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HOLLOW PISTONS IN HYDRAULICS – POSSIBILITIES OF INCREASING VOLUMETRIC EFFICIENCIES

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Abstract: *The cluster of excellence "Tailor-Made Fuels from Biomass" at RWTH Aachen University is identifying and investigating new potential biofuel candidates. Fuel candidates developed in this process greatly vary in rheological and hydrodynamic properties. One of the most important properties is the fluid viscosity. Low viscosities increase volumetric losses, friction and wear in the fuel pump, therefore lowering the overall efficiency. An optimization of the tribological contacts inside the pump is essential to ensure a sufficient performance. One of the most important tribological pairings is the piston-bushing contact which has a crucial impact on the leakage. A possibility to decrease this leakage is the hollow piston design developed for common-rail pumps. In common-rail injection pumps a reduced gap height can increase the volumetric efficiency by approximately 40% at 120 MPa injection pressure. In this paper, a first attempt is performed to transfer this specific hollow piston into hydraulic pumps. Thereby a possible application preferably operates at low viscosity and high bulk moduli. Relatively low volumetric efficiencies indicate hidden potential. Both of these cases can be found in water hydraulics. With a maximum pressure of 32 MPa radial piston units commonly operate at 25 % of the common rail system´s pressure level. Investigating possible applications of the hollow piston principle for water hydraulics, various potential piston geometries of a radial piston unit are simulated using FEM analysis. In order to ensure an optimal geometry, geometrical approaches are discussed and applied to the hollow piston as well as simulated with a Matlab model using the Reynolds equation regarding the resulting leakage and potential increase in volumetric efficiency.*

Keywords*: Pump, Piston, Common-Rail, Water Hydraulics*

1. Introduction

Within the cluster Tailor-Made Fuels from Biomass, new potential biofuel candidates are investigated and identified for a more sustainable fuel economy and lower global emissions. The primary goal is to find alternatives to current gasoline and diesel fuels. Therefore, the research focuses on two main areas of interest: Production and propulsion. The production side concentrates on finding new potential molecules as well as synthesis-processes for industry scale production, whereas the propulsion group focusses on the performance of the fuel candidates inside the combustion system with the main attention towards fuel compatibility with the current diesel and gasoline system.
In order to reduce emissions, the spray behaviour of the fuel during the injection phase plays a vital role. Here, a fast primary breakup of the fuel jet is highly important for a good atomization and homogeneous spray distribution, resulting in a combustion with lower soot emissions. One of the defining fuel characteristics for primary breakup is viscosity. Lower viscosity values tend to have a positive effect on the spray behaviour.
A promising molecule to act as a replacement fuel to diesel is di-n-butyl-ether (DnBE). Fig. 1a) shows the kinematic viscosity of DnBE in comparison to the standard hydraulic oil HLP46 and water over temperature [1]. With a viscosity value close to water, DnBE performs well in engine - tests. In order to provide the engine with enough fuel, the injection pump needs to deliver fuel at elevated pressures of up to 300 MPa when looking at compression ignition (CI). Here, the low viscosity of DnBE causes high leakage rates and low volumetric efficiencies, see Fig. 1b) [2]. Lower volumetric efficiencies and higher leakage rates would cause the need to use a bigger pump

in order to supply enough fuel. Especially in the automotive sector, an increase in size is hardly possible. Therefore, alternative ways to reduce leakage have to be investigated.

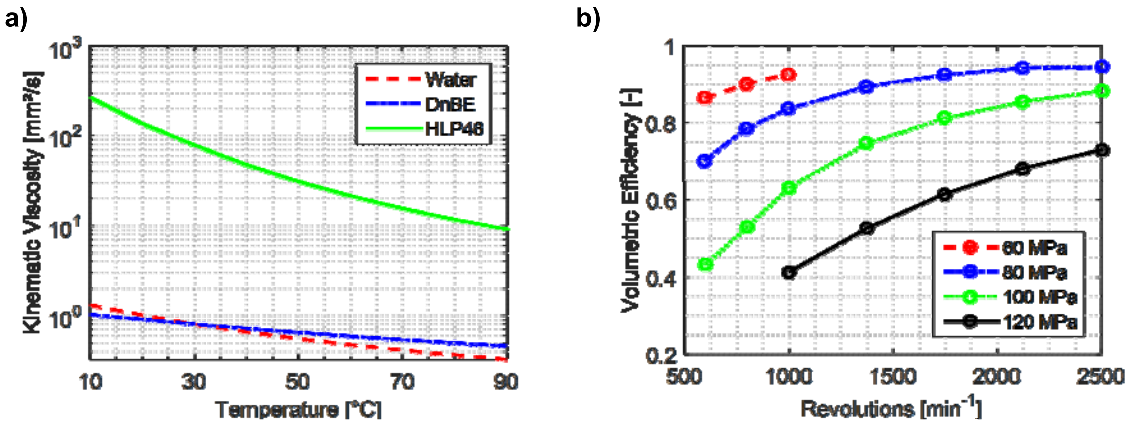


Fig. 1. a) Kinemtatic viscosity over temperature for Water, DnBE and HLP46; b) Volumetric efficiency of the BOSCH CP1 common-rail pump using DnBE

1.1 Hollow pistons in CI injection pumps

In todays common-rail injection systems, the components are developed and optimized for diesel fuel. This ensures an optimal overall system efficiency. Using different molecules with the same components can lead to reduced engine performance. In order to minimize necessary changes to the current engine design, every system component was investigated separately regarding its´ performance. High volumetric losses inside the injection pump made an adaption of the pump towards new fuel candidates essential. Due to high injection pressures of up to 300 MPa, material deformations play a major role for pump performance. Fig. 2 shows the deformation of a standard piston of a Bosch CP1 radial piston common-rail pump during operation at 135 MPa. At initial state, the gap height of the sealing gap is approximately 3 µm. On the left, the gap height between the piston and the bushing over the crankshaft-angle is shown. During the compression stroke, the pressure level rises, causing the piston to shrink, increasing the sealing gap between piston and bushing, ultimately resulting in a high leakage-rate.

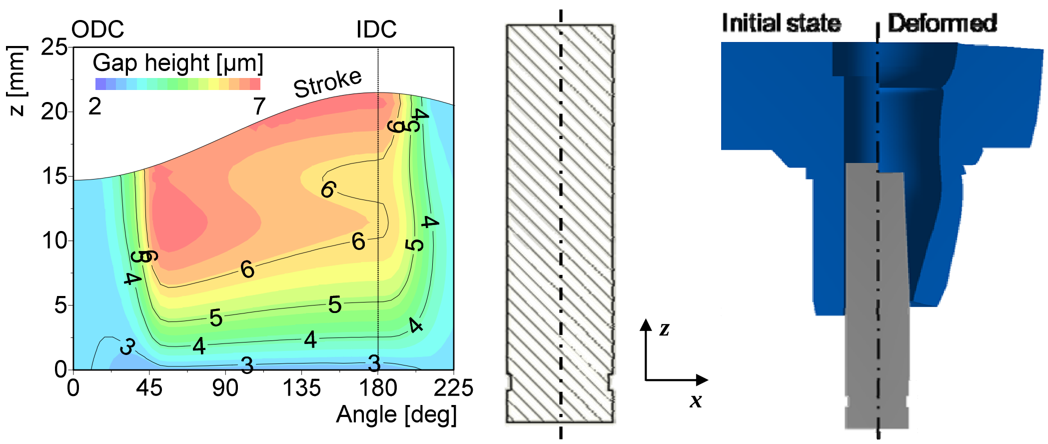


Fig. 2. Deformation of a standard piston inside a common-rail pump at 135 MPa

To reduce high volumetric losses of the injection pump, the hollow piston concept was proposed [3]. Here, the piston is hollow in the middle, see Fig. 3. The pressure within the piston equals the rail-pressure, causing a pressure difference between the piston inside and the sealing gap. The resulting deformation leads to an expansion of the piston diameter towards the bushing, see Fig. 3 right. The expansion results in a reduced gap height and therefore reduced leakage rate during the compression stroke, see Fig. 3 left in comparison with Fig. 2 left.

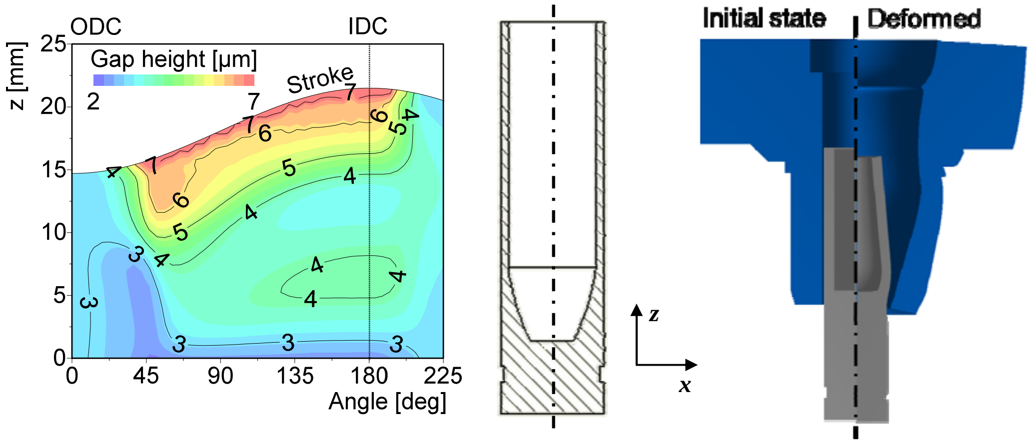


Fig. 3. Deformation of the hollow piston inside a common-rail pump at 135 MPa

The improvement of pump performance with hollow pistons of an exemplary design compared to standard pistons can be seen in Fig. 4. At 120 MPa, an increase in volumetric efficiency of up to 40 % compared to the standard piston was measured using DnBE [4].

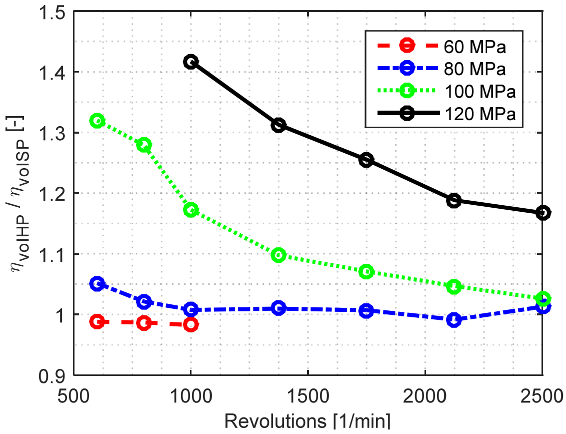
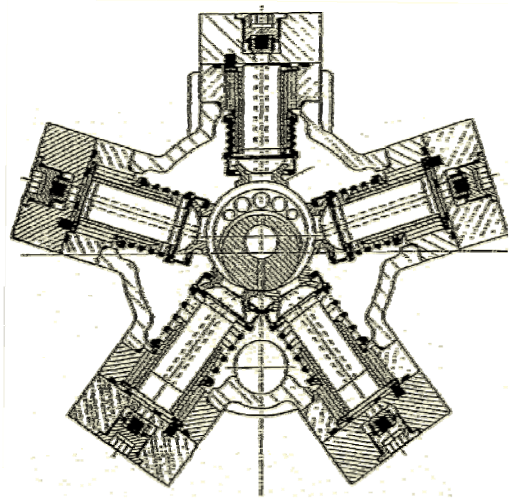


Fig. 4. Comparison of volumetric efficiencies of the standard piston (SP) and the hollow piston (HP)

1.2 Radial piston pumps in water hydraulics

The principle of radial piston pumps can also be found in water hydraulics. One advantage of radial piston pumps compared to axial piston pumps are the reduced lateral forces on the pistons. Therefore, the tendency towards an eccentricity between piston and bushing is reduced, increasing overall efficiency by reducing friction and leakage. Fig. 5 shows a cross-section of a Hauhinco RKP radial piston pump on the left [5]. The RKP pump series is check valve controlled with five or seven pistons. An eccentric drive-shaft rotates, causing the pistons to move up and down, compressing the fluid. Using high-grade steel, a separate lubrication system is avoided, with water being the lubricant. The sealing between piston and bushing is ensured through a sealing gap with no additional sealing elements. As a result, the geometric parameters of the piston-bushing contact are highly relevant, especially for volumetric losses. The right side of Fig. 5 shows a static performance diagram of a radial piston unit with a piston diameter of 30 mm. Volumetric losses cause a decrease of effective flowrate of up to 25 l/min translating into 13.3 kW additional power needed. Leakage between piston and bushing does account for parts of these losses as well as leakage through valves and housing.



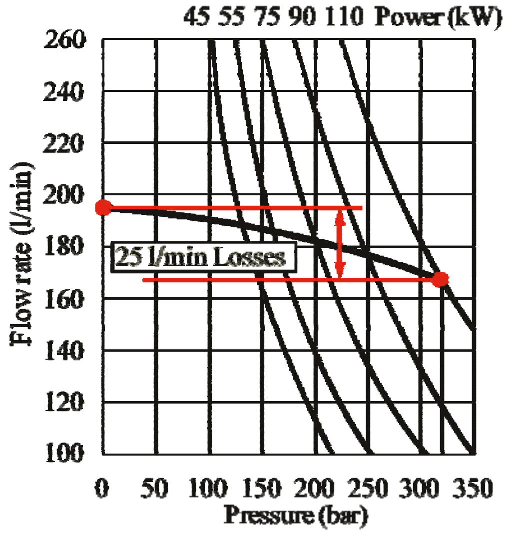


Fig. 5. Cross-section of a Hauhinco radial piston pump (left) and static performance diagram for a piston diameter of 30 mm (right) [5]

Another prominent pump principle used in water hydraulics is the axial piston pump, shown in Fig. 6 [5]. Here, hollow pistons are already in use, see enlargement of Fig. 6. The purpose of these pistons is a different one, though. With water being a relatively stiff fluid, hollow pistons increase the compression volume inside the piston chamber, therefore enhancing the controllability of the pump. Additionally, hollow pistons reduce the inertia, ensuring a more agile pump performance.

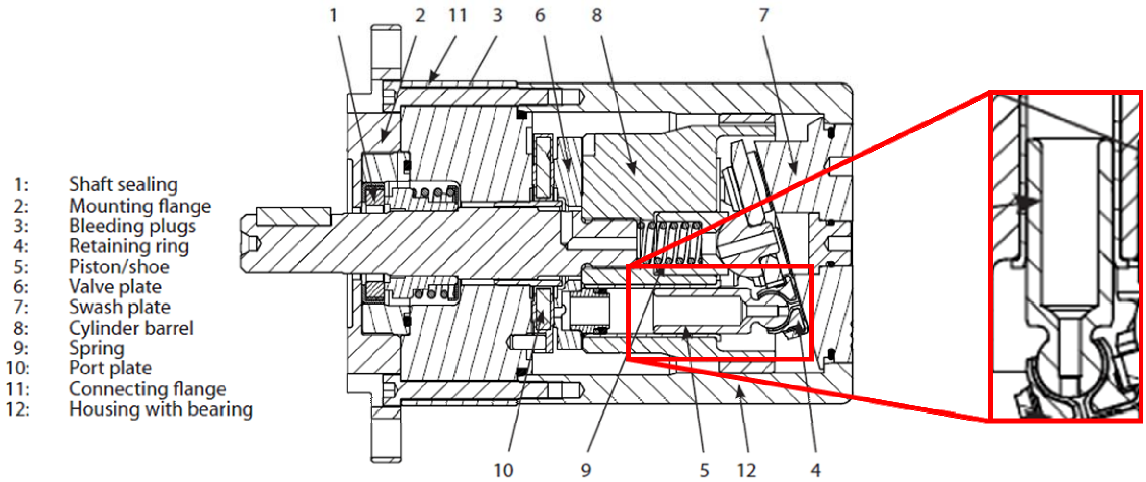
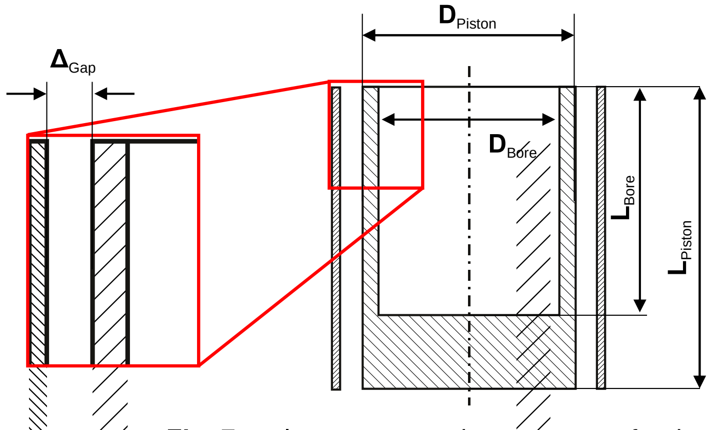


Fig. 6. Cross-section of Danfoss´s axial piston pump Nessie [5]

2. Application in water hydraulics

Looking at common-rail systems, the injection pressures go up to almost 300 MPa. Here, pressure induced deformation plays a significant role regarding the sealing gap between piston and bushing. In water hydraulics, however, pressure levels are usually in the range of up to 25 MPa. This poses the question whether the resulting forces on the piston are sufficient to cause relevant deformations. In water hydraulics, higher flow rates are needed than in fuel pumps. Therefore, the piston diameters are larger, e.g., 28 mm, see Fig. 7. It depicts the simplified piston geometry used in this work, with a stainless steel piston running inside a PEEK bushing. Piston and bushing are arranged concentrically to each other. Investigations on the performance are done at the outer dead center (ODC) where pressure and stress are at a maximum.



Parameter	Value
D_{Piston} [mm]	28
L_{Piston} [mm]	40
D_{Bore} [mm]	24
L_{Bore} [mm]	30
Δ_{Gap} [μm]	10

Fig. 7. Relevant geometric parameters for the hollow piston investigation

To evaluate the potential performance of hollow pistons in water hydraulics, two different design approaches were chosen, see Fig. 8. The first one being a bore inside the piston. This geometry can easily be manufactured by drilling. A possible downside to this design is the high mechanical stress at the bottom of the piston in the corners of the bore. This might lead to a material malfunction. As a second approach, the corners were adjusted by using splines. Manufacturing of such a geometry can also be done by drilling using an adapted drill. With the adjusted corners, stress is reduced significantly. This might lead to reduced deformations and therefore higher gap heights. In the following both design variations are investigated to compare the aforementioned advantages as well as the constraints.

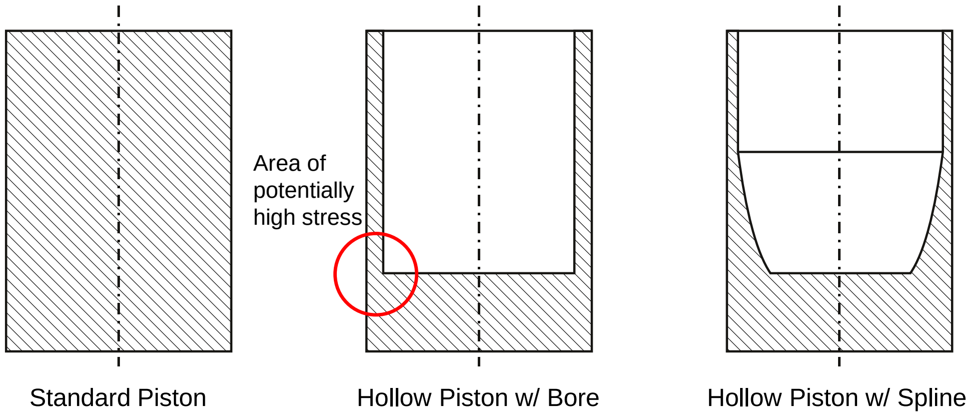


Fig. 8. Design approaches for hollow piston design

3. Simulation Approach

After the design process was done, an elasto-hydrodynamic (EHD) simulation was set up. Deformation and stress are calculated depending on the pressure ($f(p(x),\sigma)$ using FEM. Afterwards, a hydraulic reynolds-simulation determines the resulting flow and pressure profile ($Q,p(x)$). Fig. 9 shows the general simulation approach. Boundary conditions are defined for the initial design. Here, material data, pressure profile for the initial load and other parameters are set. A FEM-simulation is conducted using Ansys mechanical. Piston as well as bushing are simulated at a temperature of 293 K. Afterwards the axial deformation is evaluated and the resulting gap between piston and bushing is calculated. The gap geometry is then transferred to a Reynolds-gap-simulation. With a known pressure at the start and the end of the gap, the pressure dependend flow and the resulting pressure profile are calculated. The new pressure profile then is put into the FEM-simulation as a new boundary condition and new deformations can be calculatet. This iteration process continues until there is no change in deformation, therefore converges to a constant value.

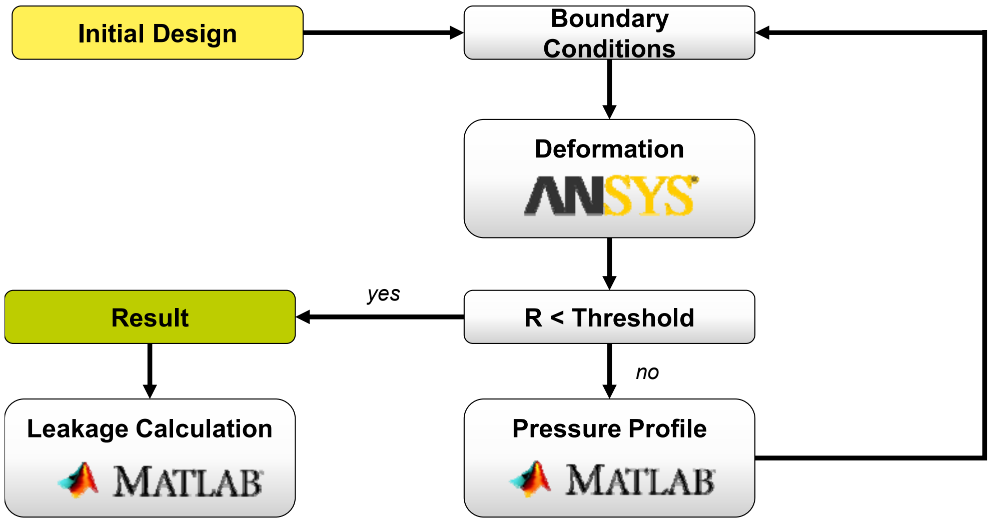


Fig. 9. Sequence diagram of the simulation process

The simulations were done using the simplified model of the piston-bushing contact shown in Fig. 7 for a static operation point at the outer dead center. Lateral forces on the piston were neglected and a concentric postion of the piston inside the bushing was asumed. This was done to get a first estimate on the achieveable improvements for comparable modes of operation. Real pump operation introduces lateral forces on the piston, causing increased wear on the one hand, and tilting and eccentricity of the piston, possibly leading to a significant increase in leakage flow between piston and bushing [6] on the other.
The convergence for the hollow piston simulation is shown in Fig. 10. Here, the maximum deformation is plotted over the position of the peak along the piston length represented in the node number. The graph shows that it takes approximately eleven iterations to reach a converged solution. For all the simulations, an investigation on the convergence was done to ensure reliable results.

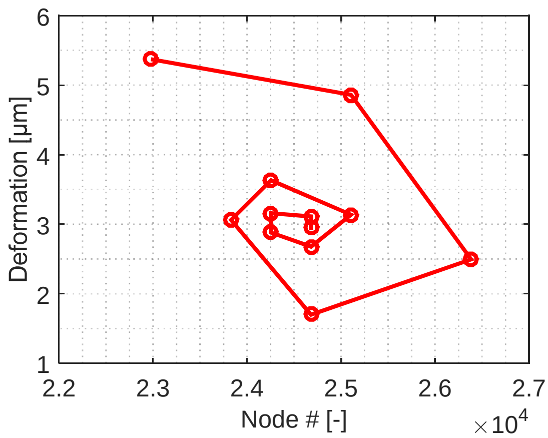


Fig. 10. Convergence graph for the hollow piston: Maximum deformation over the location on the piston

4. Results

After achieving convergence for all simulation models, the different piston designs were evaluated regarding the deformation, the resulting gap height, leakage flow and resulting mechanical stress. Fig. 11 shows the deformation of the hollow piston with bore on the left. Even though the pressure level was set to 25 MPa, significant deformations can be observed with values up to almost 3 µm towards the bushing. The sealing gap between piston and bushing is shown on the right of Fig. 11 with the pressure displayed at the top and the bottom of the gap. The mean gap width is 8.78 µm compared to 10 µm at the initial state (p=0 MPa) at inner dead center (IDC). This results in a leakage rate of 0.1401 l/min.

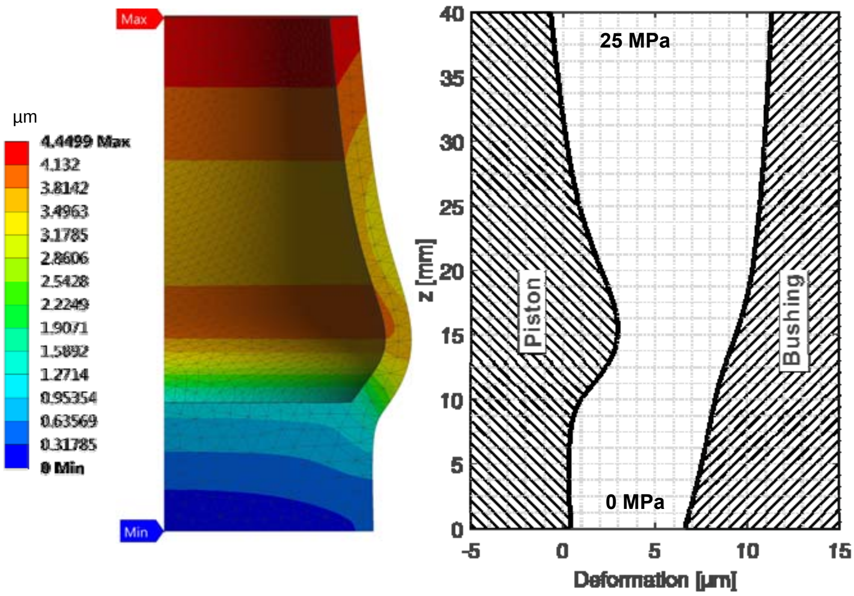


Fig. 11. Deformation and resulting gap geometry of the hollow piston with bore

The sealing gaps for the standard piston and the hollow piston with spline are shown in Fig. 12. For the standard piston, the mean gap height is 9.93 µm and 8.90 µm for the hollow piston with spline. This results in leakage flows of 0.2244 l/min and 0.1626 l/min respectively.

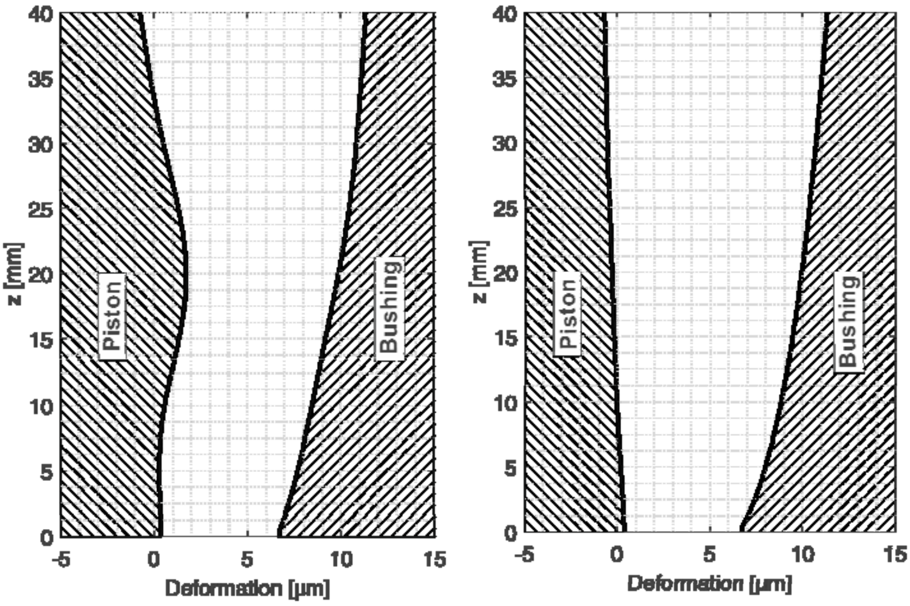


Fig. 12. Deformation and gap geometry of the standard piston (left) and the hollow piston with spline (right)

Table 1: Simulation results for different piston geometries

Pairing	ΔPiston$_{max}$ [µm]	ΔGap$_{mean}$ [µm]	ΔLeakage [l/min]	Mises stress [MPa]
Standard Piston – Bushing (SP)	-0.71	9.93	0.2244	-
Hollow piston with bore – Bushing (HP$_B$)	2.95	8.78	0.1401	76
Hollow piston with spline – Bushing (HP$_S$)	1.73	8.90	0.1626	48

With the simulated leakage rates at 25 MPa, the power loss can be calculated for all three piston designs. Fig. 13 shows the power loss in kW per piston caused by leakage at the outer dead center. The standard piston has a power loss of almost 1 kW compared to 0.68 kW for the hollow piston with spline and 0.58 kW for the hollow piston with bore. This leads to potential savings of 27.5% and 37.6% respectively at the static point investigated.

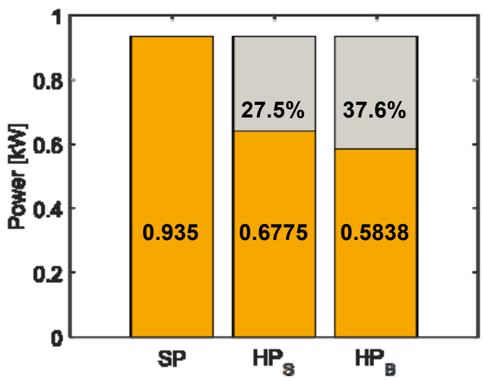


Fig. 13. Comparison of power losses caused by leakage (from left to right): Standard piston (SP), hollow piston w/ spline (HP$_S$), hollow piston w/ bore (HP$_B$)

In order to evaluate the mechanical stress on both hollow piston designs the von Mises yield criterion was applied using Ansys mechanical data, see Table 1. As expected, the hollow piston with bore shows the highest stress level at the bottom of the bore with 76 MPa. With the introduction of a spline, the maximum stress is located at the position of maximum deformation with 48 MPa. A stress reduction of almost 40% can be achieved by adding a spline to the bottom of the bore.
Fig. 14 shows a Wöhler-graph for type 316L stainless steel. In order to exceed the fatigue-limit, a specimen has to last 10^7 load-cycles. The stress at which the specimen reaches the limit indicates the stress level fit for durability. For a stress ratio of -0.2, representing alternating strain, the value is approximately 265 MPa [7]. Both of the hollow piston designs are well below that limit with a safety factor of at least 3.

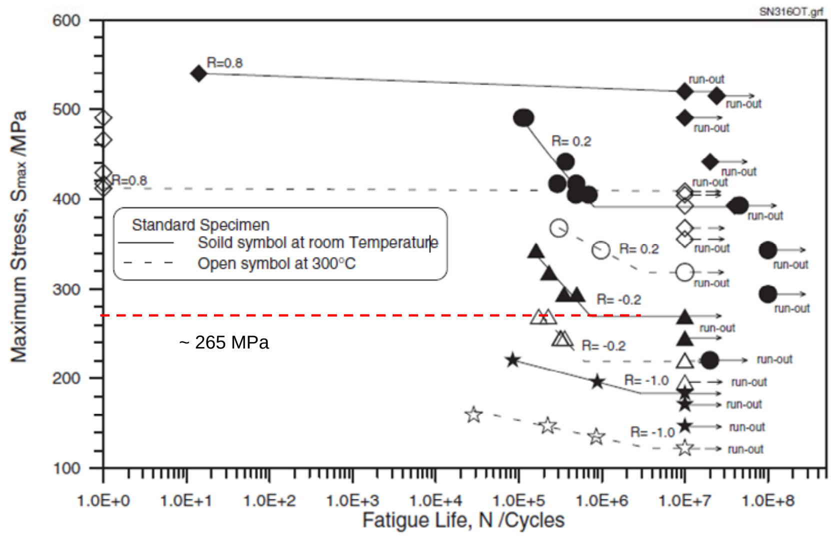


Fig. 14. Wöhler-graph for type 316L steel [7]

5. Conclusion and Outlook

In this paper, a first simulative investigation is done for the use of hollow pistons in water hydraulics to reduce the volumetric losses inside a radial piston pump. First, the knowledge of this principle is transferred from measurements on common-rail injection pumps to water hydraulics. Based on the results, two design concepts for hollow pistons in water hydraulics are derived. The concepts are simulated using Ansys and Matlab to calculate deformation and the resulting leakage flow. Afterwards the results are compared regarding the potential savings in power losses and the mechanical stress on the hollow pistons.
The results indicate a significant saving potential for both the hollow piston with bore and the hollow piston with spline with savings of up to 37.6% and 27.5% for the simulated static operation point. Additionally, the resulting stress through deformation for both designs is below the durability limit. Thus, both variants are suited for pump operation.
Hollow pistons for increasing volumetric efficiencies do show great potential. To evaluate the full potential, a transient investigation on the saving potentials has to be performed. Afterwards, a first test can provide proof of concept. In the end, an analysis on the manufacturing cost compared to the saving potential has to be carried out to ensure added value.

Acknowledgments

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