Component-based Modular System Modelling of a Spool Valve

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Translational seals are an essential part of spool valves. In order to describe the functional behaviour of such valves block-oriented system models are frequently used. First steps towards a component-based modular system modelling of a spool valve in Modelica are shown in this presentation. The different seals are modelled as components having individual contact states. These discontinuities contribute to the frictional behaviour of the system model. The friction behaviour is implemented by easily replaceable friction models. Advantages of component-oriented vs block-oriented modelling are demonstrated.

1 Introduction

Today's product development process is characterised by short development times and low costs. The traditional performance evaluation of a product, with time consuming test runs with several variants of samples, must be restructured in a more efficient way by only testing pre-selected promising designs. This requires fast and flexible development and validation methods. System modelling enables the simulation of a system behaviour at a rough design status or allows optimisation for a fully developed design by simulating and comparing a large number of variants in a comparatively short time. One way to develop a system model is the investigation of similar systems and the development of phenomenological relations. Another approach is the detailed analysis and knowledge of relevant functional components and the description of these, either by phenomenological relations or by physical models.

For spool valves, translational seals are essential parts. In these valves, several seals are in contact at the same time and the friction between seal and housing is very important for the behavior of the valve. Furthermore the contact pressure of the seal, the speed of movement and non-operation phases have a clear influence on the friction as well.

The current work focuses on a 5/2 spool valve. The test setup and experimental results are discussed in section 2, whereas different approaches to system modelling are overviewed in section 3. The built of the system model is explained in detail in section 4 and the simulation results are demonstrated in section 5.

2 Valve Test Stand Setup and Discussion of Experimental Results

The product under consideration in this work is a 5/2-way single solenoid In-line valve with pneumatic spring and internal pilot air supply. The circuit symbol is shown in Figure 1 as well as a cut in the 3D CAD model. For the experimental investigation, pressure is supplied in 1, silencers are mounted in 3 and 5, pressure sensors are attached to the working ports 2 and 4 and additionally to the chambers of the drive piston 14 and the air spring 12. The position of the piston is surveyed with laser measurement equipment. The switching signal voltage of the solenoid valve is controlled by the measuring software.

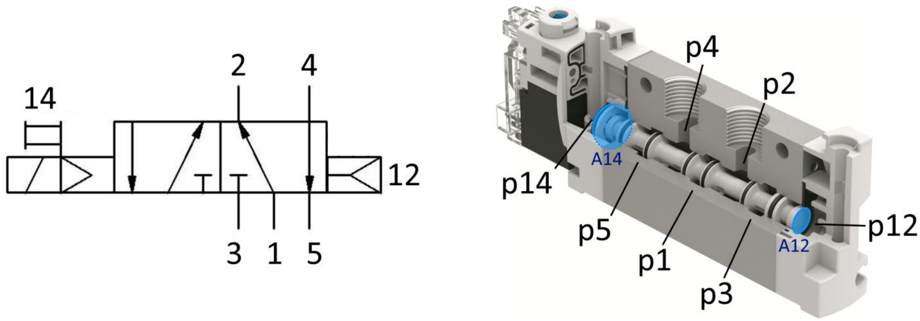

Figure 1: 5/2-way, single solenoid, pneumatic spring, In-line valve, pilot air supply internal

Figure 2 shows the recorded signals for a normal switch-on phase at a common pressure supply level. The signals are stroke s, pressures p_{12} in the chamber of the air spring, p_{14} in the chamber of the drive piston and the pressures p_2 and p_4 in the work ports. The diagrams start at time 0 s at which the switch impulse is sent to the solenoid valve that enables the pressure supply of p_{14}. At a sufficient pressure level, the piston module starts moving and the external pressures p_2 and p_4 change.

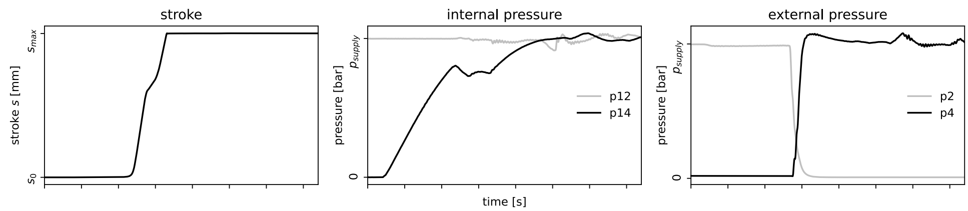

Figure 2: Data from measurement

During post-processing, other values are derived from the primary signals to interpret the performance, see Figure 3, such as the velocity $v = \Delta s/\Delta t$ and the external force on the piston module by $F_{ext} = p_{14} \cdot A_{14} - p_{12} \cdot A_{12}$. The values A_{12} and A_{14} refer to the surfaces of the air spring and the drive piston, see Figure 1. This external force allows an estimate of the friction of the seals in the system.

To obtain a broad overview of the performance of the test specimen, a series of tests was carried out with varying pressure supply from two to nine bar relative to ambient pressure and a second series with varying timespan of non-operation from 1 s to 48 hours before the switch-on test.

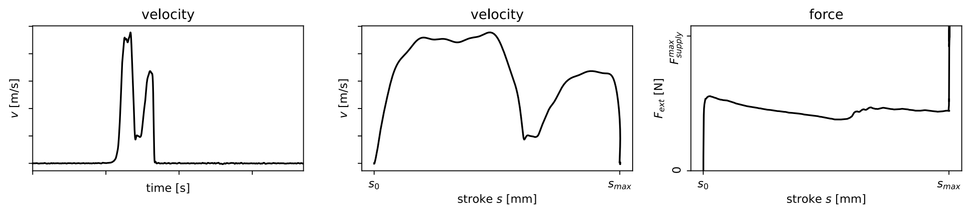

Figure 3: Derived quantities

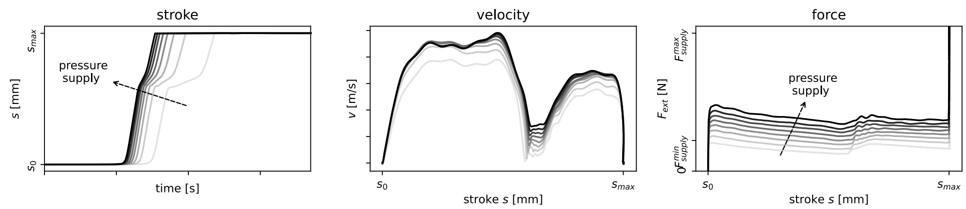

Figure 4: signals and derived quantities for varying pressure supply

Figure 4 shows the effect of varying pressure supply. At low pressure supply, the time until completion of the stroke is higher than at high pressure supply. With increasing pressure supply, the forces that are present to move the piston module increase. This leads to the conclusion that the friction of the seals increases with increasing supply pressure.

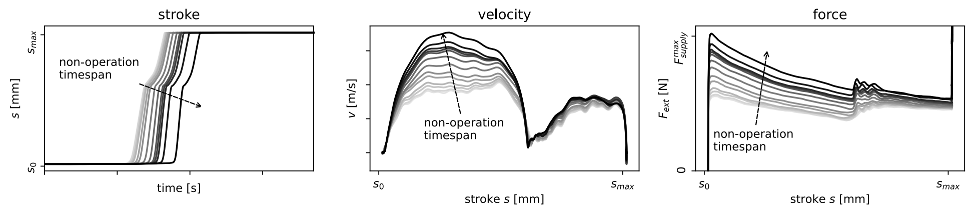

Figure 5: signals and derived quantities after timespan of non-operation from 1s to 48h

Figure 5 shows the effect of varying timespan of non-operation. With increasing timespan, the time needed for the completion of stroke increases and higher forces acting on the piston module are determined. Hence the behaviour for long timespans, i.e. ‘Monday Morning effect’, can be identified. Notice, that non-operation timespans from 1, 5 or 30 seconds already influence the performance of the test specimen. This leads to the conclusion that the friction of the seals increases with the non-operation timespan.

3 Approaches to System Modelling

Without claim of completeness, some differences between Simulink and Modelica are discussed. Instead of block diagrams with directed signals like in Simulink, Modelica allows the formulation of balance equations and other model equations. This results in a structural diagram that mimics the topology of the physical system, e.g. Figure 8.

The Modelica component models are described by algebraic equations, time-continuous differential equations and/or discrete-time equations. In a Simulink block with input and output, all equations must be displayed by block symbols. Taking the Stribeck equation for friction as an example, the Modelica implementation is given by equation (1) whereas the Simulink block diagram is shown in Figure 6.

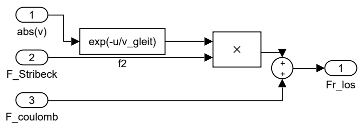

Figure 6: Block-diagram of Stribeck equation

For the Modelica system model presented in section 4, only few basic components are used with many instances which are connected to create a complex system model. The used components are a mechanic component for the moving mass, a friction component, as well as pneumatic components like a vessel, pressure source or a mass flow resistance. The pneumatic components can be described by parameters like the volume, the critical pressure ratio b or the conductance C.

These components are connected by special connectors having flow variables and potential variables. Flow variables like force or mass flow are added to zero. Potential variables like position or pressure are set equal when coupled. The coupled system of equations is solved by the Modelica solver.

4 Built of System Model

This section shows the evolution of the valve's system model from the simplest setup to a more complex model including more and more features. For the simplest case, a Simulink model is present for a comparison of modelling approaches.

4.1 Basic Model for Spool Valve

The simplest case is constructed by a moving mass with friction that has a left and a right stop and is actuated by two air springs. Figure 7 shows the block diagram of the Simulink model, which may already look confusing. The topology of the physical system cannot be identified.

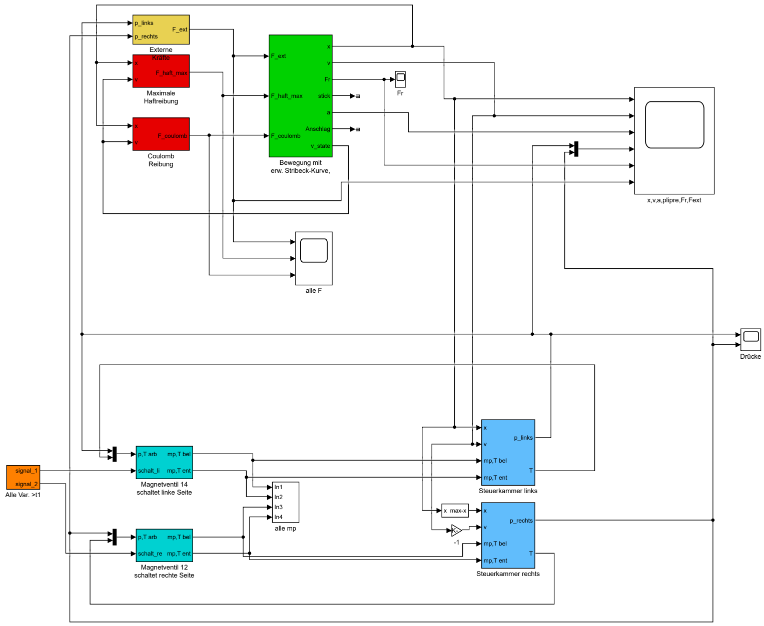

Figure 7: Basic model of a spool valve in Simulink

In Modelica, the simplest model makes use of the component 'MassWith-StopAndFriction' of the Modelica library. This component contains the method for the switching structure of the friction element proposed in [1]. It is combined with two pneumatic air springs, flow resistances and pressure sources, see Figure 8a).

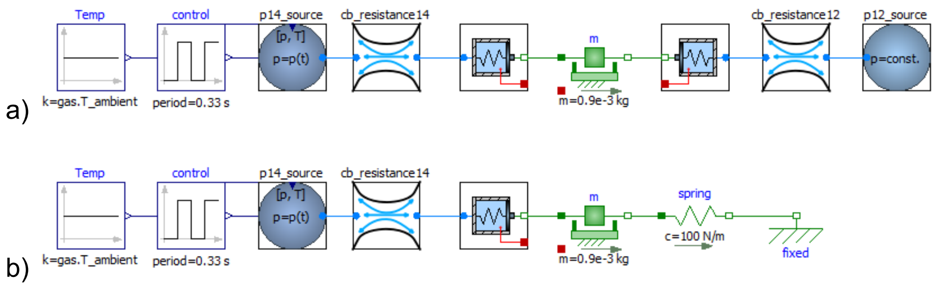

Figure 8: a) simple model with pneumatic components or b) with mechanic spring.

Parameters for flow resistance are related to the geometry of the different parts. The transition from pneumatic to mechanic domain within the air spring includes the front surface of the valve's piston module. As Figure 8 b) shows, the model can be easily changed from an air spring to a mechanical spring valve by replacing pneumatic components with mechanic ones from the standard library.

The equation describing the friction, proposed in Modelica's 'MassWithStopAndFriction' component, is the well-known Stribeck equation (1)

$$f = F_{\mathrm{Coulomb}} + F_{\mathrm{prop}} \cdot v + F_{\mathrm{Stribeck}} \cdot \exp(-fexp \cdot v) \tag{1}$$

with the Coulomb friction parameter F_{Coulomb}, the velocity v dependent friction F_{prop}, the difference between static and coulomb friction force F_{Stribeck} and the exponential decay $fexp$.

4.2 Individual Seal Component for Friction and Contact Discontinuity

To obtain the valve functionality as indicated in Figure 1, the air flow through the five pneumatic ports is controlled by six sealing rings mounted on the piston spool in combination with six sealing regions in the housing. To account for effects from each individual seal, a new 'single_seal' Modelica component is introduced with parameters for geometry information.

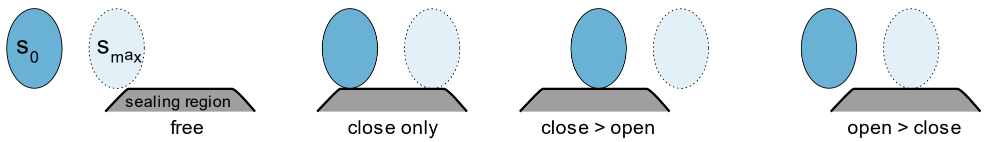

Figure 9: Possible contact states considered

A single seal can have different relations to the housing, which define the contact. From the initial position at the piston stroke s_0 to the final position at stroke s_{max}, the seal can change its contact state. The possible contact states considered in this work are: 'free', 'close_only', 'close>open' and 'open>close', shown in Figure 9. If the contact state changes from close to open, the resultant friction changes from friction f to 0. The piston stroke s_{ce}, where the seal meets the associated control edge of the housing is computed from geometry parameters.

Not only the piston stroke s_{ce} is important for the axial forces in the system, but also the geometric design of the control edge, which typically is conically shaped before the cylindrical sealing region, see Figure 10 a) for a detail view on the cross section.

Focusing on the contact between seal and control edge, e.g. with a FEM simulation, one recognises contact between seal and the conic surface of the control edge. The resulting contact normal force can be decomposed into an axial and radial component, see Figure 10 b). The axial component, 'push in force' f_{ce}, however, must be included in the equilibrium equation of forces. For a first, numerically continuous and simple implementation, the push in force is represented by a gaussian bell shaped function with the two parameters: peak value F_{ce} and a width w, see Figure 10 c).

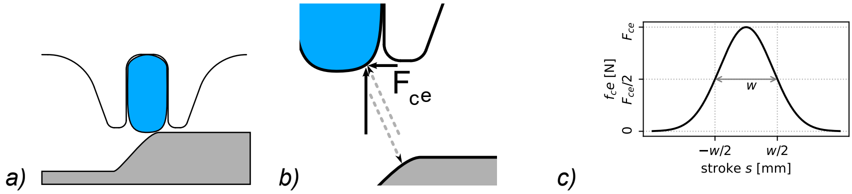

Figure 10: a) sketch of a sealing ring mounted on the piston pushed to the control edge b) contact normal force split into radial and axial (F_{ce}) component c) gaussian bell shaped function with width and height parameter

To use the 'single_seal' component in the system model, a modified component 'MassWithStop' is developed. The computation of the friction is removed but an input of forces from single seal is introduced. Furthermore, the information from the switching state are passed to the single seals using signal bus, see Figure 11a). Within the 'MassWithStop' component, the sum of friction forces from individual seals f^{sum} and the sum of push in forces f_{ce}^{sum} at the control edge is used to compute the equilibrium of forces (2)

$$0 = \text{flange\_a}.f + \text{flange\_b}.f - f^{sum} - f_{ce}^{sum} - m \cdot \dot{v} \tag{2}$$

with the mass m and the cut forces flange_a.f and flange_b.f.

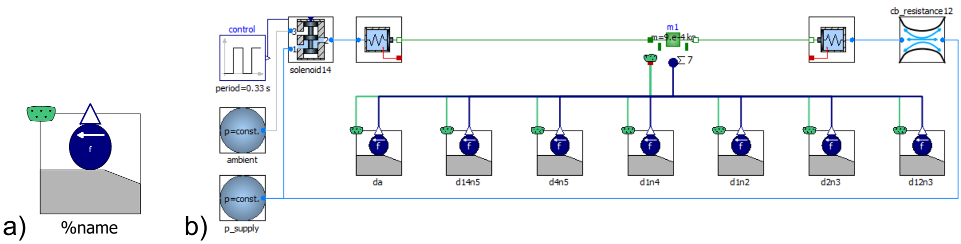

Figure 11: a) single seal with signal bus for switch information and real output b) system model for 5/2 valve with seven instances of single_seal component

Using these components, not only 5/2-way valves, see Figure 11b), but other types can easily be created.

4.3 Timespan of Non-Operation Effects on Friction Forces

As it is discussed in section 2, the timespan of non-operation influences the time until the stroke is completed and the forces acting on the piston module, see Figure 5. To enable the model to reproduce this behaviour, a parameter for the initial non-operation timespan t_{no}^{init} is introduced. During simulation, a timer is used that computes the timespan between entering and leaving the state 'locked' as continuous variable non-operation timespan t_{no}. This allows the model to capture even short-time effects e.g. from operating frequency. With this variable, it is possible to modify the friction equation by changing the Stribeck force from a parameter F_{Stribeck} to a function. In this work, $F_{\text{Stribeck}}(t_{no})$ is described by an exponential equation with upper limit $F_{\text{Stribeck}}^{\text{max}}$ and two further parameters that control the shape of the curve.

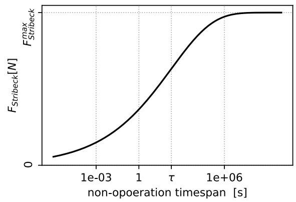

Figure 12: Friction force with increasing $F_{Stribeck}$ with non-operation time t_{no} .

A typical curve with logarithmic scale on the x-axis is shown in Figure 12, demonstrating an S-shaped growth of F_{Stribeck} with t_{no}.

4.4 Functional Pneumatic Unit of Main Valve

As mentioned in section 4.2, the seals control the connection of the pneumatic ports of the valve. To model this, the piston stroke s_{ce}, at which the seal meets the respective control edge is used to allow or block air flow. For a first, numerically continuous and simple implementation, a $tanh(s)$ function is incorporated for a transition of flow rate from 1 – 0 $[\ell/\text{bar}/\text{s}]$ at $s = s_{ce}$.This variable flow rate is used to control the flow between the pneumatic ports of the valve. A new Modelica component is created that combines the functionalities of the single friction component presented in section 4.3 and a flow resistance between two pneumatic ports, see Figure 13.

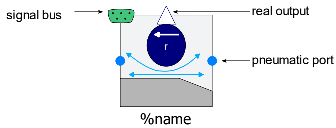

Figure 13: Friction and flow resistance component

With this new component at hand, it is possible to construct the working part of the 5/2 valve by introducing small internal volumes, that are again connected via flow resistances to working volumes or flow sensors at the relief, see Figure 14. Notice, that only few components, but many instances are used to create this complex model and it still mimics the topology of the physical system, c.f. Figure 1.

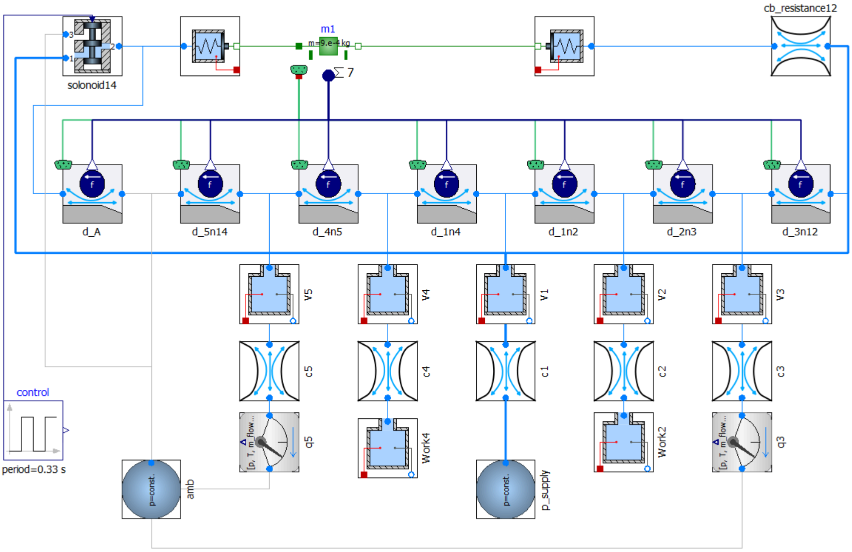

Figure 14: 5/2 way valve with five flow resistances at the pneumatic ports, one pressure supply, two 'Work' volumes and two flow sensors at the relief.

4.5 Pressure Effects on Friction Forces

Pressure supply ranging from two to nine bar relative to ambient pressure has been tested and increasing forces were observed, see section 2 or Figure 4. To account for the pressure dependent friction of the single seal, the component presented in section 4.4 is enhanced. The absolute pressure difference Δp between the two pneumatic ports is computed and used for a linear modification of the parameters $F_{\mathrm{Coulomb}}(\Delta p)$ and $F_{\mathrm{Stribeck}}(\Delta p)$. Thus, the friction forces of the system are dependent on the pressure supply because the single seal experiences an individual pressure difference.

5 Results

To demonstrate the model capability, the test situations presented in section 2 were simulated with the system model with one fixed parameter set but varying test situations.

Figure 15 shows the simulation of a normal switch-on phase with a commonly used pressure supply level. The diagrams start at time 0 s at which the switch impulse is sent to the solenoid valve that enables the pressure supply of p_{14}. At a sufficient pressure level, the piston module starts moving and the external pressure in the ports p_2 and p_4 change. The simulation resembles the measured data, cf. Figure 2.

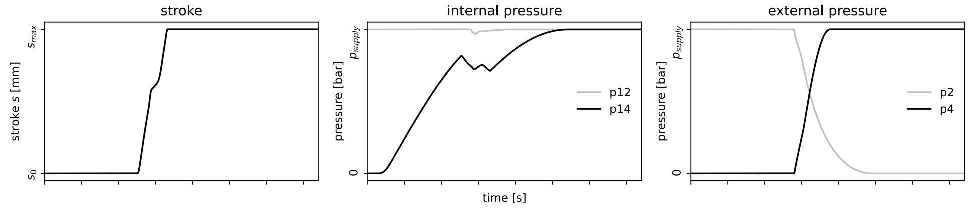

Figure 15: Simulation of a normal switch-on phase at standard pressure supply, cf. Figure 2.

The pressure dependence of the system is demonstrated in Figure 16. For the simulation only the value of the pressure supply is varied, all other parameters are constant. The main behaviour observed in Figure 4 can be reproduced by the model. The pressure dependent friction induces the increasing external forces and the slightly slower movement of the piston module.

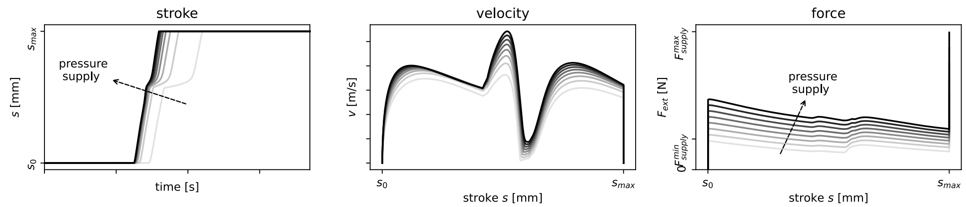

Figure 16: Simulation results for varying pressure supply cf. Figure 4

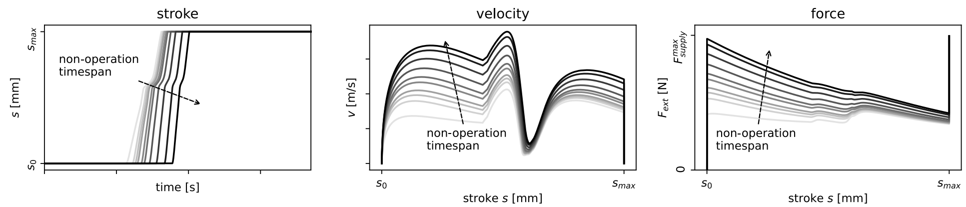

Figure 17: Simulation results for varying initial timespan of non-operation cf. Figure 5

Figure 17 shows the effect of varying timespan of non-operation. For the simulations, only the value of the initial non-operation timespan is varied, all other parameters are constant. The increasing Stribeck force yields the increasing time until the stroke is completed and higher forces acting on the piston. The simulations are in good accordance with the experimental results, cf. Figure 5.

6 Summary and Conclusion

The focus of this work was to develop a system model, based on multiple instances of basic components. The stepwise development of the model enhanced functionality by including pneumatic ports and volumes. Furthermore, the computation of friction of the system was enhanced from an overall continuous friction to a sum of non-continuous, time and pressure dependent friction forces from individual sealing rings. This system model offers the possibility to compare a large number of variants in a comparatively short time.

The task to create a specific product instance from this model is, to define all the parameters. Geometric parameters must be derived from the parts' design and air flow parameters can be determined by flow simulation. However, the identification of friction parameter is the biggest challenge. Either friction of a single seal under all conditions must be measured or determined by FE simulation using enhanced material models for the viscoelastic elastomer material.

Further improvement can be done by incorporation of air flow forces which act on the piston and can influence the motion or by replacing the Stribeck friction equation by the Lund-Grenoble [2] approach.

7 Nomenclature

Variable	***Description***	***Unit***
A_{12}	Surface area of air spring	[mm²]
A_{14}	Surface area of drive piston	[mm²]
p_{12}	Pressure in the chamber of the air spring	[bar]
p_{14}	Pressure in the chamber of the drive piston	[bar]
p_2	Pressure at working port 2	[bar]
p_4	Pressure at working port 4	[bar]
s	Piston stroke	[mm]
s_{ce}	Piston stroke where the seal meets the associated control edge	[mm]
v	Velocity	[m/s]
F_{ext}	External force on piston module	[N]
F_{Coulomb}	Coulomb friction	[N]
F_{prop}	Velocity v dependent friction	[N]
F_{Stribeck}	Difference between static and coulomb friction force	[N]
f_{ce}	'push in force'	[N]
F_{ce}	peak value of push in force	[N]
w	Width of gaussian bell shaped function of push in force	[-]
t_{no}	non-operation timespan	[s]

8 References

[1] M. Otter, H. Elmqvist and S. E. Mattsson, *Hybrid Modeling in Modelica based on the Synchronous Data Flow Principle*, In: Proceedings of the 1999 IEEE International Symposium on Computer Aided Control System Design (Cat. No.99TH8404), 1999.

[2] C. Hackl, *Dynamische Reibungsmodellierung: Das Lund-Grenoble (LuGre) Reibmodell*, In: Elektrische Antriebe - Regelung von Antriebssystemen, Springer Berlin Heidelberg, 2015, p. 1615–1657.

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