

Experimental analysis of the influence of pneumatic strut operating parameters on the effective damping value

Krzysztof Weigel-Milleret^{1,*} , Witold Grzeżożek

¹ Department of Motor Vehicles, Cracow University of Technology, Poland

* Corresponding Author: krzysztof.weigel-milleret@pk.edu.pl

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Abstract

The paper presents the results of experimental studies conducted on a measurement stand concerning the effect of selected operating parameters of a commercially available pneumatic strut with a hydraulic damper on the effective damping value. The studies were conducted at different chosen operating parameters: various harmonic excitation speeds and nominal gas pressures. The studies were conducted on a laboratory stand that enabled harmonic motion of pneumatic strut and acquisition of force, displacement, and pressure in the strut's working chamber data. The damping value was determined based on energy analysis and force-velocity characteristics. The results indicate a significant, nonlinear effect of the studied parameters on the system's damping properties. These studies allowed the collection of parameters necessary for the construction of a simplified physical model of the pneumatic strut with hydraulic damper.

Keywords: pneumatic strut, damping, dynamic damping coefficient, experimental studies

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Highlights

- Experimentally determined hysteresis loops revealed pressure-dependent energy dissipation.
- The gas spring force was separated using an experimentally derived effective area.
- The observed damping trend poses challenges for constant-frequency suspension design.
- Variable effective bellows area was identified as a function of pressure and strut length.

1. Introduction

Pneumatic suspension and damping systems are widely used in modern engineering problems, including those used in the automotive industry for springing, vibration damping, and vibration isolation. Air struts are used in suspension systems for high-end vehicles and commercial vehicles, among others. Their significant advantage is the ability to adjust dynamic characteristics, as well as vehicle ground clearance in off-road applications, by changing the amount and pressure of the gas in work chamber. Gas springs are characterized by a nonlinearity;

spring stiffness increases with load, which positively impacts vehicle safety and comfort [1,2]. The basic characteristics of air struts depend on the geometric parameters of the strut, the mechanical properties of the elastomer reinforced with fabric used to build strut bellows, and parameters such as pressure and suspension compression. Considering the complex relationship between the strut characteristics and pressure, as well as the mechanical properties of the components (rubber bellows), the whole systems are highly nonlinear [3].

Commonly used air struts with hydraulic damping do not offer adjusting the hydraulic damping force. To ensure vehicle comfort and safety, it is desirable to maintain a constant natural frequency of the system as the vehicle load changes:

$$\omega_0 = \sqrt{\frac{k}{m}} \quad (1)$$

ω_0 – natural angular frequency

k – suspension stiffness

m – mass per wheel (quarter vehicle model)

Recommended natural frequency values should be in the range of 1-2 Hz [4]. One of the advantages of air suspension is the ability to change the vehicle's ground clearance, or maintain it at a constant level by decoupling this parameter from the vehicle load. This requires varying the strut pressure, thus altering the strut's characteristics. The use of progressive springs enables the design of suspensions with adaptive characteristics.

$$f_0 = \frac{1}{2\pi} \sqrt{\frac{k(z)}{m}} \quad (2)$$

$$f_0 = \frac{\omega_0}{2\pi} \quad (3)$$

When considering the vehicle body's behavior under dynamic forces, low stiffness at small deflections provides greater comfort, while high stiffness at large deflections provides protection against bottoming out and impact load on structural and suspension components. Some of the air struts with hydraulic damping works with coil springs. Coil springs are known for their low hysteresis and negligible damping, which leads to conclusion that pneumatic struts are the primary damping elements.

Increasing suspension stiffness while maintaining comparable dynamic conditions, as it results from the formula:

$$c_c = 2\sqrt{mk} \quad (4)$$

c_c – critical damping

should lead to an increase in damping, because the aim is to maintain a similar level of damping in relation to the critical damping

In motor vehicles, it is recommended that the damping coefficient for compression stroke be within the range of (0.05-0.2) c_c , and for rebound stroke (0.2-0.4) c_c [4].

As can be deduced from the literature [5,6], real pneumatic systems may exhibit different behavior than the above theory, due to energy losses in the suspension components. For example, the damping properties of air struts may depend on friction forces and the drag force resulting from the mechanical properties of the rubber bellows. These constitute a significant component of the overall damping and influence the shape of the hysteresis loop of the damping characteristics and the level of energy dissipated in a single operating cycle [5].

In currently used automotive shock absorber solutions, the

damping force depends on a number of parameters, such as the direction of piston movement and piston position. Design solutions are currently being developed where the damping force is a function of the frequency of the driving vibrations. To assess damping, the logarithmic damping decrement is often used, which can be related to the damping coefficient. This coefficient is defined as the natural logarithm of the ratio of successive amplitudes. In this work, harmonic excitation was used, and damping was defined by introducing the concept of effective damping [7], which takes into account the actual energy losses in the system during the operating cycle. This parameter depends not only on the design of the strut and hydraulic damper, but also on operating conditions, including adjustable parameters such as the excitation speed or system pressure [2,8]. Consequently, damping is strongly non-linear and depends on both the design parameters and the dynamic conditions of suspension operation [9,10]. Fongue draws attention to the hysteresis field value, which can reduce ride comfort under low-amplitude inputs [11]. Vo discusses the issue of friction related to the material from which the bellows are made [12]. Gheorge emphasizes the material from which the bellows are made and its impact on suspension characteristics [13].

Despite numerous studies on pneumatic system modeling, which focus on constructing theoretical models [1] or simulation analysis of selected parameters [2,9,14], the literature still lacks experimental results of real struts that consider the simultaneous effect of the described operating parameters on effective damping. Modeling requires the system to be considered as a parallel system of the air spring and damper. The total force generated by the strut must be considered as a sum of components. For example, Vo and Le in their paper propose that this force should consists parts such as the thermodynamic force of the compressed air, the frictional force by the relative motion between the rubber and fabric, friction between the piston, piston rod and the seal, the rubber bellow and the surface of the piston, and the viscoelastic force of the rubber material [12].

This paper presents the results of experimental studies of a pneumatic strut equipped with hydraulic damping. The influence of operating pressure and excitation velocity on the system's damping properties was analyzed. The effective

damping coefficient was determined by the energy dissipated during the cycle. Particular attention was paid to the relationship between the change in stiffness resulting from pressure changes and the observed damping level, taking into account the effect of dry friction.

The analysis was based on the results of bench tests, where damping characteristics were determined experimentally. During the tests, force, gas pressure in the column, and column displacement were measured. Based on these data the force-displacement and force-velocity characteristics were determined.

The scope of the study included changes in initial, nominal pressure from 2 to 6.5 bar and excitation frequency from 0.5 Hz to 2 Hz, while simultaneously measuring force, displacement, and operating pressure as a function of time. This allowed for the identification of nonlinear relationships between the controlled parameters and effective damping.

2. Materials and methods

2.1. Research object

The subject of this research is a Monroe MA756 load adjusting pneumatic strut with shock absorber. In most applications, these types of struts are used in conjunction with the main suspension spring. The MA756 model provides additional suspension force under increased vehicle load. Using this element allows for the modification of the effective suspension stiffness. Suspension damping is provided by a hydraulic damper in the tested strut. This article focuses on the effect of the gas spring on strut damping. Figure 1 shows the analyzed strut. The basic strut parameters given in the Table 1.



Figure 1. Tested Monroe MA756 strut [15]

Table 1. Strut parameter.

parameter	value	unit
manufacturer	Monroe	
model	MA756	
minimum length:	10.250 / 260	" / mm
maximum length:	16.375 / 416	" / mm
travel:	6.125 / 156	" / mm
top mount:	Threaded rod	
bottom mount:	Metal-rubber bushing	

The strut air connector was redesigned to accommodate

a 16mm internal diameter pneumatic line, which is used to connect the pressure sensor and the air pressure source.

2.2. Test stand

The test stand uses a mechanical excitation via a crank system. The test object is attached to the test stand frame with an upper mount through force transducer, and the lower mount is placed in a slide guides, the movement of which is forced by the crank system. Auxiliary force transducer is placed on slide guides. The drive consists of an electric motor and a reduction gear with a gear ratio of 1:9.75. The electric motor is powered by an inverter, allowing for smooth regulation of the rotational speed (excitation frequency), which influences the achieved linear speeds of the pneumatic column. Pressure transducer is located on the pneumatic line, directly adjacent to the tested strut.

The excitation amplitude is adjustable in increments within the available range of 25-100 mm. During the tests, an amplitude of 50 mm (peak-to-peak value of 100 mm) was selected. Test stand diagram is shown in Figure 2.

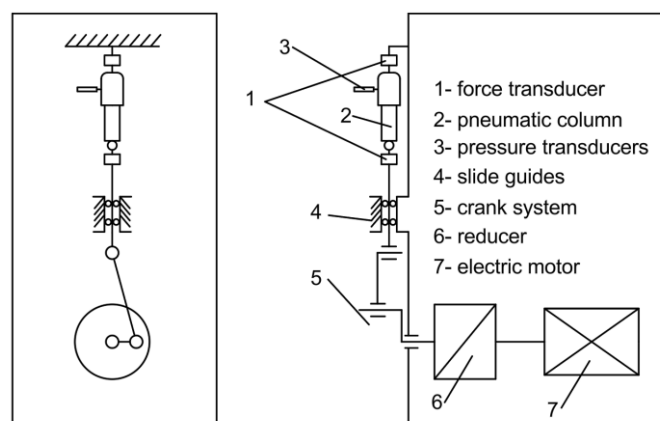


Figure 2. The diagram of the test stand and sensors.

2.3. Measuring equipment

In order to measure the force HBM U9C force transducers with a measuring range of 10 kN were used, one placed between the excitation and the lower strut mount, and the other between the upper strut mounting and the test stand frame. Strut displacement was measured using a Kubler A40 wire sensor with a measuring range of 1000 mm, mounted to the test stand frame and the lower strut mounting plate, on a sliding excitation element. Pressure was recorded using HBM P8AP pressure transducers. The measurement system was completed by a Spider 8 measuring amplifier and an data acquisition computer with HBM Catman software. Sensor parameters are provided in

the Table 2. The Figure 3. shows the test stand with the sensors.
Table 2. Sensors parameters.

Parameter	Sensor	Measuring Range	Accuracy Class/Linearity Error
Force	F_0	HBM U9C	+ - 10 kN
Displacement	x	Kübler A40	1000 mm
Pressure	p	HBM P8AP	100 bar
			0.2
			$\pm 0.35\%$
			0.3

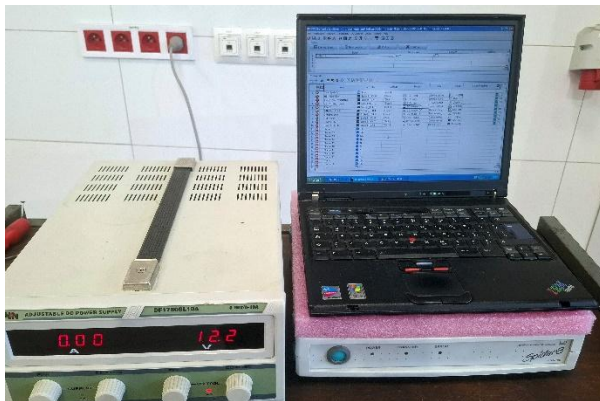


Figure 3. Test stand, measuring amplifier and data acquisition computer.

2.4. Research methodology

In the analysis of pneumatic strut with hydraulic damper with nonlinear damping properties, resultant force is a function of multiple parameters.

$$F_0(t) = F_s(l, p) + F_d(x, \dot{x}, p) + F_r \operatorname{sgn}(\dot{x}) \quad (5)$$

F_0 – resultant force of the strut,

F_s – spring force,

F_d – damping force,

F_r – frictional and viscoelastic force of the rubber bellow material,

p – current internal pressure,

l – current strut length,

x – displacement,

$\dot{x}$ – velocity,

$\operatorname{sgn}(\dot{x})$ – velocity sign.

To construct a simplified physical model, the actual, nonlinear system was replaced with a linear system in which the spring force is proportional to the displacement and the damping force is proportional to the velocity. The effect of dry friction and the rubber bellows rolling over the column surface was taken into account as constant force, dependent on the direction of motion. The analysis assumed that the total force acting in the system can be written as the sum of the elastic, viscous, and dry friction components:

$$F_0(t) = Kx(t) + C\dot{x}(t) + F_r \operatorname{sgn}(\dot{x}(t)) \quad (6)$$

K – air spring stiffness,

C – damping coefficient,

The dissipated energy was determined as the absolute value of the area enclosed by the force–displacement hysteresis loop:

$$W_d = \oint F_0 dx \quad (7)$$

where W_d is the energy dissipated by the system in one cycle.

This energy is equal to the work of the damping forces and can also be written as an integral with respect to time [6]:

$$W_d = \int_0^T F_0(t) \dot{x}(t) dt \quad (8)$$

In the damped systems with hysteresis and nonlinear dampers analysis, the hysteresis loop area is equal to energy losses and can be the basis for determining the effective damping coefficient [6,7] which combine all energy dissipation paths: viscous damping, dry friction, viscoelastic effects of rubber bellow. In the equivalent linear system, it is related to the damping force by:

$$F_d(t) = C_{eff} \dot{x}(t) \quad (9)$$

Assuming a harmonic displacement:

$$x(t) = X \sin(\omega t) \quad (10)$$

$$\dot{x}(t) = \omega X \cos(\omega t) \quad (11)$$

ω – angular frequency of the excitation,

X – displacement amplitude.

The energy dissipated in one cycle is:

$$W_{cycle} = \int_0^T C_{eff} \dot{x}^2(t) dt \quad (12)$$

What after substitution and integration over the period $T = \frac{2\pi}{\omega}$:

$$W_{cycle} = \int_0^T C_{eff} (\omega X \cos(\omega t))^2 dt = \pi C_{eff} \omega X^2 \quad (13)$$

$$C_{eff} = \frac{W_{cycle}}{\pi \omega X^2} \quad (14)$$

Above parameter can be used to quantitatively describe the performance of the struts, as well as to compare individual technical solutions of struts. The next step in the research was to determine the damping coefficients separately for the compression and rebound motions. To do this, the damping force was extracted from the total strut force. It was necessary because the presence of dry friction and viscoelastic force has a significant impact on the shape of the hysteresis loop. In particular, it:

- increases the loop area, thus increasing the dissipated energy,
- causes the appearance of a velocity-independent component, distinct from viscous damping [16].

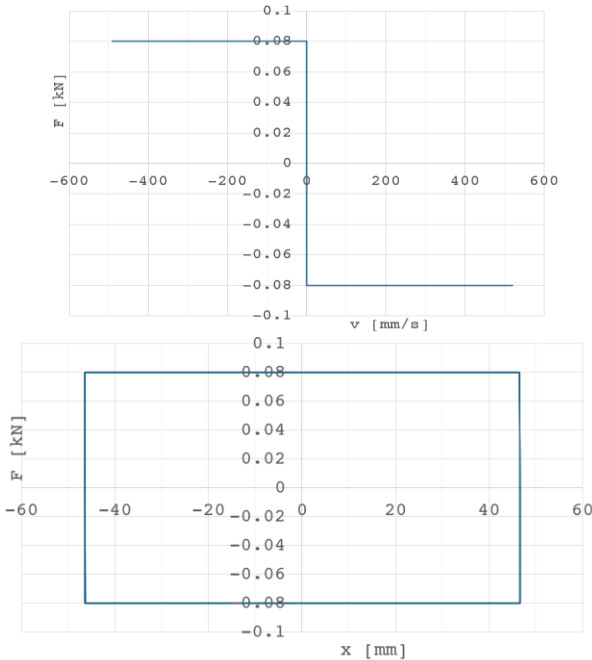


Figure 4. Friction force as a function of velocity and displacement.

$$A(l, p) = 38,96 - 0.05073 \cdot l - 1.116 \cdot p + 4.816 \cdot 10^{-5} \cdot l^2 + 0.001674 \cdot l \cdot p + 0.002897 \cdot p^2 \quad (16)$$

The test results are presented in the figure 5.

The gas spring force was calculated for each measurement sample according to the following relationship:

From an energy analysis perspective, this means that part of the energy dissipated in the cycle comes from dry friction and viscoelastic force of the rubber bellow, not solely from the pneumatic (related to gas flows and throttling at the strut outlet) and viscous damping mechanisms. In this study, it was decided to separate the friction effects so that the determined damping coefficient included only viscous effects, not dry friction. Redesigning the connection reduced flow resistance, so the pneumatic dissipation energy path can be neglected. The drag force value was measured on in the tested element, and its maximum value was ± 80 N. The friction force characteristics are shown in Fig. 4.

It was noted that the force generated by a gas spring depends not only on the operating pressure but also on the effective active area of the bellows, as was mentioned [1,14]. The force generated by a gas spring is described by the following relationship:

$$F_s(l, p) = p \cdot A(l, p) \quad (15)$$

$A(l, p)$ — effective active area of the bellow (function of strut length and current pressure).

The analysis assumed that the effective bellow's area is not a constant value. Changing the column length impacts the shape of the bellow's working part, thus changing the effective area. Furthermore, due to the deformability of the rubber element, this area may also depend on the operating pressure. The force generated by the spring was tested as a function of pressure and strut length. For tested values of strut length and pressure, the effective active area was determined. Based on the data obtained from the test, the characteristic was approximated. The Matlab Software was used to obtain the model. A second-degree polynomial (with constant, linear, and quadratic terms) was used as the function $A(l, p)$ using a script from the Curve Fitting Toolbox. The `fit(...)` script is used to fit a mathematical function to the measured data using the least squares method. The `poly22` approximation type (second-degree polynomial surface) was chosen because the coefficient of determination for this type of approximation is $R^2 = 0.9782$. The approximation results are presented as quadratic function with the following parameters:

$$F_{s,i} = p_i \cdot A(l_i, p_i) \quad (17)$$

where the i index denotes the number of measurement sample, p value is the current gauge pressure measured during

test and l is the current value of strut length. Then, in order to obtain the isolated damping force, the above components were subtracted from the total force measured on the stand, shown in Eq. (5):

$$F_{d,i} = F_{0,i} - F_{s,i} - F_{r,i} \tag{18}$$

The damping force thus extracted was used to determine the damping coefficient. The F_d force includes hydraulic damping and possible energy dissipation, which has its source in other pneumatic damping phenomena. By taking into account the variable effective bellows area, it was possible to reduce the error resulting from adopting a simplified model with a constant effective area.

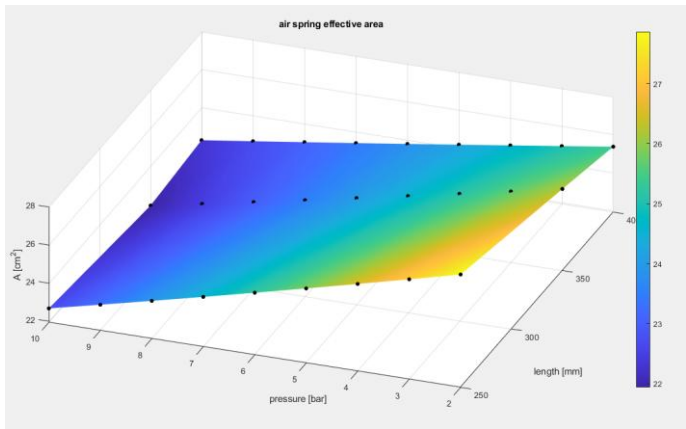


Figure 5. Pneumatic spring force as a function of pressure and strut length (black points - measurement data, surface - mathematical model).

2.5. Test plan

A series of tests was conducted on the pneumatic strut with the variable parameters listed in the Table 3.

For the considered damping in the pneumatic strut, the leading parameter, in addition to the excitation frequency, is the

piston speed, therefore the Table 3. provides the average speed calculated based on the frequency and the maximum piston speed measured during the tests during harmonic excitation for each frequency. Cyclic waveforms were recorded for multiple harmonic excitations. During data processing, the files were divided into cycles differing in strut operating parameters. Data analysis was performed using Matlab R2024a software. Statistical analysis of the determined parameters was performed to assess the repeatability of the results. The mean value and standard deviation were calculated for the vibration period, dissipated energy, and effective damping values based on subsequent system operating cycles.

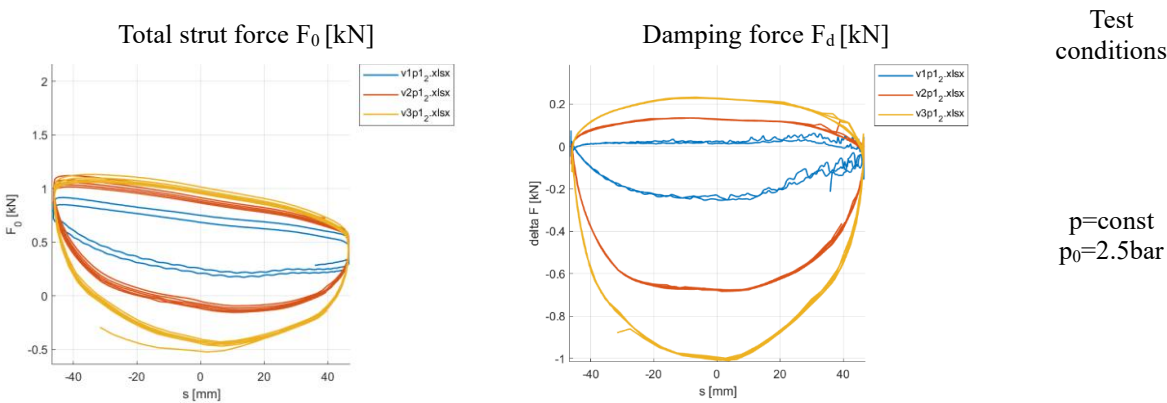
Table 3. Operating parameters.

parameter		extreme values			unit
pressure	p_0	2,5	4,5	6	bar
operating frequency	f_0	0,5	1	1.8	Hz
average piston velocity	v_a	100	200	360	mm/s
maximum piston velocity	v_{max}	150	290	520	mm/s

3. Test results

3.1. Force-displacement characteristics

The test results are presented as a function of strut displacement in Figure 6. The results collected when working with a constant nominal gas pressure in the strut (variable forcing frequency) and with a constant excitation frequency (variable gas pressure) are presented separately. The graphs in the first column represent the total strut force recorded during the test, while the graphs in the second column represent the damping force, separated as described above. For clarity, separate scales were used on the plots for the total strut force [-0.6; 2.25] kN and the damping force [-1; 0.3] kN.



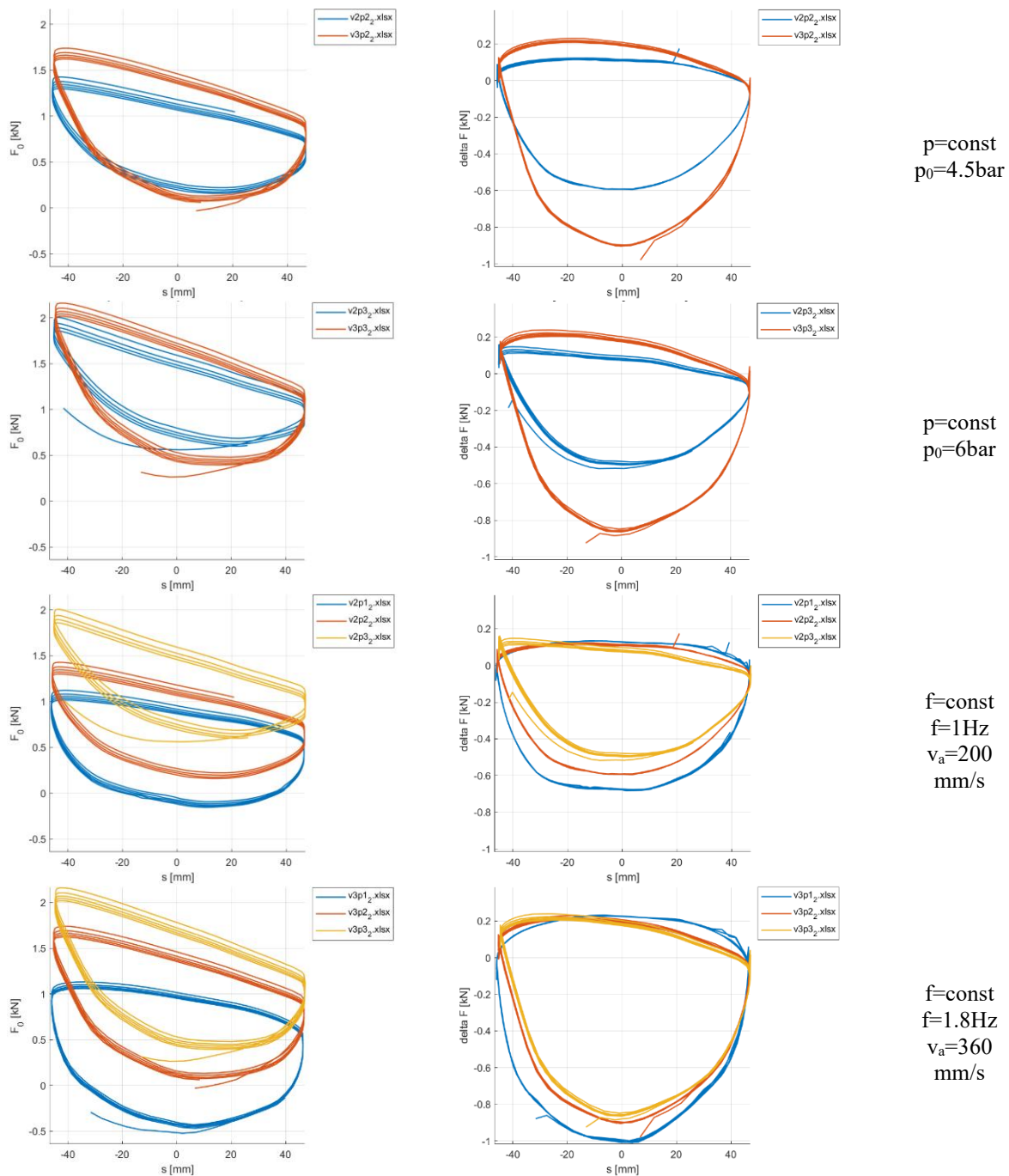


Figure 6. Force–displacement characteristics of the tested strut: total force and extracted damping force; designations in charts: v1 refers to $f_0=0,5 \text{ Hz}$ and $v_a=100 \text{ mm/s}$; v2 refers to $f_0=1 \text{ Hz}$ and $v_a=200 \text{ mm/s}$; v3 refers to $f_0=1,8 \text{ Hz}$ and $v_a=360 \text{ mm/s}$; p1 refers to $p_0=2,5 \text{ bar}$; p2 refers to $p_0=4,5 \text{ bar}$; p3 refers to $p_0=6 \text{ bar}$.

Table 4. Energy loss during one cycle and the standard deviation as a function of frequency and pressure.

p [bar]	f [Hz]	0.5	1	1.8
2.5		39.23 (0.06)	77.71 (0.3)	106.15 (0.86)
4.5		-*	66.99 (0.23)	92.45 (1.07)
6		-*	57.46 (1.31)	88.39 (1.31)

*data could not be collected due to limitations of the test stand

For the above data, the energy loss during one cycle W_d [J]

and the standard deviation (shown in brackets) are presented in the table 4.

3.2. Force-velocity characteristics

The test results are presented as a function of strut deflection velocity in Figure 7. The results collected when working with a constant nominal gas pressure in the strut (variable forcing frequency) and with a constant forcing frequency (variable nominal gas pressure) are presented separately.

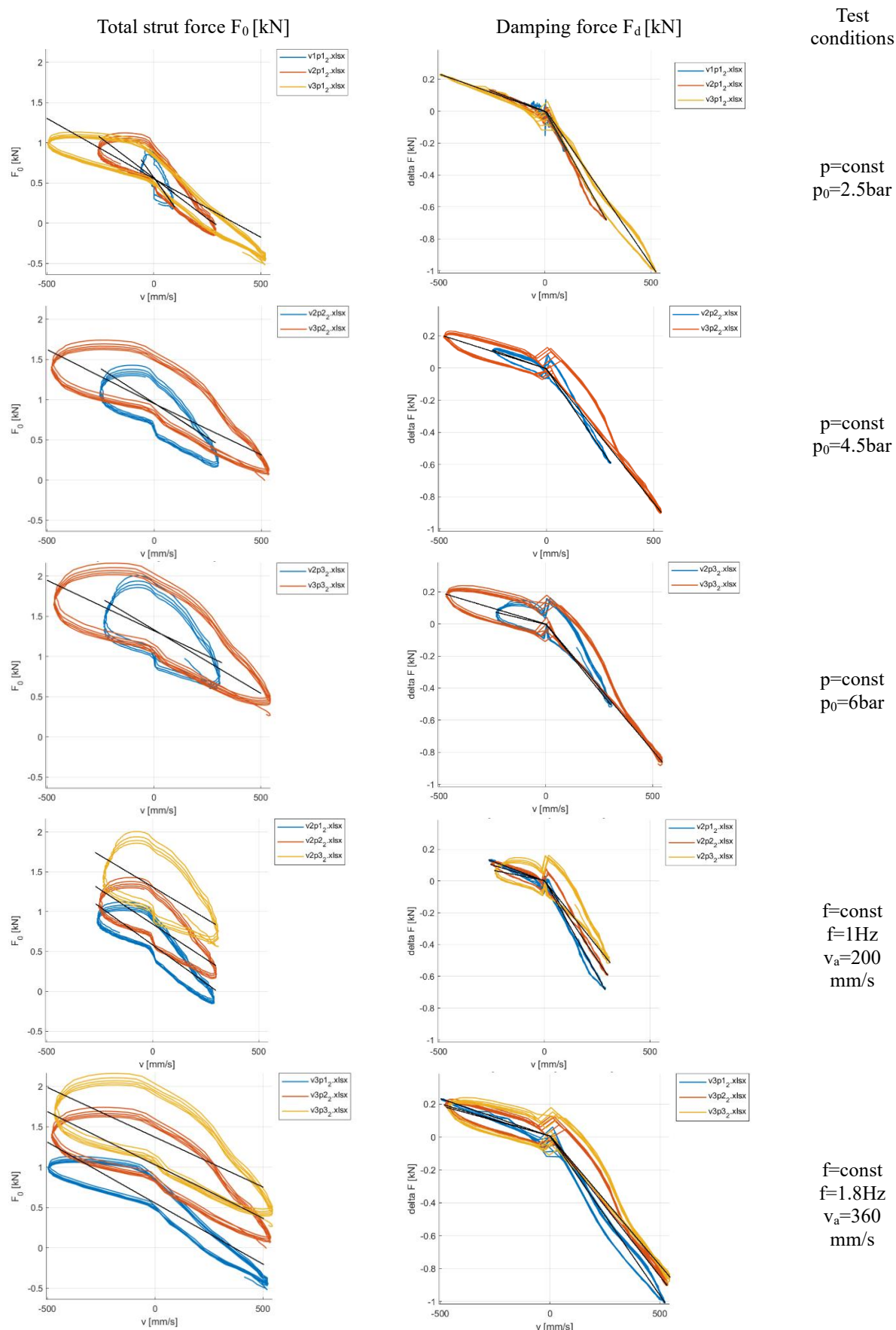


Figure 7. Total strut force and damping force characteristics as a function of velocity; designations in charts: v1 refers to $f_0=0,5$ Hz and $v_a=100$ mm/s; v2 refers to $f_0=1$ Hz and $v_a=200$ mm/s; v3 refers to $f_0=1,8$ Hz and $v_a=360$ mm/s; p1 refers to $p_0=2,5$ bar; p2 refers to $p_0=4,5$ bar; p3 refers to $p_0=6$ bar.

The graphs in the first column represent the total strut force recorded during the test, while the graphs in the second column represent the damping force, separated as described in point 2.3. The scales on the plots were selected as above, differing for the total strut force and the damping force.

Table 5. Effective damping coefficient and the standard deviation as a function of frequency and pressure.

p [bar]	f [Hz]	0.5	1	1.8
2.5		3663.51 (17.19)	1886.21 (8.99)	1416.04 (16.67)
4.5		-*	1638.76 (15.18)	1242.74 (17.99)
6		-*	1459.58 (54.66)	1203.35 (25.43)

*data could not be collected due to limitations of the test stand

Data analysis was performed in the Matlab R2024 environment using a custom procedure. Cycle detection was performed for each recorded data using the findpeaks script, imposing the MinPeakProminence condition to eliminate measurement noise. For each detected cycle, the dissipated energy (according to the Eq. (7)), the excitation amplitude, and the period were calculated. This allowed for the determination of the effective damping coefficient C_{eff} [Ns/m]. Each series of results allowed for the isolation of at least a dozen operating cycles, which allowed for statistical analysis and the calculation of the standard deviation. The calculated effective damping coefficient is shown in the Total strut force graphs in Figure 6 as black lines $F_d' = C_{eff} \cdot v$ shifted by the value of the gas spring force. For the analyzed data, the effective damping coefficient and its standard deviation (shown in brackets) are

Table 6. Damping coefficient for compression and rebound stroke as a function of frequency and pressure.

f [Hz]		0.5		1		1.8	
p [bar]	rebound	compression	rebound	compression	rebound	compression	
2.5	2480.8	413.62	2428.53	488.8	1943.76	447.86	
4.5	-*	-*	1941.01	433.22	1616.72	405.99	
6	-*	-*	1515.67	371.19	1519.97	401.62	

*data could not be collected due to limitations of the test stand

presented in the table 5. The results are presented in the form of a surface graph in Fig 8.

For each of the isolated damping force characteristics (as a function of velocity), the damping coefficient C [Ns/m] was determined. The Matlab Software was used to approximate the damping characteristics by straight lines, separately for the compression and rebound stroke. The value of the damping coefficient is equal to tangent value of the slope of the determined approximated line on the force-velocity graph [17]. The calculated damping coefficients separately for compression and rebound are shown in the damping force graphs in Figure 6 as black lines. The results are summarized in the Table 6. The results are presented in the form of a surface graph in Figure 9. separately for compression and rebound stroke.

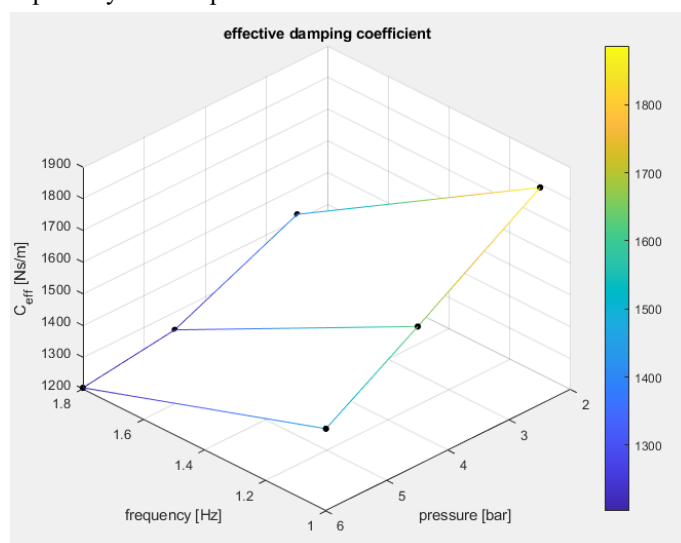


Figure 8. Effective damping coefficient C_{eff} as a function of frequency and gas pressure.

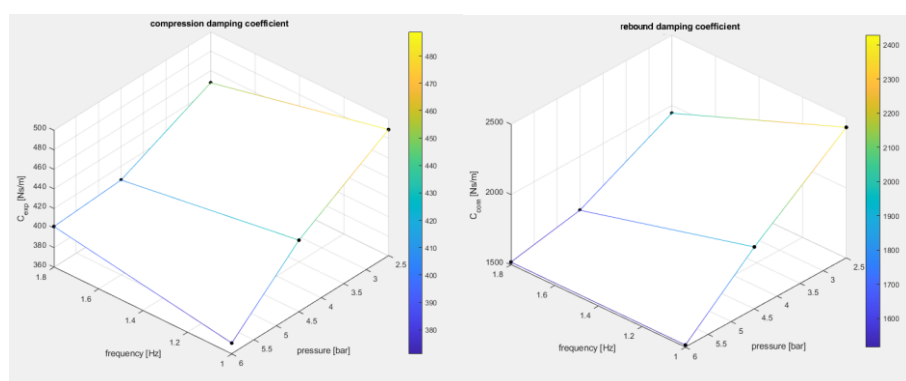


Figure 9. Damping coefficient C as a function of frequency and gas pressure for compression and rebound.

Based on the conducted tests, it can be concluded that the change in the damping coefficient for the rebound stroke is approximately twice as large as the change for the compression stroke at both tested pressures.

4. Conclusion

In the tested pneumatic strut system, stiffness and damping are not independent parameters; instead, they exhibit a coupling that is opposite to that expected from the design of vehicle suspensions, as discussed in Section 1 of this paper.

As the nominal gas pressure in the working chamber increases, a reduction in the hysteresis loop area was observed in the force-displacement characteristic, which translates to less energy dissipated during a single work cycle. The lack of user ability to influence the hydraulic damping means that increasing the gas pressure changes the damping in the pneumatic system; it is possible that this change also includes a change in the energy dissipated by friction and viscoelastic forces of rubber bellow.

Increasing the working pressure (e.g., to level the vehicle after loading) leads to a reduction in the system's ability to dissipate vibration energy. As the pressure (and therefore the stiffness of the strut) increases from 2.5 to 6 bar, a decrease in effective damping from 23% decrease of effective damping coefficient at 1 Hz excitation frequency and to 15% decrease at 1.8 Hz frequency is observed. Such effective damping coefficient decrease can be considered significant from the point of view of comfort assessment

With increasing excitation frequency, an increase in the hysteresis loop area in the force-displacement characteristic is observed, which translates to an increase in the energy dissipated in a single cycle. Increasing the excitation frequency, on the other hand, leads to a decrease in the effective damping

coefficient. Considering damping during the compression and rebound stroke separately, it can be seen that with increasing excitation frequency, a decrease in the damping coefficient, defined as the slope of the force-velocity characteristic for the rebound stroke, is observed, while a slight increase in the damping coefficient for the compression stroke is observed.

Increasing the excitation frequency, although it leads to an increase in the vibration energy dissipated in a single cycle, but also leads to a decrease in effective damping. The effective damping coefficient is proportional to energy and inversely proportional to the excitation frequency (Eq. 14.). Therefore, a 27-34% increase in energy (for different nominal pressures) with a 44% increase in the excitation frequency leads to a decrease in the effective damping coefficient value. The observed slight increase in the damping coefficient during the compression stroke at higher excitation speeds is also undesirable from the point of view of vehicle suspension design, as it leads to increased shocks and loads on structural components.

The experimental results obtained for a pneumatic strut with hydraulic damping indicate a relationship between adjustable stiffness and damping efficiency. The use of such struts as supplementary or primary, in light vehicles, spring-damping elements, according to the authors, requires analysis of damping coefficient changes to avoid values of damping coefficient outside the range recommended in the literature.

The presented research results indicate the need to construct a complex mathematical model of a pneumatic strut with hydraulic damper. In the available literature, mathematical models do not take into account the influence of pressure and frequency changes in pneumatic-hydraulic columns in a series arrangement on the damping force.

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