International Journal of Engineering Research & Technology (IJERT)
ISSN: 2278-0181
Vol. 4 Issue 09, September-2015

Performance of Automotive Air Suspension Control System

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***Abstract*—One type of vehicle suspension is the pneumatic suspension. This paper is an analytical investigation of the air spring stiffness variation under different operating conditions of input frequencies and amplitudes. Usually changing the stiffness of the air spring involves variations of the enclosed air pressure by pumping air into or out of air chamber or by changing its volume. Since, changing spring stiffness through controlling its pressure consumes power and is not instantaneous; hence controlling the stiffness through volume control is adapted in this investigation. This has been achieved by connecting the air spring volume to multiple auxiliary volumes through On-Off valves. By choosing two unequal additional volumes, four different stiffness setting are examined.**

Keywords—Pneumatic Active Suspension; Air Spring; Vehicle Ride

I. INTRODUCTION

There are conflicting requirements between vehicle ride and handling behaviors. In order to achieve good vibration isolation for sprung mass over a wide range of frequency, a soft suspension spring is required, while to maintain good road holding capability at a frequency near natural frequency of the unsprung mass, a stiff suspension spring is required. To reduce the amplitude of the sprung mass at a frequency near its natural frequency, a high damping ratio is required, while in the high frequency range, a low damping ratio is required to provide good vibration isolation. On the other hand, a high damping ratio is required to provide good road holding capability at high frequency range. These conflicting requirements cannot be met by the conventional suspension system as its characteristics are fixed [1].

The motivation of an automotive suspension system with variable stiffness and damping ratio comes involved for the conflicting requirements of comfort and handling. This paper will be concerned with studying the performance of air suspension and the effect of changing the volume on the spring stiffness due to its advantages of near ideal instantaneous response and no power input.

II. THEORETICAL BACKGROUND

Malin Presthus [2] proposed a new model for simulation of the air spring behavior of railway train. The model is a three- dimension and consists of two parts, describing vertical and horizontal behavior. The air spring model is implemented in the vehicle dynamic simulation program GENSYS. The results were compared with experimental data, finding a good agreement.

A.J. Nieto et al [3] developed an adaptive pneumatic suspension based on excitation frequency. A control strategy is proposed to avoid undesirable resonant frequencies, the control procedure is based on the pre-knowledge of incoming vibration and an efficient prediction technique is used when the incoming frequency is unknown.

Igor Ballo [4] proposed an experimental estimation of air spring characteristics in active vibration control system, and then the results were compared with theoretical considerations estimation.

Deo and Suh [5] have proposed an electromechanical suspension system capable of independent control of stiffness, damping and ride height to improve vehicle dynamics, but they find that the stiffness change requires power input and is not instantaneous. Then, they proposed a novel design for pneumatic air suspension system [6] capable of instantaneous change with no power input and no ride height change, this is done by changing the air spring volume through connecting auxiliary volumes to the air spring with On/Off valves. By adequately choosing N unequal volumes, they obtained 2N stiffness setting.

III. MATHEMATICAL MODEL

To determine the stiffness of the air spring, this is defined as the force, F acting along the centre line of the spring per unit deflection, Z in the same direction. The force F is due to the effective area, A multiplied by the gauge pressure, pg inside the air spring which is given by the equation.

$$F = p_g A \quad (1)$$

The spring stiffness can be calculated from the equation.

$$k = \frac{dF}{dZ} = \frac{\partial(p_g A)}{dZ} = A\frac{\partial p_g}{\partial Z} + p_g\frac{\partial A}{\partial Z} \quad (2)$$

Neglecting the change in the effective area, equation (2) will be:

$$k = A\frac{\partial p_g}{\partial Z} \quad (3)$$

To calculate the change of pressure inside the spring, adiabatic process is considered.

$$\text{i.e. } (pV^{\gamma} = \text{constant}).$$

$$\frac{\partial}{\partial z}\left(p_g V^{\gamma}\right) = p_g \gamma V^{\gamma-1}\frac{\partial V}{\partial z} + V^{\gamma}\frac{\partial p_g}{\partial z} = 0 \qquad (4)$$

Since the cross section area is considered constant, then the rate of change of volume per unit deflection is the effective area but with negative sign. This is due to the fact that decreasing the volume increases the deflection and vice versa.

$$\frac{\partial V}{dz} = -A \qquad (5)$$

Substituting equation (5) in equation (4) resulting in,

$$\frac{\partial p_g}{dz} = \frac{p_g \gamma A}{V} \qquad (6)$$

Substituting equation (6) into equation (3),

$$k = \frac{\gamma p_g A^2}{V} \qquad (7)$$

In the proposed design, the air spring stiffness will be calculated using Matlab Simulink program. The stiffness value is controlled by connecting two unequal auxiliary volumes V_2 and V_3 to the main air volume V_1 in successions through ON/OFF valves, as shown in *Fig. 1*. When the two valves are closed, the effective volume will be V_1 (minimum volume), so the effective stiffness will be maximum as given by equation (8).

$$K_{max} = \frac{\gamma p_o A^2}{V_1} \qquad (8)$$

Where p_o is the nominal pressure of the air spring.

For minimum stiffness, the two valves should be open for maximum effective volume. The effective stiffness in this case is given by.

$$K_{min} = \frac{\gamma p_o A^2}{V_1+V_2+V_3+V_4} \qquad (9)$$

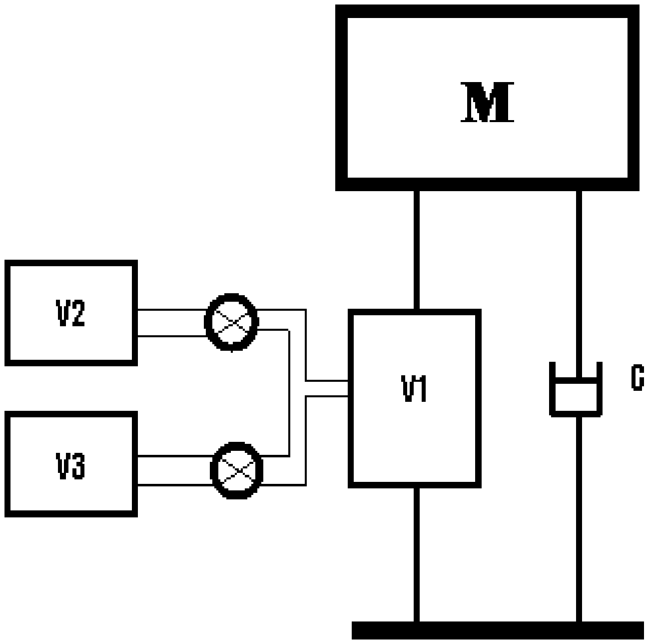

Fig. 1. The proposed suspension schematic diagram with two additional volumes connected to the air spring.

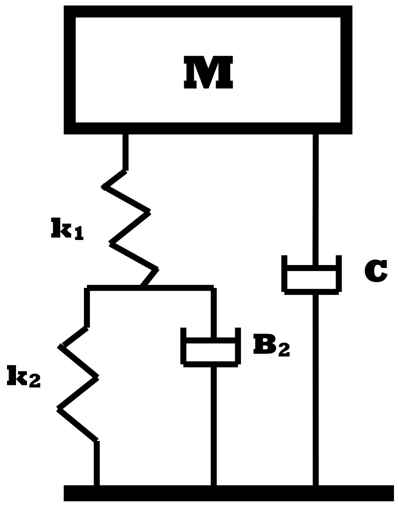

Fig. 2. Air spring model connected to the additional volumes

Assuming linear damping model, the behavior of air spring with additional volumes can be represented as spring k1 in series with parallel arrangement of spring and damper as shown in Fig. 2.

Where k1 is the stiffness of the air spring volume and k2 is the stiffness of the auxiliary volumes. The equations of stiffness k_1 and k_2 are given by

$$k_1 = \frac{\gamma p_o A^2}{V_1} \qquad (10)$$

$$k_2 = \frac{\gamma p_o A^2}{V_2+V_3} \qquad (11)$$

The complex stiffness k_{comp} of this model is given by equation (12) [6]

$$k_{comp} = \frac{k_1(k_2+B_2 s)}{k_1+k_2+B_2 s} \qquad (12)$$

Where B_2 is the orifice damping coefficient

To determine B_2, K_2/B_2 should be much greater than the system natural frequency to avoid deterioration in the performance [6].

These equations are simulated in Matlab and Fig. 3 was obtained

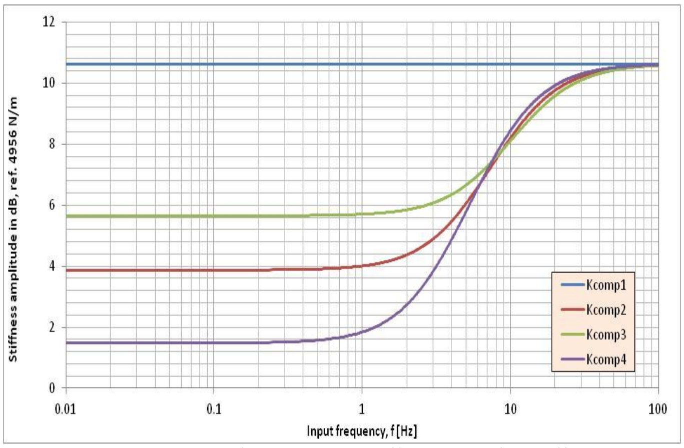

Fig. 3. Variations of the air spring stiffness at the four different volume settings with the input frequency

Fig. 3 shows the results for the four stiffness settings k_{comp1}, k_{comp2}, k_{comp3} and k_{comp4} calculated in dB with reference to 4956 N/m (minimum stiffness). The four stiffness are calculated when the volumes V_1, V_1+V_2, V_1+V_3 and $V_1+V_2+V_3$ are selected, respectively. These options are selected through the ON/OFF valves shown in *Fig. 1*.

From Fig. 3, it can be found that the system has low stiffness at low frequencies and high stiffness at high frequencies.

Further, it is obvious that the stiffness decreases as volume increases at low frequency. But at high frequencies, the change in stiffness is negligible as the air doesn't have time to go to the additional volume.

Comfort provided by the suspension system is characterized by its road induced vibration isolation. To get the transfer function to determine road vibration isolation, the model in *Fig. 2* will be simplified into *Fig. 4.*

From Fig. 4, the equation of motion is obtained as.

$$M\ddot{z_1} + C(\dot{z_1} - \dot{z_2}) + k_{comp}(z_1 - z_2) = 0$$

Assume general solution:

$$z_1 = Z_1 e^{j\omega t},\ z_2 = Z_2 e^{j\omega t}$$

$$(-M\omega^2 + k_{comp} + j\omega C)Z_1 = (k_{comp} + j\omega C)Z_2$$

$$\frac{Z_1}{Z_2} = \frac{k_{comp} + j\omega C}{-M\omega^2 + k_{comp} + j\omega C} \tag{13}$$

Equation (13) is the transfer function which describes road vibration isolation, where

Z_1: Vertical amplitude of the sprung mass

Z_2: Vertical amplitude of the road excitation

M: Sprung mass

C: Suspension damping coefficient

ω: Angular frequency

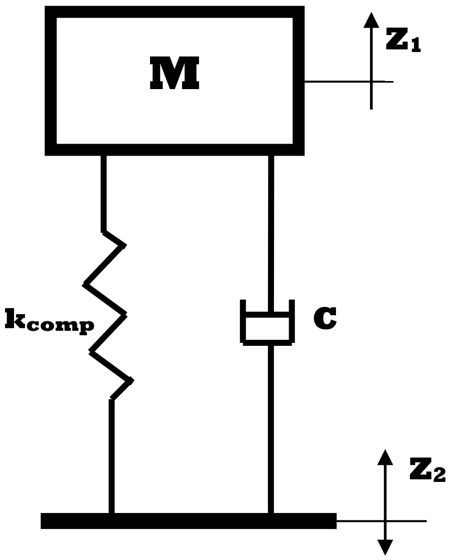

Fig. 4. Simplified air spring model

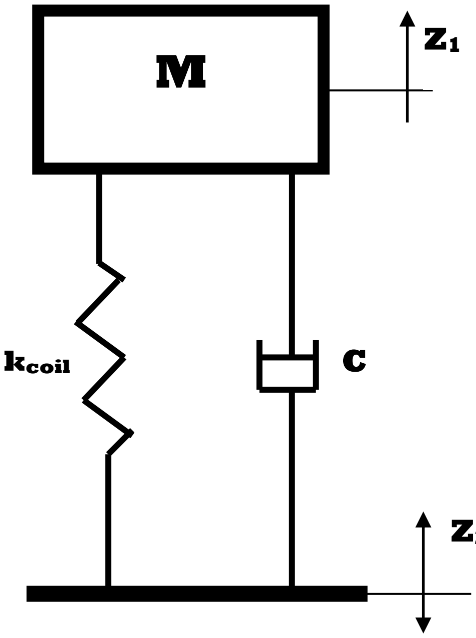

Fig. 5. Conventional coil spring suspension model

To know the enhancement in comfort behavior in the pneumatic suspension system, the road induced vibration isolation for the conventional suspension should be determined.

Similarly, the transfer function that describes the road vibration isolation for conventional coil spring suspension as shown in *Fig. 5* is

$$\frac{Z_1}{Z_2} = \frac{k_{coil} + j\omega C}{-M\omega^2 + k_{coil} + j\omega C} \tag{14}$$

Where: k_{coil} is the coil spring stiffness

To determine the suspension damping coefficient C, the damping ratio ζ has to be in the range 0.2→ 0.4 to avoid excessive magnification at resonance.

From equations (13, 14 and 15), *Fig. 6 and Fig. 7.* have been plotted.

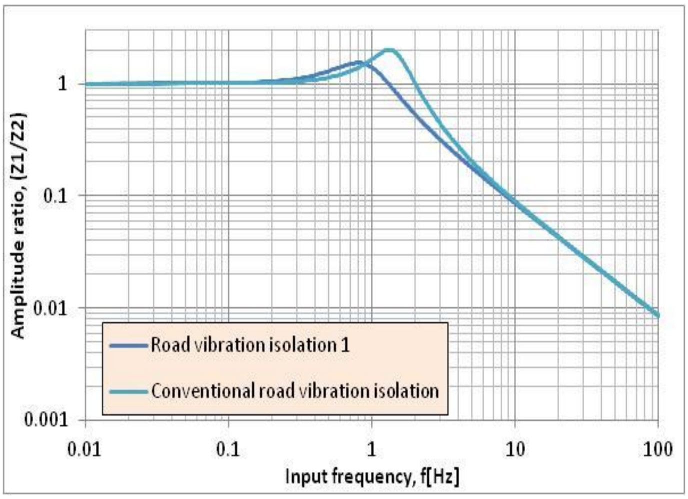

Fig.6. Comparison of road vibration isolation between a single volume air spring and conventional coil spring with sinusoidal input frequency

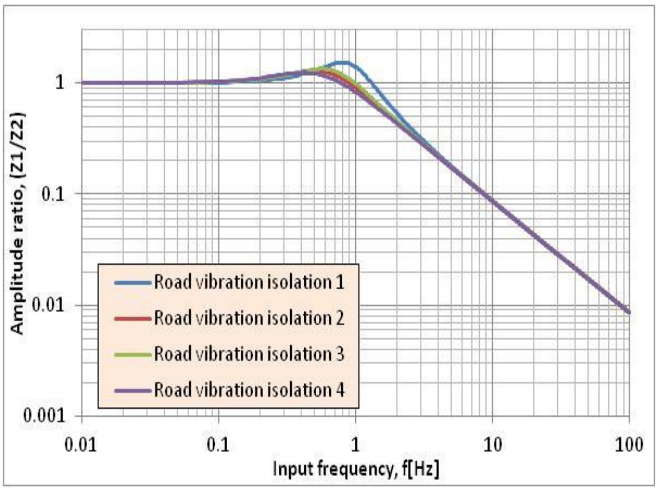

Fig. 7. Variations of road vibration isolation at four different volume settings with sinusoidal input frequency

Fig. 6 shows the road vibration isolation for the air spring without any additional volumes compared with the road vibration isolation for conventional coil spring suspension for the same damping coefficient. It was found that the natural frequency reduces by 0.52 Hz and the amplitude ratio reduces by 0.467.

The four settings in *Fig. 7* are calculated when the volumes V_1, V_1+V_2, V_1+V_3 and $V_1+V_2+V_3$ are selected, respectively. It was found that the natural frequency for the condition 1 of air spring vibration isolation is 0.81 Hz and its amplitude ratio is about 1.527, when connecting the air spring to the auxiliary volume V_3. The natural frequency of condition 3 becomes 0.57 Hz and its amplitude ratio reduces to about 1.339. When V_1 and V_2 are selected, the natural frequency for condition 2 is 0.49 Hz and its amplitude ratio becomes 1.289. Finally, when V_1, V_2 and V_3 are selected, the natural frequency for condition 4 is 0.41 Hz and its amplitude ratio becomes 1.232.

IV. CONCLUSIONS

- Replacing the coil spring in the conventional suspension by air spring reduces the system natural frequency by 0.92 and its amplitude ratio by 0.762.
- Adding auxiliary volumes to the main air spring reduces the system natural frequency and its amplitude ratio.
- Adjusting the air suspension spring stiffness can be achieved practically almost instantaneously by utilizing solenoid ON/OFF valves through intelligent logic microprocessor to provide good ride and handling behavior.

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APPENDIX

Vehicle parameters used in Matlab model

1. Adiabatic index, $\gamma = 1.4$
2. Quarter car mass, $M = 500$ kg
3. Effective area, $A = 0.01227\ m^2$
4. Air spring volume, $V_1 = 5$ liters
5. First air reservoir volume,
   $V_2 = 5$ liters
6. Second air reservoir volume,
   $V_3 = 7$ liters
7. Nominal air spring pressure,

$$p_o = \frac{M * 9.81}{A} \frac{N}{m^2}$$

8. Damping ratio, $\zeta = 0.3$
9. Coil spring stiffness, $k_{coil} = 40000$ N/m