Influence of normal force in metallic sealing

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Abstract
Metallic ball seat valves are an essential component in almost every hydraulic system in the form of check valves. Despite their wide usage in real-life applications, the physical sealing mechanism is not yet fully understood and still debated. One promising method is the contact mechanics theory which has been developed by Persson et al. for rubber seals. It can predict the leakage of rubber seals successfully, but its application for metallic seals, where plastic deformation plays an important role, has only been developed recently and is not yet fully validated. This theory is combined with surface scans of investigated seats and simulations that are performed using the finite element method (FEM) for different geometric configurations. This combination allows an efficient calculation of the leakage of ball seat valves depending on their specific design and applied forces and pressures. In this work, the predictions by simulations according to Persson’s theory are compared with experimental results. This comparison is done by analyzing the dependency of the leakage of ball seat valves in relation to applied normal forces and for different surfaces in contact. This validates the applicability of Persson’s method in this field. The presented experimental method allows the measurement of the leakage of ball seat valves under realistic conditions.

KEYWORDS
asperities, ball seat valve, leakage, sealing, surface

1 | INTRODUCTION

Ball seat valves are used as check valves in hydraulic and pneumatic systems. Their function is to prevent the flow of fluid into one direction. If the fluid is about to flow in the reverse direction, the fluid pressure will press the ball into the seat and tighten the seal. Often they are installed in reverse direction as pressure relief valves. In this case, there is a spring that applies a defined normal force onto the ball, which prevents leakage up to a specific fluid pressure.

The exact sealing mechanism is not yet well understood. If there was a good method to simulate metallic seals, one could use that method to optimize ball seat valves to be both robust and tight while keeping the costs low. A challenge in the treatment of metallic sealing is the effect of plastic deformation, which is not as important in the case of rubber seals.

Abbreviations: FEM, finite element method; HV, Vickers hardness.
Another difference to rubber seals is the larger elastic modulus $E$ of metals. This results in fewer elastic deformations and elevates the importance of long surface wave-lengths.

The sealing of a ball seat valve is determined by a large range of different factors. For example, there is the influence of the applied forces and pressures on the system. The contact pressure distribution and the contact area are determined by the elastic and plastic properties of the materials in contact as well as the valve geometry. In general it is challenging to measure the contact pressure distribution or the contact area experimentally. That is why they are usually treated with finite element based simulations, in order get the contact properties. For some geometries it can be possible to calculate the contact pressure and area analytically. The leakage is also influenced by the viscosity of the fluid, which should be sealed by the valve. These quantities are the macroscopic quantities.

On the other hand, the leakage is also influenced by the topography of the surfaces which are in contact. The materials' surfaces can be measured by a microscope. Their properties are the microscopic quantities that influence the sealing.

The purpose of this article is to test Persson’s theory for its applicability to metallic sealing. This theory is based on the surface roughness power spectrum, that can be calculated based on measured surface data. In fact, the power spectrum is the only surface characteristic that is needed for every calculation in Persson’s method. It does not make any assumptions on the distribution of surface asperities as in the case of the Greenwood-Williamson method. In the simulation method presented in this work, only two arbitrary and non-physical assumptions are needed: a mapping from a curved contact pressure distribution to a flat one and the assumption of ideal plastic behavior. Therefore, the method is consistent and computationally efficient, but it is not widely used and not yet fully experimentally validated under realistic conditions. This method has been successfully validated for some degree for rubber seals. In the case of metallic seals, there are only few comparisons between his theories predictions and experiments.

In this work, the leakage is calculated using the following method: first the contact area and contact pressure are determined by a FEM calculation. After that, the leakage is estimated using surface scans performed by an optical microscope. These results are then compared with experimental measurements using a test-rig that has been specially built for that purpose.

Similar methods, which are based on Persson’s theory, have been shown to be successful in case of hard-hard contact of mechanical seal interfaces.
In other works, the influence of the fluid pressure onto the leakage has been measured and calculated. While the application of only fluid pressure is close to the industrial application, this method measures the result of two different effects. The fluid pressure increases the pressure alongside the contact and thus increases the leakage. On the other hand, the fluid pressure also pushes the ball into the seat and decreases the leakage. The combination of both effects can lead to an increase or decrease of leakage with increasing fluid pressure depending on the surfaces and the areas of pressure. In this work, the normal force is increased independently of the fluid pressure, which is close to their application as pressure relief valves. By this, the effect can also be studied independently.

In the experiment that is presented in this work, the liquid leakage of a valve is measured, which is closely orientated on valves that are used in the industry. In other works, metallic seals have been studied under conditions and with geometries that differ greatly from industrial reality and under conditions which are not represented by the theory. In this work, the leakage of a liquid is measured rather than the leakage of a gas.

The approach in this work can be applied to other geometries and types of surfaces as well. The main limitation of this method is that it is only applicable in the elastic limit. For very high contact pressures, one of the surfaces will experience macroscopic plastic deformation. In this case, Persson’s theory does not expect any plastic deformation at all. Another problem with all sealing simulations in general is that they do not account for the influence of particles inside the fluid, which increase the leak tightness and introduce a time dependency to the leak-rate. The particles’ effects will be briefly discussed in this work.

This article is structured as follows: in Section 2, the simulational method is presented and Section 3 presents the experimental set-up and routine. In Section 4, the results of both methods are presented and compared. Afterward, there is a small outlook and conclusion.

2 | SIMULATION

The calculation of the leakage of metallic seals is separated into two different steps: a macroscopic and a microscopic calculation. In the macroscopic calculation, the macroscopic contact properties like the contact pressure distribution are calculated based on geometry and the outer influences of the system. The microscopic calculation is based on the surface properties of the materials at contact.

Although both steps can be performed mostly independently, their results have still some influences onto each other. A scheme of the simulation process is depicted in Figure 1.

![Flowchart of the whole simulation process](image_url)

**FIGURE 1** Flowchart of the whole simulation process
2.1 Macroscopic calculation

A FEM model that resembles the valve used in the experiment was used. Only the most basic parts of a ball seat valve are considered in this work. The valve consists only of a ball and a seat. They are both made of stainless steel, see Figure 2.

The exact geometric details can be seen in Table 1. In Figure 2, $R_C$ denotes the radius of the circle that is formed by the contact area. It can be found via $R_C = \sin(\varphi)R_{\text{ball}}$. The plastic properties are related to the materials' hardness. Vickers hardness of both the ball and the seat have been measured by performing micro hardness experiments with a normal load of 0.2 kp. The hardness of the seat is found to be 609 HV 0.2 and the ball 718 HV 0.2. The Vickers hardness can be used to estimate the yield stress $\sigma_Y$. They are expected to be approximately 2.0 GPa for the seat and 2.3 GPa for the ball. The large dimensions of the ball and the seat have been chosen to make the leakage more easily observable in the experiments. This is also the reason for the chosen fluid pressure, which is low compared to industrial applications.

The ball is pressed into the seat by the pressure of the fluid. The force acting onto the ball is the same as the force that can be obtained as the product of the fluid pressure and the area of the circle that is formed by the contact area with the radius $R_C$, see Figure 2. Additionally, a varying normal force is applied on top of the ball. The aim is to calculate the leakage in dependence of the total normal force.

An axisymmetric model of the valve is used in the simulation. In the contact area, a very fine mesh was chosen to correctly represent the contact properties and to ensure smooth changes between slightly different forces. A series of FEM calculations with different normal forces yield the resulting contact pressure distributions and contact areas.

The range of forces in this calculation is chosen not to lead to contact pressures that surpass one of the yield strengths. In this case, no macroscopic plastic flow will occur. One expects that both the local contact area and the maximal contact pressure increase with rising normal force.

The contact stiffness determines how the contact pressure relates to the distance of the planes in the simulation. To include the effects of surface roughness, no hard contact model can be used, where no penetration is allowed and pressures only occur at direct contact where every contact pressure is possible; instead a soft contact model is used, where the contact pressure increases with decreasing distance of planes. In this case, the contact pressure can be described as an injective function of the distance of planes. The FEM simulation uses a soft contact relation based on Persson’s model. This contact stiffness depends on the measured surface roughness power spectrum. The relation is presented later in the following section in Equation (15). With a softened contact calculation, the contact area increases and the maximal contact pressure decreases compared to a hard contact calculation.\textsuperscript{13}

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**FIGURE 2** Sketch of the ball seat valve

| Property               | Value    |
|------------------------|----------|
| radius of ball $R_{\text{ball}}$ | 20.0 mm  |
| inner seat radius $R_{\text{seat}}$ a | 7.5 mm   |
| angle of slope $\varphi$ a       | 45°      |
| fluid pressure          | 1 MPa    |
| Young's module $E$       | 209 GPa  |
| Poisson’s ratio $\nu$    | 0.3      |

\textsuperscript{a} See Figure 2.

**TABLE 1** Geometric properties of the valve in usage
The FEM calculations have been performed using the commercial software Abaqus. A 2D model with a triangular mesh is used. Second order elements are used in order to account for the roundness of the ball and the valve realistically. The element type CAX8R has been used. The mesh has been generated procedurally. The density of nodes has been selected so high, that there are no heavy fluctuations at the contact area. An example for a reduced mesh of the model can be seen in Figure 3.

The microscopic description according to Persson’s model assumes a box shaped contact pressure distribution. It has a constant contact pressure for every point of contact and is 0 at all other points, see Section 2.2. For this reason, the distribution needs to be approximated by another shape based on the results. In this work, a Gaussian fit is applied to the pressure distribution, see Equation (1), where $\sigma$ describes the deviation, $p_c$ is the maximal contact pressure, and $\mu$ is the position of the maximum.

$$p(x) = p_c e^{-\left(\frac{x-\mu}{\sigma}\right)^2}$$

The new shape has the value of the maximal contact pressure $p_c$ and a size of $2\sigma$. An example for a contact pressure distribution can be seen in Figure 4.

In Figure 4, it can be seen that the contact pressure does very closely follow a Gaussian bell curve with a coefficient of determination of $R^2 = 1 - 10^3$. Simulations with a hard contact algorithms show a different shape with a harder transition from contact region to non-contact region. The Gaussian description of the contact pressure distribution is only so close to the simulation due to using a soft contact model. In case of hard contact models, a different method has to be chosen to

**Figure 3**  Mesh of the FEM model with a different node-density, with a lower density of nodes than used in the actual simulation. At the top of the ball, a uniform load has been applied. The bottom and the right edge of the seat are fixed by boundary conditions. The whole model is constrained by the rotational symmetry.

**Figure 4**  Simulated contact pressure distribution of seat 1 (see Section 4.1) at about 2 kN additional normal force together with a Gaussian fit of the simulated data, which has very good agreement with the simulated data. The x-axis shows the position on the contacting edge of the seat, which is measured starting at the bottom of the seat.
determine the contact’s properties for usage with Persson’s method. This fitted curve allows to choose a consistent method to determine the contact length \(A_c\) and the contact pressure \(p_c\) based on the simulated data; \(A_c\) is set to be \(2\sigma\) where \(\sigma\) is the deviation of the fitted bell curve and \(p_c\) is chosen to be the maximal value of the fit of the contact pressure distribution.

### 2.2 Microscopic calculation

The microscopic calculation is based on the contact mechanics developed by Persson. In this work, the critical junction theory is used to calculate the leakage. An alternative is the effective medium approach, which delivers similar results while being more computational demanding.

The surface of the ball is very smooth in comparison to the surface of the seats. The surface of the ball has a root mean square roughness \(R_q\) (see Equation 2) of 0.05 \(\mu m\), whereas the surfaces of the seats have an \(R_q\) between 1 and 2 \(\mu m\).

\[
R_q^2 = \frac{1}{N} \sum_{i=1}^{N} (h(x_i) - \langle h \rangle)^2
\]  

(2)

where \(N\) describes the number of measured data points \(h(x_i)\). The plastic properties of the ball do not need to be considered either, because it is made out of a harder material than the seat.

In the basis of the theory lies the surface roughness power spectrum \(C(q)\), where \(q\) is the surface wave vector. \(C(q)\) describes the Fourier transformation of the auto correlation of the surface. It is defined as followed:

\[
C(q) = \frac{1}{(2\pi)^2} \int d^2x \langle h(x)h(0) \rangle e^{-i\mathbf{q} \cdot \mathbf{x}}
\]  

(3)

In this equation, \(h(x)\) denotes the surface topography as a function of the location \(x\). In practical applications, we use the fact that \(C(q)\) is also proportional to the absolute square of the Fourier transformation of the surface, which can be effectively calculated using the fast Fourier transformation (FFT) algorithm:

\[
C(q) \propto |F[h(q)]|^2
\]  

(4)

In this work, \(C(q)\) was estimated of the surface data using Welch’s method, which separates the data into different sections and takes the average of the obtained power spectra. Welch’s method allows the use of multiple line scans to calculate a single power spectrum by averaging over power spectra that have been created using different rows of data obtained from the same probe. It is important to use a window function in each case to minimize the effects of using a finite and fixed amount of data points.

If the roughness of the surface is considered to be isotropic, then the one-dimensional surface roughness power spectrum \(C(q)\) can be used, which can be obtained using a line scan. The one-dimensional \(C(q)\) is not equal to the two-dimensional isotropic \(C_{iso}(q)\), where \(q\) describes the radial component of \(q\). This is especially useful for curved surfaces like the inner surface of the seat, which is hard to measure in two dimensions. In this work, the one dimensional \(C(q)\) is obtained using a line scan \(h(x)\) in radial direction. To test the calculated \(C(q)\) for correctness one can use a relation between \(C(q)\) and \(R_q\) and compare the result with the result of Equation (2):

\[
R_q^2 = 2 \int_0^\infty dq C(q)
\]  

(5)

Persson assumes that the actual contact area differs from the apparent contact area. The ratio of the real contact area \(A(\zeta)\) and the apparent contact area \(A_0\) can be described as a function of the magnification \(\zeta\). \(\zeta\) describes a cutoff wave-length. The surface at magnification \(\zeta\) is the version of the original surface that includes only wave vectors up to a specific chosen vector:

\[
\zeta = \frac{q_{cutoff}}{q_0}
\]  

(6)
where \( q_0 \) is the smallest considered wave number. The fraction \( P \) can be found as:

\[
P(\zeta) = \frac{A(\zeta)}{A_0} = \text{erf}(\omega(q, 1)\tilde{p})
\]

(7)

where \( \tilde{p} = \frac{p}{E^*} \) is the effective contact pressure, and \( \omega \) is defined as:

\[
\omega(q, \zeta) = \left( \int_{q_0}^{q} dq' q'^2 C(q') \right)^{1/2}
\]

(8)

The effective Young’s module \( E^* \) is defined as:

\[
E^{*-1} = \frac{1 - v_1^2}{E_1} + \frac{1 - v_2^2}{E_2}
\]

(9)

Here, \( E_1 \) and \( E_2 \) denote the materials’ Young’s modules and \( v_1 \) and \( v_2 \) are the materials’ Poisson’s ratios.

The percolation theory predicts that a channel of no-contact area will percolate through a quadratic contact area as soon as \( P(\zeta) \) reaches approximately 0.4. With this information, the critical magnification at which the first leak channel through a square occurs can be found by solving \( P(\zeta_c) = 0.4 \) for the measured \( C(q) \).

In the case of metallic sealing, \( \tilde{p} \) is very small. This means that the distance of different points of \( q \) which is equal to \( \Delta q = \frac{x}{L} \) should be small for \( P(\zeta) \) to be a smooth function of \( \tilde{p} \). This means that a rough surface scan with a large total distance \( L \) and also a large distance between points \( \Delta x \) should be used. But if only such a measurement is performed, then the highest included wave vector is \( \pi / \Delta x \) for a large \( \Delta x \) and all information about higher wave vectors is lost. Therefore in this work, two different surface scans were performed for each seat: One with a high resolution and a small distance \( L \) and one with a large resolution and a large distance \( L \). Both measurement yield different surface roughness power spectra: first a \( C(q) \) with a small distance \( 2 \pi / L \) and a small range \( \pi / \Delta x \) and also one \( C(q) \) with a large distance \( 2 \pi / L \) and a large range \( \pi / \Delta x \). The \( C(q) \) used in the calculation consists of the values of the rough surface scans for low values of \( q \) and of the high resolution surface scan for the rest of the points. Using this method, the results are both accurate and efficient. An example of how a surface roughness power spectrum can be obtained from two surface scans can be seen in Figure 5.

Based on \( C(q) \), the height separating the surfaces at the critical magnification \( u_1(\zeta) \) can be obtained using:

\[
u_1(\zeta) = \bar{u}(\zeta) + \frac{\partial \bar{u}(\zeta)}{\partial \zeta} \left( P(\zeta) \frac{\partial P(\zeta)}{\partial \zeta} \right)^{-1}
\]

(10)

where \( \bar{u}(\zeta) \) is as defined as:

\[
\bar{u}(\zeta) = \pi^{-1/2} \int_{q_0}^\infty dq C(q) \omega(q, \zeta) \times \frac{1}{P(\zeta)} \int_{\beta/P(\zeta)}^\infty dp' \left[ \gamma + 3(1 - \gamma) \text{erf}^2(\omega(q, \zeta)\tilde{p}) \right] e^{-\left[ \omega(q, \zeta)\tilde{p} \right]^2}
\]

(11)

**Figure 5** \( C(q) \) calculated by two different surface scans: one with high resolution and one with low resolution.
The next step is to assume that the leakage occurs through a tube with diameter \( \bar{a}(\zeta_c) \) and length \( A_C = 2\sigma \). If we assume a Poisson flow through that channel, the leak rate \( Q \) can be found as:

\[
Q = \Delta P \frac{u_1^3(\zeta_c)}{12\eta} - \alpha
\]

(12)

where \( \Delta P \) is the pressure drop along the seal and \( \eta \) is the viscosity of the liquid and \( \alpha \) is an arbitrary factor smaller, but close to 1. The factor \( \alpha \) is added to compensate for influences from higher magnifications than the critical magnification.

The approximation of the geometry of the channel as a tube is a simplification if the pressure distribution is not similar to a step function. Despite that, experiments have shown good agreement with the theory in case of rubber seals.\(^{14}\)

Note that the factor \( \alpha \) as well as the method chosen to determine the width of the contact are both arbitrary choices and influence the leakage linearly. In this work, \( \alpha \) is chosen to be 1.

The theory is based on the leakage of a single square of contact, because squares are the basis of the percolation theory. Therefore, the leakage of a single square needs to be upscaled to account for the leakage of the whole seal. In order to do this, the leakage of a square with a side length of the radial contact area \( 2\sigma \) is multiplied by the ratio of the length of the square to the whole circumference of the contact area \( 2R_C\pi \), see Figure 2.

The method presented above allows the calculation of the leakage if there was no plastic deformation. Plastic deformation will change the surface of the softer material, in this case the seat’s surface. It reduces the surface roughness and decreases the leakage. This fact can be used to increase seal tightness in practical applications. In a method that is called impregnation, one pushes the ball into the seat with a very high pressure. This is only done once before its application. This deforms the seat’s surface once and reduces its leakage permanently. The reduction of leakage through this method has also been shown experimentally.\(^{21}\) The plastic deformation of the surface often changes slightly the reflectivity at the contact area and this can be seen with the naked eye. The surface roughness at this contact area is slightly reduced, although this effect is only slightly visible with a microscope.\(^{22}\)

According to the theory that is presented in this section, the seats plastic deformation rises with increasing magnification \( \zeta \), because the contact pressure increases with decreasing \( P \). Therefore one expects \( C(q) \) to be reduced for high wave vectors due to the plastic deformation. To calculate the leakage of a plastically deformed contact it is possible to first install the seat and apply the requested pressure and measure the surface afterwards. This will result in a surface roughness power spectrum of the plastically deformed surface \( C_{\text{plastic}} \). The calculation will predict the leakage for those surfaces at any pressure smaller or equal to the maximal applied pressures and forces.\(^{23}\)

Persson also presented a method to predict the effects of plastic deformation onto \( C(q) \) based on the undeformed surface.\(^{19}\) This transformation of \( C(q) \) has already been experimentally validated for metallic surfaces.\(^{24}\)

\[
C_{\text{plastic}}(q) = C(q) \left( 1 - \left( \frac{P_{\text{plastic}}(q)\sigma_Y}{P_c} \right)^6 \right)
\]

(13)

where \( P_{\text{plastic}} \) is defined as:

\[
P_{\text{plastic}}(q) = -\frac{2}{\pi} \sum_{n=1}^{\infty} (-1)^n \frac{\alpha_n}{n} (1 - (-\alpha_n^2 G(q)))
\]

(14a)

\[
G(q) = \frac{1}{4p^2} \int_{q_0}^{q} dq' q'^3 C(q')
\]

(14b)

\[
\alpha_n = \frac{n\pi P_c}{\sigma_Y}
\]

(14c)

Using the modified \( C(q) \) from Equation (13) in the method presented above will yield leakages that include the effects of plastic deformation with the given maximal applied pressure. The leakage will be approximately 8 times lower than for purely elastic seals.\(^9\) This is due to the surface height \( u_1 \) being only half the size of the undeformed seat. This method assumes purely plastic deformation. In general physical steels show work hardening have a smooth transition from their elastic regime to their plastic regime with a mixed regime in between those. Depending on how close the elasto-plastic behavior of the used steels is to purely plastic deformation, this approximation introduces another uncertainty to the simulation as compared to the experimental results.
The theory presented in this section is also used to determine the relation between the contact pressure and the distance of planes as presented in Section 2.1. This relation is important for the macroscopic FEM calculations.

\[ p = p_c e^{-u/u_0} \]  

\[ u_0 = \pi^{-1/2} \int_{q_0}^{\infty} dq q C(q) \omega(q, 1) \]  

\[ p_c \approx 0.749 E^* e^{-\langle \log \omega \rangle} \]  

\[ \langle \log \omega \rangle = \int_{q_0}^{\infty} dq q^2 C(q) \omega(q, 1) \log \omega(q, 1) \int_{q_0}^{\infty} dq q^2 C(q) \omega(q, 1) \]

3  |  EXPERIMENT

A test rig that is specialized on measuring the leakage is used for the experiment, which is described in the following section. Usually, particles that are in the fluid have a huge influence on the tightness of a metallic seal. This has been shown experimentally in reference.21 There are various sources of contaminating particles, for example, corrosion of the metallic surfaces, unwanted leakage of unclean hydraulic oil into the system or the remaining particles in the purified water. The particles will clog the microscopic channels through the sealing contact and will reduce the leakage over time. The effect of the particles is not included in the theory presented in Section 2.2. For this reason, the influence of the particles should be kept as small as possible in this experiment in order to measure the influence of the surfaces and of the normal forces.

3.1  |  Experimental setup

The most important part of the experimental setup is the test chamber, which contains the ball and the seat, as pictured in Figure 6, together with a sketch of the same test chamber.

The test rig to measure the leakage of ball seat valves consists of three different fluid systems.

First there is a system of purified water. The water is the leakage test fluid. In Figure 6(A), the water is colored in dark blue. Purified water has been chosen because it is expected to contain only a very small amount of particles. Also, water has a smaller viscosity as compared to hydraulic oil, which is usually used in industrial applications. The smaller viscosity leads to a higher leakage according to Equation (12) which is easier to measure. A disadvantage of water is that one has to be careful about corrosion.

The second system is a hydraulic oil system. In Figure 6(A), the oil system is colored in yellow. The oil delivers the pressure that is used to press the piston onto the ball. It is the source of the additional normal force. The oil pressure can be set to different values and thus a wide range of forces can be achieved in the experiment.

The last system is the air system, which is colored in light blue in Figure 6(A). The air pressure is used to lift the piston back up again. This is important in order to lift the ball up out of the seat and also to be able to exchange the seat. Another reason for the pneumatic system is to remove the leakage of oil into the middle chamber from the system in order to prevent it from entering in the water system.

3.2  |  Experimental procedure

Before any measurement can be done, it is important to clean the contact area between the ball and the seat from any particles. The leakage of the system always decreases in time. It is only possible to get comparable results if the contact is clean. After a long break, the water that is close to the ball should be replaced to remove the particles which have accumulated in the chamber. Right before starting a measurement, the ball is lifted out of the seat and the outcoming
FIGURE 6 Test chamber containing the ball and the seat. In the scheme, the ball is denoted with 1, the seat with 2, and the piston that enacts the force onto the ball with 3. The water system is colored in dark blue (4), the air system in light blue (5), and the oil system in yellow (6). The main dimensions are listed in Table 1.

flow of water removes the particles at the contact. Right after returning the ball into the system, the measurement has to be started quickly. Accumulators between the water chamber and the pump are used to quickly return to the requested pressure levels. Both the water pressure and the oil pressure are kept at constant values. Hydraulic accumulators help to keep the pressures constant beside the wanted and unwanted leakage. After that the measurement can be started.

Depending on the amount of leakage, there are three different methods to measure the leakage. If there is very few leakage, the best and most accurate method is to count the number of drops. The volume per drop can be determined by making multiple measurements of a fixed amounts of drops using a graduated pipette. If there is slightly more leakage, it is better to measure the volume with measuring cylinders. If the leakage is very large it can be measured using scales. All three different methods are present at the test rig and have been used for different seats and different pressures. Every one of the presented methods allow the experimenter to measure the change of leakage over time.

The leak-rate decreases over time and after a few minutes the experiment needs to be repeated. The reduction of leakage due to the particles leads to unpredictable changes in the leakage and therefore in the results. This is further discussed in Section 4.3.1. It can be seen in the data that there are also trends in the measurements between different sessions. They exist despite cleaning the contact thoroughly. A possible reason for such trends is either the influence of temperature or the corrosion of the seat or the pipe. In fact visible corrosion occurred on one of the seats that were left in the water over night. Therefore, it is also important to keep the seats dry when it is not in use.

The orientation of the ball is not important for the measurement, because the ball is much smoother than the seat.

4 | RESULTS

The experiments and the simulations have been performed for seats with different surfaces. Before the experiment is started, the initial surfaces have been examined.
4.1 | Surfaces

In total, three different surfaces are presented in this article. All three seats have first been manufactured by turning and have been sand-blasted afterwards using different times or techniques. Figure 7 shows photographs that have been taken of the seats using an microscope at 20-fold resolution.

Seat 3 (see Figure 7) is a seat that suffered to corrosion prior to the experiment. Sand-blasted surfaces are perceptible to corrosion even if they are made out of stainless steel. It did not change much during the experiment. The surfaces are mostly isotropic due to the sand-blasting. There is still a slight directional order due to the manufacturing process of the seats, which was not completely removed by sand-blasting. The corrosion has no influence on the isotropicity. The remaining anisotropicity is another source of error in the calculations.

The surface roughness power spectra of each surface are seen in Figure 8. In each case, the spectrum has a similar behavior. First $C(q)$ remains almost constant, although a few low wavelengths are more visible than others. This can be explained by the turning, which was used to create the seats. After this area $C(q)$ decreases linearly in the double logarithmic graph. This relates to a self-affine fractal behavior, which many physical surfaces show at least for small wave lengths. The clearly visible angle between both areas is not a result of the method, but a physical feature of the surfaces.

The change of resolution between the different line-scans is also clearly visible in a change of roughness of the graph. For each calculation, $C(q)$ is adjusted to include the effects of plastic deformation according to Equation (13) as if the current pressure was the highest pressure which was ever applied onto that combination of ball and seat. This also coincides with the experimental procedure.

An example for $C(q)$ of a plastically deformed surface can be seen in Figure 9.

4.2 | Simulations

The FEM calculations reveal the influence of the normal pressure onto the contact properties. The contact area in radial direction increases with rising normal pressure. In the same time, the maximal contact pressure also increases. This can be seen in Figure 10. Note that the total normal force is the sum of the force created by the fluid pressure and the additional normal force minus the force by the fluid pressure that acts on the piston.
The rise in the maximal contact pressure has several effects onto the leakage. It changes $P$, see Equation (7), and therefore raises the critical magnification. Therefore it decreases $\bar{u}$, see Equation (11), which is also directly reduced by the contact pressure. In general an exponential decline can be expected in a purely elastic setting. Additionally, a higher contact pressure also increases the plastic deformation and reduces $C(q)$, which further reduces the leakage.

The increase in the contact length also decreases the leakage linearly, see Equation (12).

The simulation shows a decline of leakage with rising normal pressure, as expected.

### 4.3 Experiments

Each measurement has been repeated for at least 5 times. With the help of this repeated measurement, the statistical measurement uncertainty can be calculated. The uncertainty on derived quantities can be calculated using Gaussian error propagation. During each experiment, the leakage was measured at multiple times. This allows the analysis of the time dependency of the leak-rate.

#### 4.3.1 Influence of particles

In each measurement, there was a decline of the leak-rate with time. An example for such a change in time can be seen in Figure 11.

Additionally, during the preparation of the measurement, it could be seen that the leakage is decreased if the water has not been exchanged for a longer time than usual. The decrease in leak-rate does not level out after any time. The influence of particles is the most likely explanation of this behavior, although further investigation is needed to further rule out other possible factors. If the decline is actually due to the particles in the fluid and the leakage actually happens through channels with a size as determined in Section 2.2, the decrease in the leak-rate accelerates with rising normal forces, because the size of the channels shrinks while the size of the particles in the water stays the same, which leads to a faster accumulation of particles.

One possible way to describe the influence of the particles is to consider the system as a filter: the seal itself is considered to be the actual filter medium. The particles that accumulate in the contact form a filter cake. The whole fluid and...
Figure 11 Change of leakage over time at 10 bar fluid pressure together with a fit onto the derivative of the filter curve seen in Equation (16a)

(A) Seat 1 with no additional force  
(B) Seat 2 with no additional force  
(C) Seat 3 with 8.4 kN additional force

smaller particles have to pass through this filter medium and the filter cake. With increasing size of the filter cake the leakage drops and the speed at which the filter cake grows decreases. Systems consisting of a filter medium and a filter cake can be described by the filter cake equation, which leads at constant fluid pressure to the following equation:

\[ Q = \frac{L_y^2 \Delta P}{\eta \delta} \left( \frac{\beta L_y}{\delta} \right)^2 + \frac{2L_y^2 \Delta P}{\eta \delta} t \right)^{-1/2} \]  

\[ L_y = 2\pi R_c \]  

In Equation (16), \( t \) is the time, \( \delta \) is a constant specific to the particles, which form the filter cake, and \( \beta \) is specific to the seal. The function seen in Equation (16a) is called the derivative of the filter curve. Using the relation for the leakage from Persson’s model (Equation 12) to describe the leakage of the unperturbed seal, one can find:

\[ Q(t = 0) = \frac{L_y \Delta P}{\eta \delta} = \alpha \frac{u_r^3(\zeta_c)}{12\eta} \frac{L_y}{A_C} \]  

\[ \beta = \frac{12 A_C}{\alpha u_r^3(\zeta_c)} \]  

Under this assumption, the leak-rate approaches 0 for large times.

When the size of the channels decreases for higher contact pressures or smoother surfaces, \( \beta \) increases to represent this. If the assumption of the accumulation of particles is true, then \( \delta \) increases as well. This represents a higher fluid resistance due to the filter cake. This is due to finer particles being blocked by the seal and accumulating at the contact.

This assumption is supported by the data presented in Figure 12. The dataset used to generate this figure is relatively small and this hypothesis still needs to be tested in future works.

4.3.2 Measured leakage

The rate of leakage is estimated by measuring the total leakage after 10 minutes and dividing the measured volume by the total time. The problem with this method is that the leak-rate is not constant over time due to the influence of the
particles as described in Section 4.3.1. This effect cannot be compensated by a constant factor, because it depends on the total normal force. The fraction between the total leak-rate and the leak-rate at the first measured time-interval lies between 1.02 and 2.10. Another effect that decreases the measured value compared to the calculated value is the factor $\alpha$, see Equation (12). We have chosen $\alpha$ to be 1, but it generally has some value smaller and close to 1. The values in the following graph (Figure 13) have all been arbitrarily increased by a factor of 1.25 for seats 1 and 2 and by 1.6 for seat 3 to make the comparison with the simulation easier.

Figure 13 shows the leakage of seat 1 and seat 2 under changing total normal force. The leakage decreases with increasing normal force, as one would expect. Both seat 1 and seat 2 have very similar surfaces and have therefore similar leakage curves. For higher normal forces, there is little change in the leakage whereas the leakage decreases very quickly in the left section of the graph. It is also clear that the influence of the particles cannot be perfectly accounted for using a constant factor. Due to this, the agreement of the simulation and the experiment differs for different pressure areas.

The simulations overestimate the leakage for seat 1. This is probably due to an insufficient surface scan. Bad or insufficient surface data can lead to a change of the relation between the distance of the surfaces and the contact pressure, see Equation (15). Here, the surface appears to be too smooth in the calculation which reduces the maximal contact pressure. The total calculated leakage is very sensitive on the contact pressure and an error in this value can have a huge impact in the total result. Using a different combined $C(q)$, which uses more data points, has only a slight impact on the final results.

The leakage of seat 3 can be seen in Figure 14. Seat 3 has a smoother surface compared to seat 2 and seat 1. It shows reduced leakage at all normal forces compared to the previous 2 seats, see Figure 13. Compared to the previous graphs, the rate of decrease is higher for high normal forces. A similar behavior will probably be observed for higher normal forces as reached in this experiment. The total leak-rate looks in all cases roughly proportional to the filter cake coefficient $\delta$ presented in Section 4.3.1, see Figure 12.

A problem in the comparison between the simulations and the experiments is the actual amount of plastic deformation. When the pump has build up the nominal oil pressure, the pressure overshoots the requested pressure and reaches...
pressures higher than the nominal pressure for a short amount of time. Depending on the time and the peak pressure, this introduces more microscopic plastic deformations onto the seat than considered in the simulations and this will reduce the amount of leakage compared to it.

Another method to compare the leakage in the simulation with the leakage of the experiment is to choose an earlier time point at which the final leakage is measured. The leak-rate decreases in time due to the influence of the particles. Therefore, the simulation, which does not include the effects of the particles, should be closer to the experiment. Due to the smaller amount of accumulated leakage, this increases the measuring uncertainty at the same time.

In Figure 15, there are graphs of the measured leak-rate after only 2 minutes, but with no semi-arbitrary compensation factor. The data are based on the same experiment, but evaluated for an early time. It can be seen that there is a good agreement for both graphs. The leakage of seat 1 does not agree in either case with the simulational data.

There are also other influences on the leakage that have not been considered in this work but still influence the leakage. For example the temperature and the vibrations of the whole system both have an influence on the leakage, but they have been kept approximately constant during the course of the experiments.

5 | CONCLUSION AND OUTLOOK

This article has shown that Persson’s contact mechanics can be used to estimate the leakage of metallic ball seat valves. The influence of the surfaces on the leakage could both be measured and calculated. The surface clearly is an important factor in determining the quality of a metallic seal. The method which has been presented in this work can be used to analyze various sizes and designs of metallic seals. For example, different types of seals can be analyzed using this method. Using this method, one can optimize a given seal type for leak tightness by calculating the leakage of many variations of a given geometry.
There is still room for improvement in the way the contact pressure distribution is introduced into the calculation. The method is still somewhat sensitive on some assumptions, which have been chosen arbitrarily. Further research will allow to make a better quantitative comparison between the simulations and the experiments while making the simulative procedure more robust.

A further progress in validating Persson’s theory for metallic seals can be achieved by isolating the influence of the normal force on leak tightness from other influences.

It became obvious in this work that particles in the fluid have a significant effect on seal tightness. The research of their influence still needs further investigations. This research will also help to distinguish the effects of the particles onto the tightness from the influence of the surface topology. This research could be used to further validate the results of this work.

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The authors have no conflict of interest relevant to this article.

AUTHOR CONTRIBUTIONS
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