DYNAMIC BEHAVIOUR OF AN AIR SPRING ELEMENT

ДИНАМИЧНОТО ПОВЕДЕНИЕ НА ВЪЗДУШНОПРУЖИНЕН ПНЕВМАТИЧЕН ДЕМФЕРЕН ЕЛЕМЕНТ

Assis. prof. dr. Gavriloski V.[1],MEng. Jovanova J.[2]
Faculty of Mechanical Engineering – Skopje, Macedonia [1,2]

Abstract: *Air springs are well-known for their low transmissibility coefficients and their ability to vary load capacities easily by changing only the gas pressure within the springs. Air springs can be used for a mechatronic approach in suspension design because of their ability to provide a controlled variable spring rate and they offer simple and inexpensive automatic levelling. Air spring dynamic model with frequency dependent characteristics has been developed for the purpose of this research. The mathematical model enables application of the model in simulation without many experimentally obtained parameters. Frequency dependence of the stiffness characteristic implemented in the new model is the main difference between the classical models and the new dynamic model. The verification of the dynamic air spring model is done by an experiment. The experimental results and results obtained by simulation in Matlab/Simulink are compared.*

Keywords: AIR SPRING, DYNAMIC MODEL, VEHICLE SUSPENSION SYSTEM, MECHATRONICS

1. Introduction

The major purpose of any vehicle suspension system is to isolate the body from road unevenness disturbances and to maintain the contact between road and the wheel. Therefore, the suspension system is responsible for the ride quality and driving stability. The design of a classical passive suspension system is a compromise between this conflict demands. However, the improvement in vertical vehicle dynamics is possible by developing an air spring suspension system.

The air spring is mainly used in commercial vehicles, but lately is also used in higher classes of passenger vehicles. The main advantages of the air spring suspension system to the classical one are:

- Simple stiffness decreasing for obtaining soft suspension for increased comfort and decreased transfer of shocks;
- Constant natural frequencies of the system for the normal loading rank;
- Constant suspension space between the sprung and unsprung mass independent of load;
- Regulation opportunity of the stiffness and achieving an adaptive stiffness coefficient to the conditions.

Despite the fact, that the air springs for passenger cars are commercially available, there are not enough researches devoted on their dynamic characteristics. Quaglia and Sorly [3] discuss the vehicular air suspensions from design aspects, but not from control viewpoint. The research in this area is mostly for commercial vehicles. In [2], detailed overview of the constructive characteristics and the theoretical assumptions for the processes in the air springs is given. There are also some results from experimental analyses.

PresthusIn [3] develops few dynamic air spring models for rail vehicles. In his paper nonlinear mathematical air spring model is developed and the results are compared with the experimental results. But this paper can not be used for air springs in passenger vehicles.

Therefore, a new dynamic model of an air spring was developed. The new dynamic model with frequency dependent characteristics was verified by experiment.

2. Modelling of air spring

The air spring system, figure 1, consists of an air balloon (primary volume) connected to a reservoir (additional volume) by a pipeline system. Since the stiffness of the air spring depends on the total volume, with an electromagnetic valve the additional volume can be included or excluded from the system, and the stiffness can be changed. When the system is disposed to vibrations, the gas gets compressed or expanded, and the pressure becomes equal in the primary and the additional volume. Considering the dimensions and the construction of the pipeline there is a phase difference between the pressures in the two volumes.

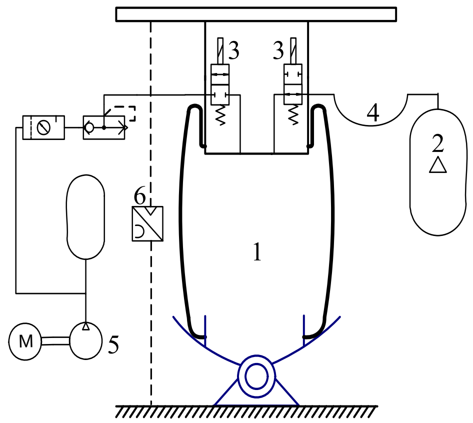

fig.1 Air spring system

1.Air balloon; 2.Additional volume; 3.Electromagnetic valve; 4.Pipeline; 5.Compressor; 6.Levelling sensor

Modelling of an air spring is based on the basics of thermodynamics and fluid dynamics. Although the process itself is quite complex, in the literature the air springs are usually presented with simplified equivalent mechanical models.

The modelling of an air spring, presented here, does not take in consideration the levelling system because these changes are very slow. Here the mathematical models incorporate the stiffness and the damping characteristics of air spring.

Under the vibrations, the behaviour of the compressed air within the air spring system is polytrophic. The minimal stiffness is when there is an isothermal change of the gas state (for frequencies f<0,1Hz), and maximal stiffness is with adiabatic state change (for frequencies f>3 Hz).

Analyses of the vehicle vertical dynamics show special interest around the frequency domain from 0 to 20 Hz. Classical dynamic models, as well as the manufacturer’s technical data is for very low frequencies from 0 to 1 Hz. But the experimental analyses show nonlinear frequency dependence of the mechanical characteristics of the air spring. The change of the stiffness of the air spring is present when there is an additional volume and it depends from the size of the balloon, the volume of the additional reservoir and the length and the diameter of the pipeline connecting the two volumes.

The difference between the classical and new dynamic model is presented on the following figure 2. From the figure can be concluded that the classical model is only valid for low frequencies.

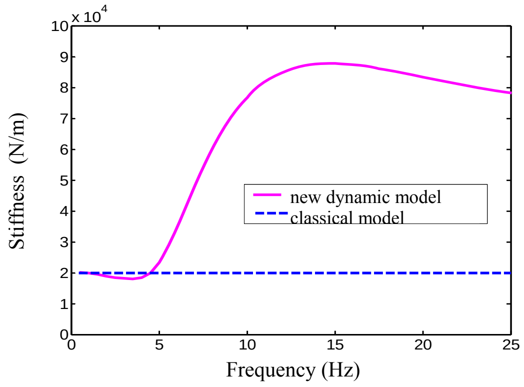

fig.2 Comparison of classical and new dynamic model

2.1. Classical model of an air spring

The absolute pressure in the air spring and the force coming from the elastic element are:

(1) $p_0 = p_B + p_{at}$

(2) $F_z = (p_0 - p_{at})A_{ef} = p_B A_{ef}$

where p_0 is the absolute pressure in the air spring, p_{at} is the pressure of the atmosphere, p_B is the measured pressure in the air spring, A_{ef} is the effective area and F_z is the vertical force.

The stiffness characteristic of the pneumatic element can be determined form the equations above:

(3) $$c_z = \frac{dF_z}{dz} = p_B \frac{dA_{ef}}{dz} + \frac{dp_B}{dz} A_{ef} = p_B \frac{dA_{ef}}{dz} + \frac{dp_0}{dz} A_{ef}$$

If the gas condition change is determined that it is polytrophic, the following equation is valid:

(4) $pV^n = const$

where n is the polytrophic coefficient.

The equation (4) is differentiated:

(5) $$\frac{d}{dz}\left(p_0 V^n\right) = p_0\, n\, V^{n-1} \frac{dV}{dz} + \frac{dp_0}{dz} V^n = 0$$

From the equations above, follows that:

(6) $$c_z = \frac{p_0\, n\, A_{ef}^2}{V} + p_B \frac{dA_{ef}}{dz} = c_{z1} + c_{z2}$$

The equivalent mechanical model according the classical approach is presented at figure 3, consisting of 2 springs with stiffness c_{z1} and c_{z2}.

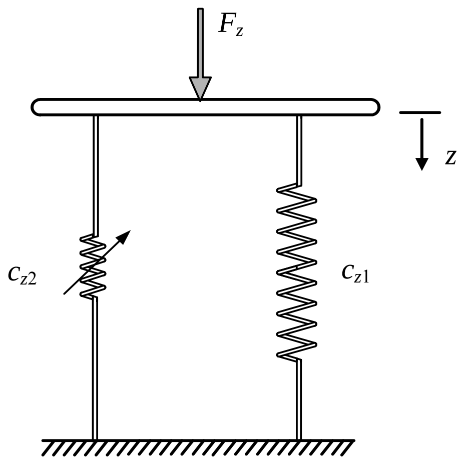

fig.3. Classical model

2.2. New dynamic model of an air spring

The derivation of a new mathematical model of an air spring is done with a physical model of a simplified air spring. The model consists of two gas volumes connected with a pipeline. In order to take in consideration the change in gas state in the two volumes, a mechanical barrier has been introduced in the pipeline as a fictive piston. The mechanical barrier is considered to be with neglected mass, and to the barrier is added equivalent fluid mass that is oscillating along the pipeline. The barrier displacement causes pressure change in the two volumes. The gas state change is polytrophic. Considerable assumptions for small changes for the gas state and appropriate linearization are applied for the simplified air spring model.

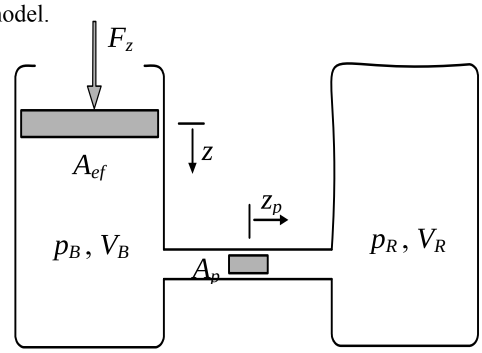

Fig.4. Air spring physical model

From the force balance that acts on the mechanical barrier from the cylinder and from the motion equation of the fiction piston, come out the following equations:

$$F_z = A_{ef}\, p_B - A_{ef}\, p_{at}$$

(7) $$m\ddot{z}_p = (\Delta p_B - \Delta p_R) A_p - b_{pp} A_p \dot{z}_p^2$$

where: p_{at} is the outside ambient pressure, b_{pp} is the coefficient for the pressure fall from the flow resistance in the pipeline, and the pressure fall is taken in consideration with quadratic change. The b_{pp} reduced to the surface of the barrier A_p gives the damping coefficient $b_p = b_{pp} A_p$.

By rearranging the expressions the result is following equations:

$$F_z = (p_0 - p_{at}) A_{ef} + \frac{p_0\, n\, A_{ef}^2}{V_{B0}} z - \frac{p_0\, n\, A_{ef}\, A_p}{V_{B0}} z_p$$

(8) $$m\ddot{z}_p = \frac{p_0\, n\, A_{ef}\, A_p}{V_{B0}} z - \frac{p_0\, n\, A_p^2}{V_{B0}} z_p - \frac{p_0\, n\, A_p^2}{V_{R0}} z_p - b_p \dot{z}_p^2$$

The equivalent mechanical model that is implemented is shown on figure 5.

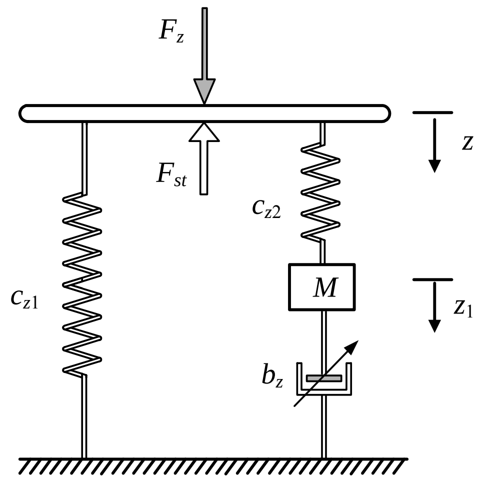

fig.5. New dynamic model

To fit the equivalent model, equations (8) need to be scaled for the piston displacement from the pipelines by scaling factor k_1:

$$\begin{cases} k_1 \dfrac{A_p}{A_{ef}} \left(\dfrac{V_{B0} + V_{R0}}{V_{R0}} \right) = 1 \\ \dfrac{k_2 \; p_0 \, n \, A_{ef} \; A_p}{V_{B0}} = \dfrac{k_1 \; p_0 \, n \, A_{ef} \; A_p}{V_{B0}} \end{cases} \quad (9)$$

The solution of this linear system of equations is:

$$\begin{cases} k_1 = \dfrac{A_{ef}}{A_p} \dfrac{V_{R0}}{(V_{B0} + V_{R0})} \\ k_2 = k_1 \end{cases} \quad (10)$$

By replacing for the constants k_1 and k_2:

$$\begin{aligned} F_z &= F_{st} + z \; c_{z1} + (z - z_1) c_{z2} \\ M \; \ddot{z}_1 &= (z - z_1) c_{z2} - b_z \; \dot{z}_1^{\;2} \end{aligned} \quad (11)$$

where:

$$F_{st} = (p_0 - p_{at}) A_{ef}$$

$$c_{z1} = \frac{p_0 \, n \, A_{ef}^{\;2}}{V_{B0} + V_{R0}}$$

$$c_{z2} = \frac{p_0 \, n \, A_{ef}^{\;2}}{V_{B0} + V_{R0}} \frac{V_{R0}}{V_{B0}}$$

$$b_z = b_p \left(\frac{A_{ef}}{A_p} \frac{V_{R0}}{(V_{B0} + V_{R0})} \right)^3$$

$$M = m \left(\frac{A_{ef}}{A_p} \frac{V_{R0}}{(V_{B0} + V_{R0})} \right)^2$$

In this dynamical model, the change of the effective area is neglected, because for the air spring the experiments were made for, this change is very small. But for certain types of air springs, this change cannot be neglected. In the new complete dynamic model a nonlinear spring was implemented with stiffness c_{z3}:

$$c_{z3} = p_B \frac{dA_{ef}}{dz} \quad (12)$$

In the new full dynamic model, friction force, experimentally determined, could also be implemented to enable more precise definition of the curve "force-displacement". The force is considered to grow slowly till it reaches maximum value following the expression:

$$\begin{aligned} F_{fr} &= \frac{z}{|z_s|} F_{fr\max} \\ z_s &= z \big|_{\dot{z}=0} \end{aligned} \quad (13)$$

The value z_s, changes every time the direction of the displacement changes, $\dot{z} = 0$.

The new dynamic model of an air spring with frequency dependent characteristics is shown on figure 6. The new full dynamic model consists of: two linear springs c_{z1} and c_{z2} that represent the stiffness of the spring; a non linear spring c_{z3} that describe change of area due to deflection; a mass M, a nonlinear viscous damper b_z and a friction damper F_{fr} which describe the inertia of the air in the pipe between air bag and auxiliary volumes.

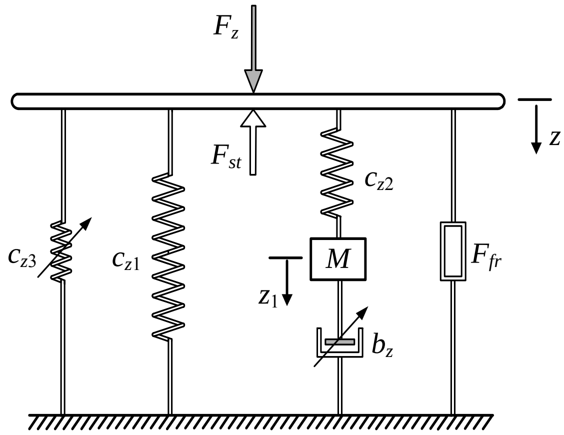

fig.6. New full dynamic model

3. Experimental verification

The verification of the dynamic air spring model is done by an experiment. Experimental results and results obtained by simulation in Matlab/Simulink are compared, and a graphical representation of the results is given in Figure 7 and Figure 8.

Experimental results and simulation results in Matlab/Simulink are obtained for two different additional volumes and for two different pressures:

- V_{R0}=0,00094 m^3, p_B=302 kPa (labelled as V1p1),
- V_{R0}=0,00094 m^3, p_B=508 kPa (labelled as V1p2),
- V_{R0}=0,0074 m^3 , p_B=297 kPa (labelled as V2p1),
- V_{R0}=0,0074 m^3 , p_B=508 kPa (labelled as V2p2).

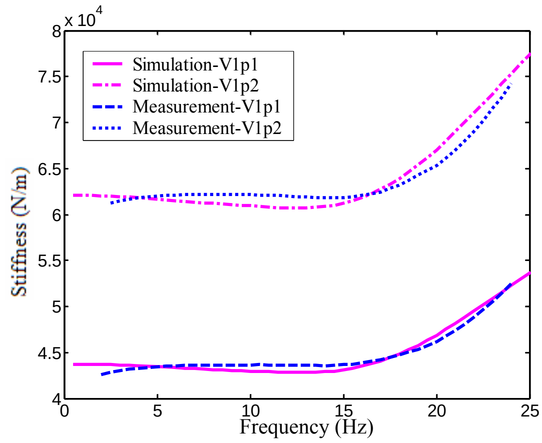

fig.7. Frequency dependent stiffness characteristics for additional volume 1

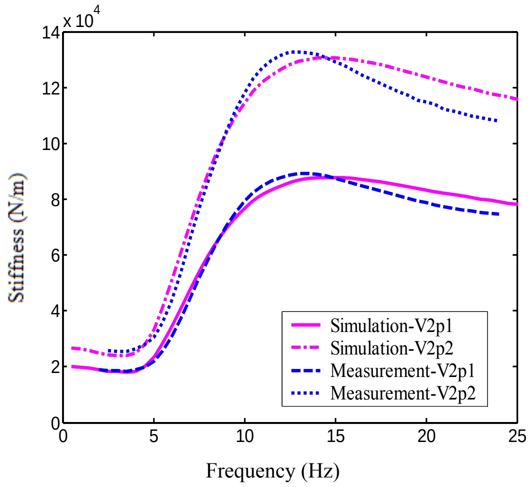

fig.8. Frequency dependent stiffness characteristics for additional volume 2

The diagrams show that the simulation results match the experimental results. This verifies the new dynamic model for air spring suspension system.

4. Conclusions

Air springs are well-known for their low transmissibility coefficients and their ability to vary load capacities easily by changing only the gas pressure within the springs.

This paper has outlined improvements of the classical model with a new dynamic model of an air spring with frequency dependent characteristics.

It is shown that connecting an additional volume to the air spring gives two values of the stiffness property and the design parameters of the surge pipe that connects two volumes influence the frequency dependence of the stiffness properties. The stiffness frequency dependence could enable design of an air spring with lower stiffness for lower frequencies and higher stiffness for higher frequencies, which will improve road holding and riding comfort at the frequencies near the sprung mass natural frequency. The proposed air spring with additional volumes has two main benefits: possibility for vehicle level control and possibility for suspension stiffness control. With the design of surge pipe that connect the volumes a possibility is given to tune the frequency range where the additional volume is operating.

5. References

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