Knuuti, Kivi; Calonius, Olof

Experimental Analysis of Friction Forces of Hydraulic Rod Seals—Effect of Pressure, Sliding Speed, Sealing Type, and Different Rod Coatings

Published in:
Advancements in Fluid Power Technology: Sustainability, Electrification, and Digitalization (GFPS 2024)

DOI:
10.1007/978-3-031-84505-5_6

Published: 01/01/2025

Document Version
Publisher's PDF, also known as Version of record

Please cite the original version:
Knuuti, K., & Calonius, O. (2025). Experimental Analysis of Friction Forces of Hydraulic Rod Seals—Effect of Pressure, Sliding Speed, Sealing Type, and Different Rod Coatings. In L. Ericson, & P. Krus (Eds.), *Advancements in Fluid Power Technology: Sustainability, Electrification, and Digitalization (GFPS 2024) : Sustainability, Electrification, and Digitalization - Proceedings of the Global Fluid Power Society PhD Symposium 2024* (pp. 83-97). (Lecture Notes in Mechanical Engineering). Springer. https://doi.org/10.1007/978-3-031-84505-5_6

Experimental Analysis of Friction Forces of Hydraulic Rod Seals—Effect of Pressure, Sliding Speed, Sealing Type, and Different Rod Coatings

Kivi Knuuti and Olof Calonius

1 Introduction

Hydraulic cylinders are extremely widely used in mobile and industrial applications for creating linear movement. All these cylinders need tight sealing to keep the highly pressurized oil inside of the cylinders and to not let it leak past the piston. Tight sealing, however, always induces friction. Kühnlein et al. [1] have measured that the friction generated by the sealing can be as much as 4% of the nominal force of the cylinder, which means that frictional forces directly decrease its performance. Additionally, friction can cause stick–slip phenomenon and worsen positioning accuracy.

Friction is a difficult phenomenon to model and even after decades of research the current models for friction in lubricated contact typically either require a lot of experimental data from the system or do not provide very accurate results. Therefore, for creating better simulation models that can be used to predict sealing behavior and to develop better seals, more data and understanding of frictional behavior is needed.

Experimental measurements of friction of hydraulic seals have been conducted using whole differential cylinders. For example, Yanada et al. [2] and Tran et al. [3] used a whole hydraulic cylinder to measure and thereby verify their computational models for sealing friction. Kühnlein et al. [1, 4] have designed a system where constant pressure measurements are much easier to obtain. In more recent years, Pan et al. [5] have compared different sealing types in varying conditions. However, when investigating a whole cylinder, it is impossible to separate friction forces induced by piston and rod seals. Often frictional forces also have to be separated from the much higher pressure forces of the piston which causes uncertainties in the measurements.

K. Knuuti (✉) · O. Calonius
Aalto University, Espoo, Finland
e-mail: kivi.knuuti@aalto.fi

L. Ericson and P. Krus (eds.), *Advancements in Fluid Power Technology: Sustainability, Electrification, and Digitalization*, Lecture Notes in Mechanical Engineering,
https://doi.org/10.1007/978-3-031-84505-5_6

Another approach is to measure friction using "non-cylinders". Angerhausen et al. [6] and Hess and Soom [7] have measured friction by pressing a strip of seal material against a rotating cylindrical surface. Muraki et al. [8] focused on stick–slip phenomenon of hydraulic seals by sliding a seal specimen on a plate.

The standard ISO 7986 [9] describes a test system for measuring hydraulic rod seal friction. Similar test rigs have been used, e.g., by Papatheodorou [10, 11] and Bhaumik et al. [12]. There are also papers studying rod seal friction with different test rigs. Bullock [13] has conducted wide-ranging measurements with constant speeds and sinusoidal movement in pressures under 80 bar. However, also his test system design requires computing out the pressure forces acting on the rod. Wang et al. [14] measured how rod surface roughness changes the frictional forces but only in pressures below 50 bar. Heipl and Murrenhoff [15] used a crank mechanism to achieve velocities up to almost 10 m/s. Crudu et al. [16] did simulations and measurements in pressures ranging from 50 to 200 bar with constant speeds ranging from 43 to 80 mm/s. Nikas et al. [17] have studied rectangular seals in very wide operating conditions: temperatures ranging from − 54 to + 135 °C and pressures ranging from 34 to 345 bar.

In this paper, friction forces of seals made of polyurethane and bronze-filled PTFE (polytetrafluoroethylene) are measured using three different rod coatings: standard hexavalent chromium, black nitride coating, and an alternative, more environmentally friendly, trivalent chromium coating. To the best of authors' knowledge, such comparison between mentioned rod coatings and sealing materials has not been conducted previously in the literature. The novelty of this research is the testing and comparison of these combinations with different relative velocities and at different pressures.

The measurements are conducted at constant pressures and with constant sliding speeds; dynamic changes in operating conditions are out of the scope of this article. The current test rig does not allow for measuring the direction dependency of the friction of the seals. Leakage and wear are also out of the scope of this article.

In chapter "Subsystem-Based Learning Control of Hydraulically Driven Nonlinear Rotary Actuators with Unknown Input Backlash" the test rig and the measurement plan are introduced. The results are shown in chapter "Evaluation of a Simple Method to Estimate the Shaft Torque in a Gerotor Pump" and discussed in chapter "Optimization-Based Energy Efficient Power Transmission Design Methodology Applied to a Compact Excavator". Chapter "Analysis of the Operating Point Method for Dimensioning of Pneumatic Drives Under Variable Loading Conditions" provides a conclusion.

2 Test Rig

The measurements in this paper are made using the test rig in Fig. 1. Section 2.1 is an overview of the physical construction of the test cylinder and the actuating cylinder. Section 2.2 goes over the seals and cylinder rods used for measurements and the measurement plan is introduced in Sect. 2.3.

2.1 Physical Construction

The test setup consists of a test cylinder and an actuating cylinder. A simplified schematic view is shown in Fig. 2 and a hydraulic schematic in Fig. 3. The actuating cylinder is custom-made to be low-friction to ensure smooth output. It does not have a piston seal and the rod seals are replaced with hydrostatic bearings. Low-friction PTFE seals act as scrapers to prevent excess leakage and air from getting into the cylinder. A proportional 4/4-type Bosch Rexroth 4WRPEH 6 C3B24L-2X/G24K0/A1M valve (Fig. 3, 2v5) is used for direction control.

The design of the test cylinder shares similarities with the one introduced in the standard ISO 7986 [9]. A section view of it is shown in Fig. 4. The design allows the

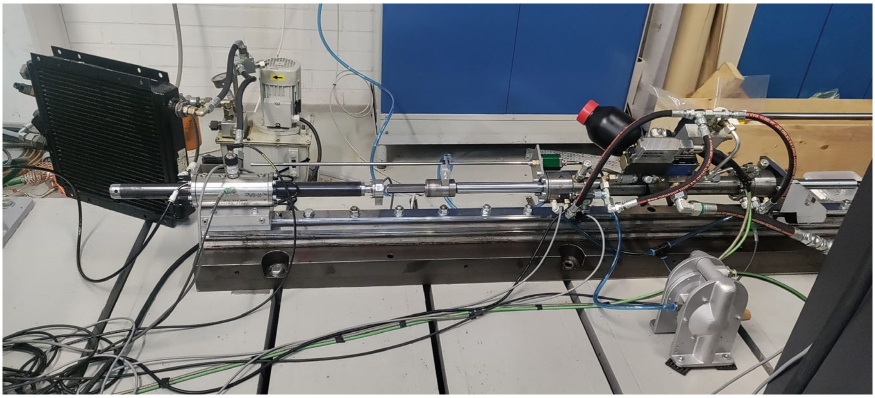

Fig. 1 Photo of the test rig

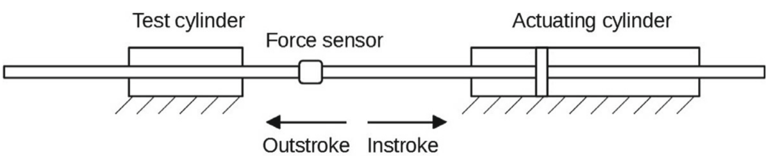

Fig. 2 Simplified schematic view of the test rig and the directions of instroke and outstroke

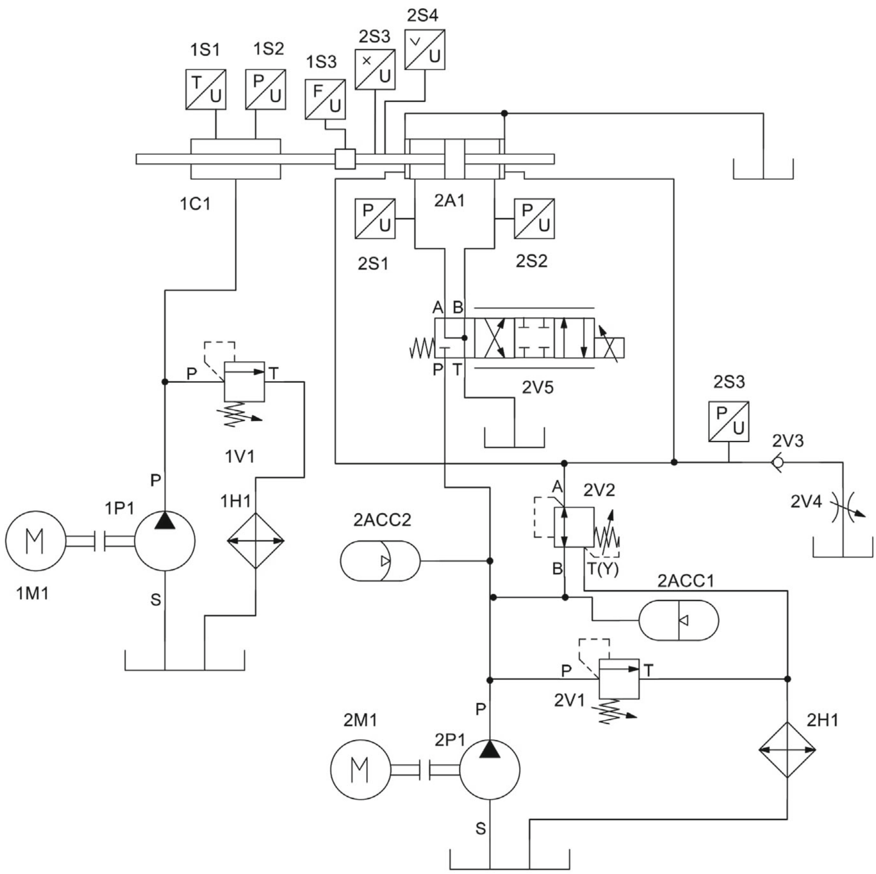

Fig. 3 Hydraulic circuit diagram of the test rig

seals to be kept in normal operating conditions (oil lubrication, pressure, temperature) while setting no limits for the stroke length. Additionally, there are no pressure forces acting on the rod in the axial direction. That means that the friction forces of the seals can be directly measured, and they do not have to be separated from pressure forces induced by a piston, for instance. The used force sensor is 5 kN HBM U2B (1S3).

Two opposing seals, one at each end of the test cylinder, means that the direction dependency of friction for the seals—which is to be expected [13]—cannot be measured using this setup and the friction value for a seal is a sum of its instroke and outstroke friction. In later chapters "instroke" and "outstroke" refer to the direction of movement of the actuating cylinder (Fig. 2).

The test seals and the guide rings are assembled in inserts machined to the manufacturer's specification. That makes it possible to use the same cylinder for testing different seals and allows for quick and easy exchange of seals.

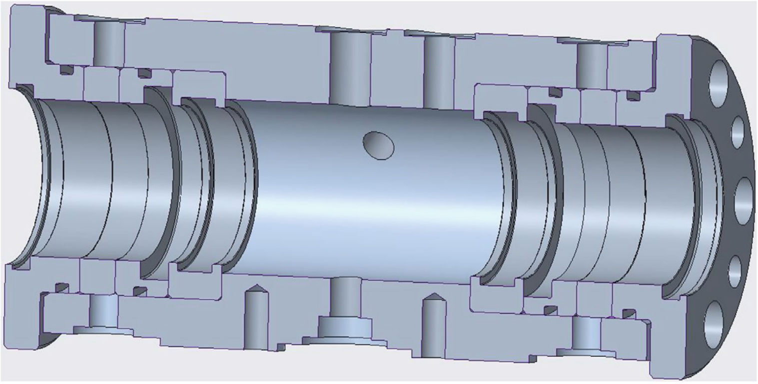

Fig. 4 Section view of the test cylinder

The test cylinder has drilled axial channels for temperature management and threaded holes for temperature and pressure sensors (Nokeval TRCP Pt100 (1S1), Hydac HDA 4746 (1S2), and Trafag 8251 (2S2, 2S3)).

Pressure inside of the test cylinder is generated by a hydraulic power pack (1M1 & 1P1, Fig. 2). The pressure is manually adjusted using a pressure relief valve 1V1. The viscosity of the oil used is ISO VG 32.

2.2 Test Seals and Rods

The measurements are made using two different seals: Merkel U-ring T20 made of polyurethane and Merkel Omegat OMS-MR made of PTFE-bronze compound. As is seen from the section views in Fig. 5, the cross-sectional profile of the seals is not similar. However, they represent a shape typical to their respective materials and therefore this study provides comparison data relevant in many applications.

The rods used are 40 mm in diameter and have a nominal length of 600 mm. Their specifications are in Table 1 and they represent what is currently commercially available. The device used for measuring the surface roughness values is Mahr MarSurf PS10.

2.3 Measurement Plan

In the standard ISO 7986, friction measurements are obtained in pressure levels of 63, 160, and 315 bar which are typical working pressures in applications [9]. However, the test cylinder used in this paper is not capable of reaching 315 bar. Therefore, 63

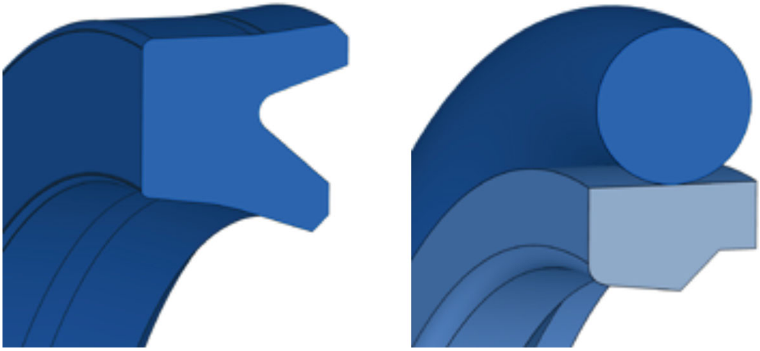

Fig. 5 Section views of the test seals. Merkel U-ring T20 (polyurethane) on the left and Merkel Omegat OMS-MR (PTFE-bronze + nitrile rubber O-ring) on the right

Table 1 Surface roughness properties of the test rods

Rod number	Rod 1		Rod 2		Rod 3	
Coating	Standard hexavalent chromium coating		Black nitride coating		Alternative coating	
Surface finish parameters before the tests	Parameter	Value (μm)	Parameter	Value (μm)	Parameter	Value (μm)
	Ra	0.140	Ra	0.411	Ra	0.251
	Rq	0.178	Rq	0.524	Rq	0.332
	Rz	1.037	Rz	3.778	Rz	2.704
	Rmax	1.350	Rmax	4.751	Rmax	3.802
	Rp	0.413	Rp	1.631	Rp	0.682
	Rt	1.350	Rt	5.839	Rt	3.993

and 160 bar are chosen from the standard and 110 bar, which is close to their average, is chosen as the third pressure point.

In the standard, data is collected in three constant sliding speeds: 50, 150, and 500 mm/s [9]. The hydraulic power pack available during the time of measurements was not capable of producing enough volume flow rate to reach 500 mm/s. Therefore, 500 mm/s was reduced to 450 mm/s. Additionally, to better capture the changes in friction as a function of velocity, an additional data point was added and data is collected in four constant velocities: 50, 150, 300, and 450 mm/s.

The speed profile for the constant speed measurements (Fig. 6) is made according to the standard. During one cycle, speed is constant for 80% of the time and for every direction change, a 10% ramp time is used. P-control of the rod position with velocity feedforward is used to implement the velocity command.

For each rod, one new set of both seals is used. Before taking the measurements, a 15-min-long run-in cycle is done. It is done at 110 bar pressure and 150 mm/s sliding speed, using the speed profile described above. After that, a measurement run lasting 150 s is done for each pressure and sliding speed combination.

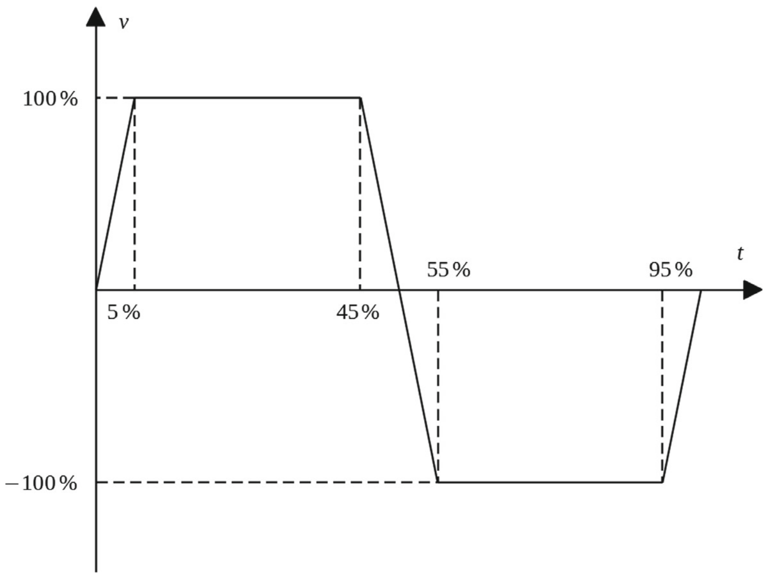

Fig. 6 Velocity command profile used in the tests. Positive velocity refers to outstroke motion as defined in Fig. 2

The temperature is held constant by controlling the temperature of water passing through the channels in the walls of the test cylinder. The controlling is done using Danfoss AVTA and AVTB thermostatic valves. The realized temperature during the tests was on average 42.4 °C.

Data acquisition is done using a National Instruments PCI 6259 card connected to a PC, where data is recorded to a MATLAB/Simulink environment at 100 Hz except for pressures and position, which are recorded at 10 Hz and 200 Hz, respectively. All signals are analog ± 10 V signals.

3 Results

A typical force–time graph of a measurement can be seen in Fig. 7. Especially with the polyurethane seal, some sticking happens in every direction change. Therefore, the friction changes into static friction for a brief moment and a peak in friction force is observed after each direction change. When the sliding speed settles to a constant value, friction stays rather constant. An arithmetic mean is taken from this rather constant section for four different cycles along the 150-s-long measurement. An example of such measurement interval is shown in the right graph in Fig. 7. An average of the four averages across the intervals is considered as the friction value for a specific measurement. The start and end points of each interval are chosen by hand.

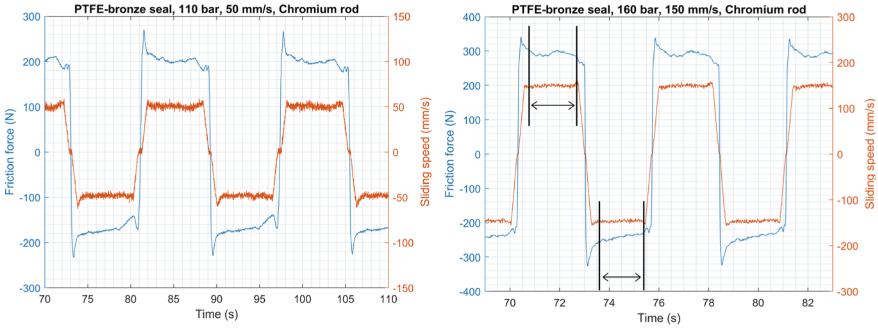

Fig. 7 Typical force–time graphs of measurements. One measurement interval for instroke and outstroke is shown in the right picture

The friction values are for an unknown reason not the same for outstroke and instroke. On average, the outstroke value of friction is 45 N higher than the instroke value. The discrepancy is of the same magnitude regardless of the friction force level but not constant enough to be treated as systematic error. Therefore, the average of absolute values of instroke and outstroke values is used.

The setup is carefully aligned using lasers but since the test rod is only supported by the soft seals and polymer wear bands, some misalignment can happen and cause the lack of symmetry of results.

3.1 Effect of Sealing Choice

Of the tested parameters, the sealing choice has the biggest effect on friction. The polyurethane seal always produces more friction force and on average the forces are 4.1 times higher. The greatest difference is 8.5 times higher force (50 mm/s, 63 bar, black nitride coating) and the smallest difference is 1.8 times higher force (450 mm/s, 160 bar, alternative coating).

No stick–slip behavior (as described, e.g., in [18]) was observed with the PTFE-bronze seals. However, with the polyurethane seal, on high pressures (110 bar and especially 160 bar) and low velocities (mainly on 50 mm/s but also on 150 mm/s with black nitride and alternative coating) different levels of stick–slip behavior ranging from minor to overwhelming was observed. Example force–time graphs are shown in Fig. 8. With the alternative coating rod, in 160 and 110 bar pressures and 50 mm/s sliding speed, no constant level of speed sliding was achieved, and these data points are omitted from calculations. Figure 8 the uneven movement also causes spikes to the force graph. Therefore, it is possible that values for high-pressure and low-speed tests are not fully representative.

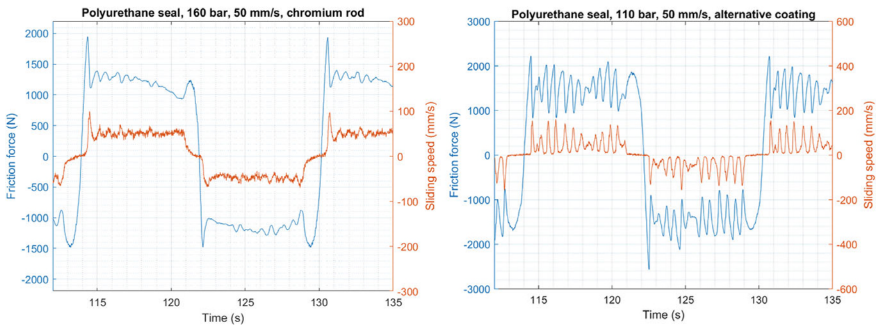

Fig. 8 Stick–slip observations. Minor on the left and major on the right

3.2 Effect of Choice of Rod Coating

The rod coating choice has a significant impact on the friction levels. In this chapter, comparisons between each coating are made for both sealing types.

Polyurethane sealing. For the polyurethane seal, the black nitride coating induces the most friction in all operating conditions, as seen in Fig. 9. On average, it generates 40.4% more friction than the standard chromium coating and 25.1% more than the alternative coating. The highest friction it generates is 1650 N at 50 mm/s sliding speed at 110 and 160 bar pressures and, at its lowest, it generates 836 N of force (450 mm/s, 63 bar).

The order between chromium and alternative coating changes depending on the sliding speed. On the slower speeds, 50 and 150 mm/s, chromium is the one with the least resistance to sliding with 49.2% less friction on average than the black nitride coating. On higher speeds, 300 and 450 mm/s, the alternative coating is the slipperiest one producing 39.8% less friction than the black nitride coating on average. The lowest friction measured with polyurethane seal was 535 N.

Interestingly, even though the black nitride coating generates the most friction, the alternative coating is more likely to induce stick–slip behavior. That could indicate that with this coating, ratio of the static and the dynamic friction is higher compared to the other tested coatings [8].

PTFE-bronze sealing. For the PTFE-bronze seal the chromium rod is the best choice in terms of friction in all measurement points. With it, the friction force values range from 120 to 315 N. In all but two measurement points, the alternative coating produces the most friction, on average 26.5% more than the chromium-coated rod. The maximum frictional force it creates is 380 N.

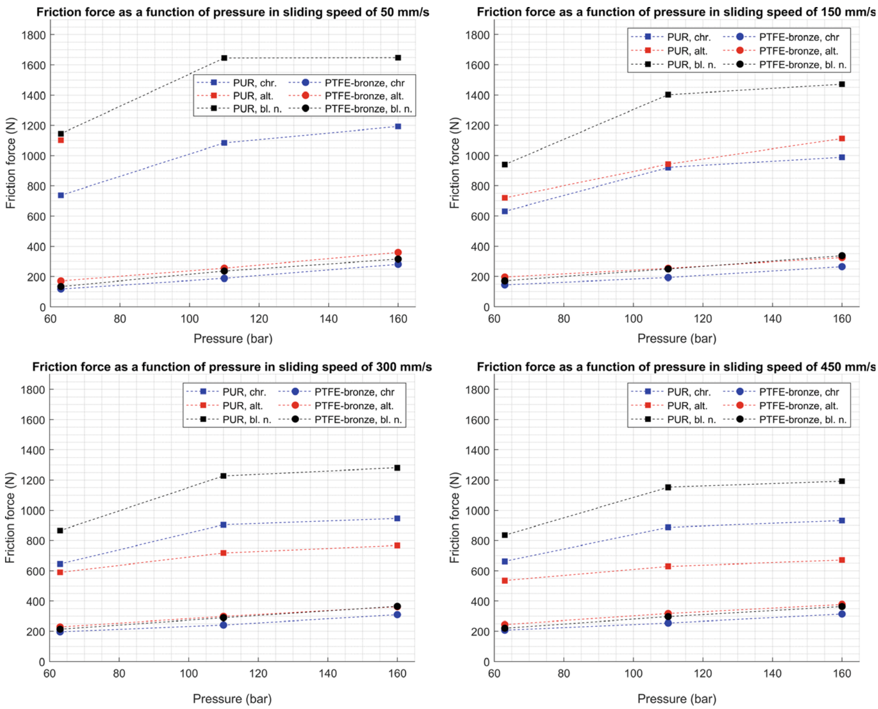

Fig. 9 Friction force as a function of pressure in different sliding speeds. Abbreviations: PUR = polyurethane sealing, “chr.” = chromium coating, “alt.” = alternative coating and “bl. n.” = black nitride coating

3.3 *Effect of Pressure*

With all speeds, seals, and rod coatings, friction increases with increasing pressure. For the PTFE-bronze seal, the effect is more significant: when the pressure increases from 63 to 160 bar, friction increases 51.6–138.7% depending on sliding speed. The least increase is with 450 mm/s sliding speed, next least with 300 mm/s and the highest increase is with 50 mm/s.

For the polyurethane seal, increase in friction over the whole pressure range is 25.3–61.7% depending on sliding speed. The sliding speed has similar impact as with PTFE-bronze seals: the least increase is with 450 mm/s and the biggest is with 50 mm/s.

With the PTFE-bronze-seal, the change over the whole pressure range is rather linear: the change from 63 to 110 bar is of the same magnitude as from 110 to 160 bar (Fig. 9). With the polyurethane seal the increase of friction saturates: from 63 to 110 bar the change is on average 37.3% but from 110 to 160 bar only 6.5%. Papatheodorou and Igers have observed similar saturation behavior with polyurethane U-cup seals [11].

It is to be noted that the absolute values of the frictional forces and therefore the absolute change in them are higher for the polyurethane seal due to the higher starting values.

The increase in friction with increasing pressure is to be expected since the contact area between the seal and the rod increases with increasing pressure [11, 19]. Especially in lower sliding speeds, when the friction is in boundary or mixed lubrication regime [18], the bigger surface area means more touching surface asperities and therefore more friction. Additionally, higher pressures lead to lower film thickness with both PTFE [19] and polyurethane seals [16] which leads to more friction [16].

3.4 Effect of Sliding Speed

For the polyurethane seal, the Stribeck effect (as described, e.g., in [18]) can be observed in the force–velocity graphs (Fig. 10). The frictional force first decreases with increasing velocity. This is due to lubrication conditions improving in the contact between the seal and rod [18]. Based on theory, when velocity increases further, full lubrication regime is reached, and friction force should become dominated by the viscous friction of the oil film between the seal and the rod, and the friction should begin to increase linearly [20]. However, in these tests, no increase of friction is observed except for the 63 bar test with the chromium rod, suggesting that full lubrication regime is not reached with the sliding speeds used in this paper. Although, for example, Pan et al. [5] and Crudu et al. [16] have observed friction forces increasing with increasing velocity with polyurethane seals in constant velocities well under 100 mm/s.

For the PTFE-bronze seal, only in 160 bar pressure with chromium and alternative coating rod, the graph has (slight) characteristics of the Stribeck curve, i.e., first decreasing and then increasing friction for both instroke and outstroke. The decrease is small, 5.9% for the chromium rod (Fig. 10, bottom graph) and 9.8% for the alternative coating.

When increasing the sliding speed from 150 mm/s, friction forces increase with all rod coatings, which is to be expected when full lubrication regime is reached [20]. The friction increase, when sliding speed changes from 150 to 300 mm/s, is on average 19% and 3.9% when going from 300 to 450 mm/s.

3.5 Other Results

Leakage was not the object of this paper and thus was not measured, but with the black nitride coating and PTFE-bronze seal, a small amount (a drop for every two stokes) of leakage was observed. The leaking oil had very fine bronze-colored dust in it, presumably from the seal.

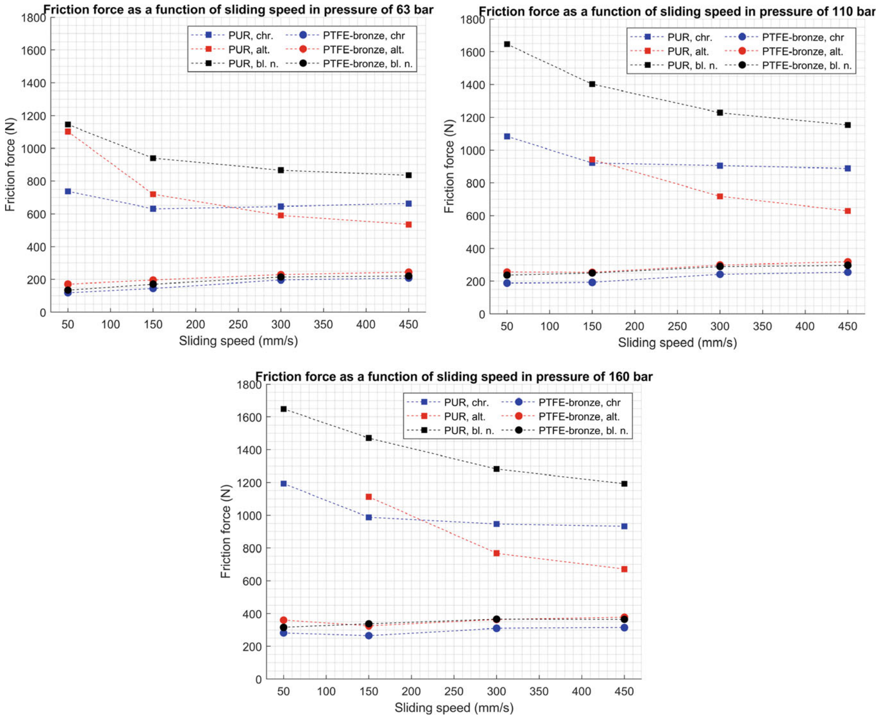

Fig. 10 Friction force as a function of sliding speed in different pressures. Abbreviations: PUR = polyurethane sealing, "chr." = chromium coating, "alt." = alternative coating and "bl. n." = black nitride coating

It was observed that especially during the runs with the polyurethane seal, the temperature in the test cylinder rose slightly (<1.5 °C). This is due to the friction heat accumulating on the contact interface [14].

4 Discussion

According to [19] and [11], PTFE-based seals can be considered as low-friction and these measurements support that claim. However, Papatheodorou and Igers [11] state that it holds true when the seals are new, but in worn-in condition and especially in high pressures and temperatures, the friction can increase a lot. In their measurements, Pan et al. [5] found polyurethane U-cup seals to generate less friction than PTFE seals. However, their measurements were done at much lower pressures (under 20 bar) which might affect the results. In tests conducted at very high sliding speeds (up to almost 10,000 mm/s), Heipl and Murrenhoff [15] found polyurethane seals to generate more friction compared to a PTFE seal.

The chromium rod is the only one filling the surface roughness requirements set by the seal manufacturer. For both seal types, Ra on the sliding surface should be 0.05–0.3 μm and Rmax should be less than 2.5 μm. The black nitride rod surpasses both of those numbers (Table 1). The Ra value of the alternative coating is within specifications, but the Rmax value is not. Increasing surface roughness causes greater frictional forces and leakage [14]. This means that some of the measured differences in friction can solely be from the greater surface roughness of the rods with black nitride and trivalent chromium coatings and not from the coating material itself. It should also be noted that the out-of-spec surface roughness can affect the lubrication conditions in the contact between the seal and the rod and cause the seal to work not as intended. Therefore, direct conclusions about comparing the frictional properties of these coating materials cannot be made based on this paper. However, these rods represent the current state of the art available commercially, making the comparison between the rods with as-purchased surface roughnesses justified.

The custom-made low-friction cylinder used in these tests proved to be quite problematic in regards to control. Possibly because the area of the piston is small compared to the diameter of the rod, or due to the lack of internal friction, stick–slip occurred easily and was hard to control. Different levels of stick–slip were observed with 50 and 150 mm/s runs at 110 and 160 bar pressures with polyurethane seals and that causes some uncertainties in the measured friction values. And some data points, such as 160 bar, 50 mm/s run with an alternative coating rod, could not be used.

The actuating cylinder can also be one of the reasons why the friction force and behavior of the system are different for the different stroke directions. One other possible reason could be the calibration of the force sensor. It was zeroed and calibrated only using tensile loads. However, since not only the friction force but also the behavior of the system is not symmetric, it is more likely that the reason has something to do with the construction of the system, for example, alignment.

Due to the lack of volume flow produced by the power pack, 500 mm/s sliding speed suggested by the standard had to be lowered to 450 mm/s. And still, with the most demanding run (450 mm/s, 160 bar, black nitride with polyurethane) the realized average sliding speed was 411 mm/s.

For the polyurethane seal, the speed value for the lowest point in the Stribeck curve, i.e., the point at which the decrease in friction stops, is quite high compared to literature. For example, in [5] the minimum point is roughly 15–30 mm/s depending on pressure for outstroke and under 20 mm/s for instroke with polyurethane U-cup seals. In [4], the minimum point is between 50 and 75 mm/s for compact seals (no material stated). In this paper, except for 63 bar tests with the chromium rod, no minimum was found, and friction forces were still decreasing at 450 mm/s. However, it is to be noted that the minimum can exist between 300 and 450 mm/s.

5 Conclusion

In this paper, friction forces of two different seal types were measured at three different pressure levels in four different constant sliding speeds using three different rod coatings. Especially the sealing material proved to have a significant effect on the friction. The friction force of a polyurethane seal was on average over 4 times higher than the friction force of a PTFE-bronze seal and therefore for an application where low friction is expected from the seal, a PTFE-bronze seal would be a better option.

Friction is seen to increase with increasing pressure with both seal types. With polyurethane, saturation behavior is observed, whereas with PTFE-bronze, the increase is more linear. Behavior as a function of sliding speed is not as uniform, however. With polyurethane seal, behavior is reminiscent of the Stribeck curve but with the PTFE-bronze seal friction increases with increasing velocity. That might, however, mean that the minimum of the Stribeck curve has been reached with lower sliding speeds and the whole measuring range is in full lubrication regime.

The surface roughness of the three tested rods differs so much that no clear conclusions can be made about comparing the frictional properties of the coating materials. However, the tested rods represent what is currently commercially available and this study shows that choice of rod coating has a significant effect on friction of the cylinder.

In further research, different rod coatings with similar surface roughness properties or seals with similar geometries could be compared. The comparisons could be expanded to dynamically changing velocities and breakout friction.

References

1. Kühnlein M et al (2011) Inner friction of large hydraulic differential cylinders. In: 12th Scandinavian international conference on fluid Power. Tampere, Finland, Tampere University of Technology
2. Yanada H, Sekikawa Y (2008) Modeling of dynanmic behaviors of friction. Mechatronics 18(7):330–339. ISSN: 0957–4158
3. Tran XB, Hafizah N, Yanada H (2012) Modeling of dynamic friction behaviors of hydraulic cylinders. Mechatronics 22(1):65–75. ISSN: 0957–4158
4. Kühnlein M et al Rapid parameterisation of a sealing friction model for hydraulic cylinders. In: 8th international fluid power conference. Dresden, Germany, Dresdner Verein zur Förderung der Fluidtechnik
5. Pan Q et al (2021) Experimental investigation of friction behaviors for double-acting hydraulic actuators with different reciprocating seals. Tribol Int 153:106506. ISSN: 0301–679X
6. Angerhausen J et al (2017) Influence of temperature and surface structure on the friction of dynamic hydraulic seals. In: The 10th JFPS international symposium on fluid power. Fukuoka, Japan, JFPS
7. Hess D, Soom A (1990) Friction at a lubricated line contact operating at oscillating sliding velocities. J Tribol 112(1):147–152. ISSN: 0742–4787
8. Muraki M, Kinbara E, Konishi T (2003) A laboratory simulation for stick-slip phenomena on the hydraulic cylinder of a construction machine. Tribol Int 36(10):739–744. ISSN: 0301–679X

9. ISO 7986: 1997(E) Standard test methods to assess the performance of seals used in oil hydraulic reciprocating applications. Geneva, Switzerland: International Organization for Standardization, 20 p
10. Papatheodorou T (2005) Influence of hard chrome plated rod surface treatments on sealing behavior of hydraulic rod seals. Seal Technol 2005(4):5–10
11. Papatheodorou T, Igers W (2011) Friction optimization on hydraulic piston rod seals. In: 52nd national conference on fluid power. Las Vegas, NV, USA. Milwaukee, WI, USA. National Fluid Power Assiociation, pp 833–840
12. Bhaumik S et al (2015) Investigation of friction in rectangular Nitrile-Butadiene Rubber (NBR) hydraulic rod seals for defence applications. J Mech Sci Technol 29(11):4793–4799. ISSN: 1738–494X
13. Bullock A (2010) Fundamental concepts associated with hydraulic seals for high bandwidth actuation. PhD Thesis. University of Bath, Department of Mechanical Engineering. Bath, Great Britain, 217 p
14. Wang B et al (2021) Experimental investigations on the effect of rod surface roughness on lubrication characteristics of a hydraulic O-ring seal. Tribol Int 156(106791):0301-679X
15. Heipl O, Murrenhoff H (2015) Friction of hydraulic rod seals at high velocities. Tribol Int 85:66–73. ISSN: 0301–679X
16. Crudu M et al A numerical and experimental friction analysis of reciprocating hydraulic 'U'rod seals. Proc Inst Mech Eng Part J J Eng Tribol 226(9):785–794. ISSN: 1350–6501
17. Nikas GK, Almond RV, Burridge G (2014) Experimental study of leakage and friction of rectangular, elastomeric hydraulic seals for reciprocating motion from −54 to + 135 °C and pressures from 3.4 to 34.5 MPa. Tribol Trans 57(5):846–865. ISSN: 1040–2004
18. Armstrong-Hélouvry B, Dupont P, De Wit CC (1994) A survey of models, analysis tools and compensation methods for the control of machines with friction. Automatica 30(7):1083–1138. ISSN: 0005–1098
19. Deaconescu A, Deaconescu T (2020) Tribological behavior of hydraulic cylinder coaxial sealing systems made from PTFE and PTFE compounds. Polymers 12(1). ISSN: 2073–4360
20. Márton L, Lantos B (2006) Identification and model-based compensation of striebeck friction. Acta Polytech Hung 3(3):45–58. ISSN: 1785–8860