Hydropneumatic Suspension Systems

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Chapter 2
Spring and Damping Characteristics of Hydropneumatic Suspension Systems

2.1 General Setup and Working Principle

The simplest hydropneumatic suspension system consists of only three components: a hydraulic cylinder, a hydropneumatic accumulator, which is directly mounted on the cylinder and, of course, the hydraulic fluid. In case cylinder and accumulator need to be separated – for example due to design space reasons – additional oil lines and fittings are necessary to provide the hydraulic connection.

After adjusting the hydraulic pressure to the required level (by adding or releasing hydraulic fluid) this system now already provides the suspension function. When displacing the piston rod, the fluid volume in the accumulator is changed and therewith the pressure ($p_1 \rightarrow p_2$). This causes a change of the force at the piston rod which, in combination with the change of the position, defines the spring rate c.

The external spring force F_F which acts upon the piston rod is always in balance with the forces resulting from the pressures onto the piston, when neglecting inertial and friction forces (Fig. 2.1a).

When the force F_F is increased to $F_F{}^*$ (Fig. 2.1b) the position of the piston changes (Δs) and therefore some hydraulic fluid is displaced into the accumulator. This change proceeds until the pressure in the accumulator (and thus on the active surface of the piston) has reached a level which again provides a balance for the system. This balance of forces is the basis for the function and the understanding of the suspension system. It will be used in the following sections for further calculations.

To allow for additional damping, a flow resistor is placed between cylinder and accumulator. It converts part of the kinetic energy of the hydraulic fluid into heat (viscous friction). This provides the desired damping in combination with the (undesirable) boundary friction caused by the cylinder sealing and guiding elements. This so called "suspension unit" consisting of cylinder, accumulator, flow resistor and hydraulic fluid already provides the suspension function and could replace the typical combination of mechanical spring and damper.

Yet with this system the major advantage of hydropneumatic suspension systems is not yet used: level control. An additional level control unit provides a constant normal position of the suspension independent from the static spring load F_F. The

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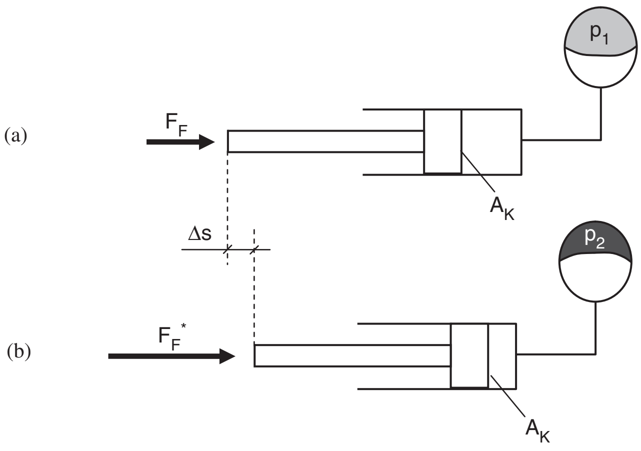


$$F_F = A_K \cdot p_1 \qquad F_F^* = A_K \cdot p_2 \qquad c = \frac{F_F - F_F^*}{\Delta s}$$

Fig. 2.1 Balance of forces at the piston of a single-acting cylinder

level control unit consists of a position sensor which, directly or via an electronic control unit, sends signals to a hydraulic control valve, which then changes the amount of hydraulic fluid in the suspension unit in order to bring the suspension back to the design position if necessary. By increasing the amount of hydraulic fluid, the level of the system is increased; reducing the amount of hydraulic fluid

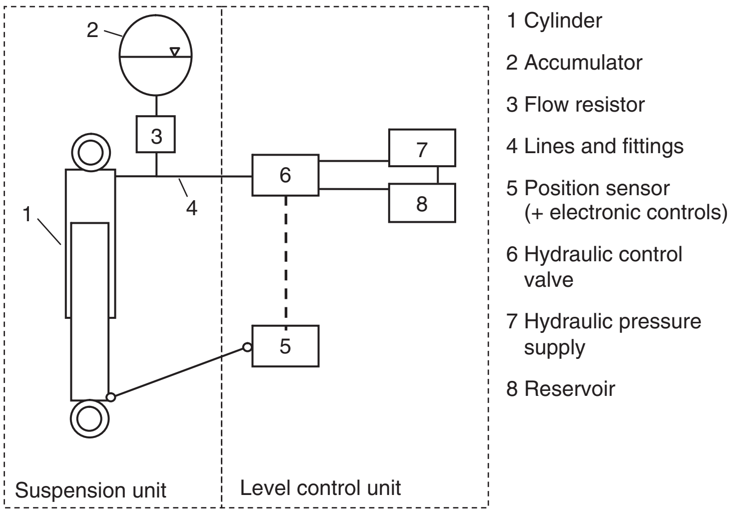


Fig. 2.2 General setup of a hydropneumatic suspension system

decreases the level of the system. Pressurized hydraulic fluid as well as the possibility to dispose of excessive fluid (to a hydraulic reservoir) need to be provided to enable that.

By combining these two parts (suspension unit and level control unit) it is possible to draw a basic schematic of a hydropneumatic suspension system (Fig. 2.2).

Section 2.2 describes two main functions of a hydropneumatic system: spring characteristics and damping characteristics regarding their basic principles and theoretical background, while Chap. 3 describes how predefined characteristics can be achieved with a certain component layout. The third main function, the level control, is described in more detail in Chap. 5.

2.2 Spring Characteristics

The spring rate of a hydropneumatic suspension system can be determined from the pure spring force–displacement curve measured at the suspension cylinder when the hydraulic flow resistor, as shown in Fig. 2.2, is removed. An increase of force onto the cylinder leads to an increase in hydraulic pressure and therefore to a change in position of the piston rod. This is due to the following reasons:

- compression of the gas in the accumulators
- widening of the (elastic) fluid lines and fittings
- compression of the hydraulic fluid

Each of these three effects causes an individual spring rate. So what is measured at the suspension cylinder is the spring rate of a spring which is made up by a sequential combination of these three individual springs. Using general laws of physics, the following Eq. (2.1) is obtained:

$$c_{\mathrm{ges}} = \frac{c_{\mathrm{G}} c_{\mathrm{L}} c_{\mathrm{F}}}{c_{\mathrm{G}} c_{\mathrm{L}} + c_{\mathrm{G}} c_{\mathrm{F}} + c_{\mathrm{L}} c_{\mathrm{F}}} \tag{2.1}$$

The stiffness of the lines and fittings as well as the compression modulus of the hydraulic fluid are usually very high, so their impact on the overall spring rate c_{ges} is low. This means that the characteristic properties of a hydropneumatic spring are mainly influenced by the properties of the gas which is enclosed in the hydraulic accumulators. So before explaining the detailed function of the various types of hydropneumatic suspensions, the properties and the thermodynamic physics of gases and their effect on the suspension function is explained.

2.2.1 Thermodynamic Background

The gas in the accumulator(s) is the medium which is responsible for the elasticity of the complete setup. Because it fulfills the most important task (spring rate) its properties are predominantly important for the behavior of the whole suspension system.

In the initial state of an accumulator – with an unpressurized hydraulic system – a certain number of gas molecules and therefore a certain gas mass m_G is trapped inside the accumulator. It is defined by the volume of the accumulator V_0 (= gas volume if no hydraulic pressure is applied) and the accumulator precharge pressure p_0. The precharge pressure always refers to the room temperature 293.15 K (or 20°C) and is set during the production process of the accumulator. For this condition the equation of state for the ideal gas is:

$$p_0 V_0 = m_G RT \tag{2.2}$$

If the gas temperature changes during the production process (for example during paint-drying), during shipping or later during the operation of the suspension system, the gas pressure changes to a new precharge pressure according to the laws for the isochoric change of state. This temperature-dependent precharge pressure must be taken into account when laying out systems that are will be operated at various temperatures.

$$p_{0,\mathrm{T}} = p_0 \frac{T}{T_0} \quad \text{(important: } T, T_0 \text{ in [K])} \tag{2.3}$$

As soon as the accumulator is integrated into the hydraulic system and the system is pressurized, the gas volume in the accumulator does not change as long as the hydraulic pressure is less than or equal to the precharge pressure. As soon as the hydraulic pressure exceeds the precharge pressure, the gas volume is compressed until a balance of forces or, for equal active areas on gas and fluid side as in diaphragm accumulators, a balance of pressure is attained. The increase of hydraulic pressure can be caused for example by loading the suspension system with the suspended mass. The compression of the gas takes place rather slowly and the new pressure level is maintained over a longer period, so in this case an isothermal change of state according to Boyle-Mariotte can be assumed. The heat generated by compression dissipates into the environment and the temperature remains constant during the process.

$$V_1 = V_0 \frac{p_0}{p_1} \tag{2.4}$$

This isothermal change of state can be used for the calculation of both the first time loading of the suspension system as well as all subsequent slow load changes: for example people getting on and off the suspended system, loading and unloading of payload, long term changes in external preloads and forces.

The suspension movement itself, when absorbing the shocks during the regular operation of the suspension system, takes place quite rapidly. The excitation frequencies that the suspension system is able to absorb usually range from below 1 Hz to sometimes over 10 Hz. These high speed changes of state leave little time for the dissipation or adsorption of heat compared to an isothermal change of state as described above. The gas will therefore change its temperature. Assuming that no heat exchange is possible, an adiabatic change of state takes place which is described by the following equation:

$$p_1 V_1^{\kappa} = p_2 V_2^{\kappa} \tag{2.5}$$

In this equation κ is the adiabatic exponent, which is the ratio of the specific heat capacity at constant pressure and the specific heat capacity at constant volume for a particular gas. In standard literature values are quoted that refer to the properties at low pressures and room temperature. These values are for example according to [DUB90]:

$\kappa \approx 1.66$ for monoatomic gases (e.g. He)
$\kappa \approx 1.40$ for biatomic gases (e.g. N_2, O_2 and therefore also air)
$\kappa \approx 1.30$ for triatomic gases (e.g. CO_2)

Although it is rarely mentioned, for hydropneumatic suspension systems it is very important to consider that κ depends significantly upon temperature and gas pressure. Figure 2.3 gives an overview of the behavior for nitrogen.

In real hydropneumatic suspensions there is always the possibility of a marginal heat exchange of the gas with its surrounding components and therefore there will never be the ideal adiabatic change of state. This means that hydropneumatic suspension processes are defined by a polytropic change of state characterized by $1 < n < \kappa$. The more heat is exchanged during the change of state, the more the polytropic exponent n for this process will go from κ towards 1; the latter defines

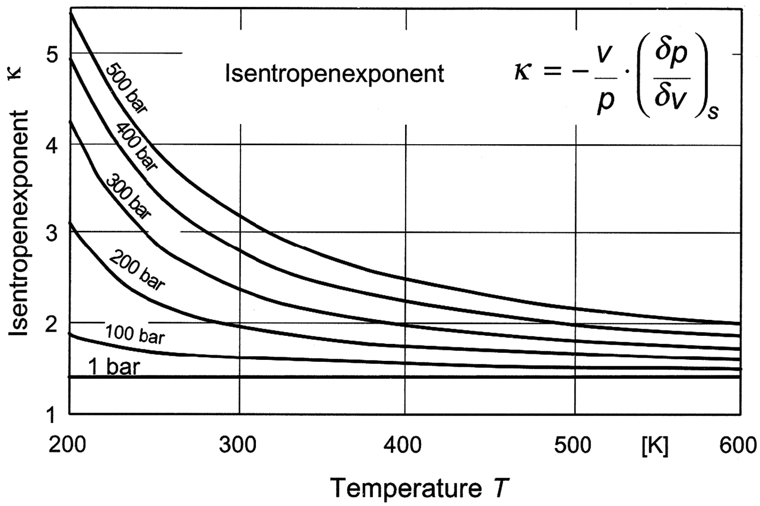


Fig. 2.3 Adiabatic exponent κ of N_2 as a function of T and p ([MUR01])

again the isothermal change of state with perfect heat exchange. The exact conditions for heat exchange are usually unknown and are very difficult to identify, which is the reason why it is extremely difficult to find out where exactly between 1 and κ the polytropic exponent needs to be chosen. Furthermore even the actual value for κ is difficult to determine due to the above mentioned effects of pressure and temperature onto κ; even more so because both parameters change constantly during the suspension processes. That is why it is only possible to estimate the polytropic exponent n for preliminary calculations.

Figure 2.4 shows how much influence n has on the pressure–volume diagram. Starting from a pressure of one bar the gas is compressed. The resulting pressure can be read from the curve progression. It is clearly visible that the pressure and therefore also the force of the cylinder increases with increasing polytropic exponent. A strong impact on the spring rate can directly be deduced.

Assuming a value of $n = 1.3$ usually works well for preliminary calculations. Only at higher pressures especially in combination with very low operating temperatures it is recommended to use $n = 1.4$ or higher (according to the behavior shown in Fig. 2.3). The most realistic average values for n can be deduced from measurements of force–displacement curves in experiments. Calculations need to be compared to the experiments and the polytropic exponent of the calculations needs to be readjusted to a level which provides best match of theoretical end experimental force–displacement curves. This empirically determined value for n can in the future be reused for the calculation/simulation of hydropneumatic systems with similar setups, especially concerning accumulators and their environment.

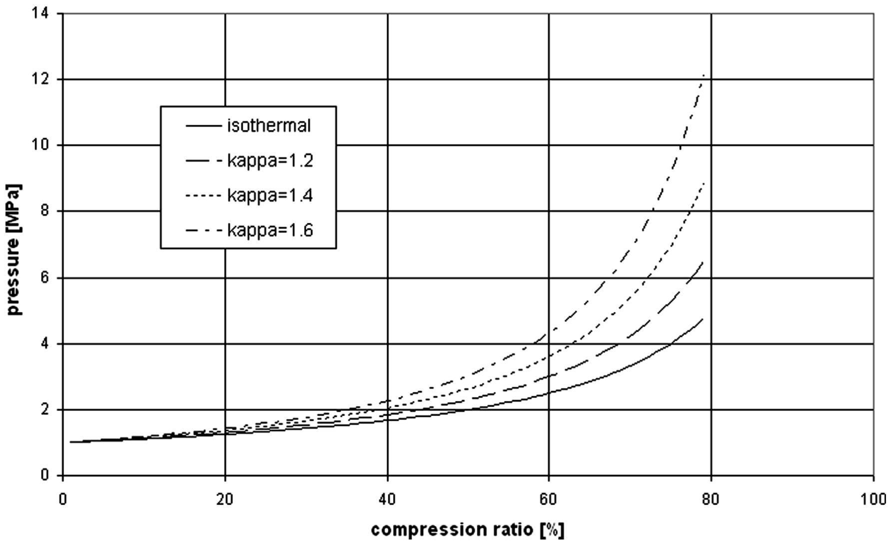


Fig. 2.4 p–V curves for different polytropic exponents

2.2.2 Calculation Predeterminations

All the calculations in this section are done under the following assumptions and conditions:

(1) The influence of the ambient pressure is neglected. This is acceptable, since the usual working pressure levels in the cylinders as well as the precharge pressures of the accumulators (commonly specified as gauge pressure above the ambient pressure) are significantly higher than the ambient pressure – usually a factor of 50 and above for the working pressure and a factor of 25 and above for the precharge pressure. If this is not the case for special hydropneumatic suspension systems it is necessary to reconsider the question whether the influence of ambient pressure is really negligible. If not, it is necessary to multiply the ambient pressure with the externally effective active cylinder area (usually the piston rod cross-sectional area) and use this force value as a preload force without additional spring rate – please refer Sects. 2.2.3, 2.2.4, 2.2.5, and 2.2.6. Furthermore in this case the absolute accumulator precharge pressure (=specified precharge pressure + ambient pressure) needs to be used for the calculations.
(2) For better comparison of the results and behaviors, all calculations are performed for a temperature of 293.15 K (or 20°C) – particularly important for the accumulator. Therefore the specified accumulator precharge pressure (referring to 293.15 K) can be directly used for the calculations. If the suspension behavior for other temperatures has to be calculated, it is necessary to first calculate the altered, temperature-dependent precharge pressure (see Sect. 2.2.1) and then use this value in the respective equations.
(3) A polytropic exponent of 1.3 is used.
(4) The suspended mass directly acts upon the suspension cylinder ($i = 1$). This means there is no mechanical linkage system that creates any kind of lever ratio of $i \neq 1$: cylinder force to gravitational force of the mass. This is often not the case for suspension systems but again for comparison of results this is assumed. Yet in Chap. 7 an example is given which explains the influence of a lever ratio $i \neq 1$.
(5) The design position of the suspension is defined to be exactly in the center of the overall stroke between the compression and rebound end stop.
(6) Damping (solid body and fluid friction) is not part of the calculations in Sect. 2.2. All hydraulic pressures do not include pressure losses (e.g. caused by flow restrictors). The focus here is purely on spring characteristics.

2.2.3 Non-preloaded Hydropneumatic Suspensions

This is the simplest type of hydropneumatic suspension. This system consists of a single acting suspension cylinder and an accumulator. The suspension cylinder

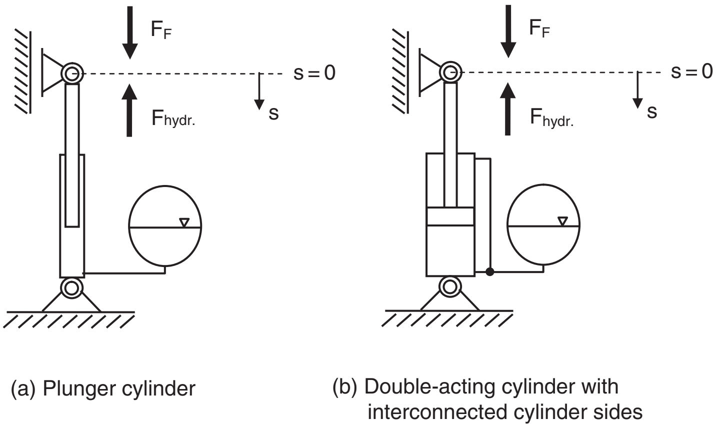


Fig. 2.5 Schematic illustration of non preloaded hydropneumatic suspensions

can be designed as a single-acting cylinder (for example, a plunger cylinder) or as a double-acting cylinder with interconnected pistonside and rodside of the cylinder. The latter system is able to provide higher amounts of rebound damping (more detailed information in Sect. 2.3). Both systems are depicted in Fig. 2.5.

It is important to consider that the externally effective active area is only the cross-sectional area of the piston rod. It is due to the interconnection of piston chamber and rod chamber (the so-called regen(erative) system) that effectively only the fluid volume displaced by the piston rod flows into the accumulator while the other portion of the fluid displaced by the piston flows back into the rodside.

The most important method to describe the behavior of a spring is the force–displacement curve for compression and rebound. It has been mentioned in Sect. 1.1 already that this curve is basically linear for a regular mechanical coil spring; other curves are possible by special winding techniques as well as parallel connection of multiple springs. The hydropneumatic spring however always has a disproportionately progressive shape of the force–displacement curve. This shape can be controlled by variation of several influencing factors. The important factors are deduced on the following pages.

Before calculating the non preloaded hydropneumatic suspension it is necessary to define some of the various states that a suspension system can be in:

State 0: Spring force $F_{F0} = 0$. The pressure in the accumulator is the precharge pressure p_0, which is defined during the production process. The gas fills out the complete internal volume V_0 of the accumulator.

State 1: Now the static suspension force F_{F1} is loading the suspension system (while $F_{F1} > F_{F0}$). The force is sufficient to compress the gas volume in the accumulator isothermally to the volume V_1 and the pressure p_1.

State 2: F_{F2} is the dynamic suspension force and oscillates around F_{F1}. Therefore the gas volume is compressed (compression) and expanded (rebound) by a polytropic change of state to the volume V_2 and the pressure p_2.

The starting point for the calculation is the correlation of the force acting onto the surface of the piston F_K and the pressure in the piston chamber p_K.

$$F_K(s) = p_K(s)A_K \tag{2.6}$$

Using the state equation for polytropic changes of state

$$p_1 V_1^n = p_2 V_2^n \tag{2.7}$$

and the definition that an increase of the displacement s causes a compression of the gas

$$V_2 = V_1 - A_K s \tag{2.8}$$

it can be deduced that:

$$p_2 = \frac{p_1 V_1^n}{V_2^n} = \frac{p_1 V_1^n}{(V_1 - A_K s)^n} \tag{2.9}$$

On the basis of the isothermal change of state from 0 to 1 it is stated

$$p_1 V_1 = p_0 V_0 \tag{2.10}$$

and therefore

$$V_1 = \frac{p_0 V_0}{p_1} \tag{2.11}$$

As well as on the basis of the balance of forces at the piston

$$F_{\mathrm{F1}} = p_1 A_K \tag{2.12}$$

and thus

$$p_1 = \frac{F_{\mathrm{F1}}}{A_K} \tag{2.13}$$

For the following calculation it can be applied that

$$p_K(s) = p_2 \tag{2.14}$$

Combining all above equations brings us to

$$F_K(s) = \frac{\frac{F_{\mathrm{F1}}}{A_K} \cdot \left(\frac{p_0 V_0}{\frac{F_{\mathrm{F1}}}{A_K}}\right)^n}{\left(\frac{p_0 V_0}{\frac{F_{\mathrm{F1}}}{A_K}} - A_K s\right)^n} A_K \tag{2.15}$$

After canceling A_{K}

$$F_K(s) = F_{\mathrm{F1}} \frac{\left(\frac{p_0 V_0}{F_{\mathrm{F1}}}\right)^n}{\left(\frac{p_0 V_0}{F_{\mathrm{F1}}} - s\right)^n} \tag{2.16}$$

by the substitution

$$\frac{p_0 V_0}{F_{\mathrm{F1}}} = h_{0\mathrm{F}} \tag{2.17}$$

the following simple relationship is obtained:

$$F_K(s) = F_{\mathrm{F1}} \cdot \frac{h_{0\mathrm{F}}^n}{(h_{0\mathrm{F}} - s)^n} \tag{2.18}$$

The dimension $h_{0\mathrm{F}}$ can be interpreted easily by a virtual image. It is equivalent to the height of a column of gas with the pressure p_0 (the precharge pressure) and the volume V_0 which has exactly the right base area so that it supports the force F_{F1}. Figure 2.6 depicts this for several different static spring loads F_{F1}.

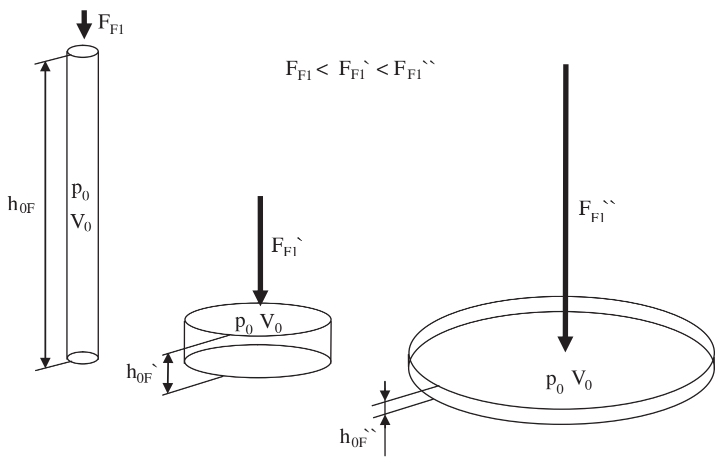


Fig. 2.6 $h_{0\mathrm{F}}$ for various static spring loads F_{F1}

This illustration demonstrates one of the most important features of a hydropneumatic suspension: the higher the static springload, the smaller the height of the gas column h_{0F} and therefore the more significant the change of the gas pressure forces on the piston at a given displacement s which then means a higher spring rate. The simple background to this is the relative change of the column height $(h_{0F} - \mathrm{s})/h_{0F}$; it becomes more meaningful with smaller h_{0F} and therefore the relative volume decrease and the relative pressure increase are more significant. This is the explanation of the increasing spring rate with increasing static springload for a hydropneumatic suspension.

In case a hydropneumatic suspension is subjected to a wide range of static springloads, another important characteristic curve needs to be considered: the dependency of the spring rate on this very static springload. This is deduced by the following calculation.

The general mathematical expression for the spring rate is:

$$c = \frac{\mathrm{d}F}{\mathrm{d}s} = \frac{\mathrm{d}(pA_K)}{\mathrm{d}s} = A_K\frac{\mathrm{d}p}{\mathrm{d}s} \tag{2.19}$$

After using Eq. (2.9) and differentiation with the chain rule:

$$\frac{\mathrm{d}p}{\mathrm{d}s} = p_1V_1^n(-n)(V_1 - A_Ks)^{-n-1}(-A_K) \tag{2.20}$$

After applying the isothermal change of state from 0 to 1 and the balance of forces at the piston according to Eq. (2.13) and furthermore Eq. (2.11):

$$c = A_K\frac{\mathrm{d}p}{\mathrm{d}s} = A_K\frac{F_{F1}}{A_K}\left(\frac{p_0A_KV_0}{F_{F1}}\right)^n(-n)\left(\frac{p_0A_KV_0}{F_{F1}} - A_Ks\right)^{-n-1}(-A_K) \tag{2.21}$$

Dissolving and again using the dimension h_{0F} Eq. (2.17) results in:

$$c(s) = F_{F1}n\frac{h_{0F}^n}{(h_{0F} - s)^{n+1}} \tag{2.22}$$

It becomes obvious that h_{0F} is also important for the spring rate. Yet we should remember that h_{0F} is depending upon F_{F1} – it is not a constant value.

For s=0 and dissolving h_{0F} we obtain the spring rate in normal (design) position:

$$c = n\frac{{F_{F1}}^2}{p_0V_0} \tag{2.23}$$

These are now the fundamental equations on which the function of every hydropneumatic suspension system is based upon. One extremely interesting consequence is that the geometry of the suspension cylinder(s) plays no role in these equations! Solely the gas fill in the accumulator as well as the suspended load determine the contour of the force–displacement curve and therefore also the spring rate.

On one hand the gas fill in the accumulator can be described by the product of accumulator precharge pressure p_0 and the accumulator volume V_0. On the other hand it can, according to the equation of state for the ideal gas, be given as $m_G RT$. This again points out clearly that, apart from the static spring load and the mass of the gas fill m_G, the spring rate is also depending upon the temperature of the gas/the accumulator. So when laying out a hydropneumatic suspension system it has to be taken into account that the precharge pressure set during the production process refers to 20°C temperature. The actual operating temperature can vary due to influences from the environment but can also be increased due to heat in the hydraulic fluid that arises from the viscous damping. The general rule is: higher temperatures soften the spring, lower temperatures make it stiffer.

Furthermore Eq. (2.23) makes it obvious that the static spring load affects the spring rate with its second power, a characteristic property of a hydropneumatic suspension. This means that the suspension behavior of the system spring-damper-mass changes with changing suspended load. This is illustrated in the following calculation using the simple example of a single-mass oscillator.

Using

$$\omega = \sqrt{\frac{c}{m_F}} \tag{2.24}$$

and

$$\omega = 2\pi f \tag{2.25}$$

and

$$F_{F1} = m_F g \tag{2.26}$$

and additionally Eq. (2.23), it is possible to calculate the natural frequency f for the non preloaded hydropneumatic suspension:

$$f = \frac{1}{2\pi}\sqrt{\frac{nF_{F1}g}{p_0 V_0}} \tag{2.27}$$

It becomes obvious that the natural frequency changes proportionally with the square root of the static springload.

$$f \sim \sqrt{F_{F1}} \tag{2.28}$$

Generally spoken and from a theoretical point of view it is a goal to keep the suspension properties of a system constant throughout all loading conditions. For vehicle suspensions this is particularly important for ride behavior, comfort and road holding. Therefore it would be ideal to have a natural frequency independent of the static springload, especially for a single-mass oscillator. Yet in many applications not only the suspended mass/static spring load but also the moment of inertia with respect to various degrees of freedom have to be taken into account. In these cases the behavior described by Eq. (2.28) is often favorable since a disproportionate change of the spring rate often results in more constant overall suspension behavior than a spring rate changing proportionally with the static spring load.

An explicit example of this is the hydropneumatic suspension for the front axle of an agricultural tractor. For lifting and moving heavy masses a so called front loader is optionally available, which consist of the hydraulically liftable boom in combination with a tiltable bucket or pallet fork attached to the end of it. When lifting a heavy mass, the center of gravity (COG) of this mass is very remote to the tractor's center of gravity which increases significantly the moment of inertia for the pitch motion of the tractor (Fig. 2.7).

The natural frequency for pitch motion needs to be kept above a certain level to prevent the tractor from becoming an uncontrollable rocking chair. Therefore it is beneficial to increase the spring rate more than it would be necessary for a constant bounce frequency. Hydropneumatic suspension systems fulfill this requirement and therefore contribute a lot to comfortable, stable and controllable ride behavior during front loader work.

The force–displacement curve of a hydropneumatic spring as well as the curves of spring rate and natural frequency as a function of the suspended static load provide essential information about the suspension properties of a particular suspension system. On the following pages of this section these curves are shown and explained for the various hydropneumatic suspension systems. To ensure good comparability

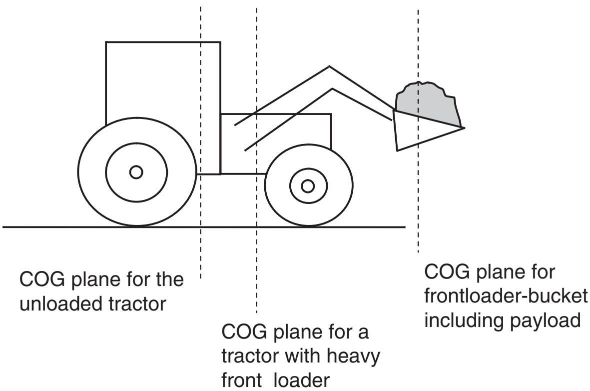


Fig. 2.7 Relocation of the tractor's center of gravity during front loader work

between the different systems, all characteristic curves are calculated using the following basic setup for the suspension system:

Single-mass oscillator
Full suspension stroke (stop-to-stop) 100 mm
Design position is the center between both end stops
Static load for the suspension is 10 kN
System tuned to a natural frequency of 2 Hz

The following Figs. 2.8, 2.9 and 2.10 show the characteristic curves for the *non preloaded* hydropneumatic spring. For comparison the graphs also show the respective characteristic curves for a linear mechanical spring (thin line).

The curves for spring rate and natural frequency are cut off below 2500 N static spring load. This range is not applicable for non preloaded hydropneumatic suspensions due to their low allowable load ratio (with the commonly used diaphragm accumulators, refer to Sect. 3.1.3) and therefore this is not relevant for real applications. In practice numerous examples can be found for applications of non preloaded hydropneumatic suspensions. For example most boom suspensions for tractors,

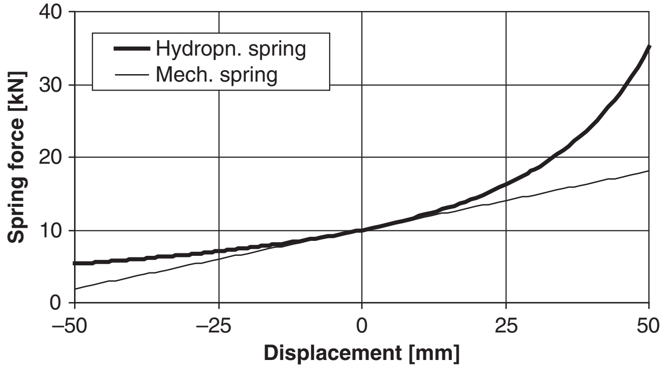


Fig. 2.8 Force–displacement curve for the non preloaded hydropneumatic spring

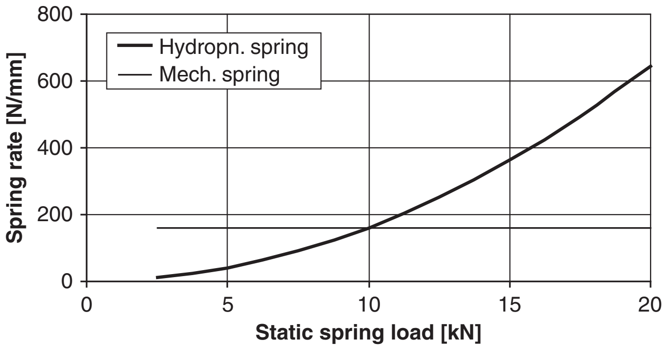


Fig. 2.9 Spring rate vs. static spring load for the non preloaded hydropn. spring

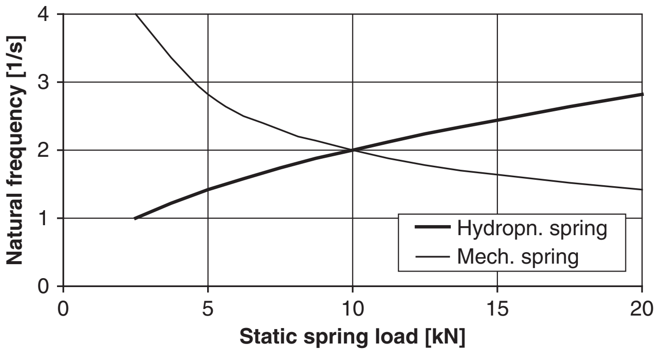


Fig. 2.10 Natural frequency vs. static spring load for the non preloaded hydropn. spring

telehandlers or wheel loaders (Fig. 2.11) – if they are suspended – are equipped with such a simple system: a cylinder whose pistonside is connected with an accumulator. This way the boom including the bucket/pallet fork etc. and the payload is suspended softly. In particular, the pitch oscillations of the usually completely unsuspended vehicle are reduced by this means. Comfort of the driver and in most cases ride stability are increased. Furthermore the reduced vibrations and accelerations at the bucket/pallet fork ensure safe transportation of the payload. In particular, bulk goods can be carried in a bucket more safely since the suspension prevents the payload from spilling and getting lost over the bucket's edge. More detailed information about boom suspensions can be found in [LAT03] and for example the patents [DE205] and [US491].

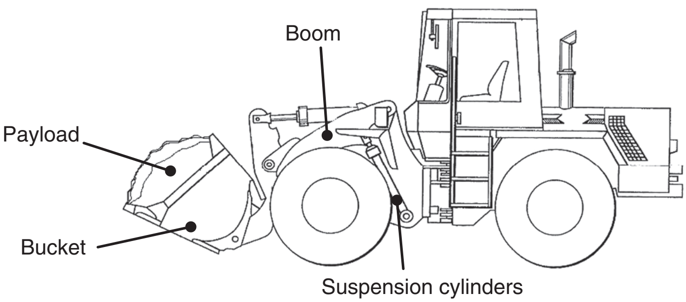


Fig. 2.11 Boom suspension of a wheel loader

The already mentioned next level of sophistication of non preloaded hydropneumatic suspensions is achieved by introducing a double-acting cylinder which is run in the regenerative mode ("regen"-mode) by connecting both cylinder sides (Fig. 2.5b). This type of system allows much higher rebound damping compared to the plunger cylinder. It can be found for example in the cab suspension of 6020series tractors produced by John Deere (HCS = hydraulic cab suspension). In the course of improvements made since the start of serial production, the external line initially used for the connection of the two cylinder chambers (as shown in Fig. 2.5b) was eliminated. It was replaced by a passage in the piston with integrated flow resistor, thus providing identical damping characteristics at better cost. In this application an increased rebound damping level is necessary due to the kinematics of the suspension linkages to prevent the cab from strong pitching due to braking or even trailing throttle in the lower gears.

The example of the HCS is shown in Fig. 2.12. It shows the integrated solution with cylinder (internal oil flow via piston cross drill) and accumulator being combined to the suspension unit. The connected hose is only used to provide oil for level control.

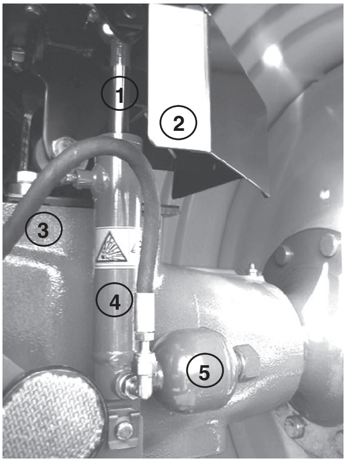


Fig. 2.12 Cab suspension cylinders of John Deere 6020series tractors

The currently highest level of development for non preloaded hydropneumatic suspensions is represented by the so called Hydractiv chassis suspension, which is used in several of Citroen's latest passenger cars. In Fig. 2.13 it is shown that the center valve (7) allows two different operational modes for the suspension. If it is energized (and this is shown in the illustration) both suspension units of an axle are interconnected via a control valve (6) and are furthermore able to displace hydraulic fluid into a third accumulator (3). Its additional gas mass is the reason for a relatively low spring rate in this operational mode. Due to the interconnection of the suspension units, no roll stability (torsional spring rate relative to the longitudinal axis) is provided by the hydropneumatic springs, so it is only the mechanical anti-roll bars which are operative in this mode.

In case the center valve is unenergized, both suspension units are separated from each other by the control valve and therefore act as individual springs. This causes additional roll stability compared to the interconnected operation. Furthermore in

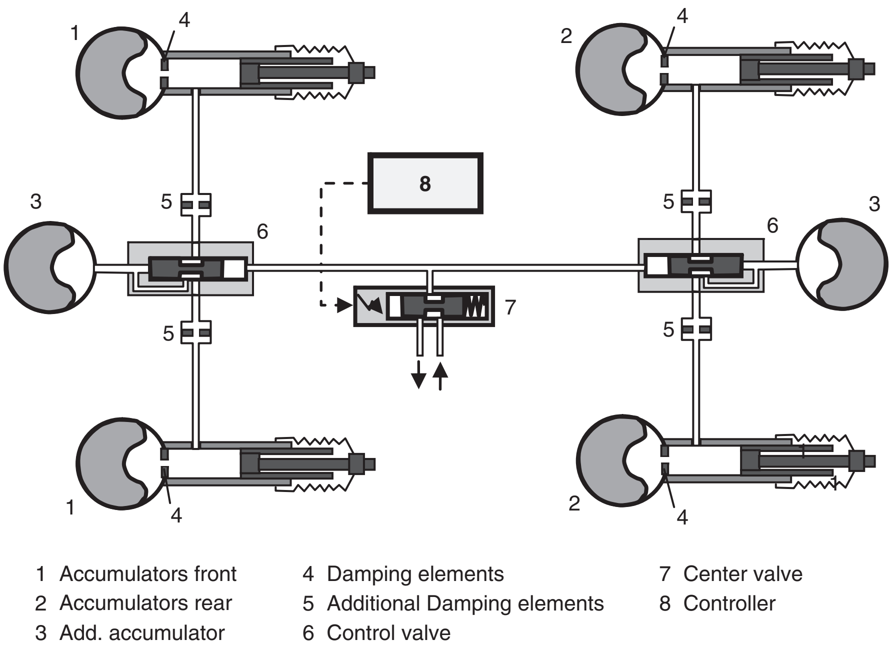


Fig. 2.13 Hydraulic schematic of the Citroen Hydractiv system (mod. from [HEN90])

the closed state, the third accumulator for each axle is decoupled from the suspension units, which causes a higher spring rate for the individual springs. So by closing the control valves, higher ride stability can be achieved especially during cornering and for "sporty" driving, while for opened control valves the soft suspension setup provides a high level of comfort which is typical for Citroen and their hydropneumatic suspensions. Detailed information about this can be found in Sect. 7.2.

2.2.4 Systems with Mechanical Preload

It is clearly visible in Fig. 2.9 that there is a quadratic relationship between spring rate and spring load. Although this characteristic feature can have positive effects, as mentioned earlier in this section, it is in many cases favorable to attenuate it. This can be achieved by saddling an additional internal preload onto the hydropneumatic springs which is already active even when no external load is applied. In this case the preload acts as a basic load and all other external loads are added onto it. This has the effect that, for a given change of the external static load, the relative load change for the spring of the preloaded suspension system is smaller than for the non preloaded system.

The practical application of the preload is done in two different ways:

(a) For the single-acting suspension cylinder the preload is applied by a mechanical spring for example by a helical coil spring or a torsional spring (mechanical preload).
(b) By using a double-acting suspension cylinder it is possible to apply a hydraulic counter-pressure on the rod side by a second hydropneumatic spring (hydraulic preload).

A third theoretical possibility is to load the isolated side with an additional mass to get the desired preload. Yet this is usually not accepted in mobile hydraulics since this would add significantly to the curb weight of the vehicle.

This section now explains the systems with mechanical preload, Fig. 2.14 shows its schematic. Since the mechanical spring preloads the hydropneumatic spring with a compressive force, both springs are positioned figuratively opposite to each other although from a functional point of view it is a parallel connection of these springs. This is why the force application point of the overall, externally effective spring force is in the connection point of both springs, in the center.

The illustration shows clearly that the mechanical spring causes, in addition to the preload, an additional spring rate for the system. This means that the system will have an effective spring rate resulting from the parallel connection of the hydropneumatic and the mechanical spring.

$$c = c_{\text{hydr}} + c_{\text{mech}} \tag{2.29}$$

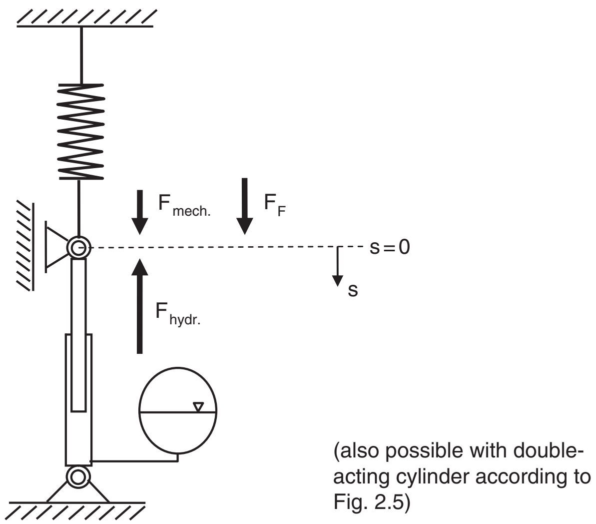


Fig. 2.14 Schematic illustration of a hydropneumatic spring with mechanical preload

The spring rate for the hydropneumatic spring c_{hydr} can be calculated using Eq. (2.23). It is important to keep in mind though that the sum of the static spring load F_{F1} and the mechanical preload $F_{\mathrm{V,mech}}$ (in the normal position of the suspension) need to be considered in the equation, replacing only F_{F1}. The spring rate for the mechanically preloaded hydropneumatic suspension therefore can be calculated by:

$$c(s=0) = n\frac{(F_{\mathrm{F1}}+F_{\mathrm{V}})^2}{p_0 V_0} + c_{\mathrm{mech}} \tag{2.30}$$

and the natural frequency for a single-mass oscillator with this system is:

$$f = \frac{1}{2\pi}\sqrt{\frac{\left(\frac{n(F_{\mathrm{F1}}+F_{\mathrm{V}})^2}{p_0 V_0} + c_{\mathrm{mech}}\right) g}{F_{\mathrm{F1}}}} \tag{2.31}$$

The calculation for the force–displacement-curve can be deduced from Fig. 2.14:

$$F_{\mathrm{F}}(s) = F_{\mathrm{hydr}}(s) - F_{\mathrm{mech}}(s) \tag{2.32}$$

The force of the mechanical spring is:

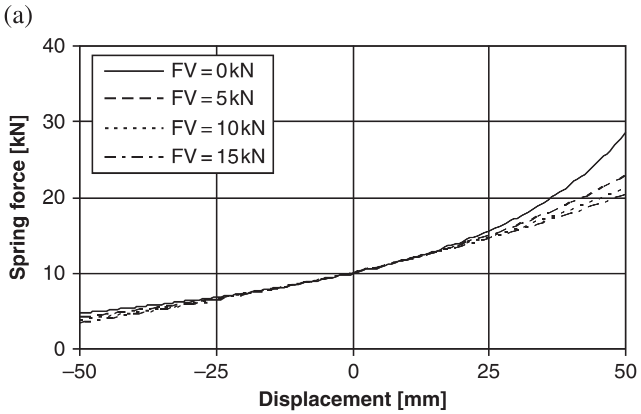
$$F_{\mathrm{mech}}(s) = F_{\mathrm{V}} - c_{\mathrm{mech}} s \tag{2.33}$$

In order to calculate $F_{\mathrm{F}}(s)$ it is necessary to use Eq. (2.16) from Sect. 2.2.3 and then insert the sum of forces $(F_{\mathrm{F1}}+F_{\mathrm{V}})$ replacing F_{F1} in Eq. (2.16):

$$F_{\mathrm{F}}(s) = (F_{\mathrm{F1}}+F_{\mathrm{V}})\frac{\left(\frac{p_0 V_0}{F_{\mathrm{F1}}+F_{\mathrm{V}}}\right)^n}{\left(\frac{p_0 V_0}{F_{\mathrm{F1}}+F_{\mathrm{V}}} - s\right)^n} - (F_{\mathrm{V}} - c_{\mathrm{mech}} s) \tag{2.34}$$

The following Figs. 2.15, 2.16 and 2.17 show the characteristic curves for the hydropneumatic suspension *with mechanical preload*. The preload force F_{V} and the spring rate of the mechanical spring c_{mech} are varied in the diagrams to show the influence of these parameters on the different curves. It is of major importance that a mechanical spring is, at a given design space, the more intricate (and therefore expensive) the higher the preload force and the higher spring rate. For this reason a preload force of 5 kN and a spring rate of 20 N/mm are chosen as a basis for the following diagrams and are varied from there. This means that for variations of the preload force, a spring rate of 20 N/mm is chosen and for the variation of spring rate in general a preload force of 5 kN is chosen. Both basis values are rather on the lower end of the optimum range, as we will see later on.

Looking at the force–displacement curves one tends to assume that the properties vary only marginally in the near range around the design point at a displacement of



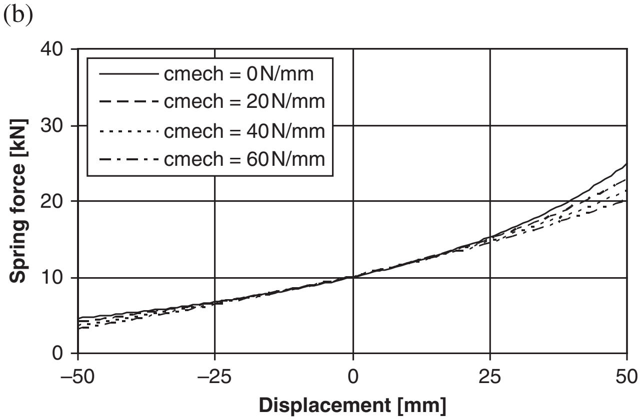


Fig. 2.15 Force–displacement curves for the hydropneumatic spring with mechanical preload. (**a**) Variable $F_V(c_{mech} = 20$ N/mm). (**b**) Variable $c_{mech}(F_V = 5$ kN)

0 mm, a static spring load of 10 kN and a natural frequency of 2 Hz. However, when looking at the areas remote from the design point and even more when looking at the curves for spring rate and natural frequency, significant differences become obvious.

Increasing mechanical preload reduces the progression of the force–displacement curve as well as the progression of the spring rate over static spring load. This means a considerable reduction of some characteristic (and sometimes unfavorable) properties of a hydropneumatic suspension. The reason for this can be found in the reduced relative load change (per kN increase of static spring load) with increasing F_V. The equation for the natural frequency indicates clearly, that an increased mechanical preload needs to be compensated by an increased gas mass $p_0 V_0$ if the natural frequency has to remain constant. Furthermore it can be deduced that the relative change of the characteristic curve (by changed mechanical preload) decreases, the higher F_V is chosen. In Figs. 2.16a and 2.17a this can be seen from the larger distance of the curves 0–5 kN compared to the distance of the curves 10 and 15 kN.

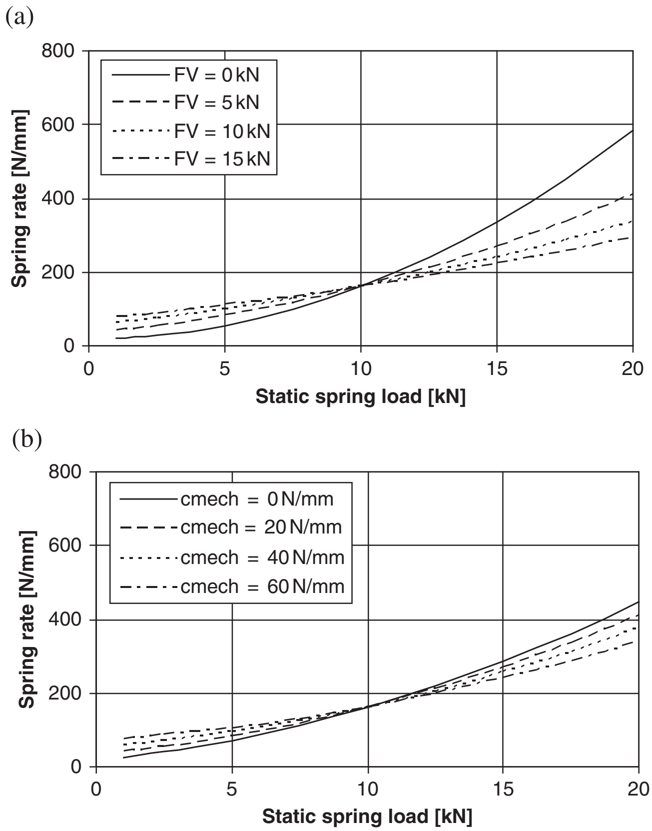


Fig. 2.16 Spring rate vs. static spring load for the hydropneumatic spring with mechanical preload. (**a**) Variable $F_V(c_{mech} = 20$ N/mm). (**b**) Variable $c_{mech}(F_V = 5$ kN)

It is also clearly visible, that the trend of the curve of spring rate vs. static spring load shows a better linearity for the preloaded hydropneumatic suspension system compared to the non preloaded system. Resulting from this is a natural frequency which is very close to the design goal of 2 Hz over a broad range of static spring load. It is interesting that the minimum of the natural frequency is moved to higher static spring loads with increasing mechanical preload F_V. In the example illustrated by Fig. 2.17a the natural frequency shows the best constancy in the range around the design point spring load (10 kN) when mechanical preloads between 5 and 10 kN are applied (reminder: $c_{mech} = 20$ N/mm).

An increase of the mechanical spring rate has a similar effect as an increase of mechanical preload. Here too, the progression of the force–displacement curve and the spring rate vs. static spring load curve decreases with increasing mechanical spring rate. This is due to the fact that more and more load is taken by the mechanical spring and off the hydropneumatic spring (same overall spring rate!) for

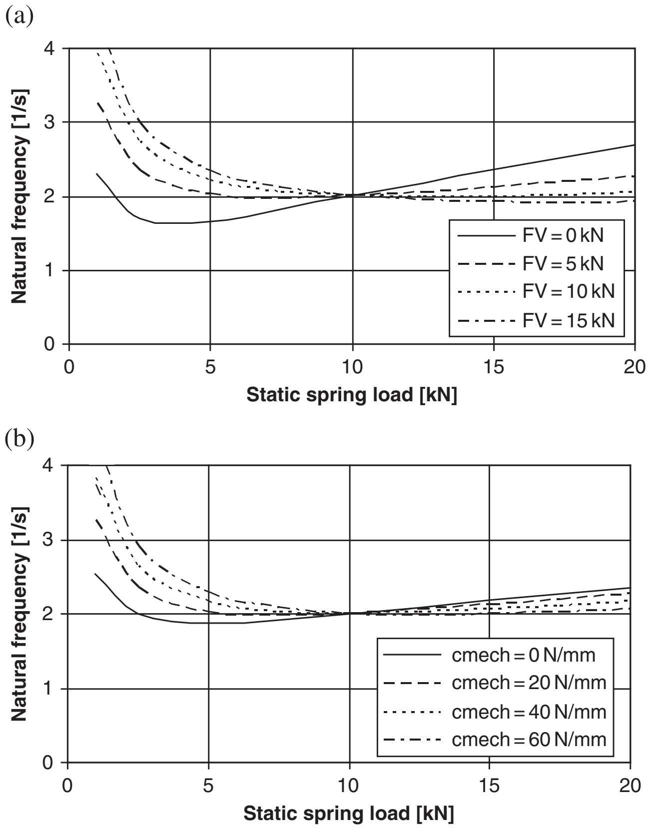
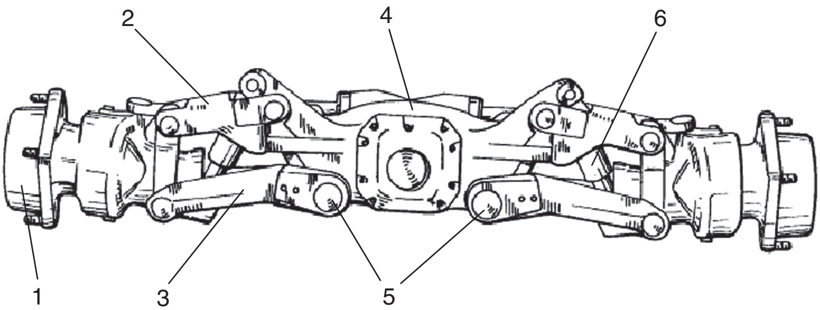


Fig. 2.17 Natural frequency vs. static spring load for the hydropneumatic spring with mechanical preload. (**a**) Variable $F_V(c_{mech} = 20\ \text{N/mm})$. (**b**) Variable $c_{mech}(F_V = 5\ \text{kN})$

increasing mechanical spring rate. So the harder the mechanical spring the softer the hydropneumatic spring and therefore the lower its influence on the overall system behavior. However even an infinitely soft spring ($c_{mech} = 0$ N/mm) cannot lower the minimum of the natural frequency as low as zero mechanical preload can. Furthermore the diagrams show that the change of the characteristic curves by increasing mechanical spring rate is independent from the original level of the mechanical spring rate. The difference between the curves for 0 and 20 N/mm is about the same as the difference between the curves for 40 and 60 N/mm. It can be easily deduced from the diagrams shown above that one drops back to the curves for a non preloaded system as of Sect. 2.2.3 if the preload F_V and the mechanical spring rate c_{mech} are both reduced to zero.

A practical example for a hydropneumatic suspension system with mechanical preload is the front axle suspension system offered by the company Carraro for agricultural tractors. It is used for example in tractors built by Claas (formerly



1 Hub
2 Upper wishbone
3 Lower wishbone
4 Central axle body
5 Inner pivot axis of lower wishbone (torsion bar invisibly behind)
6 Suspension cylinder

Fig. 2.18 Tractor front axle suspension by Carraro

Renault), Case/Steyr, Massey Fergusson and Landini. Figure 2.18 is taken from the patent US5931486. It shows an independent wheel suspension by a double wishbone arrangement (2 and 3) which is mounted on a central axle body 4. On the inner side of each lower wishbone 3 a torsional spring is placed coaxially with the respective pivot axis 5. This torsion bar creates the mechanical preload for the hydraulic cylinders 6. An adaptation of the axle to different types of vehicles is (among other measures) enabled by changing the dimensions of the torsional spring.

2.2.5 Systems with Constant Hydraulic Preload

In many cases it is difficult to integrate a mechanical preload into a hydropneumatic suspension. This is often due to the usually high demand for design space of an additional mechanical spring, which has to be able to cope with the whole stroke of the suspension. That is why it is more common to use double-acting cylinders instead and use the additional available rodside chamber to create an additional, preloading spring. This is done by pressurizing the rod chamber of the cylinder (between piston and rod end) thus applying a force which acts as the preload for the suspension. Since this active area is displaced when the piston is moved and with it the oil that is contained by the rod chamber, it too has to be connected to an accumulator, which can absorb and release the displaced hydraulic fluid of the rodside chamber. This arrangement then is basically a suspension system which consists of two individual single-acting hydropneumatic suspensions counteracting each other. The respective schematic is shown in Fig. 2.19. For a better illustration of the hydraulic

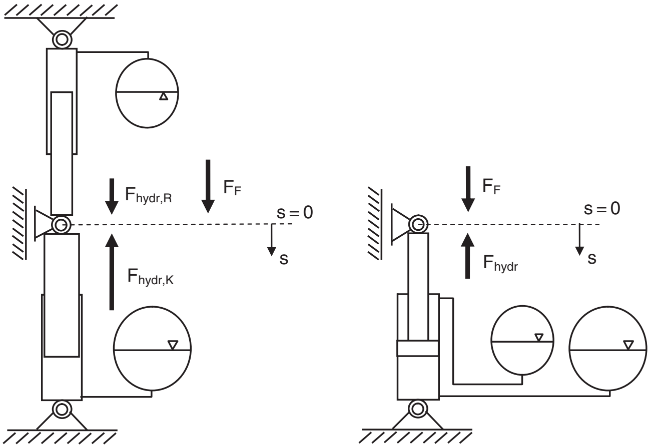


Fig. 2.19 Schematic illustration of a hydropneumatic spring with hydraulic preload

preload, pistonside and rodside are symbolically shown as separate single-acting cylinders. Again both systems are shown positioned figuratively opposite to each other although from a functional point of view it is a parallel connection of these springs.

The spring rate for the overall system can be calculated as the sum of the spring rates of the pistonside spring $c_{\mathrm{hydr,K}}$ and the rodside spring $c_{\mathrm{hydr,R}}$.

$$c = c_{\mathrm{hydr,K}} + c_{\mathrm{hydr,R}} \tag{2.35}$$

The spring rates for hydropneumatic springs can simply be taken over from Sect. 2.2.3. However that preload force needs to be taken into account. It results from the preload pressure p_{V} on the rodside and the respective hydraulically active area, the ring-shaped area of the piston.

$$F_{\mathrm{V}} = p_{\mathrm{V}} A_{\mathrm{R}} \tag{2.36}$$

The result for the overall spring rate is c:

$$c = \frac{n(F_{\mathrm{F1}} + p_{\mathrm{V}} A_{\mathrm{R}})^2}{p_{0,K} V_{0,K}} + \frac{n p_{\mathrm{V}}^2 A_{\mathrm{R}}^2}{p_{0,R} V_{0,R}} \tag{2.37}$$

And therefore the natural frequency can be calculated as:

$$f = \frac{1}{2\pi} \sqrt{\frac{\left(\frac{n(F_{\mathrm{F1}} + p_{\mathrm{V}} A_{\mathrm{R}})^2}{p_{0,K} V_{0,K}} + \frac{n p_{\mathrm{V}}^2 A_{\mathrm{R}}^2}{p_{0,\mathrm{R}} V_{0,\mathrm{R}}} \right) g}{F_{\mathrm{F1}}}} \tag{2.38}$$

Looking at Fig. 2.19 the balance of forces gives us:

$$F_{\mathrm{F}}(s) = F_{\mathrm{hydr},K}(s) - F_{\mathrm{hydr},R}(s) \tag{2.39}$$

By considering the equations for the non preloaded spring, the individual forces for pistonside and rodside can be calculated:

$$F_{\mathrm{hydr},K}(s) = (F_{\mathrm{F1}} + F_{\mathrm{V}}) \frac{\left(\frac{p_{0,K} V_{0,K}}{F_{\mathrm{F1}}+F_{\mathrm{V}}}\right)^n}{\left(\frac{p_{0,K} V_{0,K}}{F_{\mathrm{F1}}+F_{\mathrm{V}}} - s\right)^n} \tag{2.40}$$

$$F_{\mathrm{hydr},R}(s) = F_{\mathrm{V}} \frac{\left(\frac{p_{0,\mathrm{R}} V_{0,\mathrm{R}}}{F_{\mathrm{V}}}\right)^n}{\left(\frac{p_{0,\mathrm{R}} V_{0,\mathrm{R}}}{F_{\mathrm{V}}} + s\right)^n} \tag{2.41}$$

Here it is important to keep in mind that the displacement s has different preceding algebraic signs in both equations. The reason is that a positive displacement s results in a compression on the pistonside (force increases) while it leads to an expansion on the rodside (force decreases). Applying the equation for F_{V} results in:

$$F_{\mathrm{F}}(s) = (F_{\mathrm{F1}} + p_{\mathrm{V}} A_{\mathrm{R}}) \frac{\left(\frac{p_{0,K} V_{0,K}}{F_{\mathrm{F1}}+p_{\mathrm{V}} A_{\mathrm{R}}}\right)^n}{\left(\frac{p_{0,K} V_{0,K}}{F_{\mathrm{F1}}+p_{\mathrm{V}} A_{\mathrm{R}}} - s\right)^n} - p_{\mathrm{V}} A_{\mathrm{R}} \frac{\left(\frac{p_{0,\mathrm{R}} V_{0,\mathrm{R}}}{p_{\mathrm{V}} A_{\mathrm{R}}}\right)^n}{\left(\frac{p_{0,\mathrm{R}} V_{0,\mathrm{R}}}{p_{\mathrm{V}} A_{\mathrm{R}}} + s\right)^n} \tag{2.42}$$

While for the system with mechanical preload only two new parameters (F_{V} and c_{mech}) were needed for the respective equation, the number of additional parameters for a system with hydraulic preload is much higher. These new parameters $p_{0,\mathrm{R}}$, $V_{0,\mathrm{R}}$, A_{R} and p_{V} are available to tune the suspension to the desired properties. Yet, in the equations, these parameters always show up in pairs as A_{R} and p_{V}, which represent (when multiplied) the hydraulic preload force, and $p_{0,\mathrm{R}}$ and $V_{0,\mathrm{R}}$, which represent (when multiplied) the gas mass enclosed in the rodside accumulator. So the latter two parameters are (in combination with F_{V}) responsible for the spring rate of the rodside hydraulic system $c_{\mathrm{hydr,R}}$. The bottom line is that here too are two additional determining factors just like for the system with mechanical preload. These two factors can be set to the desired level by changing the parameters influencing them.

The following Figs. 2.20, 2.21 and 2.22 show the characteristic curves for the hydropneumatic spring *with hydraulic preload*. All figures are again split into two diagrams, one of them showing the influence of the preload force $A_{\mathrm{R}} p_{\mathrm{V}} = F_{\mathrm{V}}$ and the other one showing the influence of the additional rodside spring rate $c_{\mathrm{hydr,R}} = (n F_{\mathrm{V}}^2)/(p_{0,\mathrm{R}} V_{0,\mathrm{R}})$. For a better comparison with the hydropneumatic spring with mechanical preload, a 10 kN static spring force and 2 Hz for the natural

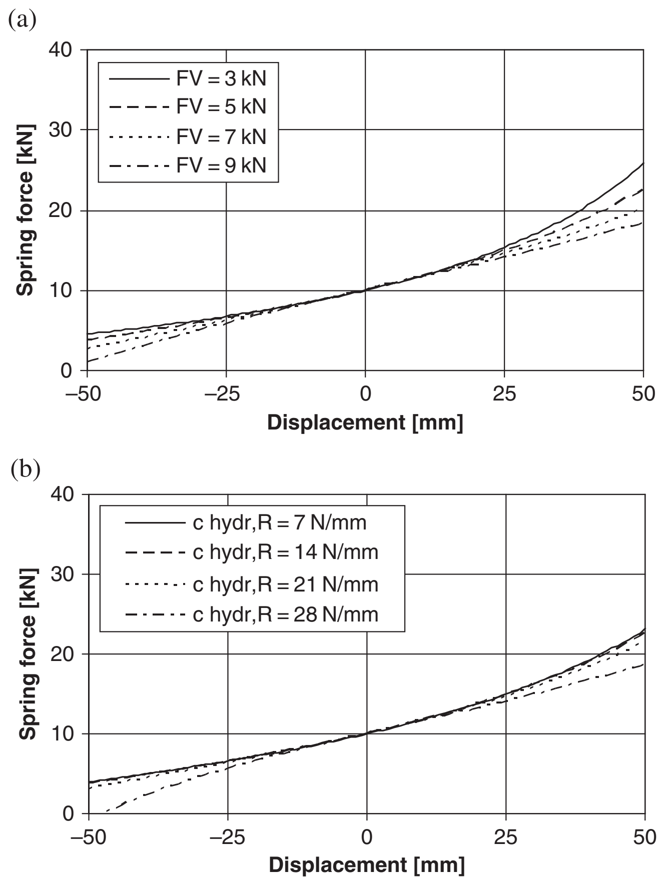


Fig. 2.20 Force–displacement curves for the hydropneumatic spring with hydraulic preload. (**a**) Variable $F_V(c = 21$ N/mm). (**b**) Variable c ($F_V = 5$ kN)

frequency are chosen as the basis for this design. The preload parameters have been chosen as:

$$A_R = 500\ \text{mm}^2$$
$$p_V = 10\ \text{MPa}$$
$$p_{0,R} = 5\ \text{MPa}$$
$$V_{0,R} = 300{,}000\ \text{mm}^3$$

This choice provides an effective preload force of 5 kN and a spring rate of the rodside hydropneumatic spring of 21 N/mm. These values are similar to those in Sect. 2.2.4, a comparison with the previous examples (Figs. 2.15, 2.16 and 2.17) is therefore easy. Please note: choosing identical values as in Sect. 2.2.4 would not show the full potential of a system with hydraulic preload, therefore this slight

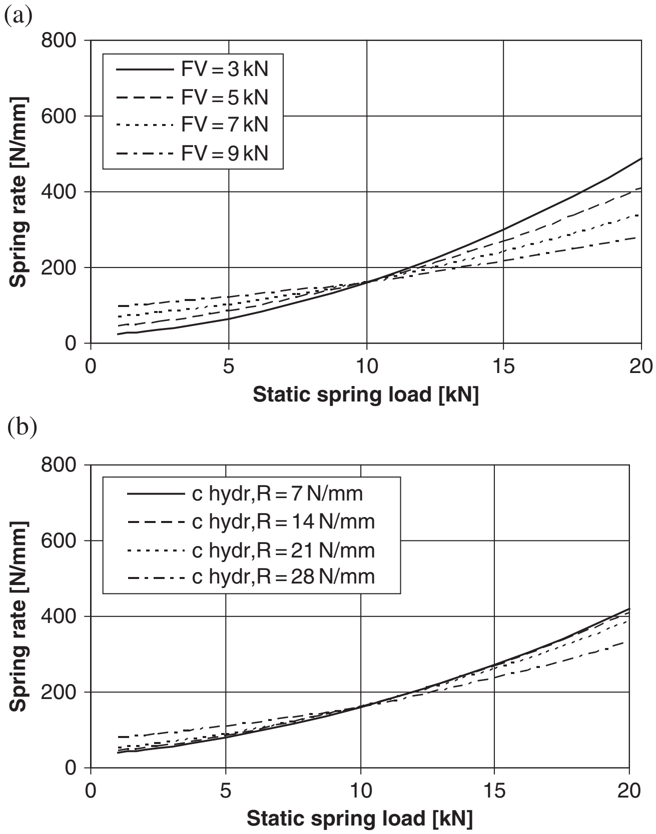


Fig. 2.21 Spring rate vs. static spring load for the hydropneumatic spring with hydraulic preload. (**a**) Variable $F_V (c = 21$ N/mm). (**b**) Variable c ($F_V = 5$ kN)

deviation was chosen. Furthermore this set of parameters provides an optimal ratio of p_V and $p_{0,R}$ as will be shown in Sect. 3.1.3.

For the hydropneumatic spring with hydraulic preload it can be stated that an increase of the rodside hydraulic spring rate $c_{hydr,R}$ and an increase of the preload force F_V have in general a very similar effect. This behavior can also be deduced from the Eq. (2.37) for the spring rate and Eq. (2.38) for the natural frequency. The precharge pressure and the rodside accumulator volume can be found in the denominator, while the preload force, represented by $p_V A_R$, is found in the numerator.

It becomes obvious that it is basically possible to create similar curves with a hydropneumatic suspension system with hydraulic preload and with mechanical preload. One special feature is characteristic to both of them: at low static spring loads the natural frequency does not drop as drastically as for non preloaded systems. It stays on the desired level over a broad range of loads and even increases when the load gets close to zero. This can be ascribed to the preload force on one

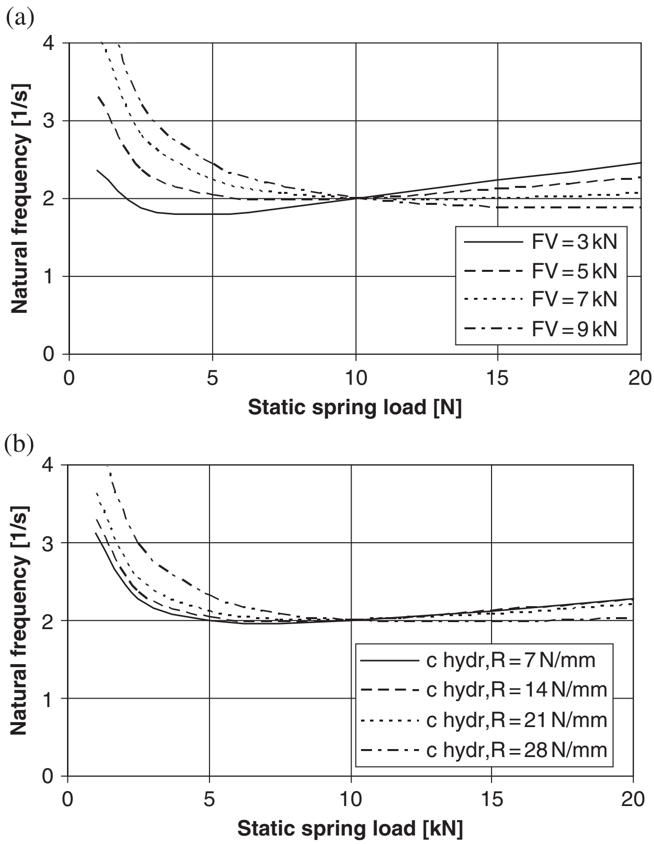


Fig. 2.22 Natural frequency vs. static spring load for the hydropneumatic spring with hydraulic preload. (**a**) Variable $F_V (c = 21$ N/mm). (**b**) Variable c ($F_V = 5$ kN)

hand, which creates the effect that, even at $F_{F1} = 0$ N, there is still a significant force on the hydraulic spring and therefore spring rate available. On the other hand it is the spring rate of the mechanical spring or of the rod side system that is still effective at zero load.

An increase in this preload force F_V, by increasing the preload pressure, results in a weakening of the characteristic curve of the non preloaded system. The setting $p_V = 0$ would represent the non preloaded system. Furthermore it is obvious that the spring rate depends less on the static spring load if the preload force is increased. In terms of the natural frequency an increase of F_V results in a shift of the minimum of the natural frequency curve towards higher static spring loads. At high preload forces an additional characteristic effect can be seen to some extent: The progression of the force–displacement curve of the rodside hydropneumatic spring causes a strong reduction in the spring force when getting close to the rebound end stop.

This effect is even more evident, if the gas mass in the rodside accumulator is changed. This is shown in the diagrams by the influence of the spring rate of the rodside $c_{\mathrm{hydr,R}}$. At high $c_{\mathrm{hydr,R}}$ the compression of the rodside gas volume is very high at the end of the stroke close to the rebound end stop. This results in a strong force increase of the rodside hydraulic spring. One can make use of this for suspension systems. With a suitable choice of parameters, the spring rate in the range around the design position (center position) can be very low, so that the suspension is soft in its main working range. Due to the progression, the spring becomes the stiffer, the closer the piston gets to its end stops. This helps to prevent the suspension from bottoming out.

The above mentioned effect can be emphasized if the preload force is reduced and a high rodside spring rate is chosen at the same time. The result is a progression of the force–displacement curve both in compression and rebound direction. The suspension can be tuned to provide this effect by the following setup of the rodside preload parameters:

$$
\begin{aligned}
A_{\mathrm{R}} &= 500\ \mathrm{mm}^2 \\
p_{\mathrm{V}} &= 4\ \mathrm{MPa} \\
p_{0,\mathrm{R}} &= 1.4\ \mathrm{MPa} \\
V_{0,\mathrm{R}} &= 100{,}000\ \mathrm{mm}^3
\end{aligned}
$$

Figure 2.23 shows the force–displacement curve for this configuration. For comparison reasons, that diagram also shows with a thin line the curve for a linear, mechanical spring. The end stop progression effect can be used without significant negative impact on the behavior of the natural frequency over static spring load. Furthermore this is also a cost effective solution since the smaller gas volume on the rodside requires only small rodside accumulators. However this can not always be used in practice since other effects and limits need to be considered – please refer to Sect. 3.1.3 for more information.

Good examples of the hydropneumatic suspension with constant hydraulic preload are the front axle suspension systems on Fendt tractors as well as on John

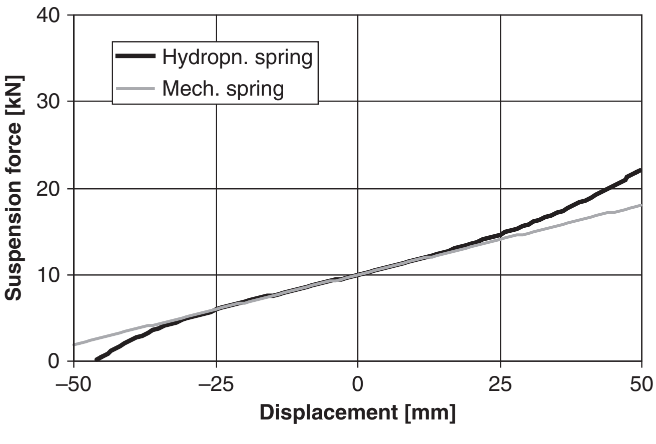


Fig. 2.23 Force–displacement curve with distinct progression close to the end stops

Deere tractors with the so called TLS I system (Triple Link Suspension I) which can be found on their 6010 and 7010 series tractors. Yet both manufacturers go different ways when it comes to how the hydraulic preload is applied. Fendt applies the full pump pressure of about 20 MPa on a rather small rodside active area whereas John Deere applies a lower, regulated pressure as the preload pressure to a respectively larger rodside piston area. Their hydraulic control blocks contain a pressure regulating valve which enables this. The advantage of this is that the variable displacement pump used on these tractors does not have to go to full pressure during corrective action of the control system. Another advantage is cylinder friction, further explained in Sect. 2.3.1. The disadvantage is the need for a larger volume accumulator on the rodside since more oil is displaced due to the larger rodside active area. More information on the design and layout of John Deere TLS I system can be found in Sect. 7.1.

2.2.6 Systems with Variable Hydraulic Preload

This is the next logical step forward from the suspension system with constant hydraulic preload. When looking at Eq. (2.37) it is easy to discover that it is possible to vary the preload and therefore the spring rate by changing a rather easily adjustable parameter: the preload pressure p_V. This possibility to influence the spring characteristics allows the control of the suspension depending on other parameters such as the static spring load or other operation parameters in an open loop or even in a closed loop.

A good example of the spring load dependent preload pressure control is the John Deere front axle suspension TLS II, which comprises a two-step adjustment of the spring rate level depending on the static spring load. This is enabled by using the pressure in the piston chamber as an input parameter for the control of the preload pressure in the rod chamber. Below a certain pistonside pressure limit $p_{K,grenz}$, the rodside pressure is set to a high level $p_{R,h}$. If the pistonside pressure is above $p_{K,grenz}$, a low level of rodside pressure $p_{R,n}$ is set. The higher pressure level on the rodside for low pistonside pressures causes an increased spring rate at low front axle loads. The control of the rodside pressure is achieved purely by hydraulic components – one pressure switch valve and one special, switchable pressure regulating valve. Figure 2.24 illustrates schematically the relationship of spring rate and static spring load. In reality there is interaction between rodside pressure and pistonside pressure. This interaction causes a difficult to define transition area around the pressure switch point. In this area it also depends on other operational conditions whether the rodside pressure is on a high or a low level. Therefore in the schematic, the vertical part of the spring rate vs. load curve line is only idealized and does not represent the behavior in reality.

There are several reasons to justify this additional effort:

(1) The positive effect of the hydraulic preload on the characteristic of the curve of natural frequency vs. static spring load is further improved (constant over a wide range).

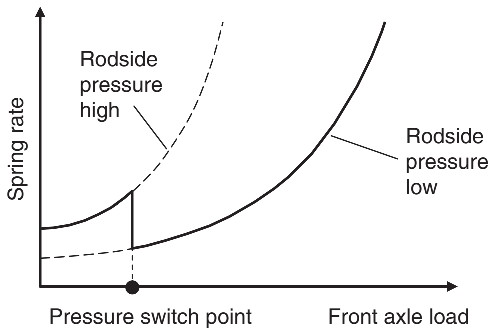


Fig. 2.24 Spring rate vs. static spring load for the John Deere system TLS II

(2) The system accounts for the fact that low front axle loads are caused by heavy implements and loads with a center of gravity far behind the tractor. These conditions increase the inertia relative to the lateral axis of the tractor which would cause a very low pitch natural frequency and therefore result in a spongy ride behavior of the tractor with frequently bottoming out suspension. The increased spring rate for these conditions compensates for this and therefore improves the suspension quality.

(3) Since diaphragm accumulators are used in this suspension system, the mentioned setup helps to keep these accumulators within the allowed pressure range of operation (refer to Sect. 2.5 for more information).

In a further step, a closed loop control of the rodside pressure provides the possibility to adapt the suspension to all kinds of different operating conditions. For this purpose, a parameter needs to be chosen as an input variable, which acts as a measure for the suspension quality. Therefore, if suspension quality worsens due to a change in operating conditions (for example an uneven road surface), the degradation in suspension quality can be detected and the spring stiffness can brought to a level which helps to re-improve the ride behavior and therefore the input variable. This is the step from a purely passive suspension system (yet with level control) towards an adaptive system. In this case it is usually necessary to use electronics as a support for the hydraulic system in order to set up a closed loop control for the rodside pressure. The electronic controller can monitor multiple input variables and can adjust the hydraulic system to the best possible setting by switching hydraulic valves and allowing oil to flow in and out of the hydraulic suspension units, especially their rodside portions. This system permits the use of the whole range of possible settings and spring rates allowed by the hardware components, in particular the accumulators. This means that there is no longer only the simple spring rate vs. static spring load curve of the constantly preloaded system and the more advanced curve for the two-step pressure adjusted system. For the system with fully variable rodside pressure, there is then a large *area* of possible operating points in the spring rate vs. spring load diagram. Figure 2.25 compares the three last-mentioned systems.

A particular advantage of the continuously variable closed loop control system is the fact that it can be easily adapted to changes in vehicle dimensions/masses etc.

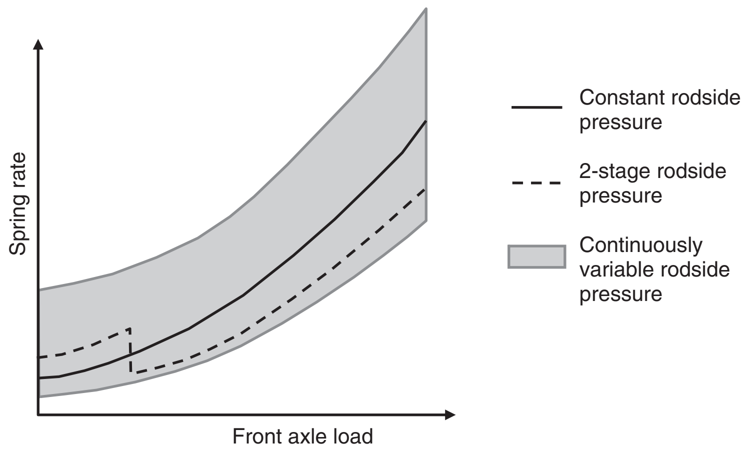


Fig. 2.25 Comparison of the spring rates of the hydraulically preloaded systems with constant, 2-stage and continuously variable rodside pressure

as well as new operating conditions. It is therefore ideal for use in different vehicle platforms since it can be used universally only with specific software and without or only minor changes in hardware.

The John Deere TLS Plus suspension system makes use of this advanced technology and therewith continues the consistent enhancement in hydropneumatic suspensions.

2.3 Damping Characteristics

The energy that is transferred into the suspension by external excitations needs to be dissipated to achieve a decay of the resulting oscillation amplitude and to avoid increasing amplitudes due to resonance. Therefore additional elements in the suspension system are necessary to transform the kinetic and/or potential energy of the suspension. In most cases kinetic energy is transformed into heat by application of a retarding force during the motion of the suspension elements. This retarding damping force usually is based upon the principle of friction. In general two different fundamental principles create the damping in a suspension system:

(1) Boundary friction, also called dry friction or solid body friction. Two solid bodies pressed onto each other with a normal force slide in their interface with a resistant force caused for example by catching of surface roughness and adhesion. The resistant force is called friction force and acts as a damping force.
(2) Fluid friction, also called viscous friction or hydrodynamic friction. A flow resistor is placed in the flow path of a fluid and causes internal fluid friction which therefore causes a pressure increase upstream of the resistor. This additional pressure is acting upon the active areas of the cylinder thus creating a retarding force, a damping force.

In addition to the above mentioned principles there are more rather exotic principles which are rarely used in suspension technology. For example there is the eddy-current principle, which is often used in vehicles as a wear-free retarder to reduce vehicle speed on downhill slopes. It is based upon the principle of induction of current in an electric conductor when it is moved through a magnetic field. Furthermore there are the so called gas-spring-damper-elements [GOL84], which provide the function of a spring as well as a damper only by their internal gas fill.

In general one tends to keep the damping forces as low as possible to get the best possible decoupling of the suspended mass on the isolated side from the excitation on the input side. Yet if the level of damping is tuned to provide optimal results under normal operating conditions, the damping will most surely be too low under extreme operating conditions. This will result in high amplitude oscillations and, as a consequence, bottoming out of the suspension. To avoid heavy accelerations when the suspension hits the mechanical end stops, another type of damping is integrated into many suspension systems which is only active when the suspension reaches the end of its stroke: the so called end-of-stroke damping or end cushioning. Additional damping elements dissipate the excessive kinetic energy before the suspension reaches the end stop. Therefore this energy is prevented from being transferred into the isolated side by a short term but very high force peak.

The following sections explain the boundary friction, the viscous friction and the special but very important area of end-of-stroke damping.

2.3.1 Boundary Friction Damping

The boundary friction is a resistive force against sliding in the interface between two solid bodies which are pressed onto each other by the normal force F_{N}. The direction of the normal force is perpendicular to the intended sliding direction, the resistive force, respectively the friction force, is acting opposed to the sliding direction (Fig. 2.26).

There will be no movement in the interface as long as the tractive force F_{Z} does not exceed a certain limit. This limit is the static friction force $F_{\mu,\mathrm{H}}$. In the case of static friction, the absolute value of the friction force F_{μ} is equal to the absolute

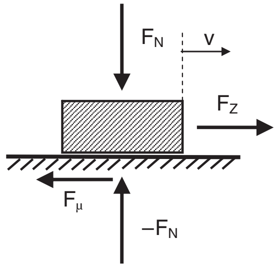


Fig. 2.26 Forces involved in boundary friction

value of the tractive force F_Z, the body is in a static balance of forces. The maximum static friction force depends upon the normal force F_N and the coefficient of static friction μ_H. The latter depends on the properties of the two involved solid bodies, in particular the type of material and the nature of their surfaces.

$$F_{\mu,H} = \mu_H \cdot F_N \tag{2.43}$$

The static friction force plays an important role in suspension systems. It is the parameter which determines the minimum level of excitation below which a suspension system cannot absorb and reduce acceleration. Up to this level of excitation, no movement between input side and isolated side is possible (hindered by the static friction, sometimes also called "stiction"). In this case the suspension components represent a fixed coupling between both sides. The level of static friction therefore very much affects what is called the response characteristic of the suspension. This describes whether it reacts sensitively on smallest excitations and irons these out or whether these excitations are passed non-elastically to the isolated side. Static friction forces are especially caused by elastomer seals in particular after long time intervals without operation and can even lead to partial damage of the seal [MUE]. It is important to notice that static friction does not contribute to the damping of the suspension system since it is only active at times without relative motion!

As soon as the tractive force exceeds the static friction force, both solid bodies start to slide on each other. During sliding the so called sliding friction force $F_{\mu,G}$ is active, which is sometimes significantly lower than the static friction force. Just like μ_H, the coefficient of sliding friction μ_G depends mainly on the properties of the two sliding surfaces. Furthermore there can be an additional dependence for example on sliding velocity (not considered in the following equation).

$$F_{\mu,G} = \mu_G F_N \tag{2.44}$$

Due to the coincidence and counteraction of motion and sliding friction force, kinetic energy is transformed into heat and therefore drawn out of the suspension system. The higher the amount of energy stored in the suspension system, the larger the amplitude of the oscillation. Due to the increased displacement, the energy drawn out of the system by friction during each oscillation increases with the stored energy. This means that a damper, which works by the principle of boundary friction, is at least partially self adapting to the needs of the system. It will be shown in Sect. 2.3.2 that fluid friction is even more capable of providing this important feature.

The first dampers which were used in suspension systems were based purely on the principle of boundary friction. Examples of these are the well known leaf springs, especially the laminated type, as well as the less well known torsional friction dampers which were even adjustable in friction by varying the normal force through the variation of the load of the (laminated) disk spring. Despite that, the pure friction dampers were not able to make it into modern suspension systems since they always had the negative side-effect of worsening the systems response characteristics.

In all suspension systems boundary friction can be found in the kinematics' bearings which are mandatory in providing the necessary suspension stroke. On top of that, there is the friction of suspension cylinders in hydropneumatic systems. Their friction originates from the guiding elements as well as the sealing elements of the cylinder.

The friction in the guiding elements can be kept low if the lateral forces in the cylinders are kept as low as possible. Internal lateral forces arise on one hand from external lateral forces, which can originate for example from a clamped fixation of one cylinder end and on the other hand from bending moments superimposed on the cylinder. These lateral forces in the rod guiding element F_{SF} and in the piston guiding element F_{KF} act as normal forces which press the guiding elements onto their friction partners and therefore cause friction forces (Fig. 2.27).

It becomes obvious that the supporting distance e between the rod guiding element and the piston guiding element becomes smaller, the further the rod moves in rebound direction. Apart from a potential risk of buckling or bending the piston rod, the normal forces in rod guide and piston guide increase and the respective friction forces with them – assuming constant external lateral forces and/or constant bending moment.

On the other hand the friction in the elastomeric sealing elements of a suspension cylinder is inevitable, inherent to their functional principle. The sealing elements have to seal off a hydraulic pressure and therefore need a normal force which presses the sealing edge onto the respective opposite surface. An unfavorable by-product of this normal force is boundary friction. In hydropneumatic suspension systems very high pressures need to be sealed off, accordingly high friction forces can be expected. Therefore it is extremely important to put a high emphasis on the optimal design of these sealing elements ([TRA90], [GES97], [FIS06]). Among other

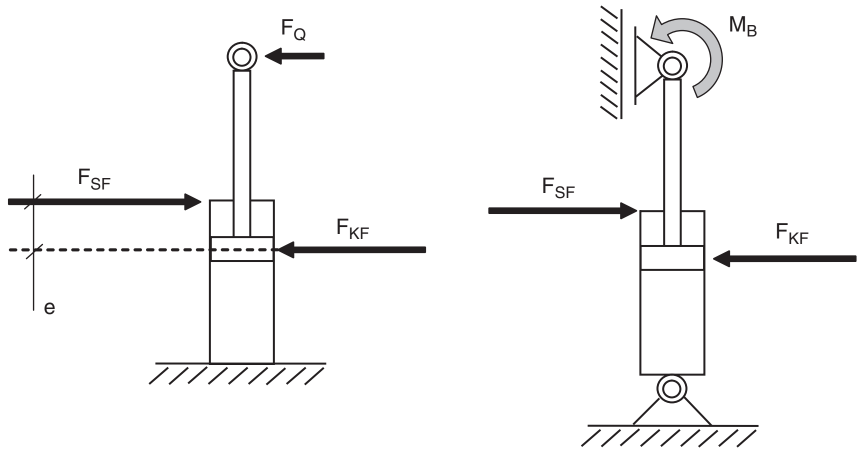


Fig. 2.27 Lateral forces in the guiding elements of a suspension cylinder

influences it is the right choice of the seal geometry and the seal material as well as, in the very beginning of the system layout, the right choice of suspension system pressures and cylinder geometry. In particular the seal diameters have an easily illustratable influence: the larger the seal diameter, the longer the length of the sealing edge(s) and the higher the friction forces.

In this context it is interesting to look at the following design task: a hydropneumatic suspension system with hydraulic preload has to be laid out. Mandatory input parameters are the piston diameter and the hydraulic preload force. For this preload force Eq. (2.36) states that

$$F_{\mathrm{V}} = p_{\mathrm{V}} A_{\mathrm{R}}.$$

This means that the same preload force can be created by a low preload pressure and a large active area on the rodside (and therefore small piston rod diameter) or with a high preload pressure and a small active area on the rodside (and therefore a large piston rod diameter). Looking at the previous explanations in this section, it becomes obvious that it makes sense to go for a small p_{V} and a large A_{R} for two important reasons: the sealed off pressure *and* the length of the sealing edge at the rod seal are lower and both effects result in lower friction forces of the rod seals.

The comparison in Sect. 2.2.5 concerning the two hydropneumatic front axle suspensions with constant hydraulic preload, one with regulated low rodside pressure and one with constantly high rodside pressure (= max. vehicle system pressure) becomes even more interesting in the light of the previous explanations.

A further important factor for friction is the mechanical layout of the suspension system, i.e. the kinematics. Assuming that the mechanical system is based on a simple rocker arm, there are various possibilities in terms of where to integrate the suspension cylinder. The further the cylinder is mounted away from the pivot point, the higher is the necessary stroke of the cylinder in order to provide a certain stroke of the suspension (at the wheels). Provided that the available hydraulic system pressure is obviously constant for all cylinder positions, the piston diameter of the suspension cylinder must increase, the closer the cylinder is positioned to the pivot point of the rocker arm – assuming, of course, the same suspended axle design load for all cylinder positions (Fig. 2.28).

Under the assumption that a certain friction force is created per millimeter of sealing edge (type of seal and sealing pressure constant), it is possible to find a mathematical proof, that the ratio of external cylinder force to cylinder friction force improves, the closer the cylinder is positioned to the pivot point of the rocker arm. The background to this is that the hydraulically active surface increases to the power of two with the piston diameter, while the perimeter and therefore the necessary length of the seal increases only linearly. Yet in reality it has to be taken into account that the above mentioned assumption is not always true if the diameter jump is chosen to be too wide, even if the same type series of seals is used. Furthermore it is important to consider, that the forces in the cylinder bearings as well as in the rocker arm pivot bearing also increase, if the

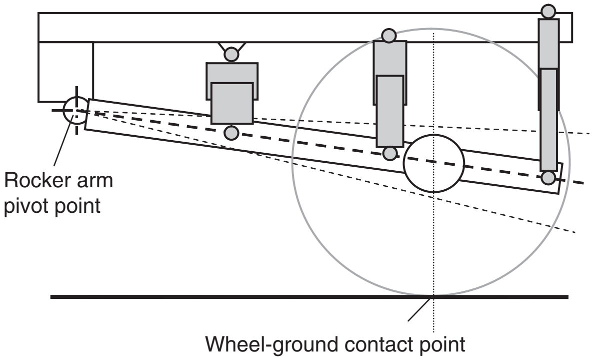


Fig. 2.28 Cylinder dimensions depending of its position relative to the pivot point

suspension cylinder is positioned close to the pivot axis. This also leads to an increase in friction. In the end it is the experiment which will give exact information about the quality of the layout of a suspension system, in particular concerning friction. Therefore friction test stands for hydropneumatic components as well as for complete systems are common in the industry. Interesting insight into latest investigations on friction by the technical university RWTH Aachen can be found in [VER08].

2.3.2 Fluid Friction Damping

The hydraulic fluid in a hydropneumatic suspension system is used as a medium to transfer the pressure on the active areas of the piston to the accumulator(s). Due to the suspension movement and therefore the displacement of the piston, the hydraulic fluid steadily flows between cylinder and accumulator with regularly changing flow direction. If a flow resistor is placed in the fluid flow, the kinetic energy of the hydraulic fluid is transformed into heat due to shear flows inside the fluid. The flow resistor creates a pressure loss, which causes, via the active areas of the piston, a force which counteracts the motion of the piston. This force is therefore taking energy out of the oscillation and hence is a damping force (Fig. 2.29).

$$F_{\mathrm{D,hyd}} = \Delta p A_K \tag{2.45}$$

$$P_{\mathrm{D,hyd}} = F_{\mathrm{D,hyd}} \mathrm{v} = \Delta p \, \dot{V} \tag{2.46}$$

It is typical for fluid friction that the pressure loss depends very much on the amount of volume flow through the flow resistor. This is the reason, why the fluid friction damping force depends, as opposed to the boundary friction, significantly on

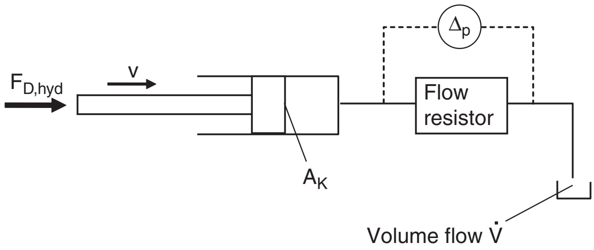


Fig. 2.29 Active principle of fluid friction damping

the speed of the suspension motion. This means that a fluid friction damper adapts even twice to the amount of energy stored in an oscillation: firstly by the amplitude of the oscillation and therefore indirectly secondly (for the same oscillation frequency) also by the velocity of the oscillation.

Simple flow resistors can be divided into two basically different types, which show a different characteristic in the dependency of pressure loss and volume flow.

(a) Throttle: The flow is decelerated by a slow transition of the flow cross-section from wide to narrow and back to wide. The cross-section of a dedicated throttle for defined additional damping usually has a circular shape and is provided by an intentionally small bore in a component in the fluid path between cylinder and accumulator. The small cross-section causes high flow velocities of the hydraulic fluid. Due to the high gradient of flow velocity from the flow center to the inner wall of the bore, high shear forces and therefore high pressure losses are generated. The latter therefore is theoretically/ideally proportional to the volume flow. Another important characteristic of the throttle therefore also is the direct dependency of pressure losses on the viscosity of the hydraulic fluid. This aspect becomes especially important since most of the common hydraulic fluids have a strongly temperature dependent viscosity (Fig. 2.30) which also makes the damping effect of a throttle temperature dependent – this is most cases very unfavorable. Another remarkable fact is the change in viscosity due to the pressure level inside the fluid. The diagram in Fig. 2.30 shows that the kinematic viscosity ν at the ISO reference temperature of 40°C increases by about 50% when the pressure is increased from 0 to 200 bar. This means another advantageous adaptation effect of fluid friction damping in throttles depending on operating conditions: higher loads mean higher hydraulic fluid pressures and therefore a higher fluid viscosity causing higher damping.

The pressure loss across a throttle bore with laminar flow can be calculated by:

$$\Delta p = \dot{V}\ \nu \rho K_{\mathrm{D}} \tag{2.47}$$

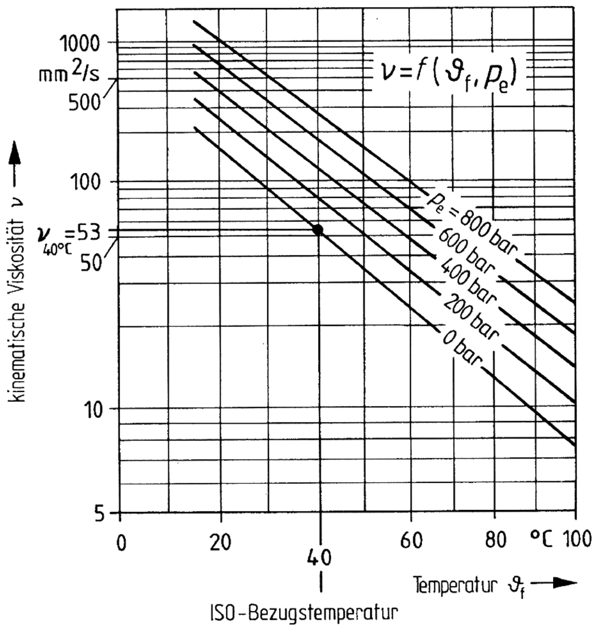
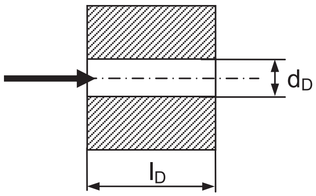


Fig. 2.30 Relationship of kinematic viscosity, temperature and pressure for a typical hydraulic fluid according to [FIN06]

Let K_D be a constant that is related to the geometry and dimensions of the throttle bore while ρ is the density of the hydraulic fluid. K_D can be calculated for the throttle geometry below:

$$K_D = \frac{128\, l_D}{\pi d_D^4} \tag{2.48}$$



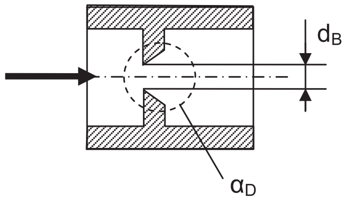
Typical hydraulic components with the character of a throttle are for example tubes, hoses and hose fittings without tight bends, bores with constant diameter in control blocks or also straight pipe fittings with constant inner diameter.

(b) Orifice: The fluid flow is subjected to one or more sudden transitions from a wide to a narrow or a narrow to a wide flow path. This causes strong turbulence in the hydraulic fluid which is the reason for internal fluid friction and hence a transformation of fluid flow energy into heat and therefore, in the end, damping. Ideally this type of flow resistor is characterized by a quadratic dependency of the pressure loss on volume flow. Opposed to the throttle only a minor amount of additional surface is in contact with the fluid flow in regions with high flow velocities. Therefore this type of resistor is ideally not depending on fluid viscosity and therefore temperature.

$$\Delta p = \dot{V}^2 \; K_B \tag{2.49}$$

Let K_B be a constant that is related to the geometry and dimension of the throttle as well as the density of the hydraulic fluid. K_B can be calculated for an orifice:

$$K_B = \frac{8\rho}{\alpha_D^2 \pi^2 d_B^4} \tag{2.50}$$



The parameter α_D is called the flow coefficient and depends mainly upon the geometry of the inlet edge and the Reynolds number.

Typical hydraulic components with the character of an orifice are for example components with changes in flow direction especially with a low turning radius (for example elbow fittings or crossdrills in control blocks), furthermore components with sudden changes in cross-section for example the bore in the cylinder wall for the hydraulic connection of pistonside/rodside or often also fittings with a wide jump in sizes on their connectors. Figure 2.31 compares the behavior of the (ideal) throttle and orifice and also shows the influence of fluid viscosity.

More information about the throttle and orifice flow resistors can be found in the respective literature for hydraulics basics (for example [MUR01], [FIN06] and [EBE74]) and is therefore not further explained here. In these books further details can be found about flow resistance of various line routing elements. These apply to hydropneumatic suspension systems, just like they apply to all other hydraulic systems.

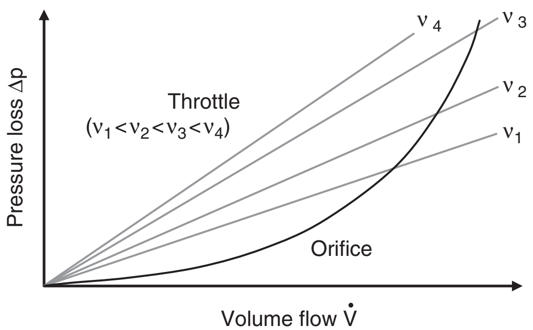


Fig. 2.31 Pressure loss Δp as a function of volume flow and viscosity

In reality you almost never have a flow resistor of purely one type but mostly a mixture of the basic throttle and orifice resistors. Therefore it is often better to assign the attribute “throttle character” or “orifice character” to a flow resistor, depending on which character prevails.

In general a damping is preferred which is not depending upon fluid temperature (viscosity). Therefore, at first sight a flow resistor with a strong orifice character seems to be the best choice. But caution is necessary here. Owing to its very nature this flow resistor dampens out oscillations with low amplitudes only very weakly and therefore causes a long reverberation time. Furthermore, especially in axle wheel suspensions, this also can cause a sensation of a “loose connection” between input side and isolated side, making the driver think that he is not fully in control of the vehicle. On the other hand the orifice reacts very strongly to heavy excitations for example when driving through a pothole. Due to the quadratic relationship of pressure drop and volume flow, the Δp will be very high, causing a high damping force and therefore high accelerations on the isolated side. A damping system of this type therefore often is only partly satisfactory.

This is the reason why special flow resistors have been contrived which often provide a rather strong basic damping already at low piston velocities by using one of the above mentioned flow resistors combined with a pressure reducing valve to avoid extreme damping forces. The high basic damping is favorable especially in axle/wheel suspensions since this provides a good feedback from the suspension system to the driver. Furthermore the higher damping forces provide better driving safety during for example evasive maneuvers, fast lane changes or “sporty” driving in general due to the reduction of the roll motion of the vehicle body. Yet to ensure that this higher basic damping does not reach extreme values for high piston velocities, the basic flow resistor is bypassed by a special kind of pressure relief valve. It opens at high differential pressures and therefore keeps the pressure loss of the overall damping valve arrangement (and with it the damping forces) on an acceptable level from a comfort perspective (Fig. 2.32, further information can be found in Sect. 4.3.2). The start of the bypassing through the pressure relief valve can be identified as the sharp bend in the damper’s characteristic curve. It is a further advantage of this valve that the opening point of the pressure relief

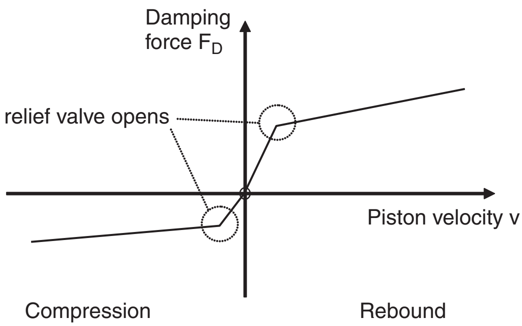


Fig. 2.32 Damping force as f(v) for an automotive shock absorber

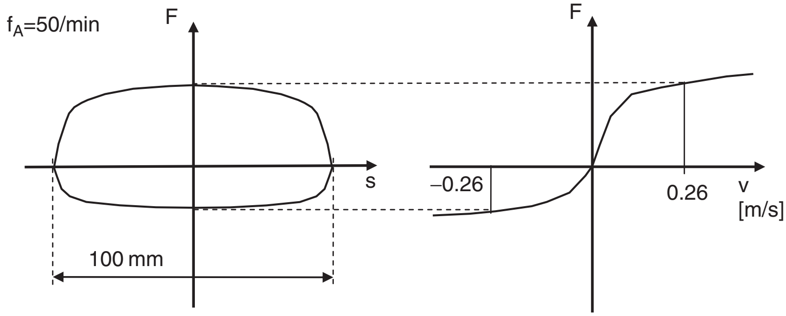


Fig. 2.33 Force–displacement curve and force–velocity curve derived from it

valve is mostly independent from fluid temperature, so the limiting of the damping forces always starts in about the same range which results in a fairly constant suspension behavior at different temperatures. Please consider that *only* in Figs. 2.32 and 2.33, opposed to the regular definition in this book, the force and the displacement during the compression stroke are negative and during the rebound stroke are positive, since this is a frequently found definition in shock absorber technology.

The force-velocity curve for a damper is typically derived from force–displacements curves recorded for an amplitude of 100 mm at different excitation frequencies f_A. The excitation frequency then determines the maximum piston velocity of the damper during the crossing of the center position between both ends of the stroke. Therefore experiments of this kind provide information about the damping force as a function of piston velocity. By extracting the maximum force (rebound) and minimum force (compression) at the center position for various excitation frequencies and transferring them into a diagram of force vs. piston velocity, the characteristic curve for the damper is obtained. Figure 2.33 shows the step from the force–displacement to the force-velocity curve for an excitation frequency of 50/min.

The above diagrams show that the damping forces for compression are lower than for rebound. This is at first sight an unsteady distribution of damping forces. Yet it takes into account that compression motions (for example when riding over an obstacle) often cause higher piston velocities than rebound motions. Furthermore it is obvious that the increasing spring force during compression adds to the damping forces and helps to decelerate the piston velocity, while during rebound motion, the spring forces decrease and therefore damping needs to take over more of the decelerating effect. The shown ratio for rebound damping force to compression damping force of about 2:1 provides a more effective and more comfortable reduction of the accelerations on the isolated side as would be possible by equal damping forces in compression and rebound.

Due to the path of the hydraulic fluid from the active area of the cylinder all the way to the accumulator diaphragm, a certain inevitable basic fluid damping of the hydropneumatic suspension system is caused by the hydraulic lines and fittings between both ends. It basically depends upon the sizing of these components. In many suspension systems on the market the fluid friction damping is purely defined by these line elements. In this case ideally they have been tested and readjusted/selected for correct layout in specific tests. However some systems show indications that this has not been or insufficiently carried out. Especially systems with long distances between suspension cylinder and accumulator show only partly the potential effect which would be achievable with shorter and/or larger diameter lines. This is why it is necessary to ensure from the very start of a design of a suspension system that suspension cylinder and accumulator are positioned close to each other. The lower the basic damping, the better the possibilities for influencing the damping characteristic of a system by targeted integration of additional damping elements.

This possibility to integrate specific damping components can be used to further adapt the damping, depending on the spring rate. This is a major advantage in particular in hydropneumatic suspension systems with their wide range of spring rates depending on spring load and preload. If the operating conditions are changed, for example by a change in static spring load, it is good to adjust the spring rate in order to get (back) to the desired natural frequency, but it is better to adjust the damping characteristics on top of that, so the dissipation of oscillation energy is also adapted to the new demands.

Such load adaptive damping systems are already available for example for trucks with air suspended axles. The damping elements are adjusted by the average pressure in the air bags and therefore adapted to the static spring load (ZF Sachs PDC – Pneumatic Damping Control [MUR98], [CAU01]). For a hydropneumatic suspension this could be done for example by an adjustable flow resistor which is either operated hydraulically directly by the pressure on the pistonside or operated electrically by an electronic controller which reads the pistonside pressure by a pressure sensor.

The electrical adjustment offers the possibility to include further information in the algorithm for the right selection of the necessary damping forces. Significant, meaningful parameters are for example the temperature of the hydraulic fluid (and therefore viscosity), certain operating conditions or driver settings. A sufficiently fast adjustable system can even provide a semi-active damping, for example via the Skyhook-algorithm, or an end-of-stroke damping (see Sect. 2.3.3). The adjustment of the flow resistor does not necessarily have to be continuous, like for example in the ZF Sachs CDC system (Continuous Damping Control, [REI05], [EUL03], [CAU01]). An adjustable damper with a stepped characteristic might also provide good results, for example like the Bilstein ADS (Adaptive Damping System) which allows for an individual two step adjustment for both compression and rebound damping [SCM00].

2.3.3 End-of-Stroke Damping

> "A passive suspension system, which never gets close to its end stops during all operating conditions, is probably not tuned to be soft enough or wastes suspension travel."

This kind of provocative sentence contains important information. Every suspension system only has a limited stroke available to isolate the excitations coming from the input side. Basically the softer a suspension system is tuned (keeping the correct relation of spring rate and damping in mind), the more it will reduce accelerations on the isolated side, yet the longer will be the displacement between input side and isolated side for certain excitations. If these excitations exceed a certain limit, the necessary displacement becomes greater than the available suspension stroke and therefore the suspension bottoms out. This causes short term high forces and accelerations which reduce subjective comfort and furthermore can overload components of the suspension system as well as components on the input side or isolated side.

One way to avoid this problem would be to tune spring and damper to a harder level, so that even under the worst conditions and most extreme excitations the available suspension stroke is always sufficient. Yet, in doing so, the result is a reduced comfort level in all other operating conditions. Therefore it is important to also consider the expected frequency and amplitude distribution of the various excitations to find an optimum level for spring rate and damping. When taking this into account it becomes obvious that it can be quite acceptable to have the suspension bottoming out slightly sometimes, if in the same turn the overall comfort level in all other operating conditions is improved by a softer setting. A suspension system can be even tuned to be softer than that if an additional end-of-stroke damping is used and the (rare) cases of bottoming out are softened by an additional damper or an additional spring. All in all this results in a remarkable gain in comfort.

To reduce the harshness of a bottoming out event at the end of the stroke it is necessary to reduce the velocity of the piston relative to the cylinder. So if the piston gets close to the end positions (for example the last 10% of the stroke in each direction) an additional decelerating force (damping or spring) needs to be activated. Most suitably this additional force creates a constant or slightly progressive gradient of velocity over displacement, meaning the deceleration is constant or increases slightly as the piston gets closer to the cylinder bottom. Ideally the end-of-stroke damping system recognizes the excessive kinetic energy which needs to be dissipated until the end of the stroke and then adapts its properties in a way such that a constant and lowest possible force level decelerates the piston until shortly before the end stop.

Many suspension systems use *elastomer elements for end-of-stroke damping*. Just before hitting the end stops, the suspension motion is decelerated by an additional elastomer spring with a minor amount of damping. So strictly speaking this is more of an end-of-stroke spring than an end-of-stroke damping. The spring force and therefore also the deceleration of the piston velocity increases from the first contact to the elastomer up to the mechanical end stop. The characteristic curve of deceleration force vs. displacement is at least a linear increase but in most cases even a disproportionately higher increase, and can be shaped for example by the outer contour of the elastomer element, by internal bores or even by collars supporting the circumference of the elastomer. This layout allows soft cushioning of minor impacts with small spring forces and, on the other hand, taking even the most extreme bumps without the bottoming out of steel parts. In passenger cars this type of end-of-stroke damping is then clearly noticeable for passengers, yet it fulfills the requirement to protect the components from overload. During the rebound motion out of the end stop, the elastomer extends back to its original shape and reintroduces most of the absorbed energy back in to the suspension system. Due to a slight damping effect, a minor amount of energy remains as heat inside the elastomer, this behavior is characterized by the loss angle of the elastomeric material.

A major disadvantage of the elastomer elements is the fact that the material is subjected to strong aging and settlement depending on the extent of use and the stresses induced therewith as well as the environmental conditions (UV-radiation, ozone, chemicals, etc.). This makes an exchange of the elements necessary in some applications. In order to prevent overloading of the elastomer elements, in some cases an additional mechanical end stop is designed into the system which limits the stroke and therefore reduces the maximum deformation of the elastomer to a level which is acceptable for the material in long term.

In hydropneumatic suspension systems another type of end-of-stroke damping is popular since it can be designed into the suspension cylinder. Theoretically a use of elastomer elements is possible here as well, but mostly it is the *hydraulic end-of-stroke damping* that is used for these cylinders. Opposed to the elastomer elements, here it is not an additional spring force with minor damping but purely an additional damping force that decelerates the piston velocity.

The effect of the hydraulic end-of-stroke damping is achieved by reducing the cross-section of the oil path out of the cylinder when the piston reaches a freely selectable distance to the end stop. So during a compression stroke the pistonside chamber is active while during a rebound stroke the rodside chamber is active for end-of-stroke damping. A pressure drop across the flow resistor is generated which then causes a pressure increase inside the respective cylinder chamber. The active area of the respective cylinder chamber is subjected to this additional pressure and therefore causes the damping force.

If the cross-section area of the additional flow resistor is designed to be variable with cylinder stroke, a possibility is created to define the effect of the flow resistor depending on piston position. This way a more constant end-of-stroke damping force level and a lower maximum force can be achieved compared to a flow resistor with constant cross-section area. The lower force peak also reduces the maximum accelerations due to end-of-stroke damping Eq. (2.34). Please consider: It is always the same amount of energy that has to be dissipated throughout the displacement of the end-of-stroke damping. Therefore the damping force – displacement integral of both curves must be identical. More information about a suitable layout of the end-of-stroke flow resistor can be found in Sect. 3.2.4.

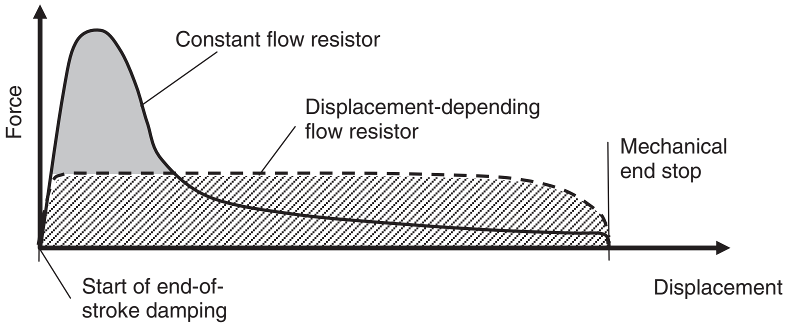


Fig. 2.34 Damping force–displacement curve for end-of-stroke damping with a constant and a displacement-depending flow resistor

2.4 Combined Operation of Spring and Damper

In Sects. 2.1, 2.2, and 2.3 the individual force components of a hydropneumatic suspension system have been described. In this section they are combined and considered as a whole system. The basis for the following explanations is a sinusoidal excitation of the input side while the isolated side is fixed. Therefore the suspension element's displacement is a sinusoidal oscillation. These conditions are chosen similar to the method for the determination of characteristic curves for regular automotive shock absorbers, described at the end of Sect. 2.3.2 (Fig. 2.33). However the main focus will be on the force–displacement curve.

The force–displacement curve can be synthesized from the individual theoretical curves for the gas spring, for boundary friction and for fluid friction – and furthermore, if applicable, the curve for end-of-stroke damping. The here mentioned example is a hydropneumatic suspension system without preload which is subjected to boundary friction and which has a simple throttle to provide fluid friction damping. The theoretical considerations of Sects. 2.1, 2.2, and 2.3 result in the individual curves shown in the Fig. 2.35.

The effect of the gas spring is idealized and therefore the same for compression and rebound which make the characteristic curve only one line. In reality this is almost true; the dissipation of spring energy due to heat rejection of the accumulators is very small so virtually no hysteresis can be detected in this curve (a). On the other hand the characteristic curves (b) and (c) show a significant hysteresis. The curve for boundary friction is exaggerated for better illustration; these friction forces should be lower in practice. In the end points (left and right) at $v = 0$ m/s there is a transition from sliding friction to static friction to sliding friction, this explains the slight force peaks there due to the higher coefficient of sliding friction. Furthermore it is visible, that the friction forces are higher, the more the suspension is compressed. This is due to the dependency of seal friction forces on hydraulic pressure. The viscous damping forces reach their extreme values when

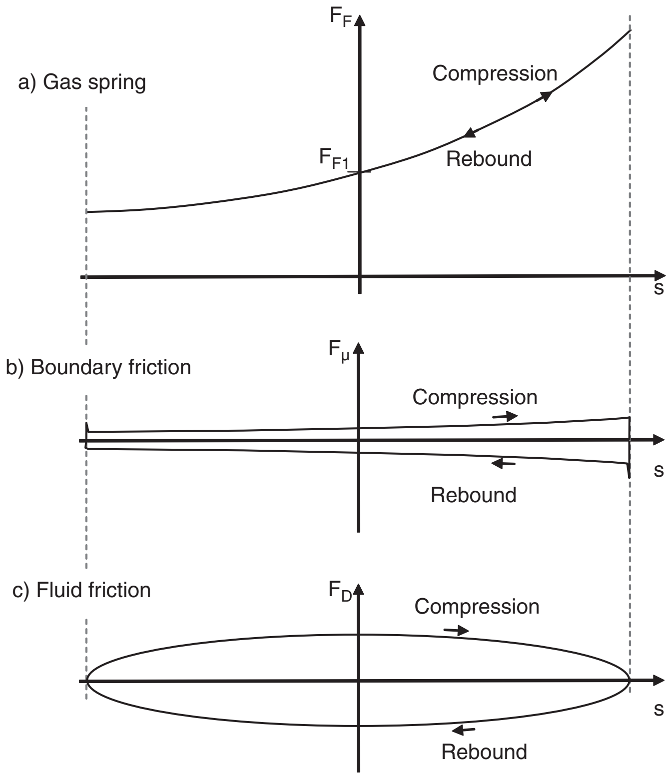


Fig. 2.35 Force–displacement curves for gas spring, boundary friction and fluid friction

the center position is crossed. Minimum and maximum have the same absolute values which indicates that the flow resistor has the same effect in both flow directions. By adding up the individual characteristic curves to one curve, the following characteristic force–displacement diagram for the hydropneumatic suspension is composed (Fig. 2.36).

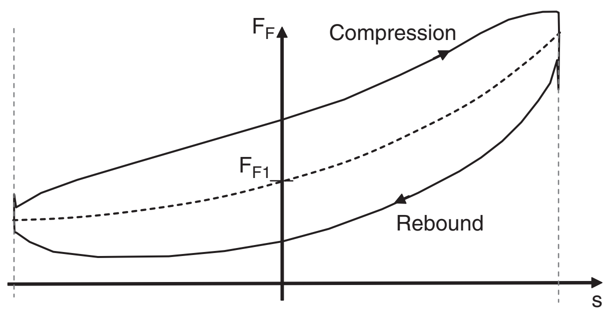


Fig. 2.36 Characteristic force–displacement diagram for hydropneumatic suspensions

This type of diagram will be found in a more or less similar shape in every experiment with a hydropneumatic suspension system when recording the force–displacement history. Such measurements provide information about the detailed character of the individual contributions spring rate, boundary friction and fluid friction. This information can be derived from the main curve by splitting it up into its individual components as shown above.

Even more information can be derived when the measurements are taken at different amplitudes, static spring loads, oil viscosities (or temperatures) or with different excitation frequencies. Then conclusions can be drawn such as:

(a) which adiabatic exponent has to be chosen for a calculation under the respective conditions (derived from the shape of the pure characteristic curve of the spring and matching it with the calculation);
(b) whether the fluid friction damping is more like an orifice or more like a throttle (derived from relationship of damping forces to viscosity and dependency of damping forces on oscillation amplitude and frequency);
(c) the magnitude of static and sliding friction and the influence of cylinder pressures on friction (derived from comparisons of measurements at different amplitudes and static spring forces).

Figure 2.37 shows an actual measured force–displacement diagram of a hydropneumatic suspension. It is possible that the actual magnitude of the static friction forces has not been completely recorded due to an insufficient sample rate. However for the detailed measurement of friction forces dedicated experiments with very low frequencies (for example 0.01 Hz or 0.1 Hz) are recommended. This fully eliminates the influence of fluid friction. Simulation results like in [HYV01] show the same shape of their force–displacement curves as shown in Fig. 2.37.

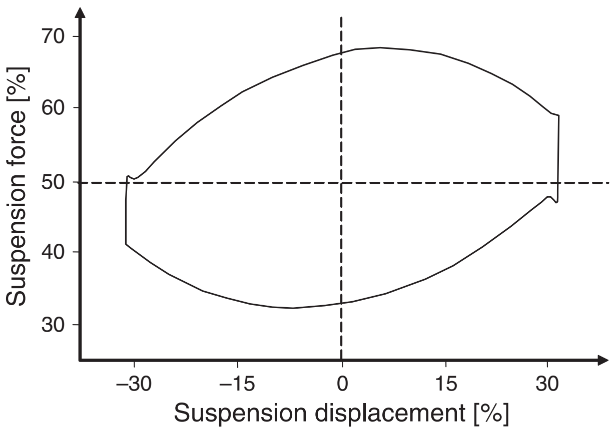


Fig. 2.37 Force–displacement diagram taken from actual measurements