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Quasi-Isobaric Compressed Air Reservoirs

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Abstract

State of the art compressed air reservoirs are designed in form of isochoric cavities. This implies that all air mass changes, either due to mass demand from the consumers or delivered air from compressors, and temperature changes lead to inherent pressure changes. Therefore, huge reservoir volumes are installed, which necessitate regular and expensive inspection works, and/or significant pressure fluctuations are tolerated, which require inefficient compressor operation at higher supply pressures.

This paper focuses on alternative compressed air storage systems enabling significantly higher capacity to volume ratios compared to conventional reservoirs. The analytically investigated concepts rely on air storage in different sorts of customized volume variable chambers acting against application-optimized arrangements of (hydro-)mechanical loading systems. The resulting benefits and weaknesses of the different variants are discussed with respect to their operational behavior, building space, as well as the requirement for recurring inspections. The paper concludes with an assessment of the required steps for realization of the best suited concept and the opportunities which arise from industrializing the investigated storage systems.

Keywords. Compressed Air Reservoir, Isobaric Storage.

1. INTRODUCTION

1.1. Usage and dimensioning of state-of-the-art reservoirs

Pneumatic reservoirs are widely used in industrial and mobile applications. In literature, two different usage scenarios are commonly distinguished: Firstly, application downstream of air compressors to stabilize network pressure for quasi continuous air demand, especially in industrial applications, and secondly, local application in pneumatic machines and networks to prevent pressure drops during short-term increased air demands.

In the first case, main task of the reservoir is to allow the pressure of the air distributing network to stay in a predefined margin despite fluctuating air supply and demand. For simple compressor controls, which only allow for switching selected compressors on and off, a minimally required cycle time for switching of the compressors usually exists to prevent the machines from inefficient operation, overheating, and excessive wear.

Varying sizing rules for the required reservoir volume can be found in different handbooks on the design of compressed air systems, e.g. in [1, 2]. These are based on the ideal gas law and a mass balance of inflowing and outflowing air of the reservoir. While the equation in [1] allows sizing for a defined compressor and specific air demand, the worst case of a 50% duty cycle is considered in most cases, as usually, the air demand of a production plants is time variant, and the system has to operate correctly under all circumstances. The equation in [2] includes this assumption of a 50% duty cycle and additionally allows to take varying inlet pressures and temperatures of the compressor and reservoir temperatures into account. By setting these values to standard conditions, both equations are identical and equal equation 1.1. The sizing of the reservoir volume V_{res} is given as a function of the allowable pressure fluctuation Δp and cycle time t_{cycle}, with the pressure at standard conditions p_0, and the volumetric flow of the biggest installed compressor at standard conditions Q_{Comp}.

$$V_{res} = \frac{p_0}{4\,\Delta p} \cdot Q_{comp} \cdot t_{cycle} \tag{1.1}$$

An alternative sizing approach is used, when pneumatic reservoirs are implemented to allow for peak flow Q_{peak} delivery over a defined (short) time Δt [1].

$$V_{res} = \frac{p_0}{\Delta p} \cdot Q_{peak} \cdot \Delta t \tag{1.2}$$

In both cases, pressure fluctuations occur, and the system pressure needs to be selected to allow for correct function of all systems downstream the reservoirs under all operational conditions. In general, this means that the maximum outlet pressure of the compressor must be higher than the required pressure in the application by the sum of all permissible pressure fluctuations of the reservoirs connected in series and the line losses. The impact of this additional pressure is discussed in the next subchapter.

1.2. Influence of pressure margins on system efficiency

It can be assumed as a rule of thumb that well-dimensioned and well-maintained compressed air stations require approximately 145 % of the energy that would be necessary for isentropic compression to generate compressed air [3]. Hence, an estimate for the mass specific energy demand w of a compressed air station, which delivers the pressure p_{out} from ambient air with pressure p_{in} and temperature T_{in} is given by equation 1.3, wherein c_p denotes the isobaric heat capacity and κ the isentropic number [4].

$$w(p_{out}) = 1.45 \cdot c_p \cdot T_{in} \cdot \left[\left(\frac{p_{out}}{p_{in}}\right)^{\frac{\kappa-1}{\kappa}} - 1\right] \tag{1.3}$$

For the simplified case of a compressor with on/off control, which delivers, when switched on, a pressure independent mass-flow into a downstream reservoir with time invariant outflow, the cyclic pressure increases and decreases in the reservoir are linear over time. The average energy demand of the compressor $\overline{w}$ at a pressure between $p_{out} = p_{req}$ and $p_{out} = p_{req} + \Delta p$ can be calculated by equation 1.4.

$$\overline{w} = \frac{\int_{p_{req}}^{p_{req}+\Delta p} w(p)\ dp}{\Delta p} \tag{1.4}$$

The following figure shows the impact of the pressure control band on specific energy demand for compressed air delivery. It becomes evident that several percent in efficiency can be gained simply by reducing the pressure control band, which is aspired by actual more complex compressor control schemes.

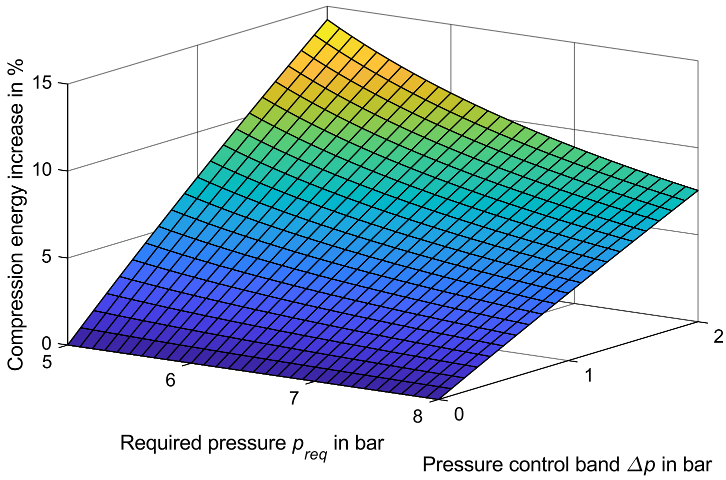

Figure 1.1. Influence of pressure control margin on energy demand for compressed air generation

Despite the derivation of the preceding equation under the assumption of a constant reservoir outflow, the above stated results are generally applicable to all kinds of systems with time-variant compressed air demand as long as no timewise coupling of reservoir outflow and compressor cycles exists, which is rarely the case in industrial systems.

1.3. Normative demands on certification and inspection of conventional reservoirs

The stored energy and associated risks of compressed air reservoirs grow with pressure and volume. Therefore, reservoir classification by means of the product of rated pressure and volume is used to specify design, manufacturing, documentation, commissioning, and recurring inspection procedures and requirements. Figure 1.2 depicts the categorization of reservoirs according to Directive 2014/68/EU by reservoir volume V and rated pressure PS. The categories are numbered in order of increasing demand in all aforementioned areas. [5]

Additionally, directive 2014/29/EU handles the making available on the market of simple pressure vessels, which cover welded pneumatic reservoirs manufactured in series for stationary use up to a PS·V product of the vessel of 10,000 bar·l and the resulting obligations of manufacturers, importers, and distributors. [6]

In the scope of this paper, the impact of quasi-isobaric reservoirs on system integrator and machine user duties are focused. In Germany, the required inspection tasks and intervals at commissioning and during the use phase are regulated in Annex 2, Table 4 of the *Betriebssicherheitsverordnung* (Operational Safety Regulation) [7]. Therein, it is distinguished between internal inspections, which require the reservoir to be depressurized and accessible, and strength tests, which require all safety valves to be dismounted and the reservoir to be entirely filled with water for pressure and leak tightness testing. Obviously, such tests are time consuming and thus expensive.

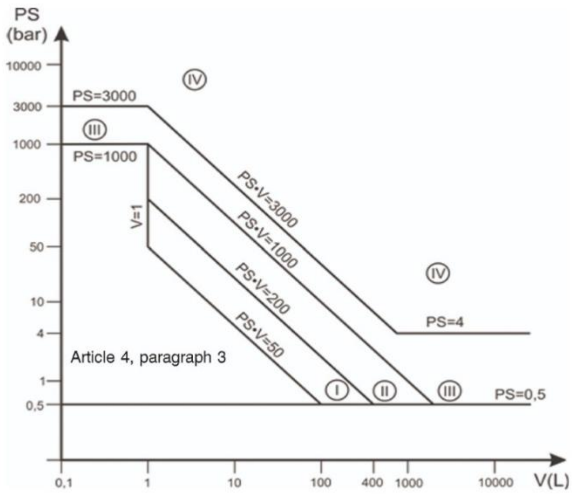

Figure 1.2. Classification of compressed air reservoirs according to [5]

The required inspection tasks for a specific reservoir depend on the categories of Directive 2014/68/EU, with the sole exception that reservoirs, which are designed to store internal relative pressures of less than 1 bar, are exempt from inspections regardless of their size. Table 1 provides an overview of mandatory inspection tasks during commissioning and in operation. Herein, it is distinguished between the inspection by notified bodies (NB) and user inspectorates (UI).

Category	**Commissioning**	**Periodic internal inspection**	**Periodic strength test**
Article 4, paragr. 3	-		
I	UI	UI (all 10 years)	UI (all 10…15 years)
II	NB		
III & IV		NB (all 5 years)	NB (all 10 years)

Table 1: Inspection duties of compressed air reservoirs

It becomes obvious that sizes, which are covered by article 4, paragraph 3 are very favorable, as no inspection duties result. As the product of pressure and volume for this category is limited to 50 bar·l, only small reservoirs fulfill this requirement, e.g. for a rated pressure of 16 bar, reservoir size is limited to 3.125 l. In case of higher required volumes, categories I and II at least do not require an external supplier for inspection. Nevertheless, training the user inspectorate and shutdown of the reservoir are required for inspection, still resulting in significant costs. For reference, pneumatic reservoirs with up to 12.5 l and 16 bar pressure ratings are covered by category I, while category II allows for up to 62.5 l.

1.4. Dimensioning of quasi-isobaric reservoirs

Quasi-isobaric storage of compressed air demands for a volume variable reservoir. Therewith, pressure fluctuations can be minimized, leading to higher efficiency of the compressed air supply or better system performance, and the installed reservoir volume can be drastically reduced. The volume change ΔV_{QIB} of a quasi-isobaric reservoir with comparable operational behavior to a conventional reservoir with Volume V_{res} is given by equation 1.5.

$$\Delta V_{QIB} = V_{res} \cdot \frac{\Delta p}{p_{req}} \tag{1.5}$$

Especially for small allowable pressure changes, huge differences in volume occur. If a quasi-isobaric reservoir is used for optimization of compressor cycling, equation 1.5 can be inserted into equation 1.1 and the variable reservoir volume in equation 1.6 results.

$$\Delta V_{QIB} = \frac{1}{4} \cdot \frac{p_0}{p_{req}} \cdot Q_{comp} \cdot t_{cycle} \tag{1.6}$$

For peak flow demands, equation 1.2 and 1.5 are combined and lead to equation 1.7.

$$\Delta V_{QIB} = \frac{p_0}{p_{req}} \cdot Q_{peak} \cdot \Delta t \tag{1.7}$$

1.5. Hysteresis losses on volume variable cavities

Several designs of volume variable cavities are used in actual pneumatic actuators. While cylinders with sliding seals are the common solution, also diaphragm arrangements exist. Seal friction or plasticity of sealing diaphragms leads to hysteresis of such actuators, which ultimately results in pressure deviations in the investigated reservoirs between filling and discharge. Moreover, the breakaway force of friction loaded sealing systems is especially relevant for the application, as it leads to dynamic excitation.

Multiple investigations demonstrate the strong impact of pressure and standstill time on the breakaway force of pneumatic cylinders [8, 9]. With the assumption of rare and short standstill periods of the accumulator piston in operation, a standstill time of less than 10 s can be assumed. In this case, *Meuser's* measurements on a cylinder with 25 mm diameter show a breakaway force per seal length of up to F_{FrL} = 0.28 N/mm at a differential pressure of 6 bar [8]. Slightly lower values have been measured by *Wakasawa et al.* for a cylinder with 32 mm in diameter [9]. Equation 1.8 estimates the friction force induced pressure hysteresis Δp_{Fr} of a cylinder based quasi-isobaric reservoir with piston diameter D.

$$\Delta p_{Fr} = 2\frac{D \cdot \pi \cdot F_{FrL}}{D^2 \cdot \frac{\pi}{4}} = 8\frac{F_{FrL}}{D} \tag{1.8}$$

The reducing effect of seal friction with increasing diameter is based on the quadratic increase in pressure loaded surface counteracting the only linearly increasing sealing length. Based on *Meuser's* data, for a piston diameter of 200 mm, a hysteresis in pressure of Δp_{Fr} = 112 mbar between filling and discharge occurs.

Even more relevant for the application is the reduction in friction force over speed at small velocities and the change in friction force between acceleration and deceleration of the piston in the mixed lubrication regime, which lead to stick-slip behavior and corresponding velocity and pressure oscillations. According to measurements from *Fujita et al.*, the friction force at deceleration to standstill can be as low as half of the breakaway force [10]. Therewith, pressure fluctuates during charging and discharging a quasi-isobaric piston reservoir by Δp_{Fr} / 4, which equals ca. 28 mbar for the before stated example of the 200 mm piston, as shown in figure 1.3.

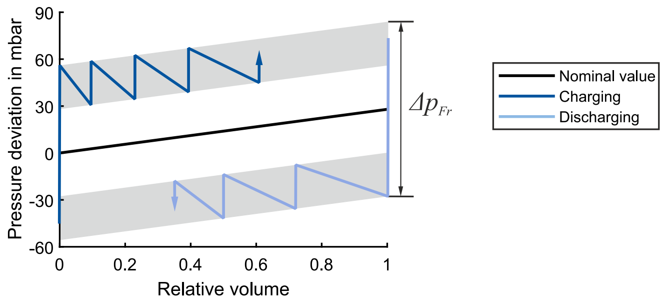

Figure 1.3. Schematic pressure cycle of a seal friction subjected quasi-isobaric reservoir

By using diaphragms for sealing of the quasi-isobaric reservoir, stick-slip effects and pressure deviations can be significantly reduced. Nevertheless, deformation of the elastomeric material with its visco-elastic material properties and possible friction between the deforming diaphragm and housing still contribute to some deviations in pressure.

A quantification of these effects in terms of pressure deviations in the aspired application is hindered by the strong impact of material selection and geometric design of the diaphragm. Therefore, only a rough estimate from comparable devices in literature can be achieved. *Bedarff* models in his PhD-thesis an active air spring for passenger cars. For model validation, a cyclic motion at constant pressure is experimentally investigated, which corresponds exactly to the application in a quasi-isobaric reservoir. The hysteresis amplitude in force amounts to ca. 0.6 % of the nominal value at an outer diameter of ca. 150 mm [11]. Hence, by using a rolling diaphragm, pressure hysteresis of the quasi-isobaric reservoir can be reduced by half in comparison to a conventional cylinder if operated at 6 bar_{rel}. Also, in the case of diaphragms, the use of higher diameters leads to reduced hysteresis.

1.6. Intermediate conclusion – Potential of quasi-isobaric reservoirs

Increasing efficiency of industrial compressed air systems requires lower pressure control bands to be realized. Thereof, the problem arises, that conventional reservoirs in compressor stations have reduced effectiveness in storing and providing compressed air to the systems, which can only be counteracted by raising reservoir size. Usage of quasi-isobaric reservoirs decouples the pressure control band from reservoir size and allows significant reduction in reservoir volume in comparison to actual design rules. Additionally, using quasi-isobaric reservoirs enables downsizing pressure loaded volumes, which can have significant effect on commissioning and inspection by the classification in lower categories according to Directive 2014/68/EU. Despite seal friction and material losses of diaphragms hinder true isobaric behavior, increasing the piston or diaphragm diameter reduces these effects to a reasonable value, thus allowing for quasi-isobaric systems.

2. CONCEPT DEVELOPMENT AND ANALYSIS

Concepts, which allow quasi-isobaric conditions in pneumatic reservoirs, require the reservoir volume to increase drastically with raising pressure in the aspired operating point. Meanwhile, the mechanical work performed by the moving walls of the reservoir under pressure needs to be stored, to allow for later deflation of the reservoir against system pressure. Out of this simple system description, the system structure for all kinds of isobaric reservoir concepts gets obvious:

Grid ↔ Variable volume with force takeoff ↔ Energy transmission ↔ Energy storage

Thereof, different concepts can be deduced and evaluated. The following table shows an overview of all possible solutions investigated in this paper.

Concept	Reservoir volume	Transmission	Energy storage
1	Pneumatic cylinder	---	Lifting of a weight
2	Pneumatic cylinder	---	Mechanical spring
3	Pneumatic cylinder	Hydraulic cylinder	Hydraulic accumulator
4	Nonlinear actuator	---	Mechanical spring

Table 2: Investigated concepts

In the following subchapters, the different concepts are presented and discussed in terms of filling state induced pressure deviations, adjustability in pressure, and building space. These findings are the basis for concept comparison and decision in the subsequent chapter 3.

2.1. Concept 1: Weight loaded pneumatic cylinder

The simplest concept of a constant pressure reservoir for pneumatic systems consists of a volume variable pneumatic chamber, e.g. a cylinder or bellow, which is loaded by a significant weight. Such arrangements can be found e.g. in the air supply of pipe organs to allow for constant pressure in the pneumatic system [12]. Also in process technology, e.g. for storage of biogas for power generation, comparable arrangements are used with huge storage volumes [13]. Both systems have in common, that the load is integrated into the reservoir piston. This approach is not feasible for typical pressures of pneumatic systems, as even for a steel weight with a density of approx. 7,800 kg/m³, a required pressure of 6 bar_{rel} would result in a height of the weight of approx. 7.8 m. Therefore, an alternative configuration as shown in figure 2.1 is further investigated. It is characterized by a weight, which surrounds a pneumatic cylinder with stroke dependent volume $V_{QIB}(s)$, thus enabling long strokes and compact design. The maximum displacement volume of the reservoir ΔV_{QIB} hence equals the difference between $V_{QIB}(s_{max})$ and $V_{QIB}(s = 0)$ with s_{max} beeing the stroke length. For this case, the width a of the weight with quadratic cross-section and length l can be estimated by equation 2.1 under the simplifying assumption, that the length of the weight l and the cylinder length s_{max} are equal. For given pneumatic parameters, weight density ρ, and gravitational constant g, the length l is the only remaining optimization parameter. For a constant displacement, the assembly gets significantly thinner and lighter with increasing stroke length while material demand and building space are reduced. On the other hand, higher strokes and smaller piston diameters lead to increased velocities, friction, and wear on the piston sealings.

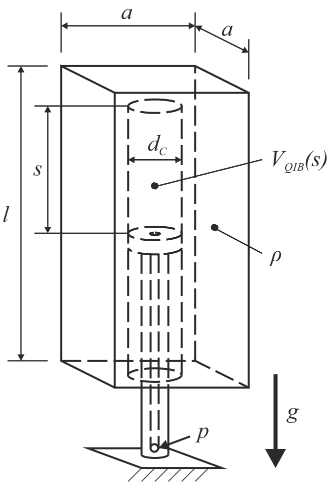

Figure 2.1. Investigated weight loaded cylinder

Equation 2.2 gives a rough approximation of the building space $V_{BS,1}$ required.

$$a = \sqrt{\Delta V_{QIB} \cdot \frac{p - p_0 + \rho g l}{\rho g l^2}} \tag{2.1}$$

$$V_{BS,1} = 2a^2 \cdot l \approx 2 \cdot \Delta V_{QIB} \cdot \frac{p - p_0}{\rho g l} \tag{2.2}$$

Regarding system dynamics, the proposed system allows to deliver constant pressure over the stroke in steady state but represents a dynamic first-order system with very low damping and small eigenfrequency. The undamped eigenfrequency in the highest position, which is the lowest eigenfrequency in operation, can be calculated by equation 2.3 and results directly from the stroke length of the system for given operational parameters [4]. For strokes of 1 to 10 m and usual pneumatic parameters, natural frequencies of the system are in the range of 0.15 to 0.7 Hz. These values are further reduced by the capacity of the connected pipework in an application.

$$f_0 = \frac{1}{2\pi} \cdot \sqrt{\frac{g}{l}} \cdot \sqrt{\frac{n \cdot p}{p - p_0}} \tag{2.3}$$

The usability of passive damping measures to optimize oscillation stability is limited, as these cause speed and thus flow dependent deviations from isobaric conditions. Therefore, weight loaded isobaric reservoirs are only applicable to systems with significantly lower dynamics than their own, which can be the case for central reservoirs in compressed air supply. An application for smoothing dynamic load peaks is not feasible. Furthermore, the huge weight of the system prevents usage in mobile compressors.

2.2. Concept 2: Pneumatic cylinder with mechanical springs

An alternative to the before-mentioned weight loading of the piston is the use of a pretensioned mechanical spring. This enables significantly higher eigenfrequencies due to the strong reduction in moving masses. The downside of this approach is that it leads to increasing pressure with reservoir filling due to the spring stiffness. In analogy to conventional reservoirs, this pressure fluctuation needs to stay in the allowable pressure margin of the application, which limits the usable spring stiffness. As the pressure margin is usually small in comparison to the absolute pressure, for conventional helical springs, enormous tensioning strokes result. These make the spring assembly with its tensioning mechanism very complex and bulky and tend to fatigue issues on the springs.

Disc springs with their varying stiffness over the stroke can be used for size and complexity reduction. The dimensions of disk springs can be tuned to allow for high stiffness at stroke begin, which is used to generate sufficient pretension force, while the force gets mostly stroke independent over a second stroke regime, which is the working stroke of the quasi-isobaric reservoir. Figure 2.2 shows the force of such an optimized disc spring over its stroke on the right-hand side, based on the force equations in [14]. This optimization is achieved by means of sheet thickness t being 0.707 times the height of the truncated cone h, which is formed by the disc in unloaded state. In the diagram, the stroke is scaled to the height h, while the force on its value at constant stiffness.

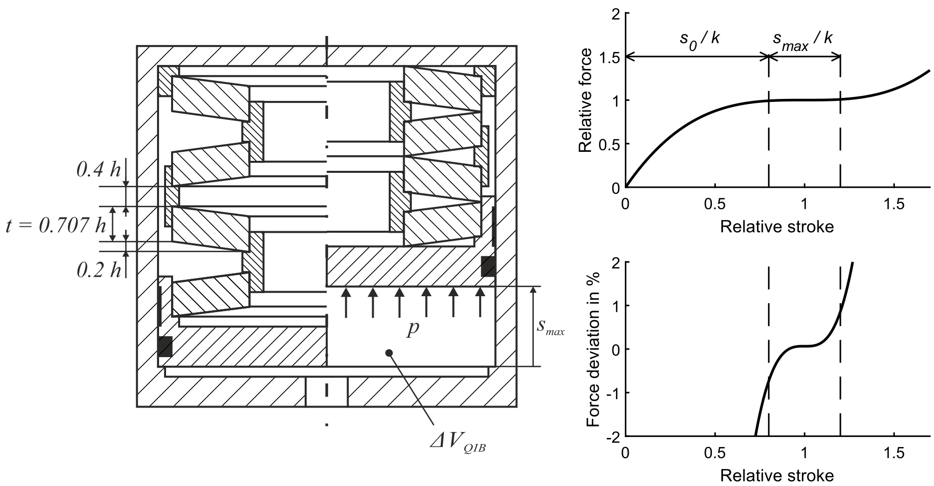

Figure 2.2. Disk spring reservoir concept and force characteristics

For disc spring compressions between 80 % and 120 % of h, the force stays constant with an accuracy of better than ±1 %. Thereof, it can be concluded that if using k disc springs, the pretension stroke s_0 of the spring assembly has twice the size of the piston stroke s_{max}. Meanwhile, very small pressure variations are ensured, which is a huge optimization in comparison to piston loading by constant stiffness helical springs.

The building space of the disk spring loaded reservoir $V_{BS,2}$ can be approximated from the volume change in operation ΔV_{QIB} by scaling it with the pretensioned length of the spring assembly divided by the stroke length s_{max}, as shown in equation 2.4. The numbers result from the schematics in figure 2.2, left and the disk spring design for constant force behavior.

$$V_{BS,2} \approx \frac{(0.4 + 0.2 + 0.707) \cdot h \cdot k}{0.4 \cdot h \cdot k} \cdot \Delta V_{QIB} \approx 3.3 \cdot \Delta V_{QIB} \tag{2.4}$$

However, the design and fabrication of the disc springs need to be very exact, as small tolerances in the design parameters, e.g. in sheet thickness, can lead to decreasing force over the stroke of the disc spring in the operational regime. This will ultimately result in unstable behavior of the arrangement, as pressure drops with increasing filling state. Furthermore, such tolerances can lead to a deviation in operating pressure of the reservoir, as the generated force in the constant force regime of the disc springs deviates from its design value.

A further drawback of the design consists in the inability to set pressure adjustments. By changing pretension, stroke regimes with significant pressure increase over the stroke are part of the operational range and the quasi-isobaric properties are lost.

2.3. Concept 3: Hydro-pneumatic reservoir

The third concept is based on a hydraulically loaded pneumatic piston. A significant reduction in building space is aspired by increasing the internal pressure of the hydraulic system in comparison to the pneumatic side and therewith its energy density. In contrast to the hydraulic bootstrap accumulator, discussed e.g. by *Ketelsen at al.* [15], the hydraulic loading is deemed to be passive. Hence, a hydraulic accumulator, which loads a hydraulic piston to act against the pneumatic cavity is investigated. The precharge level of the accumulator results in sufficient air pressure, while its size and therewith its capacity allows control of the pressure increase over the stroke. Figure 2.3 shows a schematic representation of the investigated concept. Compressed air is marked in blue, hydraulic fluid in orange.

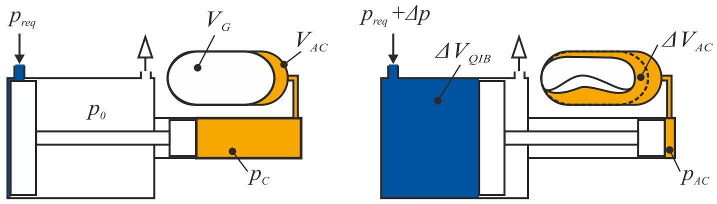

Figure 2.3. Concept of the hydro-pneumatic reservoir; left: discharged, right: charged

As a strongly simplified thermodynamic model of the hydraulic accumulator, a polytropic change of state with polytropic number n is assumed. When the pneumatic side is loaded, the liquid volume ΔV_{AC}, which is pressed by the amplifier piston into the hydraulic accumulator leads to a decrease in its gas volume and thus to a pressure increase. The accumulator pressure p_{AC} is described by equation 2.7 with the precharge pressure of the gas side p_C, and the gas volume of the accumulator V_G, which is present if the pistons stand at their left end stop. This volume is used as a reference for the precharge pressure adjustment.

$$p_{AC} = p_C \left(\frac{V_G}{V_G - \Delta V_{AC}}\right)^n \tag{2.7}$$

The piston amplifier acts as a transformer with area ratio α between the hydraulic and pneumatic system. Herein area ratio α << 1 describes the quotient of the hydraulic to the pneumatic piston surface. The force equilibrium and energy balance lead to equations 2.8 and 2.9, which both describe the energy transmission through the amplifier piston:

$$p_C = \frac{p_{req} - p_0\,(1-\alpha)}{\alpha} \approx \frac{p_{req} - p_0}{\alpha} \tag{2.8}$$

$$\Delta V_{AC} = \Delta V_{QIB} \cdot \alpha \tag{2.9}$$

Thereof, for a given required system pressure, pressure margin, and required air volume of the quasi-isobaric accumulator, the following equation 2.10 defines the accumulator dimension, as the nominal accumulator volume V_{AC} should be slightly bigger than the calculated gas volume in reference position.

$$V_{AC} > V_G \approx \frac{\Delta V_{QIB} \cdot \alpha}{1 - \sqrt[n]{\frac{p_{req} - p_0}{p_{req} + \Delta p - p_0}}} \tag{2.10}$$

The resulting building space $V_{BS,3}$ can be roughly approximated by the sum of the pneumatic and hydraulic displacement of the piston amplifier and the volume of the hydraulic accumulator, as shown in equation 2.11.

$$V_{BS,3} = \Delta V_{QIB} \left(1 + \alpha + \frac{\alpha}{1 - \sqrt[n]{\frac{p_{req} - p_0}{p_{req} + \Delta p - p_0}}} \right) \tag{2.11}$$

For an isothermal accumulator, this simplifies to equation 2.12. It is obvious that for small area ratios of the amplifier, relevant reductions in space occur.

$$V_{BS,3} = \Delta V_{QIB} \left(1 + 2\alpha + \alpha \frac{p_{req} - p_0}{\Delta p} \right) \tag{2.12}$$

A further benefit of hydro-pneumatic systems is the ability to simply adjust the pretension and thus working pressure of the system by the precharge pressure of the accumulator. This allows for pressure adjustments on a running system, e.g. to incrementally reduce operation pressure of a compressed air network for energetic optimization.

2.4. Concept 4: Nonlinear pneumatic actuator with mechanical spring

A further solution for quasi-isobaric behavior can be achieved by using unconventional pneumatic actuators for the realization of the reservoir volume with increasing active area over the stroke. As has been shown in [16] actual designs for pneumatic actuators realize either reducing forces or constant forces over the stroke at constant pressure. But, by implementing novel actuator design schemes, as shown e.g. in [17] for machine suspension, rising force over the stroke arrangements can be achieved. In the case of quasi-isobaric reservoirs, the use of such actuators as reservoir volume has the big advantage that the force characteristics of the reservoir volume can be tailored to the force stroke behavior of the loading spring. Therefore, in the following, the direct coupling of an unconventional actuator with mechanical springs, as shown in figure 2.4 is investigated.

For the quadratic correlation between volume $V(s)$ and stroke s of the reservoir in equation 2.13, the force F on the loading device is given in equation 2.14. The force balance with a loading spring with stiffness C and pretension stroke s_0 results in equation 2.15, which describes the reservoir pressure as a function of the stroke.

$$V(s) = as^2 + bs + V_0 \tag{2.13}$$

$$F(s,p) = \frac{\partial V}{\partial s} \cdot (p(s) - p_0) = (2as + b) \cdot (p(s) - p_0) \tag{2.14}$$

$$p(s) = p_0 + \frac{C}{2a} \cdot \frac{s + s_0}{s + \frac{b}{2a}} \tag{2.15}$$

For a given volume stroke characteristics with parameters a, b, and V_0 and spring stiffness C, a distinct pretension stroke exists, which allows the system pressure to stay constant over the entire volume change of the reservoir. This pressure is achieved with a pretension stroke of $s_0 = b/(2a)$. Moreover, especially for small values of b, and hence small forces at stroke begin, pretension of the springs can be very low and almost the entire working stroke of the springs can be used for energy storage, which makes this arrangement particularly material efficient regarding the mechanical spring.

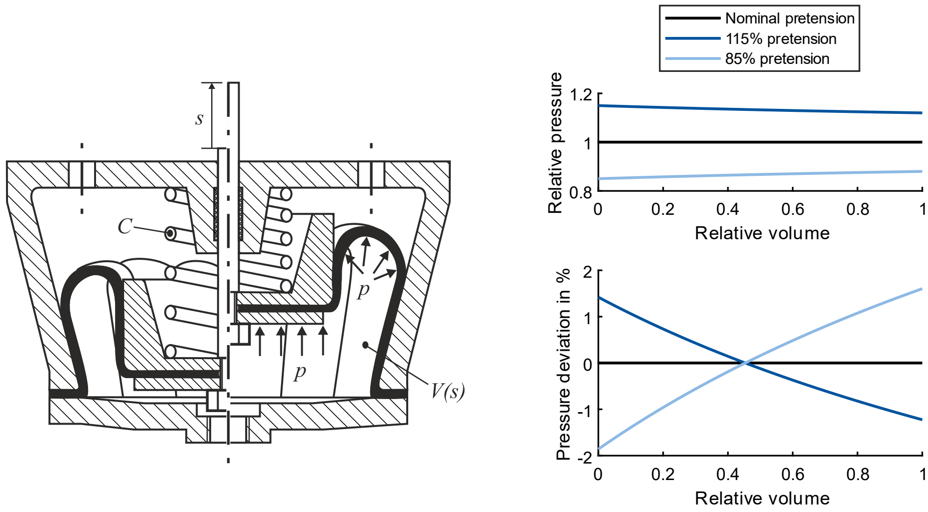

Figure 2.4. Investigated nonlinear pneumatic actuator with mechanical spring and its force deviation over the stroke for $b = 8\ a\ s_{max}$

On the other hand, if pressure adjustments by changing spring pretension are considered, the pretension stroke should be maximized to allow for small deviations in pressure over the stroke and thus stored volume. To achieve a pressure fluctuation over the stroke of less than ± 2 % at a pressure adjustment range of ± 15 %, the pretension stroke of the spring must be equal or higher than four times the piston stroke s_{max}. Hence, significant spring length and pretension stroke are required to allow for adjustability.

The change in pressure over the filling state is shown in figure 2.4 for the nominal pressure and deviating pressure settings by ± 15 %. The loss of the constant pressure over the stroke operation is clearly visible at varying pretension. Moreover, lower pressure settings can cause the system to have negative fluidic impedance, and thus to be unstable if connected to sufficient grid capacity. The impact of the slight negative impedance in the depicted case on system stability is to be further analyzed. Due to its small amount, it can probably be stabilized by sufficient viscose friction of the arrangement.

Finally, also this design has the major advantage in that it is very compact. Its required building space is estimated from figure 2.4 to be approximately four times its displacement volume, as shown in equation 2.16.

$$V_{BS,4} \approx 4 \cdot \Delta V_{QIB} = 4 \cdot (V(s_{max}) - V_0) \tag{2.16}$$

3. CONCEPT EVALUATION

First, the different concepts are compared by means of their general characteristics, as shown in table 3.

The weight loaded concept 1 has its only major benefit in being simple in realization. The required small piston diameter and long stroke and the additional guiding elements to prevent the system from tilting lead to high hysteresis and seal wear and poor dynamics.

The disc spring loaded concept 2 has advantageous dynamics, higher piston diameter, and small pressure increase over the stroke but lacks adjustability while being difficult in maintenance due to the long pretension stroke and high stored energy of the springs. Also, realization has high demands on accuracy of the disc springs to allow all sequential springs of the system to work exactly in the required working regime.

The hydro-pneumatic concept 3 is very advantageous in adjustability and dynamics, while complicating maintenance by requiring additional seals, a pressure medium, a system for flushing the fluid and for filling the gas into the accumulator, and multiple hydraulic components.

Lastly, the spring-loaded nonlinear actuator concept 4 is beneficial in all categories despite its simplicity in implementation, as this new kind of actuator first needs to be developed. Maintenance therefore is difficult to evaluate, as it is strongly design dependent.

	Concept	1	2	3	4
Criterion	Adjustability	+	- -	+ +	+
	Small hysteresis	- -	+	-	+ +
	Dynamics	- -	+ +	+ +	+ +
	Simplicity	+ +	0	+	- -
	Maintenance	0	0	-	0

Table 3: General concept comparison (Rating from poor to good: -- / - / 0 / + / ++)

In the following, the concepts are further investigated by sizing and comparing the capacity of exemplary solutions for two different applications. First, the concepts are evaluated for use in compressed air supply, with huge volume changes at lower speed. Subsequently, decentralized implementation to smoothen peak flow demands and associated pressure drops is evaluated.

3.1. Usage in compressed air supply

Reservoirs in compressed air supply are used to smoothen the grid pressure and therewith to allow the compressor control to react on differences between compressed air generation and consumption. Meanwhile, it is aspired to keep the reservoir pressure as constant as possible slightly above the required system pressure, which should be adjustable in a defined margin. By means of the quasi-isobaric reservoir, inspection duties can be minimized, which means, that in best case, a category II classification ($PS \cdot V < 1000$ bar·l) is targeted.

To compare the different concepts, their realizable displacement volumes as a function of the allowed pressure margin are calculated and depicted in figure 3.1, left. A required pressure of 7 bar_{abs} and a building space with a ground surface of 1 m² and 2 m in height are assumed. The main components and their arrangement were sized for all concepts to fit in the building space and to the allowed pressure margin. From this figure, several conclusions can be drawn:

The weight loaded system consists of one or several parallel arranged cylinders with a stroke of 1 m to fit in the height limitation and a lumped piston surface of ca. 0.1275 m². The weight is assumed to be fabricated from steel and has a mass of ca. 7.8 tons. This makes the system impractical while its displacement is rather limited compared to the other systems. The disc spring loaded system has the highest capacity at small pressure margins but does not allow any correction in system pressure. Therefore, it is also not well suited for the application.

The hydro-pneumatic reservoir achieves the highest capacity at bigger pressure margins. For smaller values, its displacement capability increases with pressure margin. In this parameter region, the restrictive parameters are the limited hydraulic accumulator size, which was set to 300 l, and its assumed maximum pressure of 300 bar. For higher pressure margins, the displacement volume of the compressed air piston limits the achievable effect, as a displacement of 1 m³ cannot be overcome in the building space under the assumption of an air piston with coaxial hydraulic loading piston and circumferential hydraulic accumulators.

By using a spring-loaded nonlinear actuator, the second-best usage of the building space at small pressure margins results. Very likely this concept has the best weight to displacement ratio, as it does not rely on heavy disk springs, a weight, or heavy hydraulic components.

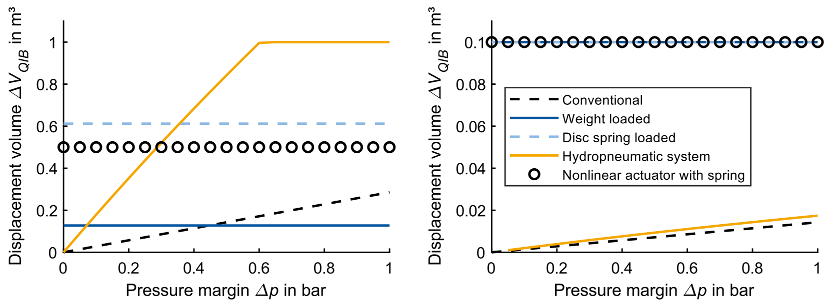

Figure 3.1. Comparison of quasi-isobaric concepts with a conventional reservoir, left: for 2 m³ build space, right: for up to 2 m³ build space and category II classification

If it is required to fulfill category II to allow for longer inspection intervals and to circumvent regular inspections by notified bodies, the rating of the different concepts changes drastically. To calculate the allowable dimensions for category II, the rated pressure of the pneumatic cavity and in case of the hydro-pneumatic system also of the hydraulic accumulator need to be defined. These were set to 10 bar_{rel} for the pneumatic and 300 bar for the hydraulic parts. The results are shown in figure 3.1, right.

To allow for category II classification, the conventional reservoir needs to stay in the PS·V = 1000 bar·l margin and diminishes in size to 100 l. Thus, it can only use a small part of the available building space, and its effectiveness drastically reduces. The cylinder chambers of the weight loaded piston, the disc spring loaded piston, and the nonlinear actuator need also to stay within this margin, which reduces their displacement capacity ΔV_{QIB} to 100 l.

The hydro-pneumatic system in contrast strongly loses in performance, when built to fit into category II. This is because, in all investigated configurations, the hydraulic accumulator needs to be strongly downsized to a nominal volume of only 3.33 l to reach category II. This significantly diminishes its feasible volumetric change of air capacity.

Finally, the missing adjustability in pressure of the disc spring device and the huge mass of the weight loaded system with its associated issues in implementation and dynamics hinder their industrial usage. Therefore, it can be concluded that for practical usage, the nonlinear actuator with spring is the most promising solution despite its required development effort.

3.2. Usage for peak demand

Usually, reservoirs for peak demand supply are smaller sized. Therefore, it is aspired to reduce the pressurized volumes to the point that an article 4, paragraph 3 classification is achieved in all cases. Meanwhile, a maximum ground surface of 0.01 m² and height of 0.5 m are assumed, which enable the conventional reservoir to use the entire building space and categorization limit in parallel. The resulting displacement volumes are shown in figure 3.2.

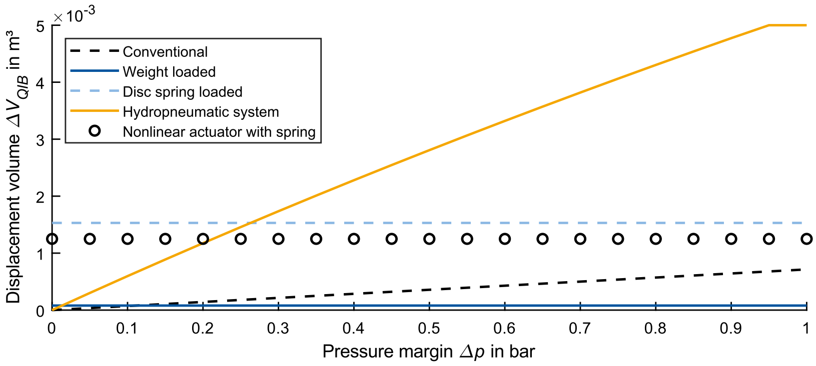

Figure 3.2. Comparison of quasi-isobaric concepts with a conventional reservoir in accordance with article 4, paragraph 3

While all kinds of quasi-isobaric reservoirs can reach a displacement volume of 5 l while falling under article 4, paragraph 3 with sufficient building space, the additional restriction in available space leads to a clear differentiation between the different designs. The hydro-pneumatic system is best suited for higher pressure margins, while the disc spring loaded cylinder and the spring-loaded nonlinear actuator are best suited for small pressure changes. In general, these three kinds of quasi-isobaric reservoirs allow for significant optimization in comparison to regular designs, while the disc spring loaded device lacks adjustability.

4. Conclusion and Outlook

Quasi-isobaric reservoirs have the potential to decrease the building space requirements in comparison to state-of-the-art designs in combination with their demands for regular inspections. Furthermore, the increased capacity of quasi-isobaric reservoirs at low pressure margins can be used to reduce pressure control bands of actual compressor stations to gain several percent in efficiency with corresponding cost savings in comparison to actual solutions. Different concepts for such reservoirs have been developed and compared by means of two different applications. Thereof, it can be concluded that the spring-loaded nonlinear actuator has the biggest potential of all designs investigated, especially for small pressure margins. In contrast, hydro-pneumatic systems have the biggest potential for higher pressure amplitudes at small scales. In future research, these findings should be experimentally validated. While the hydro-pneumatic system can be built from off-the-shelf components, the nonlinear actuator itself is part of actual research. Furthermore, system and compressor control integration of the novel reservoirs need to be investigated.

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