent control of stiffness, damping and ride-height comes from the trade-offs involved for the conflicting requirements of comfort and handling. The authors have previously proposed an electromechanical suspension system capable of independent control of stiffness, damping and ride-height [1] and discussed the application of this suspension to improved vehicle dynamics [2]. The application to improved vehicle dynamics requires the

stiffness change to be instantaneous and no change in ride-height during stiffness change (a decoupled design). In the previously proposed electromechanical suspension system [1], stiffness change requires power-input and is not instantaneous. In this paper, we propose a novel design for a pneumatic automotive suspension system capable of instantaneous stiffness change with no power input, and no ride-height change due to stiffness change.

Moreover this is robust, light-weight and low-cost. This makes the design applicable to generic vibration isolation systems with conflicting stiffness requirements. Ongoing work involves incorporation of this variable stiffness suspension system in a car.

2 PROPOSED DESIGN FOR VARIABLE STIFFNESS SUSPENSION

Assuming adiabatic compression and expansion of air, the nominal stiffness K of an air-spring is given by:

$$K = \gamma A^2 \frac{P_0}{V_1} \quad (1)$$

where P_0 , T_0 , and V_1 are the equilibrium values of pressure, temperature and volume respectively of air in the air-spring and $\gamma = C_p/C_v =$ Heat Capacity Ratio of air.

In certain existing adaptive suspensions[3], air springs are employed to achieve variable stiffness and variable ride-height by pumping in additional air into (or out of) the air springs. But this leads to coupling as the *DP: Amount of air* affects both *FR1: Control stiffness* and *FR2: Control ride-height*.

$$\begin{cases} \text{FR1: Control Ride Height} \\ \text{FR2: Control Stiffness} \end{cases} = \begin{bmatrix} X \\ X \end{bmatrix} \{ \text{DP: Amount of air } V_1 \} \quad (2)$$

Coupling due to insufficient number of DPs is a common mistake made by designers and Suh [4] proposed decoupling of such a coupled system by addition of new DPs to make the number of FRs and DPs equal (Theorem 2). We introduce additional DPs in this design through the following proposed modification.

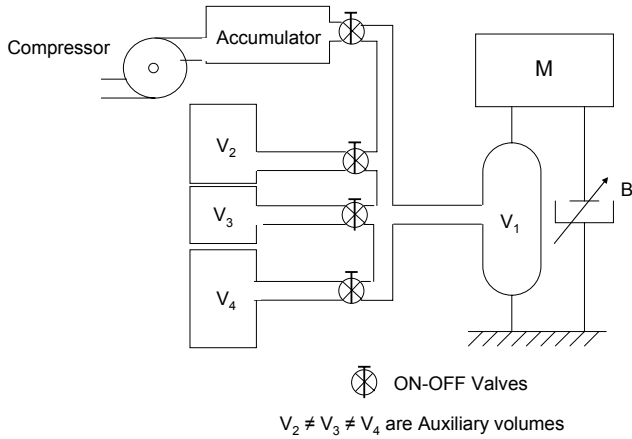


Figure 1 Proposed design with auxiliary volumes connected to air spring through On-Off Valves

In the proposed modification, we connect the air spring volume V_1 to additional (multiple) auxiliary volumes, shown by V_2 , V_3 and V_4 in the figure, through ON-OFF valves. When all three valves are closed, the effective volume is V_1 and the effective stiffness is maximum as given by equation 3.

$$K_{max} = \gamma A^2 \frac{P_0}{V_1} \quad (3)$$

When all three valves are open and all three auxiliary volumes are in communication with the air-spring volume, the effective volume is $(V_1+V_2+V_3+V_4)$ and the effective stiffness is minimum as given by equation 4.

$$K_{min} = \gamma A^2 \frac{P_0}{(V_1 + V_2 + V_3 + V_4)} \quad (4)$$

By adequately choosing non-equal volumes $V_2 \neq V_3 \neq V_4$, we can get $2^3=8$ stiffness settings between the minimum K_{min} and maximum K_{max} , by having different combinations of valves open and closed. In general, by adequately choosing N different volumes, $V_2 \neq V_3 \neq \dots \neq V_{N+1}$, we can get 2^N different stiffness settings. Note that all the volumes share the same pressure. As a result there is no change in pressure or ride-height, when the valves are opened or closed to adjust the stiffness. Secondly, opening and closing of valves is effortless and instantaneous.

Damping control is achieved through orifice control[5] or using magneto-rheological fluid[6] for damper that is in parallel with the air-spring arrangement. The decoupled design matrix is by equation 5.

$$\left\{ \begin{array}{l} \text{FR1: Control Ride Height} \\ \text{FR2: Control Stiffness} \\ \text{FR3: Control Damping} \end{array} \right\} = \left[\begin{array}{ccc} X & O & O \\ X & X & O \\ O & O & X \end{array} \right] \left\{ \begin{array}{l} \text{DP1: Compressor control } (V_1) \\ \text{DP2: Valve control } (\sum V_i) \\ \text{DP3: Orifice control} \end{array} \right\} \quad (5)$$

The set-up in Figure 1 with only one auxiliary volume connected to the main volume through an On-Off valve can be schematically represented as shown in Figure 2, where $B_2=\infty$ (for valve closed) results in stiffness $K=K_{max}$; and $B_2=0$ (for valve open) results in stiffness $K=K_{min}$. B denotes the adjustable damper in parallel with the air spring.

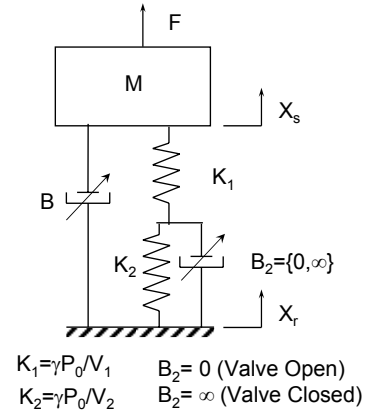


Figure 2: Schematic representation of On-Off valve-control to vary stiffness

Note that the model above assumes no pressure drop across the valve when the valve is open. This assumption may be reasonable at low frequencies, but we need a better model for behavior of this suspension system at higher frequencies.

3 THERMODYNAMIC MODELING TO ESTIMATE DAMPING DUE TO THE VALVE

With the valve closed, $B_2=\infty$ is an accurate assumption. But for the valve open, $B_2=0$ is not accurate. There is always restriction to the flow across the valve. Pressure drop in most valves is proportional to the square of the flow through the valve. This introduces damping in the system and in this section we determine the damping through thermodynamic modeling of the system.

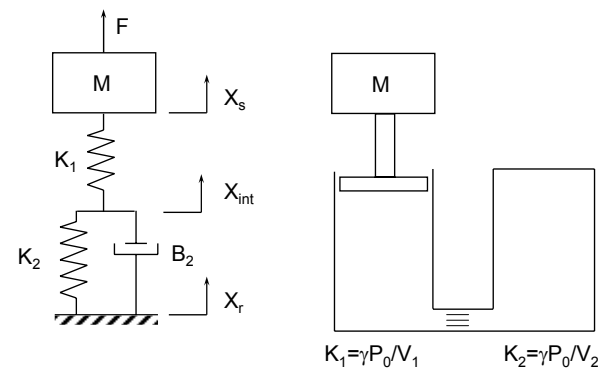


Figure 3 : Air spring with one auxiliary volume and flow resistance across the valve

The system is characterized by the (P_1, V_1, m_1, T_1) for CV1 (Control Volume 1), and (P_2, V_2, m_2, T_2) for CV2. We have made an assumption that all the gas in CV1 is in the same state, and all the gas in CV2 is in the same state. It is reasonable to assume that the properties are not a function of position within the chamber for modeling the air-spring, as the corner frequency for the suspension system is orders of magnitude lower than the frequency at which pressure waves propagate within the chamber to equalize the pressure at different points in the chamber. The

damping in the system is due the pressure drop (“effort”) across the valve which is proportional to the square of the flow through the valve. There is an additional source of irreversibility in the system, namely the mixing of air at the outlet of the valve as it flows from one chamber to another, which is also captured in the equations 12 and 16 below. There are 8 parameters and we need a relation of the form $\Delta P = f(\Delta V, \omega)$, i.e., pressure variation in response to a sinusoidal forcing of magnitude ΔV and frequency ω .

The following equations 6-17 describe the system

$$V_1 = V_0 + \Delta V \sin(\omega t) \quad (6)$$

$$V_2 = \text{constant} \quad (7)$$

Mass conservation

$$m_1 + m_2 = m \quad (8)$$

$$\dot{m}_1 + \dot{m}_2 = 0$$

Constitutive relation for gas in CV1 in algebraic and differential form

$$P_1 = \frac{m_1 R T_1}{V_1} \quad (9)$$

$$\frac{\dot{P}_1}{P_1} + \frac{\dot{V}_1}{V_1} = \frac{\dot{m}_1}{m_1} + \frac{\dot{T}_1}{T_1}$$

Constitutive relation for gas in CV2 in algebraic and differential form

$$P_2 = \frac{m_2 R T_2}{V_2} \quad (10)$$

$$\frac{\dot{P}_2}{P_2} = \frac{\dot{m}_2}{m_2} + \frac{\dot{T}_2}{T_2}$$

First Law for CV 1:

$$\frac{dE_{CV1}}{dt} = \dot{Q}_1 - \dot{W}_1 + \dot{m}_1 C_P T_{int1} \quad (11)$$

$$\dot{m}_1 C_V T_1 + m_1 C_V \dot{T}_1 = h_1 A_1 (T_a - T_1) - P_1 \dot{V}_1 + \dot{m}_1 C_P T_{int1}$$

The state of the gas leaving the CV is the same as the state of the gas in the CV ($T_{int1} = T_1$ for flow from CV1 to CV2). But the state of the gas entering the CV is different from the state of the gas in the CV. The nature of the gas entering the CV depends on the valve characteristics. If we assume that the valve is well insulated, we can assume the flow across the valve be an isenthalpic process. Hence we get:

$$T_{int1} = \begin{cases} T_1 & \text{for flow from CV1 to CV2} \\ T_2 & \text{for flow from CV2 to CV1} \end{cases} \quad (12)$$

Dividing equation 11 by $m_1 T_1$ and using equation 4, we get equation 13.

$$C_V \left(\frac{\dot{m}_1}{m_1} + \frac{\dot{T}_1}{T_1} \right) + R \frac{\dot{V}_1}{V_1} = C_P \left(\frac{T_{int1}}{T_1} \right) \frac{\dot{m}_1}{m_1} + \frac{h_1 A_1 (T_a - T_1)}{m_1 T_1} \quad (13)$$

$$C_V \frac{\dot{P}_1}{P_1} + C_P \frac{\dot{V}_1}{V_1} = C_P \frac{\dot{m}_1}{m_1} \left(\frac{T_{int1}}{T_1} \right) + \frac{h_1 A_1 (T_a - T_1)}{m_1 T_1}$$

First Law for CV2:

$$\frac{dE_{CV2}}{dt} = \dot{Q}_2 - \dot{W}_2 + \dot{m}_2 C_P T_{int2} \quad (14)$$

$$\dot{m}_2 C_V T_2 + m_2 C_V \dot{T}_2 = h_2 A_2 (T_a - T_2) + \dot{m}_2 C_P T_{int2}$$

Considering isenthalpic flow across the valve:

$$C_V \left(\frac{\dot{m}_2}{m_2} + \frac{\dot{T}_2}{T_2} \right) = C_P \left(\frac{T_{int2}}{T_2} \right) \frac{\dot{m}_2}{m_2} + \frac{h_2 A_2 (T_a - T_2)}{m_2 T_2} \quad (15)$$

$$C_V \frac{\dot{P}_2}{P_2} = C_P \frac{\dot{m}_2}{m_2} \left(\frac{T_{int2}}{T_2} \right) + \frac{h_2 A_2 (T_a - T_2)}{m_2 T_2}$$

where

$$T_{int2} = \begin{cases} T_1 & \text{for flow from CV1 to CV2} \\ T_2 & \text{for flow from CV2 to CV1} \end{cases} \quad (16)$$

The mass/volume flow rate is related to the pressure drop across the valve by the valve flow equation

$$\dot{m}_1 = -\dot{m}_2 = f(P_1 - P_2)$$

$$q = c_v \sqrt{\frac{P_1 - P_2}{SG}} \quad (17)$$

$$\dot{m}_1 = \rho q$$

Equations 6 to 17 present the complete set of equations that describe the behavior of this spring. This is a set of non-linear differential equations and it not possible to obtain an analytical solution for the purpose of determining the effect of the DPs on FRs. We simulated these non-linear equations in Matlab and the relations of the form $\Delta P_1 = f(\Delta V_1, \omega)$ are obtained as shown in Figure 4. From these graphs, we can obtain the magnitude and phase of $|\Delta P_1 / \Delta V_1|$ as a function of $|\Delta V_1|$ and ω .

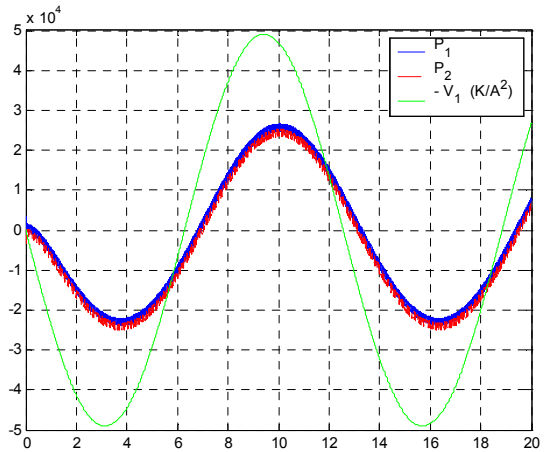


Figure 4: Results of the simulation (for $\omega=1$ rad/sec) showing pressure variation in the two chambers and the weighted volume variation.

The following observations can be made from the simulation results:

1. At low frequencies, the pressure variation ΔP_1 was in phase with the volume variation ($-\Delta V_1$) which means that the system behaves as a pure spring at low frequencies. Moreover the ratio $A^2 \Delta P_1 / \Delta V_1$ is equal to the minimum

stiffness $K_{\min} = \gamma A^2 P_0 / (V_1 + V_2)$. The pressure variation ΔP_2 is the same as ΔP_1 which means the two volumes are in communication.

2. In the mid-frequency range, the pressure variation ΔP_1 leads in phase with respect to the volume variation ($-\Delta V_1$). This means that the system exhibits dissipation in the mid-frequency range. Moreover the ratio $A^2 \Delta P_1 / \Delta V_1$ increases from the minimum stiffness $K_{\min} = \gamma A^2 P_0 / (V_1 + V_2)$ to the maximum stiffness $K_{\max} = \gamma A^2 P_0 / V_1$ as the frequency ω increases. The pressure variation ΔP_2 has smaller magnitude than ΔP_1 and lags in phase.
3. At high frequencies, again the pressure variation ΔP_1 is in phase with the volume variation ($-\Delta V_1$) which means that the system behaves as a pure spring at high frequencies. Moreover the ratio $A^2 \Delta P_1 / \Delta V_1$ is equal to the maximum stiffness $K_{\max} = \gamma A^2 P_0 / V_1$. The pressure variation ΔP_2 is very small in comparison with ΔP_1 which means that flow across the valve chokes for higher frequencies and the auxiliary volume becomes ineffective at higher frequencies, as manifested by high stiffness at higher frequencies.
4. The flow resistance is non-linear. As a result, the frequency range where the transition from low stiffness to high stiffness occurs depends on the magnitude of the ΔV in addition to the valve flow-coefficient c_v .
5. These trends can be captured in the following linearized model shown in Figure 6 which can be used for setting the valve DPs.

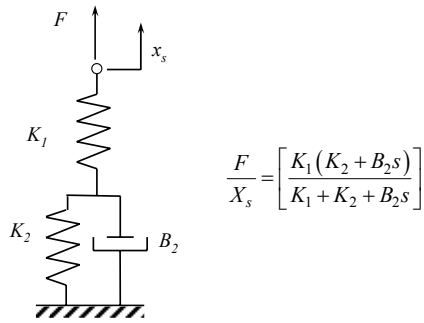


Figure 5: Linearized model of air spring connected to auxiliary volume through a valve

This qualitative behavior of the air spring with auxiliary volumes can be represented as lead compensator element, employing a spring in series with a parallel arrangement of spring and damper as shown in Figure 5. The complex stiffness of this element is given as

$$K_{comp} = \frac{K_1 (K_2 + B_2 s)}{K_1 + K_2 + B_2 s} \quad (18)$$

The Bode plot in Figure 6 shows that such element has low stiffness at low frequency and high stiffness at high frequency, with the two corner frequencies being K_2/B_2 and $(K_1+K_2)/B_2$. The corner frequencies can be moved by changing B_2 or the flow-resistance offered by the valve.

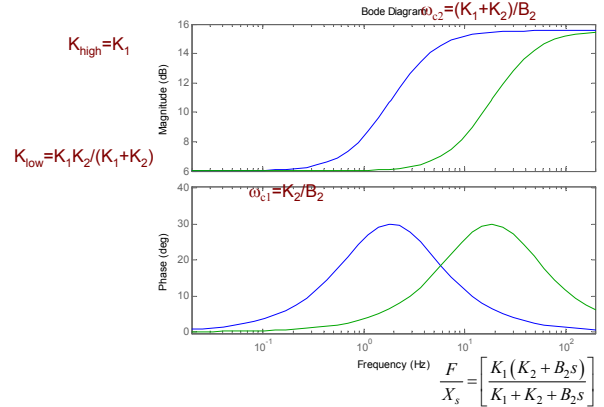


Figure 6: Bode plot showing the frequency response of the linearized model

4 DESIGN OF DAMPING IN LINEARIZED MODEL

The previous section discussed that air-spring with auxiliary volumes with the valve open has low stiffness K_{low} at low frequencies, and high stiffness K_{high} at high frequencies and the corner frequency depends on the damping coefficient related to the flow resistance of the valve. The original intention was to get a low stiffness at all frequencies as indicated by equation 5. Comfort in a suspension system is characterized by the road noise isolation, which is quantified by the transfer function X_s/X_r . In this section we show that the deviation from a soft suspension performance (in terms of comfort) is marginal provided that the corner frequency of the lead compensator is high enough (or in other words, the damping due the valve is low).

We use the linearized model of the air-spring with an auxiliary volume and evaluate the deviation of its performance from the performance of a soft and stiff suspension as shown in Figure 8(a) and (b).

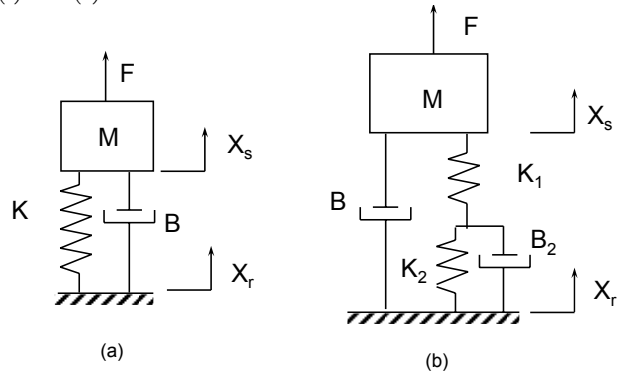


Figure 7: Comparison of comfort metric (road-noise isolation) for the two systems

Figures 8-11 plots the Bode plots for the transfer function X_s/X_r for conventional soft suspension (given by equation 19), conventional stiff suspension (with same ζ given by equation 20) and the air-spring with auxiliary volume for different values of B_2 (given by equation 24).

$$\frac{x_s}{x_r} = \frac{B_{low}s + K_{low}}{Ms^2 + B_{low}s + K_{low}} \quad (19)$$

$$\frac{x_s}{x_r} = \frac{B_{high}s + K_{high}}{Ms^2 + B_{high}s + K_{high}} \quad (20)$$

The asymptotic behavior of these transfer functions at high frequency are given by

$$\left| \frac{x_s}{x_r} \right| \rightarrow \frac{B_{low}}{Ms} = \frac{2\zeta\sqrt{K_{low}M}}{Ms} = \frac{2\zeta\omega_{n1}}{s} \quad (21)$$

$$\left| \frac{x_s}{x_r} \right| \rightarrow \frac{B_{high}}{Ms} = \frac{2\zeta\sqrt{K_{high}M}}{Ms} = \frac{2\zeta\omega_{n2}}{s} \quad (22)$$

The asymptotic behavior explicitly depends on B and M, but since $B = 2\zeta\sqrt{KM}$ and ζ has to close to 0.4 to avoid excessive magnification at resonance, the asymptotic behavior does depend on K (although not explicitly).

For the air spring with auxiliary volumes, the transfer function is given by:

$$\frac{x_s}{x_r} = \frac{B_{low}s + K_{comp}}{Ms^2 + B_{low}s + K_{comp}} \quad (23)$$

where the complex stiffness K_{comp} is given by equation 18. The transfer function then becomes:

$$\frac{x_s}{x_r} = \frac{B_{low}s(K_1 + K_2 + B_2s) + K_1(K_2 + B_2s)}{(Ms^2 + B_{low}s)(K_1 + K_2 + B_2s) + K_1(K_2 + B_2s)} \quad (24)$$

Figure 8 shows that if the flow-resistance due to the valve is too high ($K_2/B_2=0.1\omega_n$), then the auxiliary volume does not reduce the stiffness in the frequency region of interest. As the damping reduces ($K_2/B_2=\omega_n$), we still observe a deterioration in the comfort. For $K_2/B_2=10\omega_n$, we see that the peak is actually reduced as the lead compensator introduces damping near the resonant frequency, but not anywhere else. For $K_2/B_2=100\omega_n$, the two transfer functions are indistinguishable.

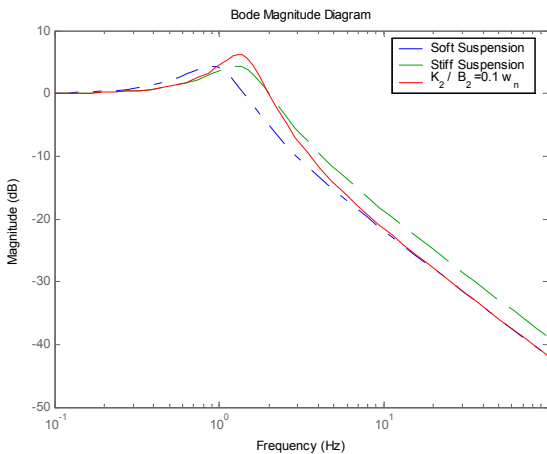


Figure 8 Road noise isolation for high damping
 $K_2/B_2 = 0.1\omega_n$

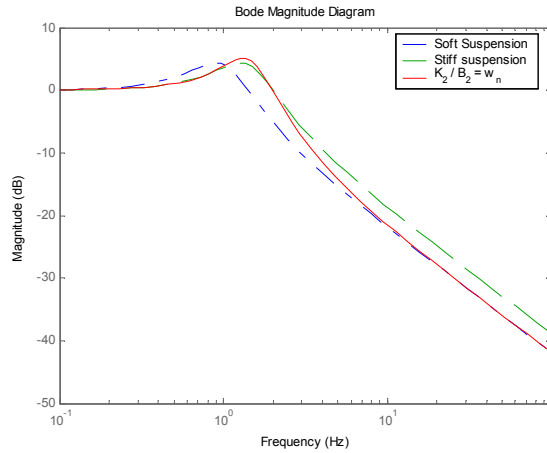


Figure 9 Road noise isolation for moderate damping
 $K_2/B_2 = \omega_n$

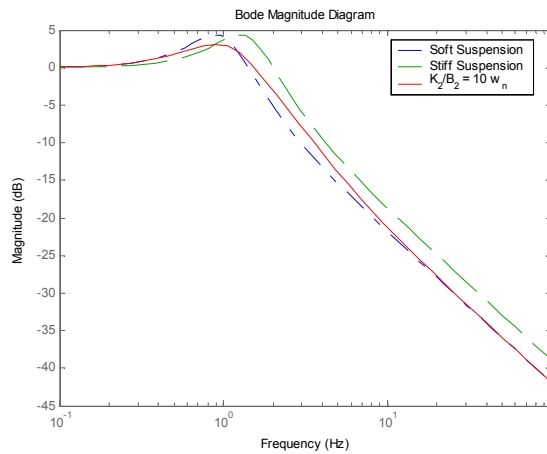


Figure 10 Road noise isolation for low damping
 $K_2/B_2 = 10\omega_n$

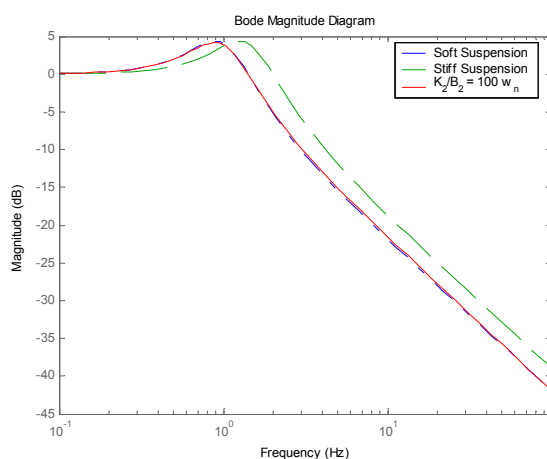


Figure 11 Road noise isolation for very low damping
 $K_2/B_2 = 100\omega_n$

From the plots, we can see that the air spring with auxiliary volume has the same asymptotic behavior irrespective of the

valve flow resistance. This is because we can use low damping B_{low} with the system.

If the condition $K_2/B_2 \gg \omega_n$ is satisfied, there is no deterioration in performance due to the valve resistance.

5 APPLICATIONS

Application of variable stiffness isolator to automotive suspension systems has been extensively discussed in an earlier publication [2]. The proposed vibration isolator is capable of instantaneous stiffness change with no power input and no ride-height change due to stiffness change. Moreover this design is robust, light-weight and low-cost and hence can be used in generic vibration isolation stages with conflicting stiffness requirements or time-varying stiffness requirements. An example of conflicting stiffness requirements is a precision motion stage for semiconductor metrology in which a stiff isolation system is required to isolate from the inertial forces during the rapid motion of the stage and a soft isolation system is desired for isolation from fab ground vibrations during the optical measurements. An example of time-varying stiffness requirements could be for isolation in prosthetic limbs, where the required stiffness changes with gait, walking style and speed, terrain and other factors. Currently we are incorporating this air suspension into a real car.

6 CONCLUSIONS

A novel design for a pneumatic automotive suspension system with independent control of stiffness, damping and ride-height is proposed. In this design, we connect the air spring volume to multiple auxiliary volumes through On-Off valves. Opening and closing of the valves, changes the effective volume of the air spring and thereby changes the stiffness of the air spring. By adequately choosing N unequal auxiliary volumes we can get 2^N stiffness settings. The stiffness change is instantaneous, requires no power input, and involves no ride-height change, making this a decoupled design. Some applications of this design are discussed. Thermodynamic modeling of the air-spring with the auxiliary volumes is presented to develop guidelines for adequate valve-sizing.

7 LIST OF ABBREVIATIONS

P_0 = Nominal Pressure of air in the air-spring
 V_0 = Nominal Volume of the air-spring
 T_0 = Nominal Temperature of the air-spring
 C_p = Specific Heat Capacity of air at constant pressure
 C_v = Specific Heat Capacity of air at constant volume
 c_v = Valve flow coefficient
 ρ = Density of air
 q = Flow-rate of air through valve
 γ = Heat Capacity Ratio of air.
CV1, CV2 = Control Volume 1, Control Volume 2
 P_1 = Pressure of air in CV1
 V_1 = Volume of CV1
 m_1 = Mass of air in CV1
 T_1 = Temperature of air in CV1
 P_2 = Pressure of air in CV2

V_2 = Volume of CV2
 m_2 = Mass of air in CV2
 T_2 = Temperature of air in CV2
 K_1 = Stiffness of chamber 1
 K_2 = Stiffness of chamber 2
 B_2 = Linearized damping coefficient due to the valve
 B = Damping Coefficient for the SDOF car model
 ω_{c1} = Lower corner frequency for the lead compensator
 ω_{c2} = Upper corner frequency for the lead compensator
 ω_n = Natural frequency of the soft suspension

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