



# Analysis and characterization of the friction of vehicle body vibration dampers

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## Abstract

The friction at the contact surfaces of a vehicle body vibration damper, which are moved relatively to each other, influences its transmission behavior at the start of movement (breakaway force) as well as with excitation signals of higher velocity and thus has an impact on the comfort properties of the damper. According to Vibracoustic (Die wichtigsten Kriterien für deutsche Autofahrer beim Autokauf, Springer Fachmedien Wiesbaden GmbH, Wiesbaden, 2019), for most German drivers (63%) comfort (in addition to brand and appearance) before driving dynamics (53%) and environmental compatibility (48%) is the most important criteria when evaluating a new car, which explains the importance of this vehicle characteristics. Furthermore, the friction is present with any relative movement of the damper and is, therefore, relevant for the design of the damper and the associated vertical dynamics. The friction is generally determined in the fully assembled state of the damper, including oil filling and gas pressure at a very low movement velocity to eliminate the influence of the damping force. This measurement method allows no or only inadequate statements about the friction behavior at, e.g. more dynamic excitation scenarios. As a result, the aim should be to characterize the friction properties without the influence of hydraulic damping at the start of movement or reversal of movement, as well as at higher movement velocities. Another goal is to evaluate the influence of the internal pressure of the damper on its friction behavior. The test damper used here is a commercially available monotube damper that has been modified in accordance with the requirements for these tests. The results shown below can be used as starting variables for further investigations for the targeted optimization of the friction properties and thus for the improvement of driving comfort. The reduction in damper friction promises an increase in comfort due to the improved decoupling of the vehicle body from the road excitation. Furthermore, the data obtained enable the level of detail of simulation models to be increased and serve as a basis for comparing different friction pairings and contact surfaces in the damper. For the substitution of coatings (chrome-free piston rods → environmental protection) or tube materials (aluminum matrix composites → lightweight construction) as well as for changes in the surface structure and roughness, the results enable an evaluation of the friction properties compared to conventional dampers and the adjustment of the friction pairings in the sense of the best possible functionality.

**Keywords** Damper · Friction · NVH · Response characteristic

## 1 Introduction

The transmission path for roadway-induced vibrations from the wheel-roadway contact to the vehicle body and thus the passenger is largely defined by the stiffness, masses and

damping in the chassis. Under the greatly simplified, widespread and here later refuted assumption of a velocity or frequency-independent, constant friction force, the friction within the damper leads to an increase in the stiffness in the entire excitation spectrum and thus tends to improve the transmission of vibrations with the result of corresponding coupling into the body. Since in HEV,<sup>1</sup> the internal combustion engine is out of service in certain driving situations and in BEV<sup>2</sup> not existing, therefore, only very little noise or vibration masking can be expected from the powertrain. For

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this reason, road and chassis-induced noise and vibrations are even more clearly the focus of NVH<sup>3</sup> optimization in this type of vehicle. A precise design of a vehicle with regard to its vibration properties and the resulting NVH behavior needs exact knowledge of the transmission behavior of the vibration transmission links. Furthermore, for reasons of efficiency, vehicle design is being increasingly supported by simulations. A detailed understanding of the physical relationships is necessary for the simulation models for NVH tuning as well as for the mechanical component design. In addition to many other areas, this also applies to the friction in the damper and in particular to its behavior in the entire frequency spectrum of the transmitted vibrations. Assuming damper friction as constant and independent from velocity or frequency is not sufficient for a good design of a modern

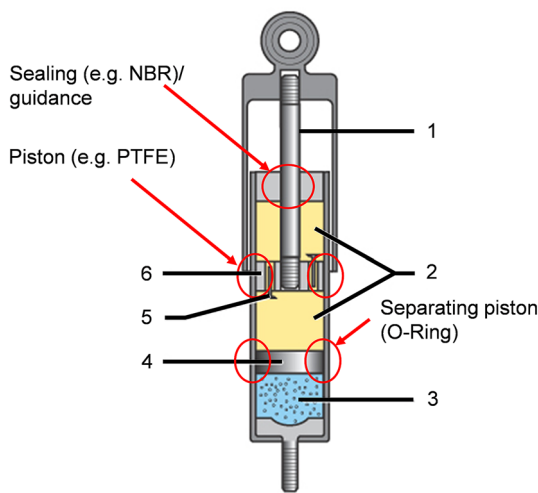
vehicle. For this reason, the relationships and properties of the friction in the damper are considered in more detail below.

## 2 Damper friction, cause and effects

The friction in a telescopic damper (standard design in vehicle construction) arises at the contact surfaces that slide relatively to each other in the sealing guidance unit at the part where the piston rod emerges from the damper tube and at the contact surface of the damper piston with the damper tube. The monotube damper (Fig. 1) also creates friction on the separating piston, which isolates the oil-filled working area from the gas-filled compensation area.

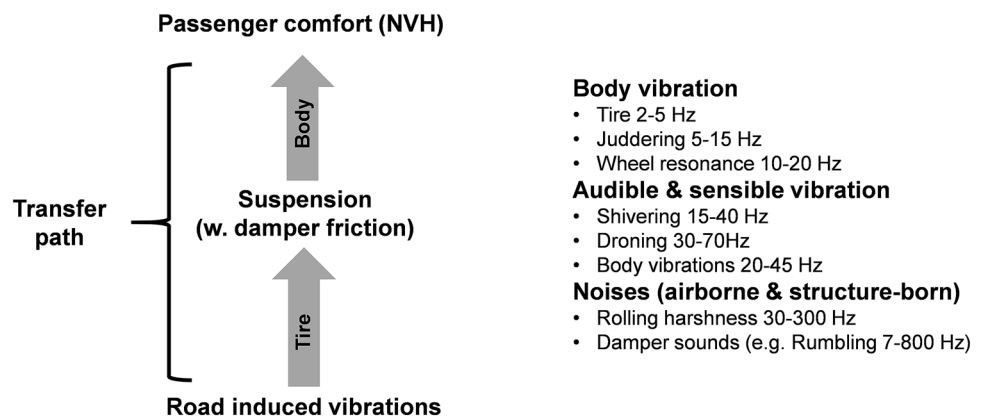
The friction occurring at these contact surfaces always counteracts the direction of movement. This frictional force is to be added to the damping force and consequently leads to an increase in the force applied by the damper beyond the designed hydraulic force. As part of the transmission path (Fig. 2), this results in a slight change in the transmission behavior or a stiffening of the system compared to the consideration of purely hydraulic forces, which can have a negative influence on certain vibration and noise phenomena in different frequency ranges.

This property gives rise to the above-mentioned influence on the NVH behavior of the vehicle, which tends to be adversely affected by the increase of stiffness and possible stick–slip effects, particularly towards higher frequencies. Since the influence of the friction can predominate especially at low strokes and thus also at higher frequencies with higher velocities and more movement reversals per time step (high frequencies and relatively large strokes are practically not found in the suspension when used as intended), the “harshness” and “noise” areas play a particularly important role here.



**Fig. 1** Monotube damper (schematically) and representation of the contact surfaces at which friction occurs. 1. Piston rod; 2. oil; 3. gas; 4. separating piston; 5. piston valve; 6. piston; based on [5]

**Fig. 2** Schematic representation of the transmission path of the road excitation to the occupant (left), examples of vibration and noise phenomena occurring in the vehicle, (right, based on [2])



<sup>3</sup> Noise Vibration Harshness.

Figure 3 shows the commonly used differentiation of the frequency ranges in the vehicle. Vibrations below approx.

15 Hz can only be felt (vibration), whereas vibrations greater than 100 Hz are primarily only audible (noise). The transition area in between is the so-called harshness area. With low-frequency vibrations and large amplitude, e.g. brake nodding, rolling when cornering or excitations in the body’s natural frequency range (around 1 Hz), the occurring damping and spring forces clearly outweigh the frictional forces. The focus here is on the importance of the damper in terms of optimum traction through low wheel load fluctuations. Higher frequencies, especially beyond the natural wheel frequency (approx. 10–15 Hz) have a minor influence on the driving safety [4], but are important for passenger comfort. From this, it can be seen that the friction in the damper primarily affects driving comfort.

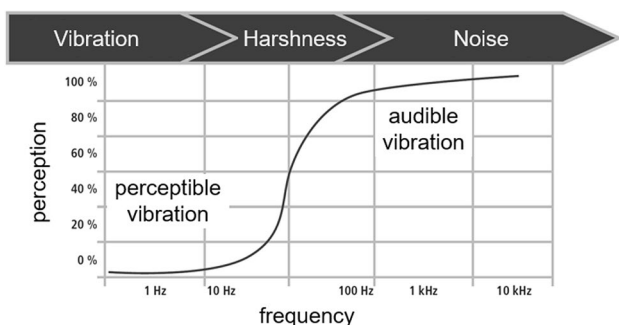


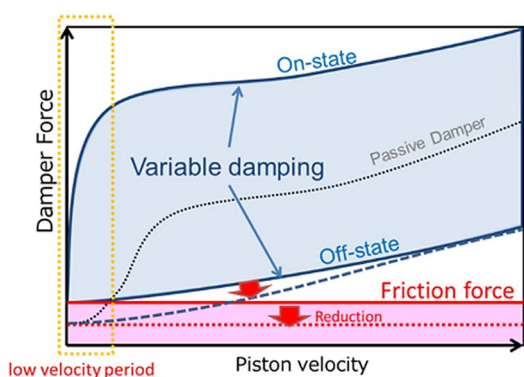
Fig. 3 Frequency ranges of NVH, based on [6]

### 3 State of the art

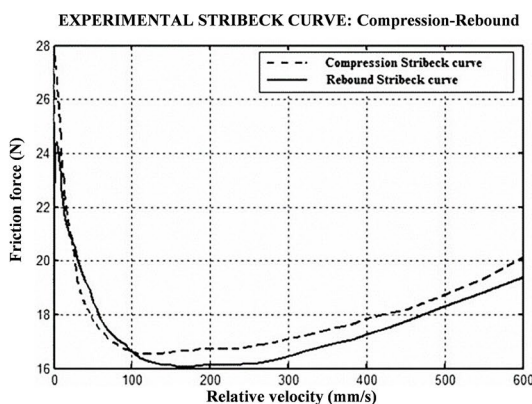
The following Fig. 4 shows excerpts from two publications, which are based on the subject of friction in the damper. In the diagram on the left, a problem of magneto-rheological dampers is illustrated, in which, when the lowest damping forces are set, the frictional force in the lower velocity range can even exceed the damping force, as a result of which a very soft characteristic curve cannot be fully realized here. Due to the necessary particles in the damper oil, the friction in magneto-rheological dampers is naturally higher than in conventional dampers with pure oil filling.

The right part of Fig. 4 shows the frictional force of a twin-tube damper over the relative velocity. As can be seen, this characteristic curve was determined up to an excitation velocity of 0.6 m/s and furthermore with an oil-filled damper without valve plating. The increase in force towards higher velocities, therefore, leads to the assumption that this increase in force is not the result of increased friction, but of hydraulic forces. At this point, reference is also made to [2], which deals with a similar topic, but focuses on the twin-tube damper and the measurement method and data processing differ from the following representations or are not presented in detail.

Hence, there is a need of corresponding investigations to characterize the friction force under clearly defined boundary conditions and without influences due to hydraulic forces on the piston and at higher movement velocities. Furthermore, the methodology used is not or only to a very limited extent presented in the existing sources.



Honda: Effect of friction reduction of magneto-rheological semi-active damper on ride comfort and vehicle dynamics



KYBSE: Modelling of friction phenomena in sliding conditions in suspension shock absorbers

Fig. 4 Excerpt from literature research, previous studies on friction in the damper, their approach and background, left magneto-rheological damper [1], right twin-tube damper [3]

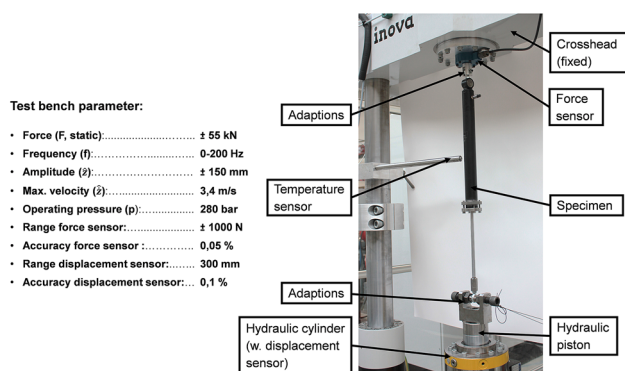


Fig. 5 Test bench setup and technical data (extract)

## 4 Experimental set-up

### 4.1 Testbench

The test bench used for the experiments is a hydropulse test bench from the manufacturer Inova consisting of a conventional hydraulic system, which parameters are designed for testing a wide range of different vehicle body vibration dampers. With its variable structure and with the largely customizable control system, the test bench is designated for operation in a scientific environment. The centerpiece is a hydraulic cylinder, which provides the necessary displacements/movements of the damper and also the required forces. The force generated in the damper is supported by a height-variable, massive crosshead on two posts (see Fig. 5). Both main components are screwed onto a rigid test bed, which is itself placed on elastomer bearings. The hydraulic power is provided by a hydraulic unit in combination with appropriately designed bladder accumulators. The most important parameters of the test bench are shown in Fig. 5 below.

The measurements explained below are carried out in the operating mode “position control”. The displacement measurement is realized via an inductive displacement sensor implemented in the hydraulic cylinder. The (friction) force resulting from the displacement of the damper is recorded by a force sensor based on temperature-compensated strain gauges at the crosshead (sensor data see Fig. 5). For the experiments, depending on the operating mode, a frequency–amplitude or velocity–amplitude combination is specified in the control software of the test bench according to which the movement sequence of the hydraulic cylinder is regulated. Depending on the test program, data are recorded as a mean value storage from a defined number of measuring cycles or as continuous storage with acquisition rates up to 10 kHz, which is indicated in the respective section of the measurements.

### 4.2 Tested specimen

The vehicle body vibration damper used for the measurements presented below is a commercially available mono-tube damper from the manufacturer ZF Friedrichshafen AG. Designed for the rear axle of a mid-range vehicle, it has a piston rod diameter of 11 mm, a damper tube diameter of 36 mm and an internal pressure of approx. 26 bar. As a standard, the damper piston is coated with PTFE<sup>4</sup> to reduce friction. The damper oil used here is TITAN SAF 1579 EU 175 from the manufacturer Fuchs Lubricants GmbH with a density of 0.833 g/ml at 15 °C, a kinematic viscosity of 23.0 mm<sup>2</sup>/s at 20 °C and 3.3 mm<sup>2</sup>/s at 100 °C. For the necessary modifications of the damper (as seen in Fig. 6), the damper is opened a little below the end of the tube on the piston rod outlet side, the tube is provided with a thread, and an appropriate device for realizing a reclosability is attached. The damper tube is equipped with a pressure connection and a check valve to create the required internal pressure after reassembling. The pressure connection is applied using a laser welding process to avoid any tension or distortion in the damper tube due to the lowest possible heat input. The pressure is applied with undried air by a hand pump and is adjusted by measuring the gas pressure force directly on the test bench.

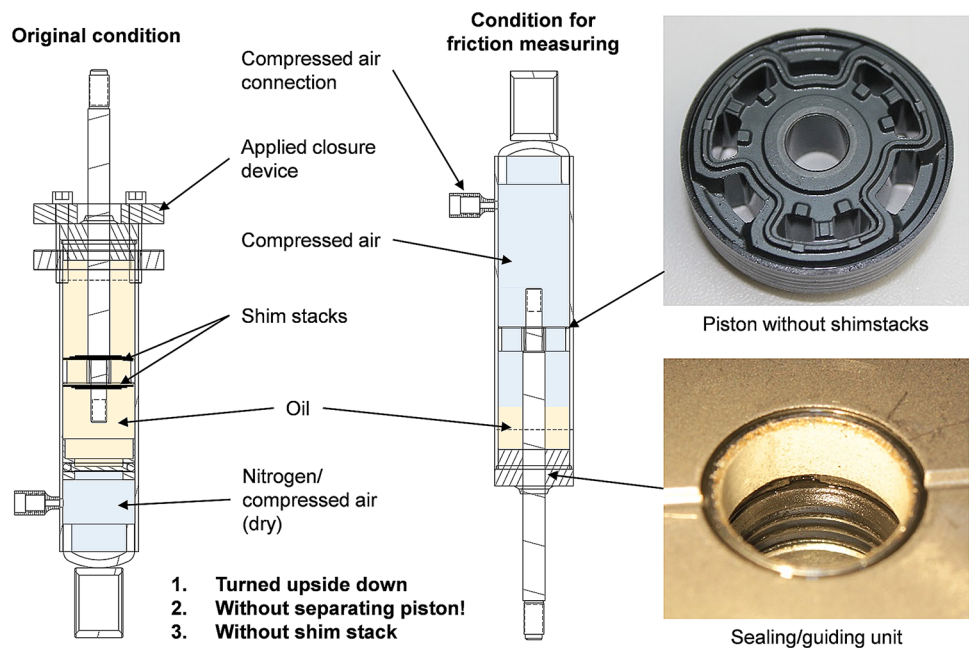
The aim of these modifications shown in Fig. 6 below is to reduce all other forces occurring to a minimum, apart from the friction forces to be measured. For this reason, the valve plating on the test damper is dismantled to largely eliminate flow losses due to the narrowing of the cross-section in the valves. Only the through holes represent a throttling point. To further minimize the flow resistance, the tests are carried out with a significantly reduced amount of oil, as can be seen in Fig. 6 (center).

As a result of the reduced amount of oil, the piston moves exclusively through air during the tests, which means that the flow resistance, particularly at high velocities of movement, is significantly reduced due to the considerably lower viscosity. To avoid inadequate lubrication of the sealing guidance unit, a small amount of oil is filled into the damper tube and the damper is installed in the “upside-down” arrangement at the test bench (see Fig. 5), so that oil is permanently at the contact surfaces between the piston rod and sealing guidance unit. To lubricate the contact surface between the damper piston and the tube, the damper is moved close to its rebound stop before each measurement, so that it is immersed in the aforementioned amount of oil. The damper is then moved close to the compression stop, as a result of which the tube is wetted with oil over the entire measuring stroke. The measurements themselves begin in the geometric center position

<sup>4</sup> Polytetrafluoroethylene.



**Fig. 6** Original condition and modification of the measured single-tube damper for friction measurements (schematically), details piston (without shim stack, right above) and seal guide package (right below)



of the damper and have a maximum amplitude of 50 mm. The amount of oil inserted is calculated so that the piston does not hit its surface in any measuring program. Preliminary investigations, not to be explained here, show that even when the test damper is operated for several seconds at a higher frequency, the frictional force does not increase, from which it can be concluded that the lubricating film remains intact at least for the duration of a measurement. The tube is wetted again before each measurement. It can also be seen in Fig. 6 (center) that the tests are initially carried out without the separating piston. This procedure is based on the fact that no separating piston is required for the construction in this form and, due to the significantly reduced oil quantity and the difference in compressibility between air and oil, it does not move as in the original state and would, therefore, have an indefinite negative effect on the measurements. However, since the separating piston has due to its short displacement a minor influence on the overall friction of the damper, the friction behavior of the separating piston will be examined separately in future tests. Furthermore, the test damper is provided with ball joints at its upper and lower connection points to the test bench, which allows an assembly stiff in the direction of force but free of tension in all other directions to prevent the effects of lateral forces.

## 5 Test procedure and results

For the precise, metrological characterization of the friction of the test damper described in the previous section, a number of different measurements are carried out on the

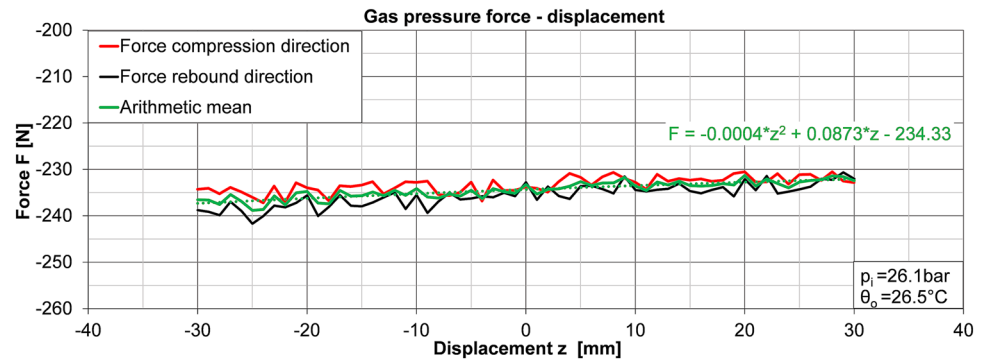
hydraulic test bench, the test sequence, boundary conditions and results of which are explained in more detail below.

### 5.1 Preliminary investigations

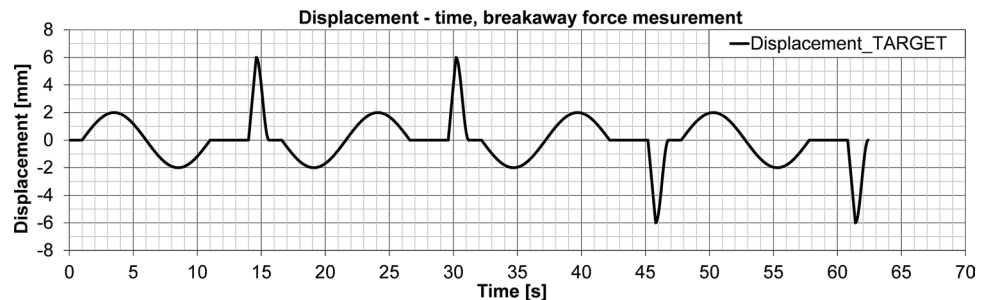
Since the test damper is opened for reconfiguration, the internal pressure must be re-established after reclosing, as this has a direct influence on the friction properties, especially of the sealing at the outlet. Furthermore, measurements to quantify this relationship between internal pressure and friction force require a variation of the internal pressure. Because of this internal pressure, there is a gas pressure force pushing the piston rod in direction of the rebound stop. Since this force is inevitably also measured during each measurement on the test bench, it must be eliminated from the total force measured. For this purpose, the gas pressure force is measured at least after every change in internal pressure or refilling, to increase accuracy even before individual measurements. The procedure for measuring the gas pressure force is almost standardized and done statically, i.e. step by step (step size: 1 mm) without any relative movement between the piston rod and the damper tube during the measurement.

Figure 7 shows the result of such a gas pressure force measurement as an example for a total stroke of 60 mm. The diagram also shows that the measurements are carried out in both the rebound and compression directions, and the gas pressure force characteristic curve results from the arithmetic mean of the two curves thus created. This eliminates any influence of the friction on the measurement of the gas pressure force even at a standstill (creeping of the seal,

**Fig. 7** Gas pressure force over damper displacement, static measurement of gas pressure force



**Fig. 8** Measurement program (displacement target) of the breakaway force examination



etc.). The fundamental characteristics of the gas pressure force over the displacement match the known characteristics of a fully assembled damper and thus the Boyle–Mariotte law, with a slightly lower gradient than the original damper. This can be explained with the smaller amount of oil and, therefore, the much larger gas volume in the modified test damper resulting in a smaller pressure change with the same volume change as in the original damper. There is a slight difference between the adjusted internal gas pressure affecting on the diameter of the piston rod and the measured gas pressure force of the whole damper. This difference is related to the weight of the damper tube with oil filling and every adaption up to the force sensor inevitable counteracting the gas pressure force while measuring with the intended sensor. This measured gas pressure force characteristic curve is subtracted from the measured force in the following dynamic investigations for friction measurement.

## 5.2 Main investigations

The main examinations for the characterization of friction follow the preliminary examinations. For this characterization, two different test series are carried out, which are explained in more detail below and the results are considered.

### 5.2.1 Breakaway force characteristics

In the first of these test series, the main focus is on examining the breakaway force of the damper, which is mainly influenced by the friction. These play a role both at the beginning of each damper movement and at the reversal points of cyclical movements. The measurement program shown in Fig. 8 below is used for this investigation, varying different parameters. The main part of this measuring program consists of sudden movements of the hydraulic cylinder in the form of a triangular ramp, in which the velocity of movement is kept constant over the entire displacement of 6 mm. The movement, like all the measurements presented here, starts in the geometric center position of the damper. The selected displacement results from preliminary examinations which showed that a steady state of the measured force is reliably achieved on this displacement with each measurement.

The triangular ramps are carried out both in the rebound direction and in the compression direction of the damper to be able to evaluate the directional dependence of the breakaway force. Furthermore, a defined state of the friction partners, in particular, the piston rod seal, is adjusted before the start of each triangular ramp. For this purpose, a sinusoidal displacement signal with an amplitude of 2 mm is inserted in front of each triangular ramp for preconditioning. This is done in such a way that with the first triangular ramp the last movement direction of the damper is the same direction as the following ramp, with the second in each case in the

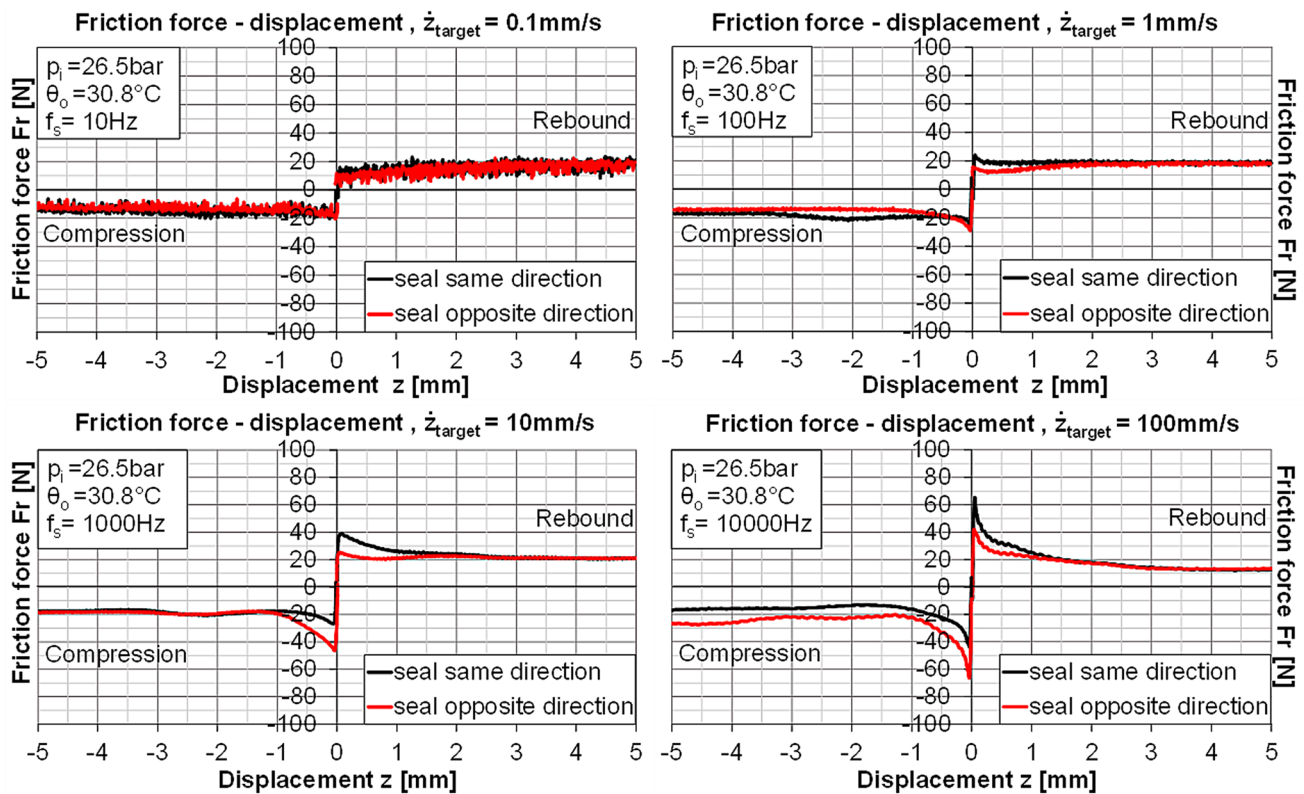


Fig. 9 Friction force over the damper displacement for different target velocities of a triangular excitation with constant velocity

opposite direction. In this way, in addition to establishing a defined initial state, a possible direction reversal behavior of the damper can also be considered. The parameter varied during this measurements is the target velocity of the triangular ramp function (0.1–100 mm/s). The sampling rate of the measured force is adjusted depending on the target velocity between 10 Hz and 10 kHz. Figure 9 below shows selected results of this investigation. The diagrams show the breakaway forces at different target velocities of the triangular ramp with a constant internal pressure of 26.5 bar (near original internal pressure). At the lowest possible movement velocity (left above in Fig. 9), no significant breakaway force can be seen, but here, it becomes clear that the control quality of the hydraulic cylinder still has room for improvement at very low velocities of  $< 1 \text{ mm/s}$ . Even at a target velocity of  $1 \text{ mm/s}$ , a slightly increased force can be seen immediately after the start of the movement (coordinate origin). This is important insofar as a friction measurement is usually carried out at this velocity in the case of completely assembled dampers, as a result of which this breakaway force has to be measured at the reversal points and interpreted accordingly. At the higher target velocities shown (below in Fig. 9), a significantly increased force is determined immediately after the start of the uniform movement, which can be seen here as a breakaway force. However, this breakaway force

is not fully present at the start of the movement, but only a short time later. Here, it becomes clear that damper friction is not pure Coulomb friction between solid bodies, which can be clearly distinguished in static and sliding friction, but rather elasto-hydrodynamic friction. After the piston rod and damper piston have traveled a short distance (approx. 2 mm at  $10 \text{ mm/s}$  and approx. 4 mm at  $100 \text{ mm/s}$ ), a steady state arises in which the frictional force remains at a constant, low level, similar to the values at lower velocities. A significant directional dependence (pulling or pushing direction) can hardly be recognized, only at  $1 \text{ mm/s}$  and  $10 \text{ mm/s}$  the frictional force in the compression direction is somewhat higher than in the rebound direction. This is the result of the asymmetrical structure of the seal and (to a lesser extent) the PTFE coating of the damper piston. The sealing used in hydraulic dampers is usually asymmetrical to its function to keep the pressurized oil inside and dust or other environmental influences outside. A dependence of the breakaway force on the direction last traveled (preconditioning) can be seen in particular at the higher velocities in the two lower diagrams in Fig. 9. The frictional force is higher in the direction of rebound when the damper was last moved in the same direction. In the direction of pressure, the frictional force is exactly the opposite in relation to the sealing preconditioning. This can be explained on one hand by the asymmetry

of the elastic friction partners, in particular the seal, on the other hand by the fact that the preconditioning results in a minimal tension in the seal, which results in an offset of the force already in the starting position of the movement.

However, the gas force compensation must be observed in detail. This means, that between measuring the gas force and compensating it in the Friction measuring there is a certain time, in which the gas pressure force can slightly change (e.g. due to minimal changes in temperature) with larger impact on the results of the friction measuring. Also, the minimal tension of the seal due to the preconditioning are at the moment not exactly reproducible which leads to minimal differences in friction force between two following measurements. On this topics, there are further investigations and a resulting improvement of the test procedure and the test bench needed and planned. This could be e.g. a real time pressure measuring with high resolution and accuracy, which is not provided yet. It should also be mentioned that the velocity of the hydraulic piston cannot increase analogous to the ideal triangular ramp immediately from standstill to the target velocity without delay. Related to the inertia of the moving masses from e.g. the piston of the hydraulic cylinder there is only a finite acceleration possible at the beginning of the movement, as a result of which a more harmonious increase in velocity than with an ideal triangular signal results. As a result of that, the breakaway force tends to be lower than with an ideal ramp, but is in every measuring related to the measured velocity of the hydraulic piston on which the tested specimen is mounted with a very high stiffness. Relative movement between the specimen and the testbench except the wanted one can be excluded for this results. Nevertheless, the acceleration at the beginning of the movement, especially at great target velocities, is very high, which means that the test bench is operated in a highly unsteady manner. Future studies will quantify the impact of high acceleration on the measurement result with the use of e.g. accelerometers. The measurement with the target velocity of 1000 mm/s is not shown and evaluated here, since in this case force signal oscillations occur from the shock-like excitation of the spring-mass system consisting of damper tube and adapter mass as well as the rigidity of the force sensor due to the high target velocity.

### 5.2.2 Friction force characteristics over wide velocity range

The investigations presented below serve to characterize the friction of the experimental damper over a large range of movement velocity. The basis for the selection of the velocity range and the measured amplitudes are the specifications of the VDA<sup>5</sup> damper test, expanded by support points for better resolution of discontinuities in the course of the

frictional force. In contrast to the previous measurement program, a sinusoidal displacement signal with constant amplitude and variable frequency is run, which results in different movement velocities. However, only the maximum of the velocity per cycle is used as the design variable. The amplitude of the measurements is 25 mm and the velocity at the zero crossing varies from 1 mm/s to approx. 1800 mm/s. The friction force measurement is also carried out in the displacement zero crossing of the movement, i.e. at the highest velocity per cycle. In this way, the frictional force can be measured without being influenced by the breakaway forces, since the respective measuring point is sufficiently far from the reversal point of the movement (see breakaway force measurement → steady state). One settling cycle and three measuring cycles are carried out at low velocities, three settling cycles at high velocities and five measuring cycles immediately afterwards. The measured friction force is averaged over the number of measuring cycles. In this series of measurements, there is also a variation of the internal pressure to determine the influence of this on the frictional force of the test damper. The diagrams in Fig. 10 show the frictional force of the experimental damper versus the velocity of movement. For better comparability, the frictional force in the pressure direction is shown as absolute values, since the measured values have a negative sign due to the opposite direction of force. Furthermore, the force measurement values are not compensated by the gas pressure force measurement described at the beginning (as in the other measurements shown here) but are normalized to the force measurement value at a selected reference velocity of 100 mm/s. The formulas (1) to (3) show the calculation of the normalized friction forces in rebound,  $F_{r,reb,norm}(\dot{z})$  and compression direction  $F_{r,comp,norm}(\dot{z})$ .  $F_{norm}(100 \text{ mm/s})$  represents the force at the relative velocity of 100 mm/s by which the measured values (gas force and frictional force)  $F_{comp}(\dot{z})$  and  $F_{reb}(\dot{z})$  are supplemented, with the gas force being compensated accordingly.

$$F_{r,reb,norm}(\dot{z}) = F_{reb}(\dot{z}) - F_{norm}\left(\frac{100 \text{ mm}}{s}\right), \quad (1)$$

$$F_{r,comp,norm}(\dot{z}) = \left| F_{comp}(\dot{z}) - F_{norm}\left(\frac{100 \text{ mm}}{s}\right) \right|, \quad (2)$$

$$F_{norm}(100 \text{ mm/s}) = F_{comp}(100 \text{ mm/s}) - \left( \frac{F_{comp}(100 \text{ mm/s}) - F_{reb}(100 \text{ mm/s})}{2} \right) \quad (3)$$

The background to this procedure is that the results shown in Fig. 10 react very sensitively to even slight deviations or changes in the gas pressure force (e.g. as a result of temperature changes) and a direct comparability of the

<sup>5</sup> Verband der Automobilindustrie.



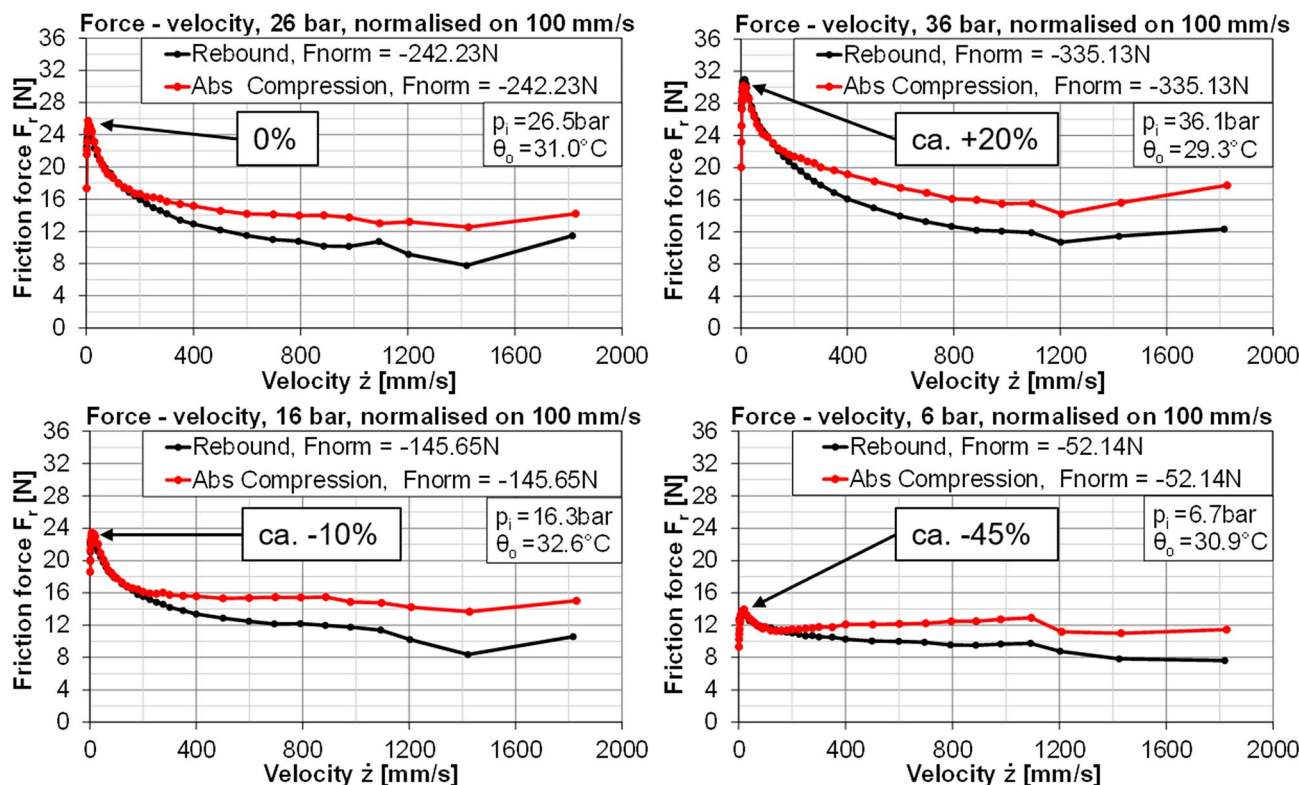


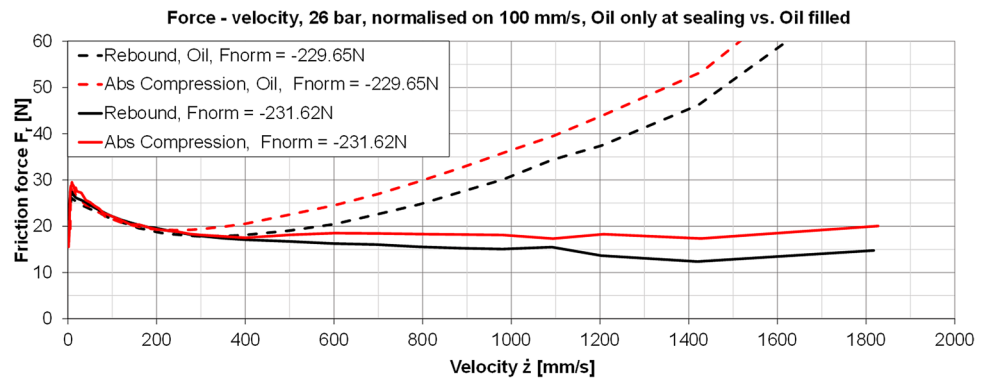
Fig. 10 Frictional force over movement velocity at different internal pressures

curves is not possible in this way. The reference velocity was determined after preliminary evaluation of the raw data, since a good approximation of the measurement results of both force directions can be seen in the force curve. At this velocity, both vibration-related inconsistencies (at higher velocities/frequencies) and control inaccuracies (at lower velocities/frequencies) of the test bench reach a minimum. Since the measurements are only made in the geometric center position of the damper, the course of the gas pressure force over the stroke is irrelevant here, so that compensation by a fixed value (noted in the respective diagram in Fig. 10), which corresponds to the gas pressure force at the normalizing point, is legitimate. The graphs in Fig. 10 show a course that is very similar to that of a Stribeck curve. This course is fundamentally to be expected in the case of hydrodynamic friction, but it differs from the curve known from plain bearings that there is no pronounced boundary or static friction at very low velocities. On the contrary, the friction force at low velocities initially shows very low values, which reach a global maximum at movement velocities of between 5 and 20 mm/s depending on the series of measurements and then, like the Stribeck curve, decrease again with increasing movement velocity. The discernible discontinuities of the curves around 1000–1200 mm/s are the result of resonance vibrations of the entire test bench on its elastomer

bearings. As mentioned before a variation of the internal pressure can be seen in the diagrams below (Fig. 10). That variation serves to figure out the obviously major impact of that particular property of the damper on its frictional behavior in detail and numerical value. The major interest on this property has two reasons. The first one is to be able to anticipate the friction while designing a damper and, therefore, determine the gas pressure in relation to the calculated loads and the construction design of the damper, especially the diameters of the piston rod and tube. The second one has its reason in the fact, that the internal pressure of a damper is constantly changing while being used in a vehicle under cyclical and fluctuating load and knowing the correlation between internal pressure and friction makes it possible to quantify the frictional behavior in the field of application.

The internal pressure variation starting from the original internal pressure (left above in Fig. 10) shows an increase in the maximum friction force by 20% when the pressure increases by 10 bar (right above) and a reduction in friction force by approx. 10% when the pressure is reduced by 10 bar. A further reduction in the internal pressure by 20 bar relative to the original internal pressure results in a reduction of 45% (based on the maximum frictional force). It can be concluded from this values, that the frictional force varies constantly during normal operation of the damper between

**Fig. 11** Frictional force over velocity, comparison with (dashed) and without original oil quantity (each without valve plating)



approximately 20% above and 45% below the static measured value related to the maximum frictional force. In addition to the influence on the maximum frictional force it can also be seen that the pressure as well has an impact on the friction at any other value of the displacement velocity but to a lesser extent. This relation between the frictional force, the displacement velocity and the internal pressure is important for understanding the influence of the friction on normal operation of a vehicle. This cannot be provided by the standard friction measuring with very low displacement velocities and under constant internal pressure since during operation of the damper, the displacement velocity and therefore the load and with that the internal pressure are changing constantly in a large range. The background of the correlation between internal pressure and frictional force can be found in the sealing of the damper, which is pressed more powerful to the piston rod with increasing internal pressure. The possible influence of a pressure gradient between the two sides of the piston on the frictional force cannot be investigated with the described setup due to the missing oil and valves. Another conclusion from the results of the pressure variation can be found in the possibility to lower the frictional force by lowering the internal pressure. Since the internal pressure of a monotube damper serves not only to prevent the formation of bubbles under high loads but also to support the compression force, this cannot be reduced as desired without, for example, increasing the effective piston area (damper tube diameter). An increase in the damper tube diameter by, for example, 10 mm in the case of the experimental damper would allow a possible reduction of the internal pressure by approx. 10 bar at a constant maximum damper force of 2650 N in the highest dimensioning point of the VDA cycle. This would result in an average reduction of the frictional force of approx. 2.8% over the entire velocity range and a maximum reduction of 10% at the highest frictional force. Furthermore, the pressure variation gives a qualitative insight into the friction in the fully assembled damper (original condition), because the pressure and thus the friction can vary briefly, depending on the level and direction of the damping forces, but is not or only with great effort

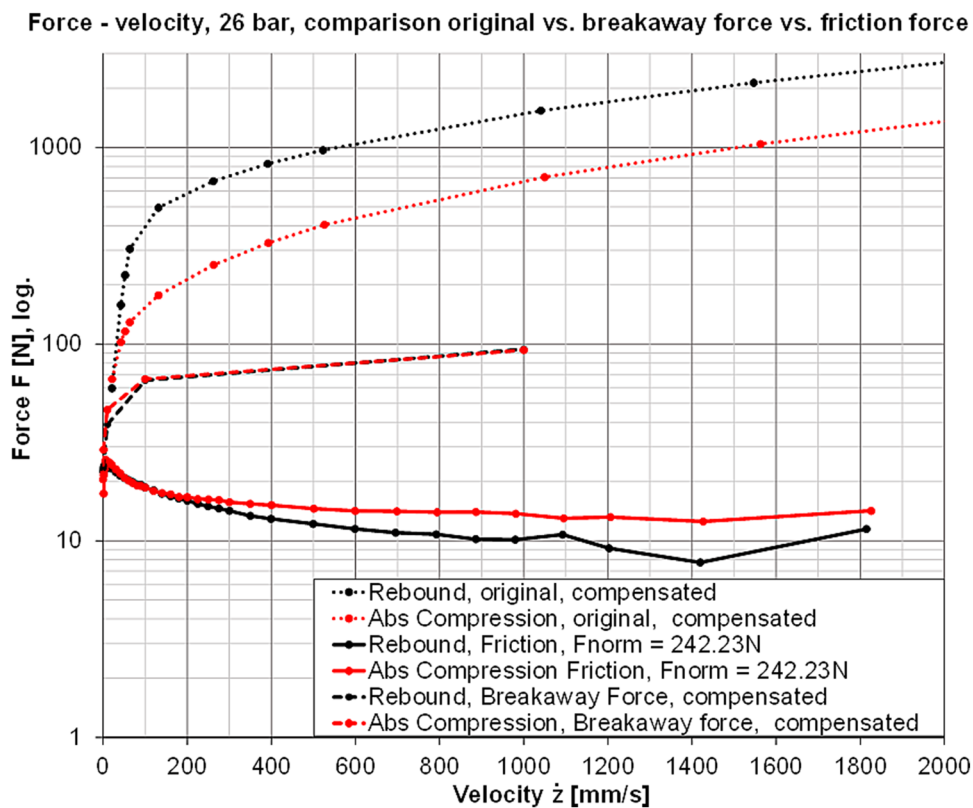
measurable in the original condition and under dynamic load. Moreover, it can be seen in the diagrams from Fig. 10 that the frictional force tends to decrease from its maximum with increasing velocity, which at least up to the highest measured velocity also does not correspond to the classic course of the Stribeck curve, which increases again towards higher velocities. The course showed in Sect. 3 above (Fig. 4, right) is quite similar to the one shown here, but shows this increase towards higher velocities. This increase can, however, be due to a measurement with the original oil filling, because in this case hydraulic pressure losses and thus damping forces occur despite the valve plating being removed, as shown in the following diagram in Fig. 11.

The diagram above shows the friction force measurement results of the test damper both with the standard oil fill quantity (dashed lines) and with the reduced oil fill quantity for the tests shown here. Here, it becomes clear that from a relative velocity of 200–300 mm/s there is a significant increase in the measured force, which can be attributed to the hydraulic forces due to pressure loss at the piston even without valve plating. The discrepancies in the characteristic curves for the direction of rebound and compression at velocities above approx. 200 mm/s, which can be seen in this section, are attributable to the gas pressure force in the damper tube, since these cannot be observed with the system open to the environment. The precise analysis of this problem is the subject of future studies.

## 6 Conclusion

As part of the investigations presented here, the friction properties of a commercially available monotube damper were characterized in various theoretical excitation scenarios. In terms of signal form, amplitude and velocity, these are based on real excitation scenarios both in the damper test and during driving. Thus, the frictional properties of a monotube damper with regard to the breakaway force in the case of shock-like excitations and the frictional force over a wide velocity range (cf. VDA cycle) can be quantified using

**Fig. 12** Comparison of damping forces, breakaway forces and friction forces for the test damper



the results obtained. For this purpose, a process was developed which, based on a significantly reduced amount of oil in the damper in combination with removed valve plating, enables the detection of the frictional forces without disturbing influences of hydraulic forces on the damper piston. The results confirm this approach, since measurements with oil filling, even without valve plating, are prone to errors at least at higher velocities ( $> 200$  mm/s). It can also be found that, as soon as the damper is moving, the frictional forces did not reach a maximum at the lowest velocity (see standard friction force measurement and Stribeck curve), but only at low to medium damper velocities (5–20 mm/s). In contrast, with shock-like excitations of the damper from the rest position, frictional force maxima occur, which significantly exceed those of the frictional force with almost uniform movement and are relevant for the transmission behavior of the damper, since shocks from the wheel-road contact are transmitted more strongly to the body. It should be noted here, however, that the hydraulic damping forces in the normal, oil-filled damper also increase significantly, particularly in the case of rapid, impulsive movements. A further finding of the investigations is the clear influence of the internal damper pressure on the friction forces, which could be determined with the help of the results, at least for the type of damper used, in numerical values. Figure 12 finally shows the determined frictional forces in relation to the damping force of the experimental damper in its original state. The logarithmic

representation of the forces clearly shows that the breakaway force above 10 mm/s clearly exceeds the friction force in motion.

In principle, however, the friction forces that occur generally remain well below the damping forces, so that, as expected, their influence is to be rated as low here. In the case of dampers with a softer characteristic curve, variable dampers and frequency-selective dampers etc., the influence of the friction in relation to the damping force can be considerably more pronounced.

## 7 outlook

In the course of the investigations presented here, individual questions have arisen which, as already indicated in part, will become the subject of further test series. First of all, the frictional force investigation at high excitation frequencies and low amplitudes should be mentioned. Furthermore, the directional dependence of the frictional force, which cannot yet be fully described with the available measurement results, will be the subject of further investigations. In this context, a more precise measurement of the gas pressure force is required. Since the measurement results are sensitive to even minimal movements in the submillimeter range, instead of measuring the gas pressure force (as described here), the aim is to calculate it. In this case, an increase in

the gas volume by connecting an additional volume is also to be assessed as practicable, since pressure changes due to changes in the stroke of the damper can thereby be considerably reduced. This has two advantages: on the one hand, the change in gas pressure is minimized over the stroke and, on the other hand, the pressure influence on the friction remains constant over the stroke. By increasing the gas volume, the measurement inaccuracy regarding the discrepancy of the force values between the rebound and compressive direction at high velocities ( $> 200$  mm/s) is also eliminated with high probability, since these are not found in the unpressurized system that is open to the environment. Furthermore, future investigations will deal with the influence of the separating piston on the frictional forces in the monotube damper, since this was not initially considered in the investigations presented here for the reasons already mentioned. In addition, future studies will examine the influence of the surface quality and material of the friction partners. In this context, measurements of the surface roughness of the test damper that have already been carried out, using the piston rod as an example, show that its surface differs significantly between almost unused areas with  $R_z \approx 0.6$  and areas of frequent contact with the sealing guidance unit with  $R_z \approx 0.2$ . Another subject of investigation will be the dependence of the frictional force on other boundary conditions than the ones mentioned here. The test bench behavior itself is examined more closely and further optimized, particularly at high accelerations (breakaway force tests) and very slow movements of low amplitudes. A supplementary object of investigation will also be the influence of lateral forces on the damper in the real chassis on its friction properties. To classify the influence of frictional forces on driving comfort, both simulation-based investigations adapted from the data obtained here for validation are sought, as well as driving tests. The results generated in this way then allow a statement as to the area in which the frictional force has a noticeable influence on the vibration and noise behavior of the overall vehicle. Conversely, there is the possibility of designing the friction in a targeted manner and at an early stage in the development process, to avoid unnecessary effort to reduce friction and, at the same time guarantee a very high level of occupant comfort.

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