Testing and calculation of vehicle adaptive suspension air springs

D Yurlin, S Bakhmutov, V Kulagin, R Kuleshov
FSUE "NAMI", Moscow, Russian Federation

E-mail: dmitry.yurlin@nami.ru

Abstract. The paper is dedicated to describing the methods of measuring the characteristics of suspension air springs (elastic pneumatic elements) on a test bench, as well as to the methods of evaluating their effect on vehicle parameters. The results of the measurement of air spring characteristics on the test bench under various conditions are presented together with the analysis of changes in their properties. A method for setting a universal force (load bearing) characteristic of the air spring, as well as a method for simulating it in the vehicle are suggested. The results of the vehicle simulation with characteristics of the air springs of different stiffness are shown, a numerical interval of the possible change in the vehicle behaviour parameters in case of change of suspension stiffness in the process of driving is specified.

1. Current trends in the development of suspension air springs
Current leaders of the automotive industry switch to adjustable suspension springs, whereby the adjustment is performed not only by the road clearance but also by their stiffness. An example is the suspension stiffness adjustment system implemented by Porsche in such models as the Panamera and Cayenne. The new versions of these vehicles are equipped with the system of air springs with three chambers (volumes) switching between which allows achieving 3 different stiffnesses in one air spring unit. The description of the Porsche system is given in figure 1.

![Figure 1. Design of multi-chamber (polyvolume) suspension air spring by Porsche.](image)
Within the projects which are being implemented by FSUE "NAMI", there is a need of analysis of capability and feasibility of such systems and of the advantage of their application. As a result, works on experimental, analytical and computational analysis of the air suspension systems were carried out.
2. Measurement of the air spring stiffness on the test bench
In order to check the feasibility of considerable change in the air spring stiffness by reducing its internal volume, an experiment was carried out where the air spring stiffness was measured with various internal volumes, and also the effect of the internal system pressure and compression velocity (rate) on the stiffness were investigated. The external view of the test object and of the experimental test bench installation is shown in figure 2. The installation control system is presented in figure 3.

The test setup included special washers the removal of which leads to the increase of the internal volume and reduction of the stiffness. The test bench is equipped with a hydraulic installation for compressing the air spring with various frequencies, as well as with a sensor measuring the force generated by the sleeve-type element in the axial direction.

![Figure 2. Experimental installation view.](image)

![Figure 3. Experimental installation control system.](image)

The graphs of dependence between the axial force and the piston displacement are shown in figure 4. The record is made for 3 variants of frequencies: 0.1 Hz, 1 Hz and 2.5 Hz, as well as for different internal volumes. To simplify the visualization, the results are shown only for 2 volumes of 3.68 l and
2.93 l. The resulting stiffness for different frequencies, as well as the dynamic stiffness coefficient are shown in Table 1. The measurement was carried out under the pressure providing the force of 10500 N in the design position.

Figure 4. Record of force on sleeve-type element with different volumes and frequencies

| V_intern | V_washer | Number of washers | Stiffness at different frequencies, N/mm | Coeff_dyn_stiffness |
|----------|----------|-------------------|----------------------------------------|---------------------|
| 1        | 1        | -                 | 0 Hz                                   | 1 Hz                |
| 3.68     | 0        | 0                 | 29.3                                   | 37.8                |
| 2.93     | 0.75     | 10                | 39.2                                   | 50.4                |

Table 1. Stiffness of sleeve-type element with different volumes.

Figure 5 shows the dependence between the stiffness of the air spring and the internal volume and vibration frequency. Analysis of the results shows that increase of vibration frequency above 1 Hz does not result in additional increase of stiffness. With the amplitude used, the frequency of 1 Hz corresponds to the relative velocity of 0.22 m/s. The coefficient of dynamic stiffness (ratio of dynamic stiffness to quasistatic one) is 1.28 on average.

Figure 5. Effect of variable volume on the air spring stiffness.
To simplify the tests, the measured internal pressure in the initial (design) position is replaced by the "installation (mounting) force" - the axial force, developed by the air spring with the initial length at the moment of installation on the bench. Figure 6 provides the recorded force data for different installation force (internal pressure) and two variants of frequencies: 0.1 Hz and 1 Hz.

Based on the data from Figure 6, the data according to Table 2 can be obtained. The stiffness of the air spring increases along with the increase of the installation force.

Table 2. Measurement results for different installation force.

| Pressure, MPa | Force, N | Frequency, Hz | Stiffness, N/mm |
|--------------|---------|--------------|-----------------|
| 9.68         | 11550   | 0 Hz         | 39.43           |
| 9.69         | 11550   | 1 Hz         | 51.07           |
| 8.81         | 10500   | 0 Hz         | 38.16           |
| 8.84         | 10500   | 1 Hz         | 48.95           |
| 7.98         | 9450    | 0 Hz         | 35.57           |
| 7.94         | 9450    | 1 Hz         | 45.99           |
| 7.01         | 8400    | 0 Hz         | 34.30           |
| 7.02         | 8400    | 1 Hz         | 43.34           |

Figure 7 provides measurement results with different installation forces. The stiffness increases along with the internal pressure increase, the dynamic stiffness is higher than the quasistatic one by 28% on average.
3. Universal description of the air spring force

Based on the performed measurements, a universal description of the sleeve-type element force depending on the internal volume, pressure and piston velocity may be suggested. A polynomial function obtained when measuring the air spring characteristics and accepted as a basic one (\(F_{baz}(h)\)) may be used for that. The design position of the air spring is considered to be its axial length when installed in a vehicle and when the vehicle is motionless. The design force is considered to be the value of the load borne by the air spring under these conditions. Accordingly, on the test bench, the design position of the air spring is reproduced and the initial load considered as the design force is generated. In order to simplify the comparison of the functions obtained with different parameters, all the graphs are brought to the zero initial value – the measurement is carried out with different design forces which are then subtracted from the obtained values. The dependence is built based on the piston displacement, similar to the measurements on the test bench. In order to take into account the effect of the volume, pressure and velocity, special coefficients are introduced which also depend on the piston displacement, as the effect of these parameters changes depending on the sleeve-type element compression ratio (degree) (\(K_{v}(h), K_{f}(h), K_{velos}(h,velos)\)). The coefficients are built by way of identifying correlations between the basic characteristic, auxiliary characteristic and current properties of the system. Based on the test results, different characteristics which can be divided into "basic" and "auxiliary" are obtained for individual parameters. Knowing the value of the defining parameter change which led to the transition of the basic characteristic to the auxiliary one, and knowing also the parameter value for the "new" – "current" – state of the system, the value of the coefficient, by which the basic characteristic values should be multiplied in order to obtain its current condition (Figure 8), may be defined.

**Figure 7.** Effect of internal pressure on the air spring stiffness.

**Figure 8.** Characteristics for different states of the system.

Upon that, the progression of the characteristic growth for different piston positions may vary, which should be considered in case of rearrangement. The \(K_{vsp}\) auxiliary coefficient which is defined
for each position of the piston within the analyzed range (corresponding to the possible piston displacement in real systems) is used for this. The auxiliary coefficient takes on values less than one in accordance with the ratio (correlation) of the values of the investigated characteristics (Figure 9).

![Figure 9. Auxiliary coefficient characteristic.](image)

To carry the basic and auxiliary characteristics to the current state of the system, the Kosn main coefficient is used which corresponds to the ratio of the differences of characteristic values in the design position – the basic characteristic and the current one to the basic and auxiliary ones – Equation (3).

The basic characteristic was chosen from the test results, with the parameters which were used in every test (during the tests with different volume, different pressure and velocity). The basic characteristic corresponds to the volume of 3 litre (9 auxiliary washers), the force of 10500 N and the velocity close to 0 m/s. The auxiliary characteristic corresponds to the test results with the system properties considerably differing from the basic one in terms of the investigated parameter. The characteristic for 10500 N, 0 m/s and 3.68 l is used for the volume coefficient. The characteristic of 8300 N, 0 m/s and 3 l is used for the force (pressure) coefficient. 10500 N, 0.22 m/s (1 Hz) and 3 l are used for the velocity coefficient.

The equation of each of the 3 coefficients is made by an identical pattern:

\[
K = 1 +/− Kvsp * Kosn
\]  

(1)

\[
Kvsp = \frac{Ybaz(z) − Yref(z)}{Ybaz(z)}
\]

(2)

\[
Kosn = \frac{Ybaz(z) − Ytec(z)}{Ybaz − Yref}
\]

(3)

According to the provided analytics and formulas (1), (2) and (3), the coefficients of pressure (force), volume and velocity will be as follows.

Volume coefficient:

\[
Kv = 1 − KvspV * KosnV
\]

(4)

\[
KvspV = \frac{FbazV(z) − FrefV(z)}{FbazV(z)}
\]

(5)

\[
KosnV = \frac{Vbaz − Vtec}{Vbaz − Vref}
\]

(6)

Force (pressure) coefficient:

\[
Kf = 1 − KvspF * KosnF
\]

(7)
\[ KVspV = \frac{Fbaz(z) - Fref(z)}{Fbaz(z)} \]  
(8)

\[ KosnF = \frac{Fbaz - Ftec}{Fbaz - Fref} \]  
(9)

Velocity coefficient:

\[ KVels = 1 + KVspVel * KosnVel(Velos) \]  
(10)

\[ KVspVel = \frac{FbazVel(z) - Fref(z)}{FbazVel(z)} \]  
(11)

\[ KosnVel = \begin{cases} 
VelosTec & \text{if } Velos \leq 0,22 \\
VelRef & \text{if } Velos > 0,22 
\end{cases} \]  
(12)

Thus, the equation of force of the sleeve-type element will correspond to the function of the piston displacement (13).

\[ Ffin = Ftec + Fbaz(h) * Kf(z) * KV(z) * KVels(z, velos) \]  
(13)

According to the previously presented system parameters, the volume coefficient took on the values according to Figure 10, the force coefficient – according to Figure 11 and the velocity coefficient – according to Figure 12. At the same time, the main coefficient took on the value of 0.71 for the volume (Vtec = 3.48 l) and 0.3 for the force (Ftec = 9790 N). For the velocity, it amounted to 1 (as the quasistatic situation was considered). According to Equation 13, a curve characteristic of the air spring was built (Figure 13). Figure 13 A shows the curve according to the specified parameters, Figure 13B provides the curve at the velocity above 0.22 m/s.

![Figure 10. Resulting value of the volume coefficient.](image-url)
Figure 11. Resulting value of the force coefficient.

Figure 12. Resulting value of the velocity coefficient.
Figure 13. Resulting curve of the air spring. A – for quasistatic state, B – for dynamic state.

To check the accuracy of the calculation performed, the data obtained by calculation shall be compared with the data obtained during tests for the same system properties. Figures 14 A and B provide the comparison of simulation (computed) and experimental data for the installation force of 9450 N, volume of 3 l and two variants of velocity – quasistatic and 1 m/s.
It is evident from the tolerance graphs analysis that the data deviation for the displacement of \( \pm 50 \) mm does not exceed 5%. With bigger displacements, the tolerance increases along with the value of the force itself, however, no high calculation accuracy is required within these ranges due to no need to assess the ride comfort. It is the progressivity value that is estimated, which corresponds to the required level. In general, it is obvious from the graphs and diagrams that the simulation (computed) and experimental characteristics almost match, which indicates high calculation accuracy.
4. Description of the prototype design of multi-chamber (polyvolume) air spring system

To test the system in a real vehicle, a prototype system with two gas chambers (volumes) in each air spring is used. Its characteristics and calculated stiffness are given in Table 3.

| Axle       | Front            | Rear             |
|------------|------------------|------------------|
| Static force on the air spring F=9790 N | F=8080 N         |
| Volume     | 3.48             | 3.32             |
| Stiffness 0 Hz | 31               | 31               |
| Stiffness 1 Hz | 41               | 42               |

The prototype air spring view is given in Figure 15. The curve for the front air spring with the volume of 3.48 l is used as an example in Figure 13.

5. Computer analysis of the air spring influence on the vehicle behaviour

To assess the vehicle behaviour with the real characteristic of the air suspension, a complex model is developed with the vehicle description in the form of a multibody model in the ADAMS environment and the air spring description by a separate program in the Matlab environment. The system parameters and the piston speed obtained from the multibody model are used at the input to the air spring program. The layout of the Matlab model is given in Figure 16.
Three coefficients are calculated in this model with the subsequent calculation of the force on the air spring. The $K_{vsp}$ coefficients are presented in the form of polynomials in accordance with the basic and auxiliary characteristic graphs.

To assess the stiffness of the in-vehicle air springs, it is necessary to conduct a number of special maneuvers with measuring the key parameters, the value of which will give an opportunity to estimate to which extent the selected stiffness suits the vehicle.

It is customary to use the maneuvers specified below for the vehicle properties assessment.

For stability and steerability assessment:
- driving with a constant radius;
- driving with a constant radius and steering angle;
- sine driving;
- slalom;
- "fish hook";
- lane change;
- double lane change (elk test);
- acceleration;
- braking.

For vibration load assessment:
- cobblestone driving;
- single bumping test.

Upon that, not all the specified maneuvers are indicative for assessing the contribution of the air springs only. In the dynamic maneuvers, dampers (shock absorbers) are more significant, while during the maneuvers related to big lateral pitches antiroll bars contribute more. Among the stability and steerability maneuvers, the air spring contribution can be detected when changing the lane. Increase in the air spring stiffness by 30% leads to decrease in the maximum roll angle by 2.5% at the first peak and 8% at the second one or by 3% at the first one and 12.5% at the second one in case of the deactivated antiroll bar (Figure 17). The graphs where both air spring chambers are used (soft characteristic) are designated as "Air soft". For the variant with one chamber (hard characteristic), the "Air hard"
The designation is used. The graph with the activated antiroll bar is designated as "ARB on", the one with the deactivated antiroll bar is designated as "ARB off".

**Figure 17.** Dependence of roll angle and lateral displacement on longitudinal displacement for lane change at 80 km/h. Lateral vehicle displacement on the right axis. Longitudinal vehicle displacement on the x-axis.

The air springs show themselves most clearly during braking and acceleration. Without switching off the antiroll bars, the decrease in the longitudinal pitching reaches 10%, irrespective of the "soft" or "hard" characteristic of the dampers (Figure 18). The graphs with low resistance of the dampers are designated as "Damper soft", the graphs with high resistance – as "Damper hard".

**Figure 18.** Dependence of longitudinal pitching angle and longitudinal acceleration on time for braking with 0.25g acceleration from 100 km/h
At the same time, when driving on cobblestone pavement, the decrease in maximum values of accelerations reaches 34% (8% for hard dampers) and that of the mean square deviation of acceleration is by 17% (5% for hard dampers) – Figure 19.

![Graph of vertical vibration acceleration dynamics at front passenger seat rails.](image)

**Figure 19.** Graph of vertical vibration acceleration dynamics at front passenger seat rails.

### 6. Conclusion

The paper presents a method to measure the force (load bearing) characteristic of adaptive suspension air springs. The provided data enable making a conclusion and obtaining polynomial descriptions of properties of the air springs when changing their internal volume, pressure in the design position and piston velocity. It was discovered that the dynamic stiffness of the suspension is 28-30% higher than the quasistatic one. At the same time, the velocity increase for more than 0.22 m/s does not result in any additional increase in stiffness. A method of universal description of a sleeve-type element force by multiplying the basic characteristic polynomial by 3 coefficients defining the main properties of the system (internal volume, initial pressure, piston velocity) is presented.

A method of investigating the air springs installed in a vehicle is described, also a method of simulating the vehicle behaviour for performing such research is presented. Upon that, the real air spring characteristics and the most distinctive maneuvers are used in the model. The air spring volumes (chambers) themselves were selected according to the real prototype of such system manufactured for the vehicle behaviour study. The simulation results with the numerical parameters indicative of expediency of implementing air springs with two and more chambers in the vehicle suspension have been obtained.

### References

[1] Gantikow M, Boyraz E, Kallert N, Legierski R *The new multi-chamber air spring by Porsche – future innovation in chassis mechatronics and integration* 7th International Munich Chassis Symposium 2016 Wiesbaden: Spinger Fachmedien 2017 pp 437-456

[2] GOST 31191.1-2004 (ISO 2631-1:1997) Vibration and shock. Measurement and evaluation of human exposure to whole-body vibration

[3] GOST 31507 – 2012 Handling and stability. Technical requirements. Test methods

[4] Pevzner Ya M, Gorelik A M *Pneumatic and hydro-pneumatic suspensions* Ed. Gelfgat D B Moscow Leningradskaya printing house of Gosgortechnizdat Publ, 1963 p 321

[5] Chassis Handbook Fundamentals, Driving Dynamics, Components, Mechatronics, Perspectives
This book is based on the 2nd edition of the German book a hrwerkhandbuch edited by Bernd Heiβing and Metin Ersoy 1st ed

[6]  
*Spring design manual* prepared under the auspices of the SAE spring committee 2nd ed 1996

[7]  
Peyman K E *Interconnected Air Suspensions with Independent Height and Stiffness Tuning*. A thesis presented to the University of Waterloo in fulfillment of the thesis requirement for the degree of Master of Science in Mechanical Engineering Waterloo Ontario Canada 2014

[8]  
Nieto A J, Morales A L, Chicharro J M, Pintado P *An adaptive pneumatic suspension system for improving ride comfort and handling* Journal of Vibration and Control 2016 Vol. 22(6) pp 1492–1503

[9]  
Nahvi H *Evaluation of Whole-Body Vibration and Ride Comfort in a Passenger Car*. Mechanical Eng Dept Isfahan University of Tech Isfahan 84156-83111 Iran 2009

[10]  
Khettou N et al, *Contribution to the modelling of a pneumatic semi-active Control of vehicle suspension* Vojnotehnički glasnik / Military Technical Courier Vol LXIII No 4 2015, pp 99–115