Cylinder Block Deformation as Indicator for Cylinder Pressure in Axial Piston Units: A Novel Method Using Strain Gauges

Karl Stübig, Luca Löding, Frank Geuer, Lars Leonhard,
Dr.-Ing. Matthias Liermann, Sumeru Srinivasamurthy, Carsten Fiebing,
Dr.-Ing. Robert Rahmfeld

Danfoss Power Solutions GmbH & Co. OHG, Krokamp 35, 24539 Neumünster
mliermann@danfoss.com

Abstract

Cylinder bore pressure prediction is the foundation for designing timing notches in valve plates of axial piston units. The timing notches in a valve plate affect not only efficiency of the axial piston unit, but also its controllability and NVH characteristics. It is state of the art to use cylinder bore pressure measurement for verification of simulation results. With a telemetry system, the bore pressure data can be retrieved from the rotating unit to a stationary data acquisition system. This data can be used for comparison with simulation results, to support the optimization of different timing designs of the valve plate and operating conditions.

The feasibility of integrating a pressure sensor into the rotating kit of a production unit depends on space constraints between housing and kit and is further complicated by routing of wires through the shaft to the telemetry unit. As the units become smaller, it is increasingly difficult, if not impossible, to fit the pressure sensor and cables securely.

This paper investigates a novel approach to use a strain gauge applied to the outside of the cylinder block as an indication of the pressure inside the cylinder block. This approach addresses a major obstacle to the experimental investigation of smaller size displacement units, since the strain gauge is mounted flat on the surface of the block in an area, where most deformation is to be expected.

Measurement results show the correlation of the strain gauge signal in comparison with a conventional pressure measurement. The signal is affected by additional parameters, such as shaft speed and angle, line delta pressure, and temperature.

Keywords. Telemetry, simulation validation, data acquisition

1. Introduction

Hydrostatic pumps and motors convert mechanical energy into hydraulic energy and vice versa. In a swash plate type axial piston pump, see schematic in Figure 1, the cylinder block is driven by the drive shaft. The pistons located in the cylinder bores perform a translational movement when sliding with their slippers on the inclined swash plate. By changing the displacement angle β, the unit displacement volume can be adjusted. In pumping mode, during half of one revolution, the pistons move from inner dead point (IDP) to outer dead point (ODP), while fluid is drawn into the displacement chamber from the low-pressure port. Moving in the opposite direction from ODP to IDP during the second half of the revolution, the pistons push out the fluid from the displacement chamber into the high-pressure port. The unit used during this investigation has an odd number of pistons ($k = 9$), resulting in a periodic change of either four- or five-cylinder bores connected to high pressure.

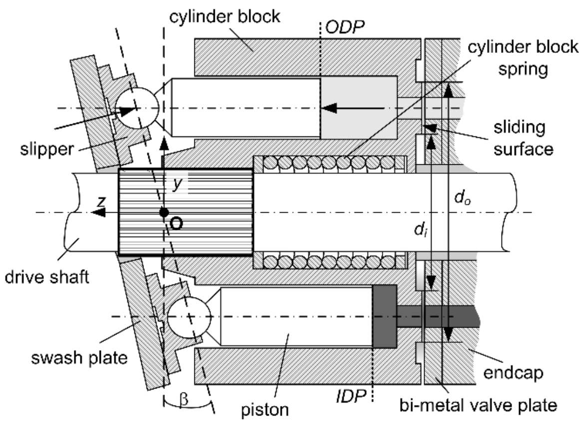

Figure 1 Principle of a swash plate type axial piston hydrostatic pump

The transient pressure in the cylinder bores of axial piston pumps is the result of several flows, see Figure 2. They are defined by the kinematics of the moving mechanical parts which drive the compression and relaxation of the fluid and cause the opening of flow passages between different fluid cavities within the pump and the connected hoses. The most important flow passages defined by the moving mechanical parts are the overlap between cylinder block kidneys and the valve plate [1]. Minor flow passages that influence the cylinder pressure exist in form of leakage paths across valve plate sealing lands, piston and bore bushing, and through the hydrostatically balanced slippers.

The valve plate is the crucial component connecting the rotating cylinder block with the stationary pump endcap. It determines the timing and shape of the pressure transition. A steep transition without cross-port leakage is efficient. However, with some degree of cross-port leakage permitted, the transition is smoother and leads to a quieter pump. The swash plate moment is directly related to the cylinder pressures. The swash moment $M_{S,x}$ around the x-axis can be written as

$$M_{S,x} = \frac{R}{cos^2\beta}\sum_{i}^{9} F_{piston,i} \cos\varphi_i \tag{2.1}$$

where R is the pitch radius, β the swash plate angle, $F_{piston,i}$ the axial force exerted by the i-th piston, and φ the shaft rotational angle [2]. The axial piston force $F_{piston,i}$ is primarily dependent on the cylinder pressure. Secondarily it is composed of inertia and friction forces.

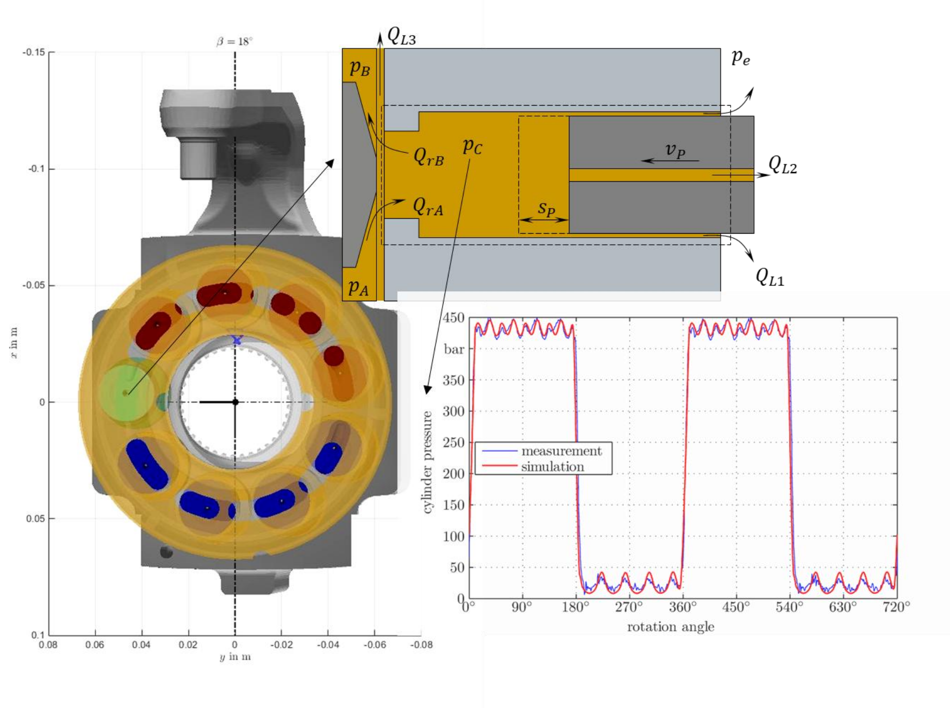

Figure 2 Transient cylinder pressure and control volume with in- and outbound flows

Lumped-parameter system simulation can be used to predict the transient pressure signal in axial piston units. It is an important tool for the design engineer to optimize the valve plate area characteristic [3] [4]. The simulation gives insight not only into the transition of cylinder pressures, but also into flow ripples, volumetric efficiency and swash plate moments. This type of simulation is state-of-the-art and basic equations are common knowledge. However, there are frequently cases in practice where model predictions do not align with measured performance. To find the root cause for this mismatch, experiments are needed that give insight into the transient nature of the cylinder pressure. This paper presents a novel experimental method of using a strain gauge to measure the strain caused by deformation of the cylinder block to get a high-frequency signal that correlates with cylinder pressure. The main advantages of this method are the reduced cost of the sensor, speed and ease of sensor application and the reduced required installation space. With this method, cylinder pressure measurement in small displacement pumps becomes feasible where it was not possible before.

1.1. Pressure measurement of cylinder bore pressure in axial piston units

The study of hydraulic-mechanical performance and the transient nature of the cylinder pressure of axial piston units have a long history [5]. The cylinder pressure is of special interest because it determines directly or indirectly several performance characteristics.

Kleist [6] and Ivantysynova [7] used specially designed single-piston test setups to investigate temperature and pressure distributions in the gap between piston and cylinder. These setups used inverted kinematics so that sensors did not have to be part of a rotating unit. Inverted kinematics means that the kinematic chain of the axial piston unit is reversed compared to conventional operation. In this configuration, the rotating group is held stationary while the surrounding components reproduce the relative motion. This allows all sensors to remain in a stationary reference frame, eliminating the need for integration into rotating components and simplifying measurement setup and signal acquisition.

But cylinder pressures in rotating cylinder blocks have also been measured, for example in Olems [8] and Helgestad et al. [9], who modified a commercial pump by fitting a miniature piezoelectric pressure sensor into the cylinder block. Ivantysynova et al. [10] designed a special tribo-pump with the ability to measure cylinder pressure and piston friction force in the rotating cylinder block. Challenges of fitting miniature pressure sensors into the cylinder block are the limited space in the cylinder block to fit sensor and cables. Another challenge is that the bore hole that connects the pressure sensor with the cylinder creates additional dead volume of the piston or changes the flow pattern and therefore influences the measurement. Bergemann [11] addressed this challenge with a custom-applied manganin wire inside the cylinder bore. Manganin is a trademarked name for an alloy that changes its electrical resistance with temperature as well as with hydrostatic pressure. With a second measurement coil it is possible to compensate for the temperature dependency and with appropriate amplification to produce a high-responsive pressure signal.

Experimental work is still important today alongside simulation efforts. More recently, Ivantysyn et al. [12] [13] developed a complex measurement setup which included 24 temperature sensors at 20 distinct positions and a pressure sensor in one of the chambers of a rotating cylinder block to confirm a temperature/efficiency relationship predicted by simulation. They concluded that while simulation results were largely confirmed in target operating conditions, new insights were gained in border-line conditions, which are hard to predict. Wieczorek et al. [14] similarly conclude that simulation tools cannot replace experiments but are another approach to support development. The literature survey shows that the experimental effort to measure physical quantities in the rotating cylinder block is considerable, to the extent that only in two published studies [8] [9] sensors could be placed into series hardware; however, even in this case, significant modifications to the hardware were required to accommodate sensors, connectors, cables, amplifiers, and the telemetry unit.

Pressure sensors have limitations regarding installation space, cost, and durability in harsh operating conditions. This study explores an alternative approach to measure cylinder pressure by a surface-attached device. We propose to use a strain gauge to sense the surface tension caused by deformation of the cylinder block and to derive the cylinder pressure by correlation. The main objective of this work is to develop and validate a measurement setup capable of tracking pressure changes based on cylinder block deformation and to identify the key challenges and limitations of this method. Experimental results demonstrate that the method reliably captures pressure trends and dynamic effects, making it a promising tool for both research and industrial testing, particularly in small-displacement axial piston units.

2. Methodology

The proposition of this paper is that the deformation of the cylinder block can be used as indication of the pressure inside the cylinder bore. Figure 3 shows a FEM analysis of the deformation of a cylinder block of a Danfoss H1P 280 pump with 420 bar pressure applied in static condition. The pressure load is applied uniformly across the cylinder surface from the dead volume region near the block kidneys up to the piston in its current stroking position. The number of high-pressure bores corresponds to the instantaneous porting condition of the machine. In the current condition 4 cylinders are connected to high pressure while 5 cylinders are connected to low pressure. The deformation leads to surface strain which can be measured with strain gauges applied close to the bottom of a particular cylinder. But other parameters may have an influence on the measurement as well, such as rotational speed and angle, line delta pressure, and temperature. For example, the lateral force of the piston that is supported in the cylinder bore adds to the block deformation. It is therefore expected that the strain gauge signal also depends on the block rotational angle. The influences of these parameters is shown in chapter 3 and discussed in chapter 4.

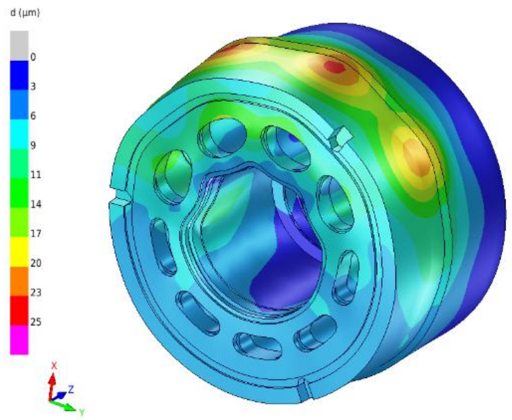

Figure 3 H1P280 cylinder block deformation at $p_{HP} = 420$ bar and $p_{LP} = 20$ bar

2.1. Measurement program

Goal of the measurement program is to verify the proposition of this paper, that a strain gauge applied to the lateral surface of the cylinder block can deliver a signal that corresponds to the cylinder pressure. To do this, it is necessary to apply a pressure sensor to the cylinder chamber for verification and test several operating conditions. Table 1 lists the conducted experiments to investigate the influence of speed, pressure, displacement and temperature on the strain gauge signal.

Table 1: Test plan

Test run	Purpose	Temp [°C]	Speed [RPM]	Δp [bar]	swash plate angle [%]	Figure
1	Proof of function, pressure steps	34-54	<0.1	0-400, steps	0%	6
2	Speed ramp I	53-58	200-1500	100	39%	7
3	Speed ramp II	49-54	200-1500	0	0%	8
4	Pressure ramp	59-60	1500	0-450, ramp	42%	9
5	Influence of swash plate angle I	36-42	1	200	0%	10
6	Influence of swash plate angle II	31-35	1	200	100%	10
7	Influence of temperature	25-90	1000	250	50%	11

2.2. Test setup

A pump with large displacement (H1P 280) is selected as test unit to give sufficient space for sensors and cables, see Figure 4. Pressure, strain gauge and temperature signals are transferred from the rotating cylinder block to a stationary data acquisition system via a telemetry unit. An IMC MTP-NT telemetry unit is used for this purpose. Several components of the pump had to be reworked, and some new parts had to be designed and added to create this measurement setup. A type J thermocouple and Mesurex PM6 Series pressure sensor were mounted into threaded bore holes in such a way as to get as close to the bottom of the cylinder as possible to avoid influence on the chamber dead volume. Two HBK VA71K1.6/350_E strain gauges were applied onto two adjacent cylinder chambers. Two strain gauges were applied to have signal redundancy in case one is damaged and to investigate whether the pressure rise in one cylinder may affect the signal of the adjacent cylinder strain gauge. All cables are glued into a groove that was manufactured into the cylinder block for protection from mechanical damage caused by slushing in the oil-filled case as the cylinder block rotates. The cables are then guided towards the shaft and routed through a hole in the drive shaft, across the shaft coupling which connects the charge pumps and further on through the custom-made telemetry shaft into the telemetry unit.

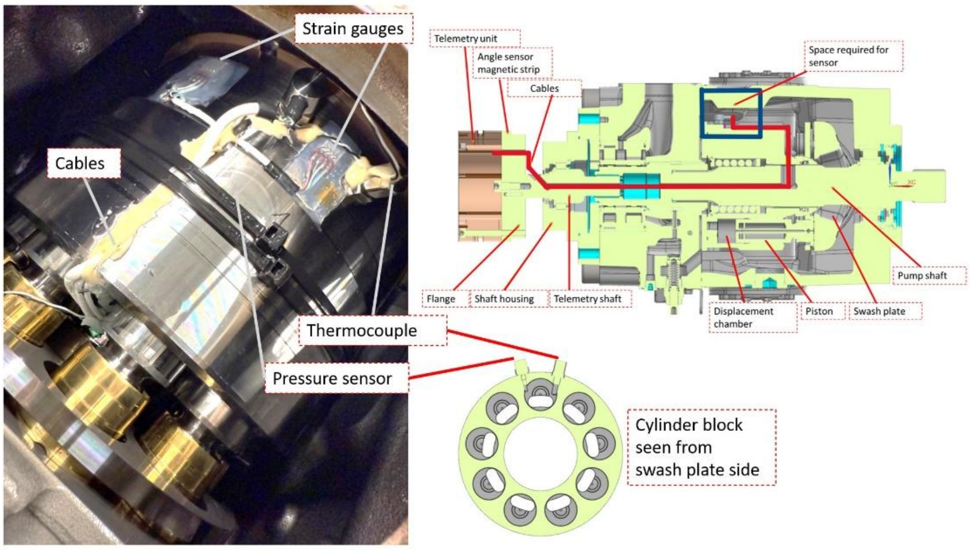

Figure 4 Sensor integration in H1P 280. Two strain gauges, a pressure sensor, and a thermocouple are fitted onto the cylinder block with cables that are guided through a customized hollow shaft to a telemetry unit.

Figure 5 shows the assembled pump with telemetry unit on the test bench. The telemetry unit houses four integrated signal amplifiers and A-D converters for the two strain gauges, the pressure sensor and thermocouple. Their digital signals are transmitted as pulse code modulated bit stream signals via inductive transmitter to the pick-up head. Power to the sensor modules is provided via induction from the power head. A rotational angle read-head that reads the pump rotational angle from a magnetic scale is mounted to the top of the telemetry shaft housing. Pressure and temperature sensors are connected to hydraulic return and feed line ports.

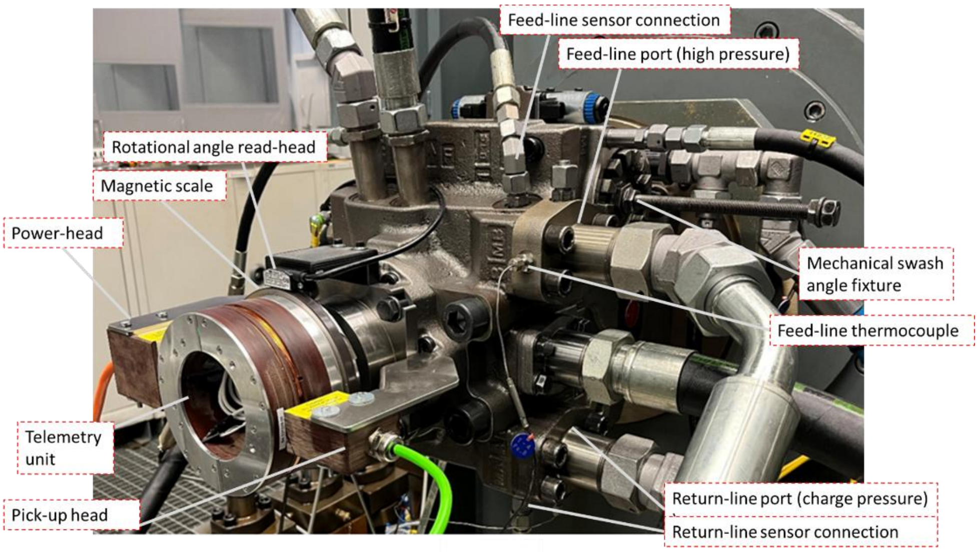

Figure 5 Pump under test with telemetry unit, rotational angle sensor and further sensors connected to the pump port connections

3. Results

The following test results show to which extent the strain gauge signal corresponds to the cylinder pressure. The main take-away is that the strain gauge is primarily sensitive to cylinder pressure but picks up other influences as well, which will be hard to compensate for in practice. Some potential causes and what this sensitivity to secondary effects means for the applicability of this measurement principle will be discussed in chapter 4.

3.1. Proof of function

Figure 6 shows how cylinder pressure measurement and strain gauge signal correlate after an initial static calibration procedure. In the test, the shaft angle is held fixed, and the pump is pressurized externally in steps from 50 to 400 bar. The tank oil temperature is controlled but the cylinder temperature is not kept at a constant level because of a lack of fluid exchange in the cylinder. It varies between 34 and 54 °C.

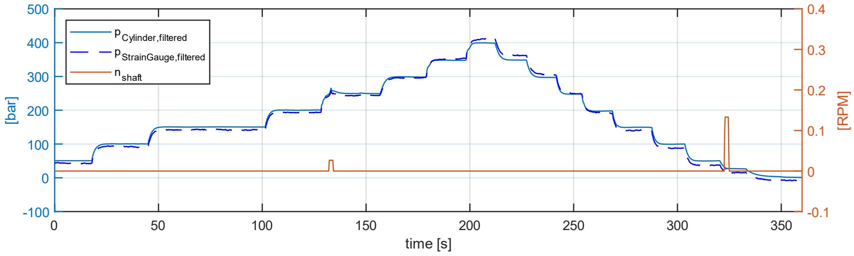

Figure 6 Comparison of cylinder and strain gauge pressure signals. Pump shaft is held fixed; pressure is applied externally in steps; temperature varies between 34-54 °C.

The correlation between pressure and strain gauge signal is qualitatively given. There is a quantitative difference in the strain gauge signal compared to the pressure sensor signal as the pressure steps up and down. For example, between $t = 190$ s and $t = 220$ s the pressure sensor signal differs only 1 bar, whereas the strain gauge signal differs more than 12 bar.

An unexpected artefact in the pressure signal is observed at $t = 133$ s. Both, pressure sensor and strain gauge signal spike briefly and then abate to their previous level. The explanation for this can be given as follows: The applied pressure impacts the load on the servo system. Due to swash plate moments, the swash plate angle experiences a sudden change. The pump displacement causes torque to be applied by the pump and a slight motion of the shaft against the fixture. The motion also impacts the cylinder pressure. This is picked up by the pressure sensor and the strain gauge alike.

3.2. Influence of shaft speed, load, and displacement angle

Figure 7 and Figure 8 show the cylinder pressure and strain gauge signals, each over one rotation at different speeds. In Figure 7 the pump is set to 40% displacement and against 100 bar line delta pressure. Figure 8 shows results for zero displacement and no-load condition.

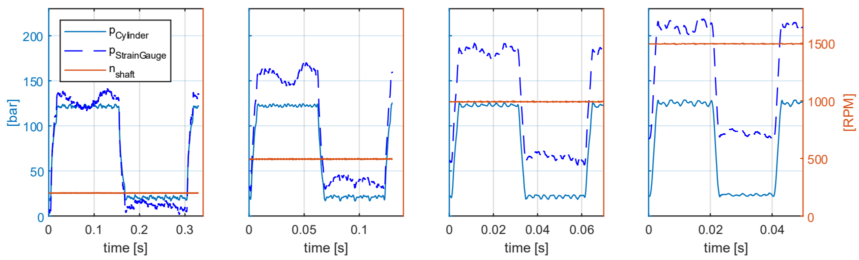

Figure 7 Comparison of cylinder pressure and strain gauge signal at
$n = [200,500,1000,1500]$ RPM; Line delta pressure 100 bar, 7° displacement, 55 °C

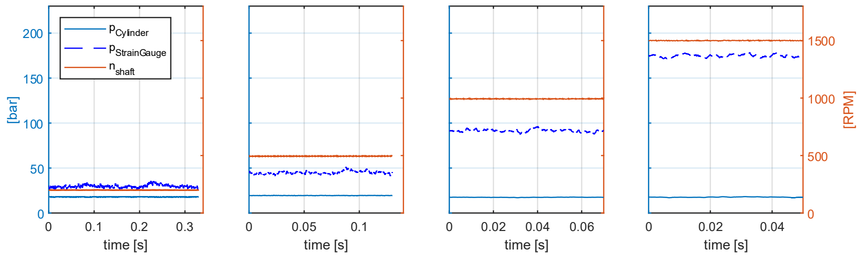

Figure 8 Comparison of cylinder pressure and strain gauge signal at
$n = [200,500,1000,1500]$ RPM; Line delta pressure 0 bar, 0° displacement, 52 °C

The dynamic response is excellent, but quantitatively the signal is hard to interpret. Evidently, shaft speed strongly influences the strain gauge signal. To a lesser extent, line delta pressure and displacement impact the signal. Where the pressure signal reports a plateau between the high- and low-pressure transitions, the strain gauge signal bends inward. It seems that the signal is shaft angle dependent.

In Figure 9 the effect of different levels of line pressure is shown for fixed speed. The strain gauge high- and low-pressure plateaus spreads outward as the line delta pressure increases.

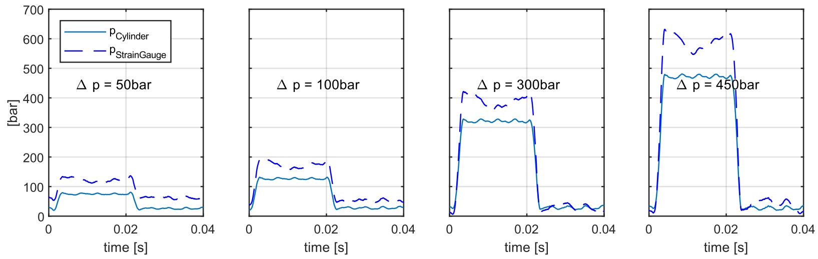

Figure 9 Comparison of cylinder pressure and strain gauge signal at fixed speed 1500 RPM and different line delta pressure $\Delta p = [50,100,300,450]$ bar, 7.5° displacement

Figure 10 shows the influence of shaft angle and displacement. It shows the result of two tests in one diagram. In both tests the shaft is driven precisely with 1 RPM and the line delta pressure was held constant at 200 bar. The comparison shows the strain gauge signals over one rotation for both cases, as the displacement is changed between 0% and 100%. The influence of pump displacement angle is evident, but comparatively minor. The time for one rotation is 60 seconds, therefore the periodic patterns in the high-pressure plateau are no measurement artefacts but clearly show the effect of adjacent pistons entering and exiting the high-pressure kidney.

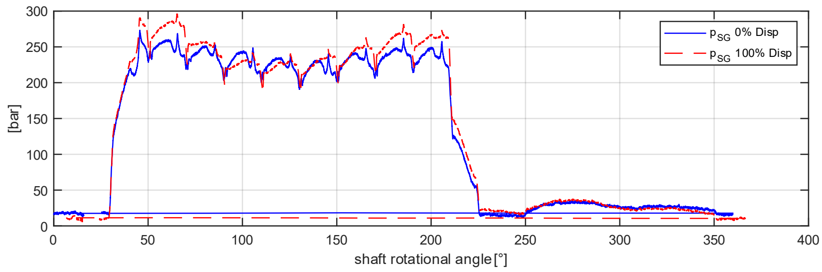

Figure 10 Influence of swash plate angle on strain gauge signal; strain gauge signal plotted over shaft rotational angle for 0 and 100% displacement at 1 RPM and 200 bar line delta pressure.

3.3. Influence of temperature

To investigate the influence of temperature alone, the pump is operated at constant speed against a fixed load. The cooling system is switched off, and the circuit allowed to heat up. At $t = 2271$ s the cooling system is activated again and the temperature decreased. The strain gauge and cylinder pressure signals are shown by their envelopes because they oscillate with high frequency between low and high pressure. Also plotted are the line pressures p_B, p_A at the inlet and outlet of the pump. They are constant throughout the test, whereas both, the cylinder sensor and strain gauge signal vary over test duration. A correlation with temperature cannot be explained easily for both, the cylinder pressure sensor and the strain gauge.

The pressure sensor signal should be compared to the inlet temperature because the cylinder is cooled by the incoming flow. The pressure signal drops at the start of the test and rises later. But it stays constant during much of the test, where the inlet temperature clearly changes. The strain gauge signal should be compared to the case temperature because it is mounted to the surface of the cylinder and in contact with case oil. However, it does not correlate in trend with the case temperature at all. It rises in the beginning, drops during the middle but starts increasing again before the cooling is switched on. At the moment of cooling down the oil, the strain gauge signal remains roughly on the same level.

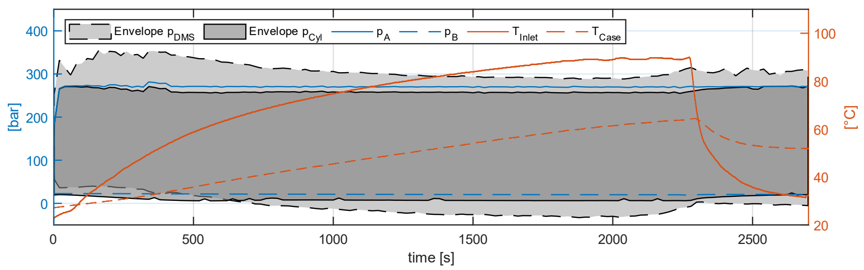

Figure 11 Influence of temperature on signals from cylinder pressure sensor and strain gauge. Speed is constant at 1000 RPM; line delta pressure is constant at 250 bar; pump displacement is 50%.

4. Discussion

The measurement results show high correlation between strain gauge and pressure signal with no noticeable delay. However, the signal is influenced strongly by secondary effects summarized in Table 2.

Table 2: Influence factors on strain gauge signal

Effect	Influence	Quantification
Shaft speed	Strongly positive	0.04 - 0.16 bar/rpm
Line delta pressure	Strong, different for high and low pressures	-0.2 - 1.5 bar/bar
Displacement	Moderate, different for high and low pressures	-0.07 - 0.2 bar/percent
Shaft angle	Pressure dips inward between transitions	10-30%
Temperature	Strong, incongruent	

No simulation was done to date that could confirm the root cause for each of the influence factors. At this point we can only hypothesize.

With regards to the influence of shaft speed, see Figure 7 and Figure 8, it is quite hard to imagine that centrifugal forces on the pistons cause deformation of the cylinder block bore that is picked up by the strain gauge. This is still a matter of investigation.

The influence of line delta pressure on the strain gauge signal is not easily explained. Figure 9 shows that the signal during the high-pressure phase is affected differently than during the low-pressure phase. By offsetting the strain gauge signal by -30 bar and multiplication with a gain of 0.8 it is possible to improve, but not to eliminate this effect, as Figure 12 shows. This gain and offset would be dependent on other parameters such as speed and temperature as can be easily shown.

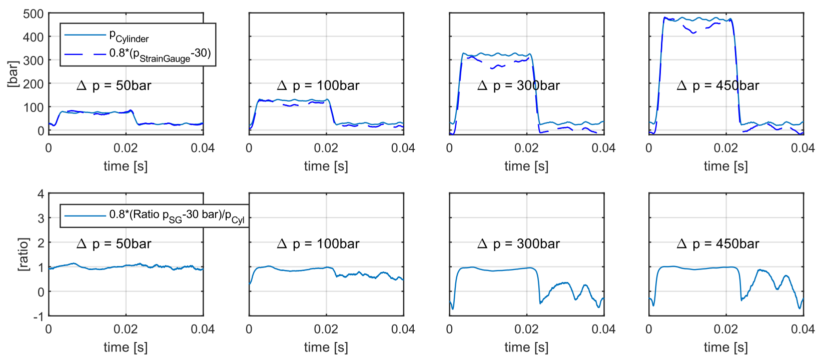

Figure 12 Results in Figure 9 with alternative scaling of strain gauge signal.
Scaling $y = 0.8(x - 30)$ shows improved matching.
Top row: absolute comparison; Bottom row: relative comparison

An interesting observation is that the strain gauge signal bends towards lower values during the high-pressure phase as the piston moves from outer dead center to inner dead center. The underlying cause of this behavior, however, is not yet fully understood.

The temperature influence is apparent, see Figure 11. However, a mapping to any of the temperature readings could not be found. This indicates that the effects of several parameters are not independent of each other.

The study shows that the strain gauge signal cannot be simply used as a pressure measurement, but certainly as an indication. It is reasonable to expect that some level of compensation for speed and pressure can be achieved through state-dependent scaling and offsets. But this scaling would not cover all influences, such as temperature. For speed and pressure, it would likely be product-specific and therefore not very useful.

Is it, after all, of any value to use the strain gauge signal as indication of the cylinder pressure? It was noted in chapter 3.2 that the high-frequency content of the strain gauge signal follows the actual pressure very closely. To illustrate this, Figure 13 shows the measurement of Figure 7 with a high-pass filter applied. It removes the low-frequency biases below 20 Hz. The bottom row plots of Figure 13 show the difference between strain gauge and pressure sensor signals. During the high to low and the low to high pressure transitions, the signal amplitude is high, while the error low. In other words, the strain gauge signal gives a good indication about what happens dynamically during the transitions. The high frequency content of the strain gauge signal by itself shows good correlation.

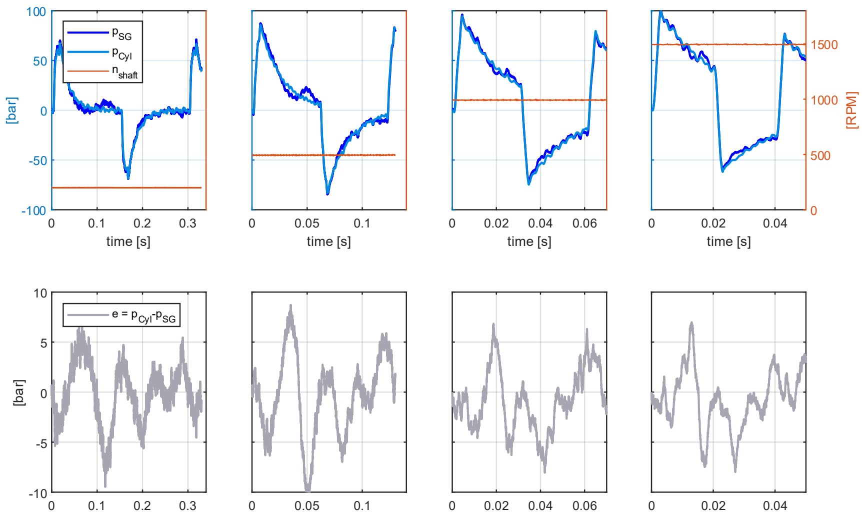

Figure 13 Comparison of cylinder pressure and strain gauge signal after high-pass filtering. Top: High-pass filtered data from Figure 7. Bottom: Difference between both signals. Error small compared to signal magnitude during transitions.

5. Conclusion and Outlook

The measurement of physical signals from rotating parts requires some form of telemetry unit. To measure the cylinder pressure in an axial piston pump, miniature sensors, delicate wiring, and connecting in very narrow spaces are required. This study shows that a strain gauge mounted to the outside of a cylinder block is sensitive to pressure changes, but also to shaft speed, shaft angle, line delta pressure, displacement and temperature. We conclude that it cannot be used as a replacement for a pressure sensor in a general sense. However, the high-frequency content of the strain gauge signal corresponds very well with the cylinder pressure and may be used in cases, where the focus of study is on the fast high to low and low to high pressure transition. Also, for verification of simulation models this type of measurement is useful because it is the transient response between low and high pressure in the cylinder bore that needs validation, not the plateau phases.

6. References

[1] R. Kornicki, R. Rahmfeld, S. Srinivasamurthy, Q. Hongbo and L. Minghui, "On-Board Cylinder Bore Pressure Measurement of Axial-Piston Hydrostatic Machines," in *International Conference on Fluid Power Transmission and Control (ICFP2017)*, Hanzhou, China, 2017.

[2] J. Ivantysyn and M. Ivantysynova, Hydrostatic pumps and motors : principles, design, performance, modelling, analysis, control and testing, New Delhi: Tech Books International, 2003, 2003.

[3] J. Tvarůžek, R. Rahmfeld, C. Fiebing, A. Krahn and W. Göllner, "Dynamic Valve Plate Design for an Axial Piston Pump (Servo-Less Pump)," in *14th International Fluid Power Conference*, Dresden, 2024.

[4] N. J. Fulbright, S. Hall, M. Creswick, S. Smolka and R. Rahmfeld, "Efficiency Improvements in a Servoless Swash Plate Pump: Experimental and Simulation Results," in *2024 International Maha Fluid Power Conference*, Purdue, 2024.

[5] K.-H. Titscher, "Untersuchung von Einflüssen auf den Druckverlauf in einem Zylinder einer Hochdruck-Hydraulik-Axialkolbenpumpe," RWTH Aachen University Disseration, Aachen, 1970.

[6] A. Kleist, "Berechnung von hydrostatischen Dichtstellen in hydraulischen Maschinen," *Ölhydraulik und Pneumatik,* vol. 39, pp. 767-771, 1995.

[7] M. Ivantysynova, "Temperaturfeld im Schmierspalt zwischen Kolben und Zylinder einer Axialkolbenmaschine," *Maschinenbautechnik,* vol. 34, pp. 532-535, 1985.

[8] L. Olems, "Investigations of the Temperature Behavior of the Piston Cylinder Assembly in axial Piston Pumps," *International Journal of Fluid Power,* vol. 1, no. 1, pp. 27-38, March 2000.

[9] B. O. Helgestad, K. Fostesr and F. Bannister, "Pressure Transients in an Axial Piston Hydraulic Pump," *Proceedings of the Institution of Mechanical Engineers,* vol. 188, no. 17/74, pp. 189-200, 1974.

[10] M. Ivantysynova and R. Lasaar, "An Investigation into Micro- and Macrogeometric Design of Piston/Cylinder Assembly of Swash Plate Machines," *International Journal of Fluid Power,* vol. 5, no. 1, pp. 23-36, 2004.

[11] M. Bergemann, Systematische Untersuchung des Geräuschverhaltens von Kolbenpumpen mit ungerader und mit gerader Kolbenzahl, Aachen: Mainz, 1994.

[12] R. Ivantysyn and J. Weber, "Advancing Thermal Monitoring in Axial Piston Pumps: Simulation, Measurement, and Boundary Condition Analysis for Efficiency Enhancement," *International Journal of Fluid Power,* vol. 25, no. 4, pp. 547-590, 2024.

[13] R. Ivantysyn and J. Weber, "Efficiency Forecasting in Axial Piston Pumps: Leveraging High Speed Fluid Temperature Correlations for Condition Monitoring," in *The 12th International Symposium on Fluid Power*, Hiroshima, Japan, 2024.

[14] U. Wieczorek and M. Ivantysynova, "Computer Aided Optimization of Bearing and Sealing Gaps in Hydrostatic Machines - the Simulation Tool Caspar," *International Journal of Fluid Power (IJFP),* vol. 3, no. 1, pp. 7-20, 2002.