Highly Integrated Rotational Drives for Servopneumatic Applications

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Abstract

This paper summarises the development of pneumatic rotational drives under aspects of miniaturisation and sensor integration. At this, an ample miniaturisation can only be achieved by significantly minimising leakage and friction while these often are contradictory optimisation goals, requiring a methodical development of novel actuator principles. Furthermore the integration of customised sensor and gear modules is necessary to meet the miniaturisation requirements. After a simulative optimisation of several novel drive concepts, the best suited concept and a test rig for its characterisation are built up. The paper closes with a discussion of the obtained measurement results and an outlook on ongoing research activities on the described subject.

KEYWORDS: Miniaturisation, Integration, Bend test, Optimisation

1. Introduction

In the scope of several projects sponsored by the German Research Foundation (DFG), a servo-pneumatic robotic hand was developed at the Institute for Fluid Power Drives and Controls (IFAS) of RWTH Aachen University. However, the implemented pneumatic drives, using common principles for the realisation of swivel movements, inhibit a further optimisation and miniaturisation of the system. This is due to friction forces and leakage at the seals as well as the displacement which is needed to convert the pressure energy stored in the fluid to the required output power.

To overcome this, novel concepts for swivel drives are investigated at IFAS in the scope of a subsequent project named „Highly integrated drives for a servo-pneumatic robotic hand” sponsored by the German Research Foundation (DFG) as well. The objective is to develop drives with integrated sensors, which possess a maximised power density to afford the required miniaturisation for the use in a robotic hand.

A higher power density can only be achieved by using rotational drives and gears for swivel applications. In this way, the displacement of the drive can be used several times while the output shaft is pivoting from one end position to the other. Hence, the required working space decreases while keeping the output torque on a constant level. On the other hand, the needed reversing usually leads to additional constructed space, which has to be considered and compensated by a smaller displacement and thus an ample gear transmission ratio.

An analysis of the underlying function structure in rotational fluid power drives and a synthesis of physical effects realising these functions regarding aspects of miniaturisation and friction minimisation has been conducted in a first project phase /1, 2/. Herein, the gathered information on the function structure and physical effects is not only able to represent common rotational fluid power drives, but also to show novel drive concepts. Solutions eliminating sliding seals by use of revolving sealing systems or membranes are from special interest, as these lead to a reduced friction torque and leakage rate of the drive and thus improve its dynamics.

After a short discussion of some developed concepts, the best promising solution is simulated and optimised to get further information on power density and dynamic behaviour. To parameterise simulation, materials of flexible structures are characterised by bend tests. An analysis of the simulation results focusing on the use in servo-pneumatic applications and the miniaturisation potential, defines the best suited design parameters. Finally, first measurement results of the drive and its components are discussed.

2. Investigated Concepts

The development of novel drive concepts has been carried out by use of the methodology described in Pahl and Beitz /3/ and VDI 2221 /4/. The functional analysis of existing drive principles leads to a uniform function structure consisting of different communicating functional entities. These are e.g. power takeoff, reversing, gear, angular sensor and synchronisation of reversing and power takeoff. Herein, this structure is not only able to represent the investigated drives but also to develop new drive principles by using alternative physical effects to realise the single functional entities. Therefore a morphological box was developed to systematically investigate all possible effects and its combinations for each functional entity. The box was filled by use of effect catalogues /5/ and analogy observations.

The use of similar physical effects for different functions, as e.g. power takeoff and reversing, enables the design of highly integrated systems with a minimised number of parts and thus an ample miniaturisation potential. Also, the use of common parts for different effects can improve miniaturisation.

After first evaluation of the huge amount of resulting concepts concerning friction and leakage, the three concepts described below seem to be well suited for the application.

The membrane drive (see **figure 1**) uses membranes on a conical support to generate radial forces on a wobbling assembly which turns an eccentric commutation valve. Thus, it synchronises itself as motion and pressurisation of the wobbling assembly and its membranes are coupled by the commutation valve's movement. Inside of this rotating commutation valve an electromagnetically actuated valve spool is allocated which is able to change the rotational direction and thus to realise a torque or position control.

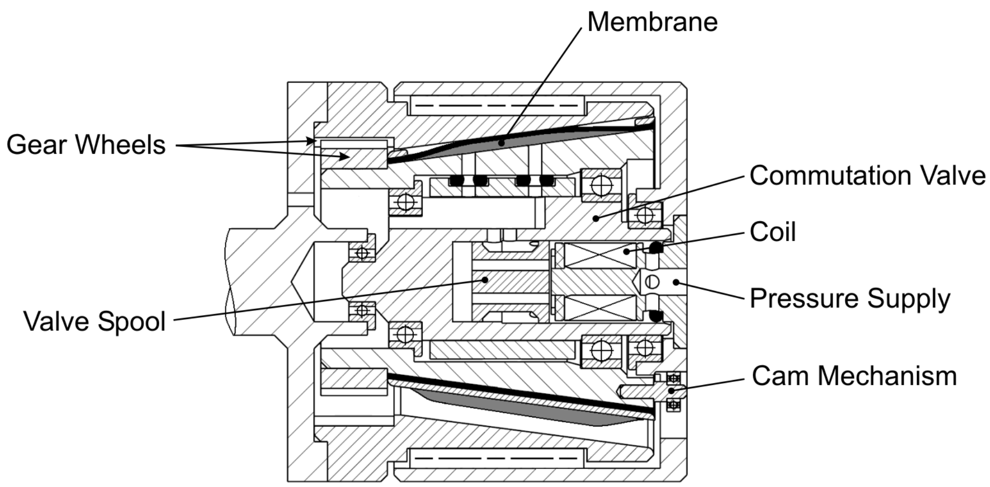

Figure 1: Membrane Drive

The motion of the wobbling assembly is captured by an eccentric drive, consisting of a gear wheel fixed on the wobbling assembly and an internal geared wheel on the output shaft. Due to the small eccentricity of the commutation valve, the two geared wheels have only slightly different numbers of teeth leading to an ample transmission ratio. To avoid a rotation of the wobbling assembly and thus an undesired motion of the output shaft, the wobbling assembly is guided by a cam mechanism consisting of three small roller bearings rolling up in corresponding bores in the housing.

The Harmonic Hose Drive shown in **figure 2** consists of a Harmonic Drive which is driven by a hose surrounding it. Contrary to conventional harmonic drives, the stator in form of an outer internal geared wheel is designed as flexible cup, so that the hose can act on the elliptic rotor. Both the stator and the flex spline are three dimensionally deformed. Due to the rotor's 180° symmetry, the commutation is achieved by the rotor itself and thus no additional parts are required. Valve integration is not possible as the hoses and thus the air inputs are allocated on the outer side of the drive.

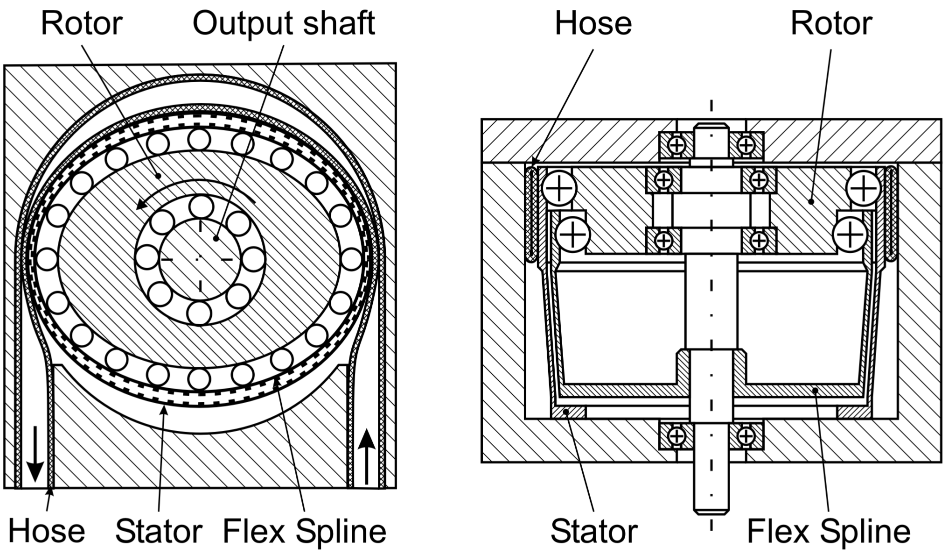

Figure 2: Harmonic Hose Drive

The third concept shown in **figure 3** also consists of a Hose drive but is not using the harmonic drive principle. Instead, a planetary drive is integrated in the elliptic rotor. Thus, less flexible structures are required as the flexible internal geared wheel is omitted compared to the harmonic hose drive. Also the three-dimensional flexion of the flex spline is reduced to planar deformation of the rotor sleeve.

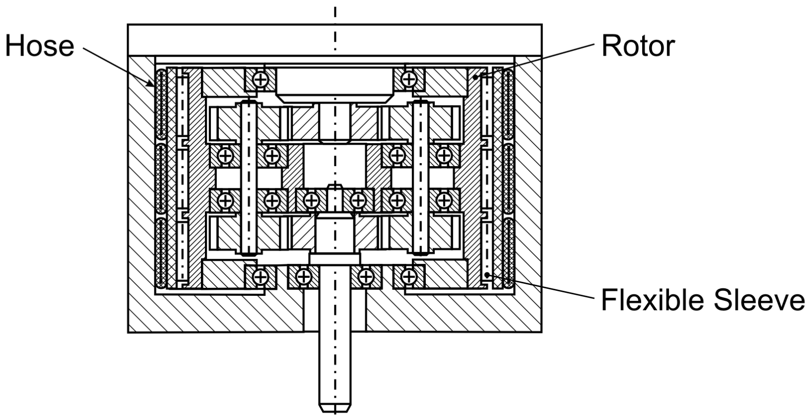

Figure 3: Planar Hose Drive

3. Concept Evaluation

The Membrane Drive obviously possesses the highest integration potential due to its internal control valve. However, it only allows small wobbling strokes and provides due to the small eccentricity small driving torques at high radial loads. Thus, friction between commutation valve and wobbling assembly is significantly reducing the output torque or even inhibits motion, while this gap has to be minimised to reduce leakage. The use of a pressure balanced commutation valve reduces radial forces in the gap, but nevertheless friction takes big influence on the behaviour of the drive.

In contrast to this, the Harmonic Hose Drive imposes a big challenge on its material due to its large required degree of deformation. The cyclic radial deformation has to be big enough to flatten the hose and thus create two independent working chambers while allowing a significant inflation volume of the chambers. In addition, a compact design in axial direction leads to an increased deformation, as the changeover from the elliptic cross section on the rotor to the round shaped cross section at the output shaft junction (flex spline) or the housing (stator) has to be realised on a small distance.

This problem is avoided by the Planar Hose Drive which does not need a connection between rotor sleeve and housing or output shaft. Hence, the axial length of the drive can be reduced to afford miniaturisation. Beyond that, contrarily to the Harmonic Hose Drive, the planetary drive's gear ratio in the Planar Hose Drive is not influenced by other geometric parameters and thus can be optimised for a specific application.

Due to these advantages, the Planar Hose Drive is further developed by simulation in the next chapter.

4. Simulation and Optimisation

Simulating the Planar Hose Drive requires a model which involves the hose deformation and the resulting pressure areas with their effective directions. Additionally, radial forces due to the rotor sleeve deformation have to be considered to analyse friction losses in the roller bearing separating rotor and rotor sleeve. In the following the model and parameter identification as well as the optimisation of the drive design are shown. All simulations are realised in MATLAB.

4.1. Modelling

The modelling of the hose requires a mathematical description of the active pressure area's width h as a function of the gap size b between housing and flexible sleeve. Neglecting the hose's wall stiffness, this can be achieved by considering the form shown in **figure 4** on the left, consisting of two semi circles and two line segments, while keeping the perimeter equal to the original circumference of the hose cross-section. On the right, the calculated three-dimensional form of the hose is represented.

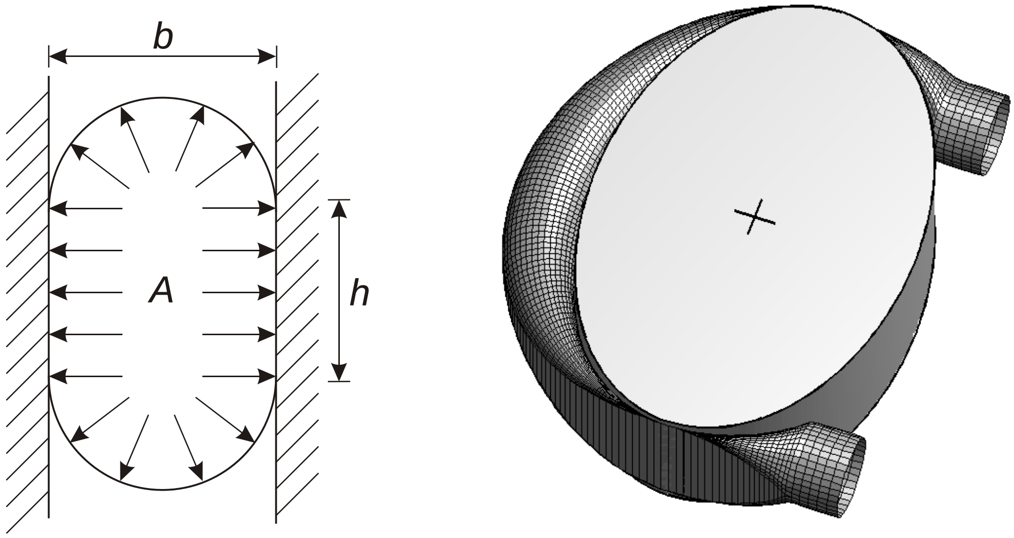

Figure 4: Hose modelling

In this manner, first simulation results are achieved showing the need of multiple parallel hoses to overcome torque pulsation. Herein, a combination of four parallel hoses seems to be a good compromise for the application regarding torque pulsation, complexity and miniaturisation.

The flexible sleeve’s stiffness is considered by a modified beam theory, which was validated by FEM simulations. Herein, the sleeve is discretised in tangential direction to calculate the local bending moments due to its deformation. In a second step, the required external loads to achieve this bending moments are determined. Contrary to a standard beam the structure consists of a closed curve and thus does not possess endpoints where the boundary conditions are well known. Therefore, it has to be opened and the boundary conditions of both sides have to be iteratively matched.

Figure 5 shows an exemplary simulation result. The inner arrows are representing the mechanical loads on the roller bearing due to the deformation of the originally round shaped elastic sleeve, while the outer arrows are visualising the pressure loads due to unequal pressurisation of the working ports ($p_A > p_B$). The dashed lines show the neutral axis of the sleeve in round and elliptically deformed state.

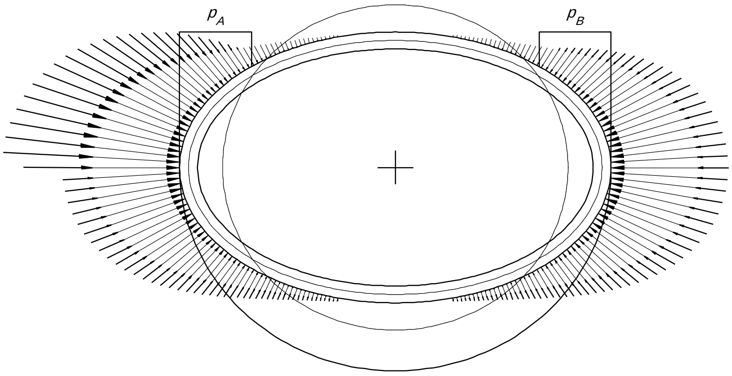

Figure 5: Simulation result

4.2. Parameter Identification

The identification of material parameters for the flexible sleeve is carried out by simplified three-point bend tests on the basis of ISO 178 /6/. As the sleeve is round shaped it can be manufactured by turning or as the thickness has to be minimized by laminating foils. While a metallic sleeve must be very thin to avoid high radial forces on the roller bearing and to allow a sufficient endurance, the use of polymers allows higher wall thicknesses. On the other hand viscoelastic deformation of plastics leads to energy absorption and thus reduces the output power of the drive. This also applies to adhesives used for lamination. Hence hysteresis has to be evaluated in bend tests too. The test rig is shown in **figure 6**.

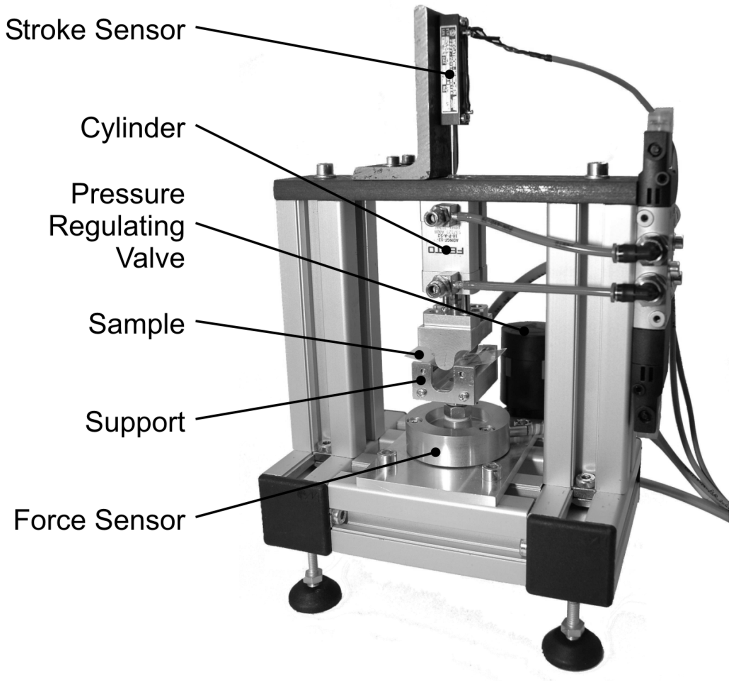

Figure 6: Bend test rig

A support holds the sample on both sides of its centre point, where the load is applied by a vertically arranged exhaust air side throttled pneumatic cylinder through a half cylindrical thrust piece. The sample is analysed by measuring the cylinder's stroke and the resulting force on the support. To adjust the load, the maximum exerted force can be set by a pressure regulating valve.

Some characteristic results can be found in **figure 7**. As a reference a 0.103 mm thick steel strip is represented. The small hysteresis at higher deflexions is caused by friction between support and test strip. As a negative example possessing a huge hysteresis, a fibre reinforced polyamide laminate, consisting of two 0.250 mm thick glass fibre reinforced polyamide foils which are laminated by an adhesive film is shown. Its poor performance is due to the high tensile strength of the fibre reinforced foils compared to the used adhesive, leading to a relative movement in the laminate's partially viscoelastic adhesive layer. The third measurement result corresponds to a polyester laminate build from two 0.125 mm thick polyester foils which are laminated by a hot melt adhesive. It absorbs less energy as the hot melt adhesive of the polyester laminate possesses a higher stiffness and lower dampening. From all investigated probes, the polyester laminate is best suited for building up a first prototype.

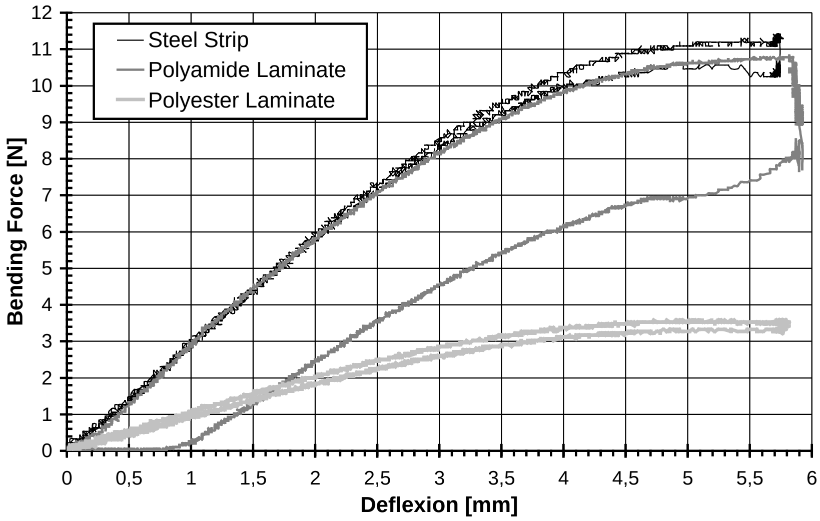

Figure 7: Characteristic results from bend tests

For the parameterisation of the simulation, a bend modulus of the sleeve is required. It can be calculated from the measurement results by the following equation using measurement values for ΔF and Δx at small deflexions. The equation accords to ISO 178 /6/ and was deduced from the bending line.

$$E = \frac{d^3}{4\,w\,t^3} \cdot \frac{\Delta F}{\Delta x} \tag{1}$$

The derived parameters for the simulation are an average bend modulus of 5,000 N/mm² and a thickness of 0.25 mm for the polyester sleeve.

4.3. Simulation Results

Using above mentioned models and parameters with an inner hose diameter of 4 mm and a friction coefficient of 0.04 for the roller bearing the behaviour depicted in **figure 8** is achieved. Surface 1 shows the result for an upstream pressure of 6 bar $_{rel}$ and a downstream pressure of 0 bar $_{rel}$. Surfaces 2, 3 and 4 show respectively 5 bar against 1 bar, 4 bar against 2 bar and 3 bar on both sides. While at small semi-minor axis the volumetric pulsation of the drive becomes obvious, friction losses are dominant at higher length. The discontinuous behaviour of the friction torque is due to the change of the effective areas of the up- and downstream pressures. Therefore it disappears in case four, as the pressures are equal and the sum of both areas stays constant.

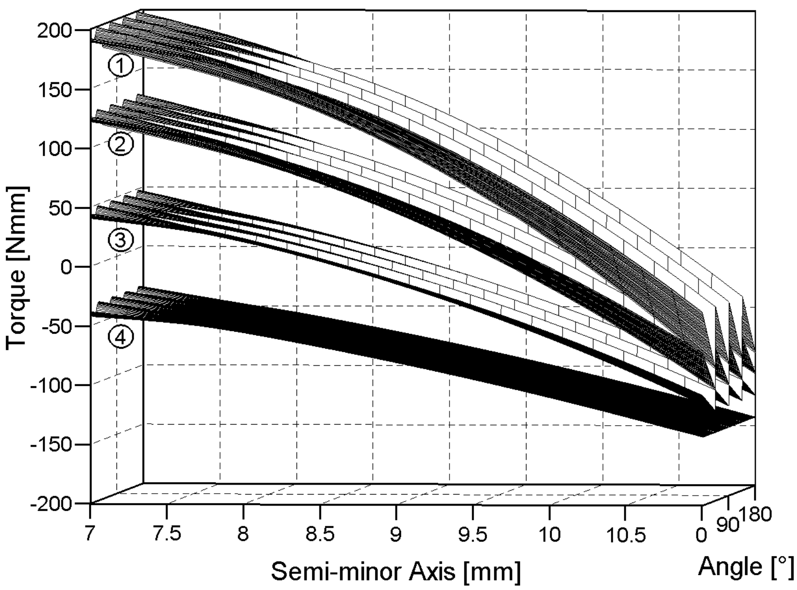

Figure 8: Simulation Results at different pressures

To prevent the rotor sleeve from damage provoked by excessive bending, a small semi minor axis length of 8 mm is realised in the prototype.

5. Prototype measurement

After building up a first prototype according to the above specified geometry, it was learned that the simulated torque could not be achieved. Moreover, already the drag torque of the unpressurised drive is comparatively high regarding the maximum output torque. Possible causes are friction losses on the bearings and gears as well as the squeezing of the hose. As the hose's wall stiffness and three-dimensional deformation was only roughly described in the simulation model, its influence on the exerted force is investigated by modifying the bend test rig. This is achieved by changing the support in a flat measuring plane so that the thrust piece squeezes the hose while measuring the required squeezing force at different pressures. Measurement results (cf. **figure 10**) show, that the used hose consisting of fibre glass reinforced silicon underlies a major diameter expansion while shortening in axial direction when it is pressurised. This has two different impacts on the drive's function:

- The simulated torque characteristic is not achieved due to changing pressure areas.
- On the squeezing major-axis of the elliptic rotor, the axial length of the drive is shorter than the space required by the flattened hoses. Thus the required axial compression of the hoses counteracts motion.

To validate the simulation model of the hose, measurement data can be compared to simulation results. As shown in **figure 9**, the diameter expansion of the hose is regarded in a modified simulation model by introducing a linear elastic model in the cutting plane. Its tensile strength was adapted to fit the base points of the measured data.

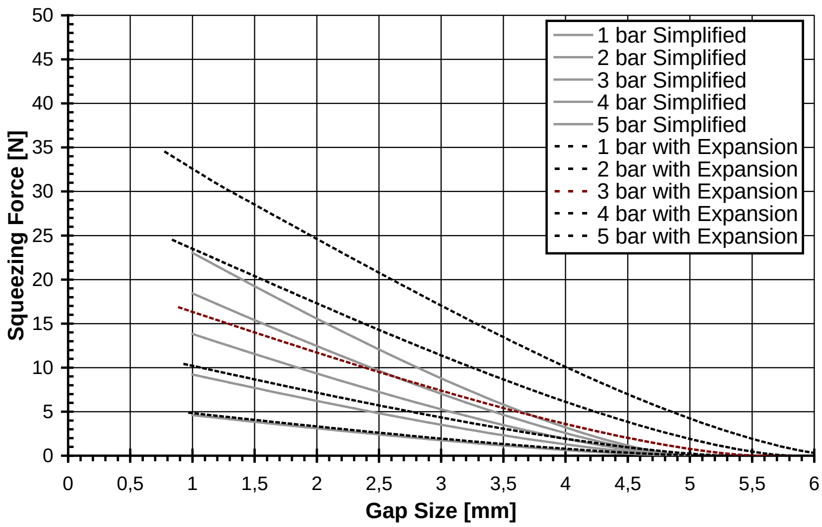

Figure 9: Comparison of simplified and expansion considering model

With this enhanced model, a good accordance between measurements and simulation results can be achieved. The comparison of simulation and measurements is represented in figure 10. The most significant difference between measurement and simulation occurs in unpressurised state, as the squeezing losses are not considered in the simulation model.

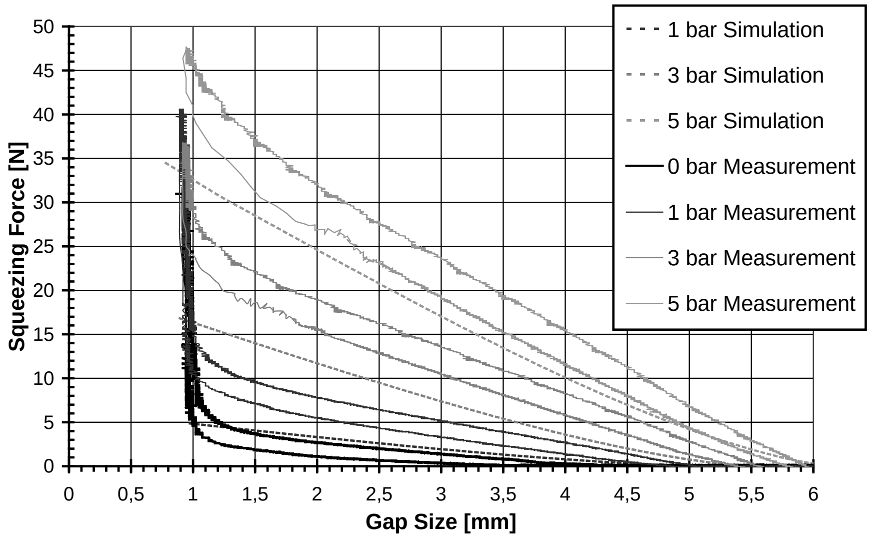

Figure 10: Comparison of measurements and expansion considering model

6. Conclusion and Outlook

In the scope of this paper, a methodological development of novel drive principles, a corresponding simulation model for the best suited alternative, its validation, as well as a first prototype evaluation are shown. While the simulation model of the drive predicts a well suited torque characteristic, it can actually not be achieved in measurements due to a suboptimal hose behaviour, which was analysed by measurements.

For the further development of the drive, better suited alternatives for the used hose have to be found to optimise the output torque of the prototype. Herein, especially the diameter expansion of the hose has to be regarded. Furthermore, the investigation of the hose's behaviour on a test rig has revealed optimisation potential in the simulation model of the drive by introducing a linear elastic model for its diameter expansion. In addition, while optimising the hose model, the integration of wall stiffness can be considered to minimise simulation errors.

7. Acknowledgments

The work described in this article is funded by the German Research Foundation. The authors would like to thank the German Research Foundation for its support.

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9. Nomenclature

b	Gap size	mm
d	Distance between supports	mm
E	Bend modulus	N/mm²
F	Bend force	N
h	Width of the active area	Mm
t	Thickness of the test object	Mm
w	Width of the test object	Mm
x	Deflexion	Mm