Rapid Parameterisation of a Sealing Friction Model for Hydraulic Cylinders

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Abstract

This paper presents experimental results of a sealing friction test rig for different cylinder sizes, sealing types, pressure conditions, and load cases. The test rig, the test procedure, test parameters, and measurement results are shown. Based on the obtained measurements a physically based sealing friction model is set-up which accounts for the investigated parameter variations. The friction model and the respective parameters are given. In the last part of the paper the developed sealing friction model is compared to measured results for various parameter sets. The presented model is appropriate to consider sealing friction for hydraulic cylinders. From an engineering point of view the major advantage is the instant accessibility of all required parameters to fit the model for the specific case.

KEYWORDS: sealing friction, cylinder friction, friction model, geometry based parameterisation

1. Introduction

During the design and dimensioning process of hydraulic systems the system or design engineer often encounters challenges when estimating the friction of hydraulic cylinders. Different approaches are common practice; one of them is the coulomb friction model. In some cases a friction of 10% of the nominal load force is assumed /1/. The Stribeck model is more detailed and precise and it is widely applied. However, the parameterisation of this model is a major challenge as basically four parameters need to be known: the breakaway force, the minimum friction force and its corresponding velocity, and the coefficient for lubricated friction /2/. If a knowledge base is available the Stribeck model is suitable to account for sealing friction. On the contrary, if no expert knowledge is at hand such friction models are difficult to apply. To overcome this issue extensive cylinder sealing friction measurements were carried out by RWTH Aachen University, Institute for Fluid Power Drives and Controls and Merkel Freudenberg Fluidtechnic GmbH with hydraulic differential cylinders (see also /3/). The experiments included investigations for different cylinder sizes and two types of sealing systems. Based on the obtained results a friction model is set up. This friction model is dependent on system and geometric parameters which are instantly available at the design phase of a hydraulic system. In the following the test rig and the testing procedure is presented after which the focus is set on the friction model.

2. Sealing Friction Test Rig

The test rig for sealing friction investigations is shown in **Figure 1**. It consists of a frame to which two test cylinders, aligned against each other, are mounted. These test cylinders are connected via a cross-head. The driving cylinders realise the desired motion profile by applying external forces to the cross-head. During the measurements the piston side chambers of upper and lower test cylinder are connected via hoses. The rod side chambers of both test cylinders are also connected.

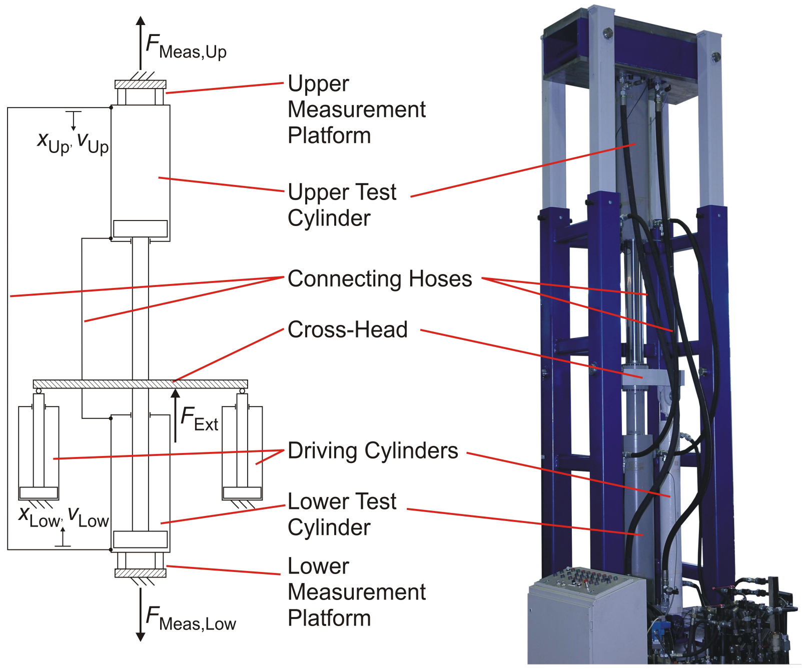

Figure 1: Cylinder Friction Test Rig

For the experiments the desired pressure conditions are imposed on the test cylinder chambers with an external hydraulic power pack. After the pressures are set the external hydraulic unit is disconnected from the test equipment with ball valves. During movement within the test procedure the pressures in the test cylinder chambers stay constant as only oil volume is exchanged and no compression or decompression occurs. The friction force is recorded with two measurement platforms mounted to each test cylinder barrel. The measured friction force is a sum of the sealing friction induced at the rod sealing system and the piston sealing system.

One full test procedure includes the cylinder outstroke and the cylinder instroke. A roughly sinusoidal motion profile was imposed by the driving cylinders to allow a smooth movement (see **Figure 2**).

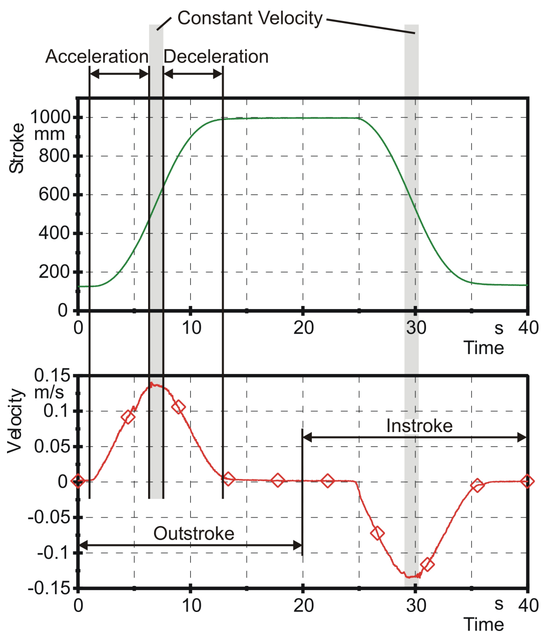

Figure 2: Velocity Profile

The maximum test velocity is about 0.15 m/s and with the applied position and velocity profile the acceleration and deceleration can be individually investigated (see also /2/ for further information).

With the described test rig, two different sets of differential cylinders were tested.

- Test cylinder pair 1: Rod diameter 50 mm, Piston diameter 100 mm
- Test cylinder pair 2: Rod diameter 160 mm, Piston diameter 200 mm

Besides different cylinder sizes, compact and chevron sealing systems were examined. Each cylinder was first tested with the compact sealing systems, then the cylinders were modified and the chevron sealing systems were mounted. As only the seal carriers were exchanged, the surfaces of rod and cylinder barrel are identical and so the experimental results of each cylinder pair can be directly compared. The sealing types used at the rod and piston were identical, meaning that for the case of the compact sealing system compact seals were mounted to rod and piston sealing system during one set of experiments.

For each of the resulting four combinations of cylinder size and sealing type, the pressure combinations listed in **Table 1** were measured.

Pressure Case	Pressure in bar				
Piston Side (p_{Rod} = 0 bar)	50	100	150	200	250
Rod Side (p_{Piston} = 0 bar)	50	100	150	200	250

Table 1: Measured Pressure Combinations

These pressure combinations are divided into two cases: The first case is the piston side case, during which the pressure in the rod side chamber is kept at 0 bar and the pressure in the piston side chamber is increased in steps of 50 bar. The second case is called the rod side case, for which the piston side chamber is kept at 0 bar and the rod side chamber pressure is increased. **Figure 3** shows a qualitative sectional view of a differential cylinder. Main difference between piston and rod side case is, that for the rod side pressure combination case both sealing systems are pressurised. For a differential cylinder and the piston side pressure combination case only the piston sealing system is pressurised.

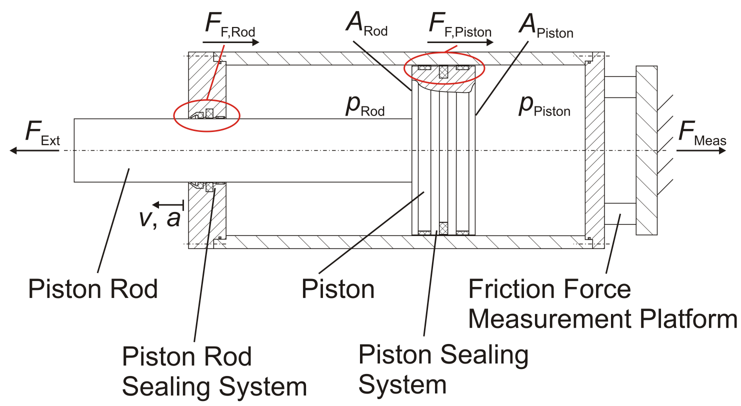

Figure 3: Sealing Systems in Differential Cylinders

After the test rig and the experimental procedure were outlined the next chapter focuses on the sealing friction model development.

3. Sealing Friction Modelling

With the described test rig experimental results were obtained. A typical result is depicted in **Figure 4**. The diagram is basically divided into outstroke (positive velocity) and instroke (negative velocity). For each stroke direction the measured curves can be divided into two parts. In case of the outstroke the one with the lower friction values characterises the acceleration phase, the curve with the slightly higher values the deceleration phase.

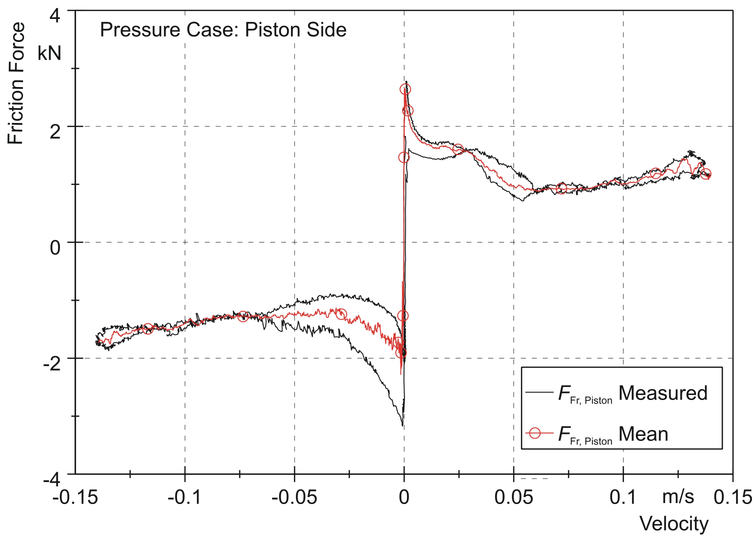

Figure 4: Typical Experimental Result

To develop a practicable friction model a mean curve was calculated, which is the mean of acceleration and deceleration phase. Such a mean curve therefore represents one measurement. For all measured curves of one parameter set the mean curves are calculated, for which an example is shown in **Figure 5**.

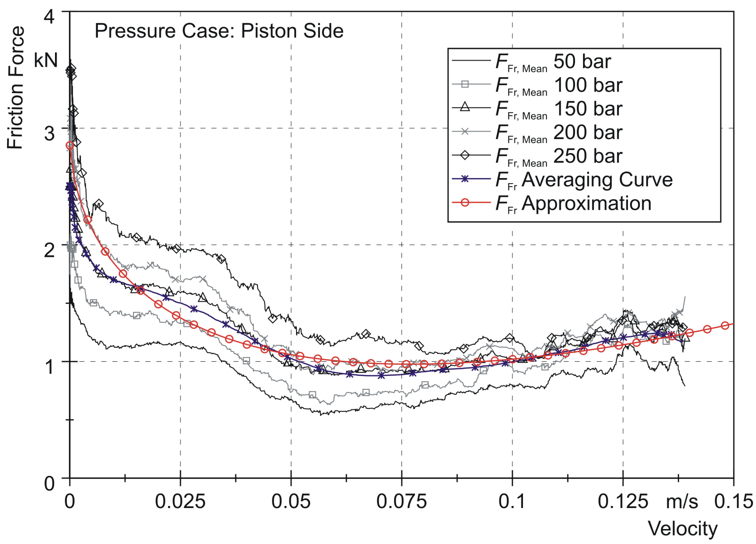

Figure 5: Mean Curves and Approximation, Compact Sealing System

Figure 5 above shows the obtained friction force curves dependent on the velocity for the case of piston side pressurisation. The curves show the mean results and the piston side pressure is increased in steps of 50 bar. The shape of the measured curves is similar and therefore an averaging curve is calculated. For modelling, this averaging curve is approximated by root functions and the following Eq. (1), given for a pressure of 0 bar, fits the averaging curve with a coefficient of determination of 0.976.

$$F_{\text{Fr, Piston, Compact}} = 1.651\,\text{kN} + 17.626\,v\,\frac{\text{kNs}}{\text{m}} - 11.733\,\sqrt{v}\,\text{kN}\sqrt{\frac{\text{s}}{\text{m}}} + 1.913 \times \times \sqrt[3]{v}\,\text{kN}\sqrt[3]{\frac{\text{s}}{\text{m}}} \qquad (1)$$

The curve of Eq. (1) is also shown in Figure 5 for a piston side pressure of 150 bar and is used to model the F-v behaviour of the compact sealing system.

Another set of mean curves for piston side pressurisation of the chevron sealing system is depicted in **Figure 6**.

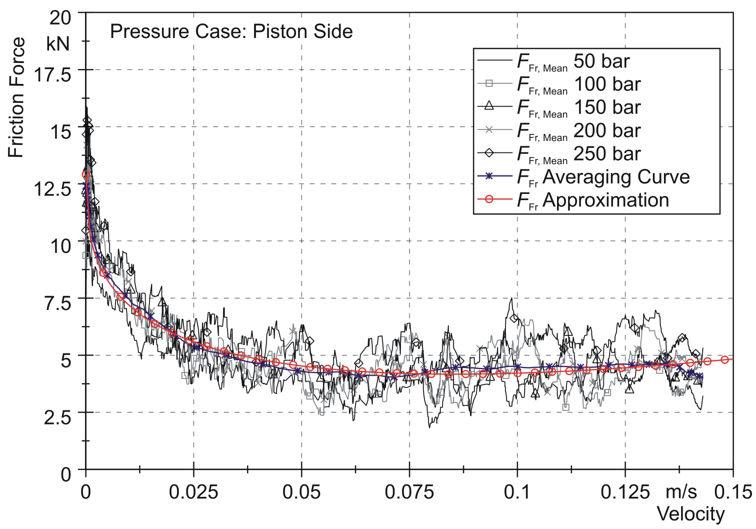

Figure 6: Mean Curves and Approximation, Chevron Sealing System

Figure 6 contains friction measurements with increasing piston side pressure. Again, the curve shape is similar and an averaging curve is calculated. This averaging curve is approximated by root functions and is used for F-v behaviour modelling. Figure 6 also illustrates the averaging curve for a pressure of 150 bar. The approximation formula for the chevron sealing system and a pressure of 0 bar is given in Eq. (2).

$$F_{\text{Fr, Piston, Chevron}} = 7.474\ \text{kN} + 35.297\, v\ \frac{\text{kNs}}{\text{m}} - 6.591\sqrt{v}\ \text{kN}\sqrt{\frac{\text{s}}{\text{m}}} - 13.961\times$$
$$\times\sqrt[3]{v}\ \text{kN}\sqrt[3]{\frac{\text{s}}{\text{m}}} \tag{2}$$

The coefficient of determination with respect to the averaging curve is 0.984. The shown formulas of Eq. (1) and Eq. (2) are valid for cylinder chamber pressures of 0 bar.

To examine the pressure dependency of the friction force, subsequent curves are directly compared. This comparison of the subsequent pressure set-up measurement data results in a pressure coefficient K_p, which is constant for both compact seals and chevron seals and independent of the load case. The coefficient K_p was found to be

$$K_p = 0.004\ \frac{1}{\text{bar}} \tag{3}$$

With the coefficient K_p the friction force for every chamber pressure can be calculated. It was expected to also identify a geometric or diameter influence of the piston sealing system induced friction. This geometry dependency of the piston friction force could not be identified in the measurement results and the piston system friction was rather constant and independent of the piston diameter. One reason is that only one pair of test cylinders per cylinder size was investigated until now and so far no statistically verified statements concerning the piston diameter can be given. To account for the geometric dependence of the piston sealing system induced friction on the piston diameter one could use approaches stated in literature.

As the pressure combinations were carried out according to Table 1, it is possible to determine the influence of the rod sealing system. When referring to Figure 3 it is obvious that the friction force for the rod side case is equal to the friction force of the corresponding piston side case plus the friction force portion of the rod sealing system. In general, four dependencies of the rod system friction were expected: Pressure dependency, geometric influence, velocity dependency, sealing type influence.

To determine these influences quantitatively the measured forces of the piston side case were subtracted from the rod side case. This yielded the friction force portion of the rod sealing system. These calculations were performed for all measurement data and it was found that the rod sealing friction pressure dependency is described by the coefficient K_p given in Eq. (3). In addition to the pressure dependency the rod system friction scales linearly with the rod diameter. The coefficient K_d characterises the geometric influence. Surprisingly, no velocity behaviour was identified and the rod sealing friction is widely constant over the velocity range instead. Eq. (4) describes the rod sealing system friction force, where d is the rod diameter and p_1 the rod side chamber pressure.

$$F_{\mathrm{Fr,\,Rod}} = K_{\mathrm{d}} K_{\mathrm{p}} d p_1 \tag{4}$$

The coefficient K_{d} differs for the compact sealing system and the chevron sealing system, which has a stronger geometric influence.

$$K_{\mathrm{d}} = \begin{cases} K_{\mathrm{d,\,Compact}} = 0.008625\ \dfrac{\mathrm{kN}}{\mathrm{mm}} \\ K_{\mathrm{d,\,Chevron}} = 0.0212\ \dfrac{\mathrm{kN}}{\mathrm{mm}} \end{cases} \tag{5}$$

To scale the friction curves with respect to the pressure dependency, the maximum cylinder differential pressure $\left|p_1 - p_2\right|$ is needed. As the portion of the rod sealing system is clearly identifiable, it is possible to model the friction force of both differential cylinders and double rod cylinders. Parameter definitions for both cylinder types are illustrated in **Figure 7**.

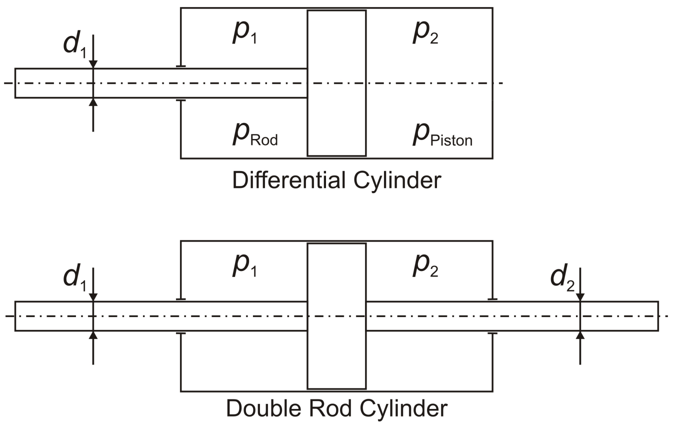

Figure 7: Designation of Chamber Pressures

Considering Eq. (1) and Eq. (2) and applying the parameter designation of Figure 7 the friction force F_{Fr} is formulated. It consists of the piston sealing system friction $F_{\mathrm{Fr,\,Piston}}$ and the rod system sealing friction $F_{\mathrm{Fr,\,Rod,\,i}}$. Scaling is accomplished with the pressures p_1 and p_2, the pressure coefficient K_{p}, the rod diameters d_{i}, and the diameter coefficient K_{d}.

$$F_{\mathrm{Fr}} = F_{\mathrm{Fr,\,Piston}} K_{\mathrm{p}} \left|p_1 - p_2\right| + \sum_{i=1}^{2} F_{\mathrm{Fr,\,Rod,\,i}} \tag{6}$$

With Eq. (4) follows the formula for the sealing friction model.

$$F_{\mathrm{Fr}} = F_{\mathrm{Fr,\,Piston}} K_{\mathrm{p}} \left| p_1 - p_2 \right| + \sum_{i=1}^{2} K_{\mathrm{d}} K_{\mathrm{p}} d_i p_i \tag{7}$$

Piston sealing friction $F_{\mathrm{Fr,\,Piston}}$, the pressure coefficient K_{p}, and the diameter coefficient K_{d} depend on the sealing type. To sum up all parameters are given in **Table 2**.

Sealing Type	Parameter	Value / Formula
Compact Sealing System	$F_{\mathrm{Fr,\,Piston}}$	$1.651\,\mathrm{kN} + 17.626\,v\,\frac{\mathrm{kNs}}{\mathrm{m}} - 11.733\sqrt{v}\,\mathrm{kN}\sqrt{\frac{\mathrm{s}}{\mathrm{m}}} + 1.913 \times \times \sqrt[3]{v}\,\mathrm{kN}\sqrt[3]{\frac{\mathrm{s}}{\mathrm{m}}}$
	K_{p}	$0.004\,\frac{1}{\mathrm{bar}}$
	K_{d}	$0.008625\,\frac{\mathrm{kN}}{\mathrm{mm}}$
Chevron Sealing System	$F_{\mathrm{Fr,\,Piston}}$	$7.474\,\mathrm{kN} + 35.297\,v\,\frac{\mathrm{kNs}}{\mathrm{m}} - 6.591\sqrt{v}\,\mathrm{kN}\sqrt{\frac{\mathrm{s}}{\mathrm{m}}} - 13.961 \times \times \sqrt[3]{v}\,\mathrm{kN}\sqrt[3]{\frac{\mathrm{s}}{\mathrm{m}}}$
	K_{p}	$0.004\,\frac{1}{\mathrm{bar}}$
	K_{d}	$0.0212\,\frac{\mathrm{kN}}{\mathrm{mm}}$

Table 2: Overview of Parameters

To show the applicability of the developed friction model the model is compared with experimental results. **Figure 8** illustrates the model for the compact sealing system and the cylinder with a piston diameter of 100 mm when the piston side is pressurised. An appropriate fit is visible for the middle and upper velocity ranges, whereas the breakaway force of the shown measured curves is about 250 N lower than the model (roughly 10% deviation). In Figure 8 the mean curves of acceleration and deceleration are compared. The breakaway force of the deceleration motion is usually above the mean curve (see Figure 4) and so the model is considered adequate.

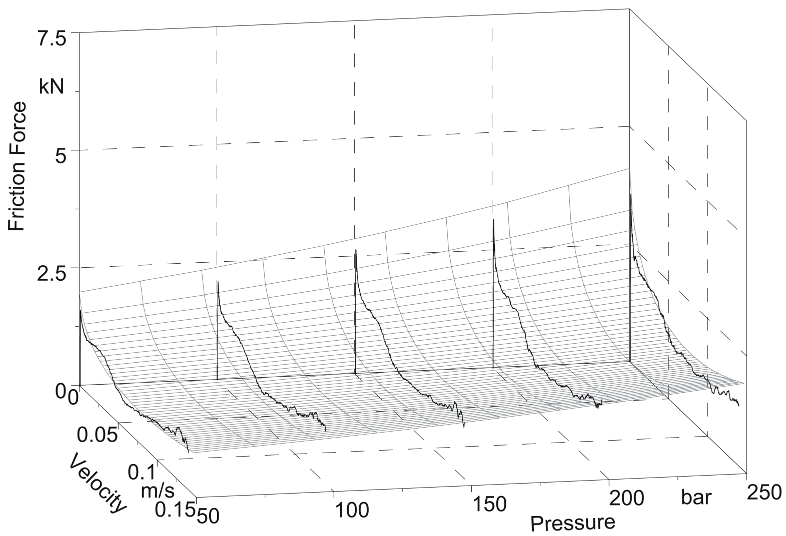

Figure 8: Compact Seals, Piston Diameter 100 mm, Pressure Case: Piston Side

In **Figure 9** experimental results and the friction model are illustrated for compact seals and a piston diameter of 200 mm when the rod side chamber is pressurised. For this case both sealing systems are pressurised, resulting in a higher friction force level. The F-v behaviour of the model is acceptable and especially the pressure dependency is appropriate. For this case the breakaway forces of measurements and model also differ with a value of about 500 N.

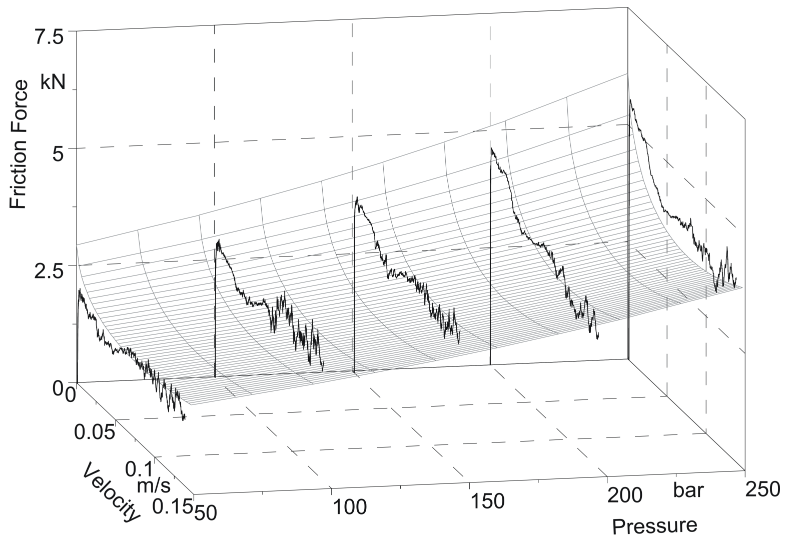

Figure 9: Compact Seals, Piston Diameter 200 mm, Pressure Case: Rod Side

Figure 10 depicts the comparison of experiments and model for the cylinder with a piston diameter of 100 mm with chevron sealing system. In this figure the piston side pressurisation is shown for which the model describes the F-v behaviour, the pressure dependency, as well as the breakaway friction force with a high accuracy.

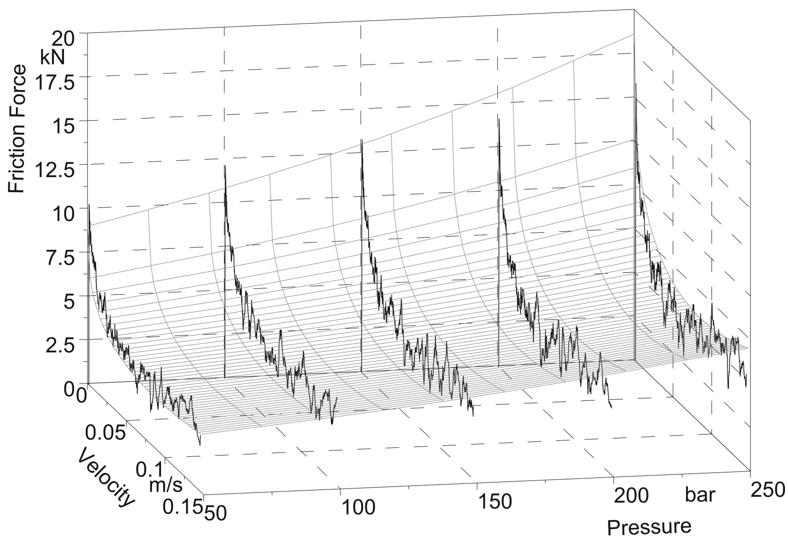

Figure 10: Chevron Seals, Piston Diameter 100 mm, Pressure Case: Piston Side

The developed friction model based on geometric parameters is a promising approach for system engineers to account for sealing friction in hydraulic cylinders. Main advantage is the instant accessibility of parameters. To fit the model to the application the following information is necessary: type of cylinder, piston and rod diameters, expected pressures, velocity, type of sealing system. If commercially available system simulation programs are used, the system inherent parameters, i.e. pressure and velocity, are implicitly known. Furthermore, the type of cylinder as well as the piston and rod diameters are parameterised anyway. The only additional information needed is the type of sealing system, which is instantly available for the design and system engineer.

4. Conclusion and Outlook

In this paper sealing friction measurements are shown that were obtained with an especially developed test rig for differential cylinders of larger diameters. Two test cylinders are measured at the same time and a sinusoidal velocity motion profile is applied for the test procedure. Maximum speeds of 0.15 m/s were investigated and the test cycles included outstroke and instroke. For the acceleration and deceleration phase of the motion differing characteristics were observed. Mean curves were calculated from acceleration and deceleration phase and a friction model was developed based on these curves. The model is dependent on the sealing type, the predominant cylinder chamber pressures, the load case, and the diameter of the piston rod. Finally, the developed model is directly compared to measurement data sets and is an appropriate tool to precisely account for sealing friction. As the presented sealing friction model is exclusively based on system and geometry parameters it is easy to handle especially in the planning phase of a hydraulic system.

5. References

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6. Nomenclature

a	Test Cylinder Acceleration	m²/s
d	Rod Diameter	mm
F_{Ext}	External Force	kN
F_{Fr}	Friction Force	kN
F_{Meas}	Measured Force	kN
K_d	Rod Diameter Coefficient	kN/mm
K_p	Pressure Coefficient	1/bar
p_i	Chamber Pressures	bar
p_{Piston}	Piston Side Chamber Pressure	bar
p_{Rod}	Rod Side Chamber Pressure	bar
v	Test Cylinder Velocity	m/s
x	Test Cylinder Position	m