
Numerical Simulation of Fluid-Structure Interaction in the Design Process for a New Axial Hydraulic Pump

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Abstract: With the help of numerical simulation a new high-pressure hydraulic axial pump has been developed at Gdansk University. The pump is used in different applications demanding high power, e. g. construction machinery or military applications. Its unique feature is the total independence of a pressure switching mechanism, which saves weight and provides the possibility to control the pump by computer. Prototypes have been built and tested, for example in lifting devices on ships in an extreme temperature environment. However, when the pressure changes from high to low via a commutative bushing harmful pressure peaks were observed. To avoid these peaks a compensation chamber between the high and low pressure areas was introduced. Next to prototype testing interdisciplinary numerical simulation was used to find the optimal layout of the chamber and the pump. The hydraulic oil and the movement of the pistons were modeled in Fluent. The wall force that is exerted by the fluid on the chamber wall leads to deformations of the wall, which in turn changes the fluid domain. The deformation of the wall was computed with Abaqus/Explicit. To catch the interaction between the movement of the wall and the fluid Fraunhofer SCAI's coupling software MpCCI was used. The results of the coupled simulation proved to be very insightful for the development of the new pump and will be presented in this paper.

Keywords: CFD Coupling, Coupled Analysis, Design Optimization, Experimental Verification, Hydraulics, Fatigue

1 Introduction

Axial pumps with cam-driven commutation units – so-called PWK pumps – emerged as a result of a research project conducted in the Department of Hydraulics and Pneumatics at the Gdansk University of Technology.

As for all axial hydraulic piston pumps several cylinder chambers are positioned around the rotating shaft of the pump. The rotation of the shaft and the attached swash plate leads to movement of the pistons which decreases and increases the fluid volume of the chambers alternately. A window – which is part of the control sleeve or commutating bushing – connects the

chamber between the pistons with the low pressure and high pressure in- and outtake channels. The main elements of the pump are shown in Figure 3.

The axial piston pumps with constant displacement show a very good performance with a working pressure of up to 55 MPa, an overall efficiency of 94 % and good power density. In constant displacement pumps the pistons always travel the same distance inside the cylinder chamber resulting in a constant amount of output fluid.

To control the fluid amount variable displacement pumps are used. Usually, the piston displacement of



Figure 1 Constant displacement pumps PWK-78 (top) and PWK-27.

these pumps is controlled with a complicated hydraulic servomechanism. Theoretical analysis shows that the displacement of the PWK pump can be controlled by a low-energy actuator. This is the major advantage of the new developed pumps because it reduces the pumps cost and dimension significantly, thereby simplifying the whole process chain. To steer the displacement of the pump – and thereby the amount of fluid that is put through – the angular position of the control cam in relation to the pumps shaft has to be controlled. As both parts are rotating fast a special planetary gearbox (cf. Figure 2), that ensures precise control over the displacement in the operating range, was developed. This can be driven by a stepping motor. First prototypes of the variable displacement PWK pumps have been built and tested and showed good performance. Unfortunately, harmful pressure peaks, that could lead to pump damage, were observed. With the help of a compensation chamber these peaks could be significantly reduced. To find the optimal layout of the pump and the compensation chamber experiments and numerical simulation were used.



Figure 2 Planetary gearbox driven by stepping motor.

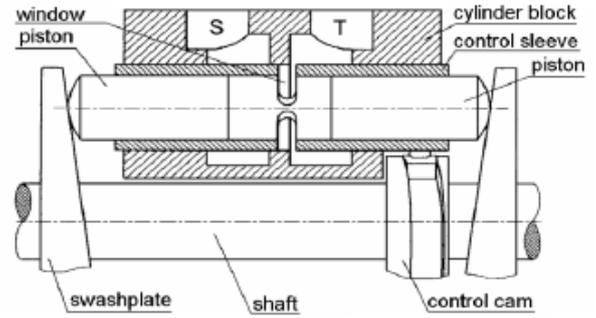


Figure 3 Main elements of PWK pump. Cross section of one chamber, the shaft and steering elements.

2 Pressure Peaks and their Compensation

2.1 Pressure Peaks

First tests of the control mechanism for the variable displacement pump confirmed the feasibility of the concept. However, harmful pressure peaks in the cylinder chambers were observed when the cylinder chamber is disconnected from the inlet and the outlet channel. The pressure reaches very high values – 20 MPa above the average pressure in the chamber – for a short time (cf. Figure 5).

These peaks are influenced by many different factors. The most important ones are the displacement adjustment, the rotational speed and leakage. When the displacement of the pump is decreased, the pressure peaks increase. This is due to the higher velocity of the pistons when the chamber is disconnected from the intake and outtake channels. A higher rotational speed of the shaft also leads to higher pressure peak values; mainly because it reduces the effect of leakage. Oil leakage reduces the amount of fluid to be compressed and thereby decreases pressure peak value. For fast rotating pumps the piston movement is fast – shortening the time during which the fluid is compressed and reducing the influence of leakage. This increases the pressure peak values. The compressibility of the hydraulic oil is directly connected to the occurring leakage and thereby also plays an important role in the investigation of pressure peaks. Leakage is influenced significantly by

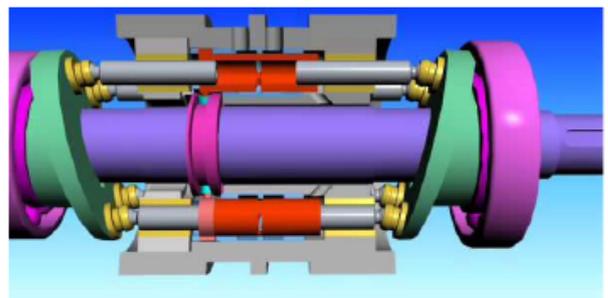


Figure 4 Constant displacement pump with four chambers.

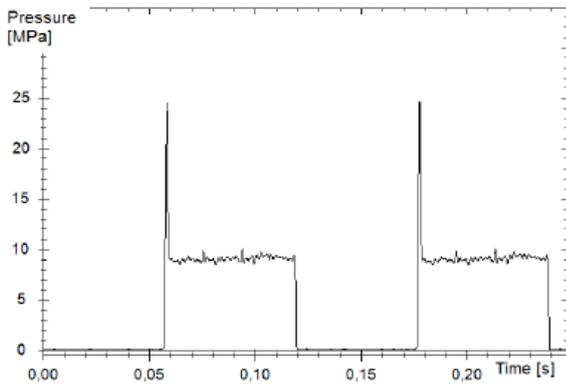


Figure 5 Pressure variation in pumps chamber (rotational speed 500 rpm, oil temperature 33°, displacement adjustment 20 %.)

the pressure of the fluid. The small gaps in the pump through which small amounts of fluid can leave the pump cycle get considerably larger during pressure increase. This aggravates the leakage which in turn influences the pressure – resulting in a very complicated feedback.

In [Osi2012] an equation – using the pumping pressure, the volume of the chambers and the compressibility modulus of the fluid – to estimate the pressure peak value is presented. The real volume of fluid that has to be compressed – which is directly linked to the amount of leakage – can be integrated into the equation.

2.2 Pressure Peak Compensation

The pressure peaks of up to 20 MPa could damage the pump and produce a lot of noise. To get a more robust pump these pressure peaks need to be compensated.

The peaks occur when the cylinder chamber is disconnected from the intake and outtake channels and the fluid is compressed. To shorten the period of disconnection from the channels and to give the fluid more room an additional compensation chamber with an elastic wall was introduced. The cylinder chamber connects to the compensation chamber when it is disconnected from the pumps main channels.

In the phase of compression (which can be seen in Figure 6a) the compensation chamber is connected to the fluid volume in the cylinder: the surplus fluid can flow into the compensation chamber which results in a reduction of the pressure peaks. When the pistons move back, increasing the volume of the cylinder chamber, the compensation chamber again connects to the cylinder and the fluid can flow back (cf. Figure 6b).

As the different cylinder chambers (in most cases seven, nine or eleven cylinder chambers are used) compress in an alternating cycle one compensation chamber can be used for all cylinders. First experimental tests showed that the introduction of the compensation chamber reduced the pressure peak value by 50 % without decreasing the efficiency of the pump.

The main objective of the interdisciplinary numerical simulation is to find the optimal layout of the compensation chamber. Differences in shape, elasticity

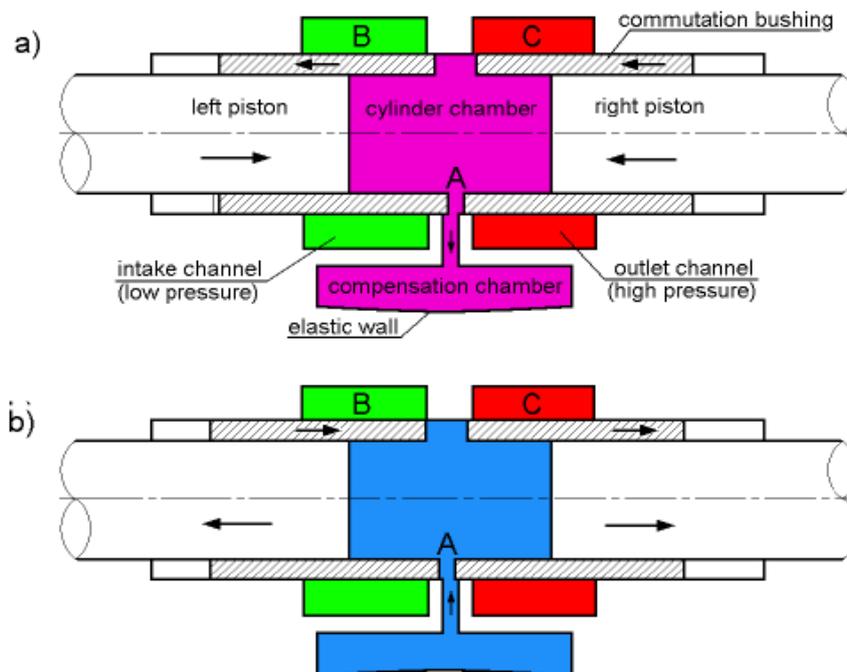


Figure 6 Pressure peak compensation with a compensation chamber with an elastic wall. a) shows high pressure in the cylinder chamber which causes the fluid to flow into the compensation chamber. In b) the pistons are moving outwards and the fluid flows back into the cylinder chamber.

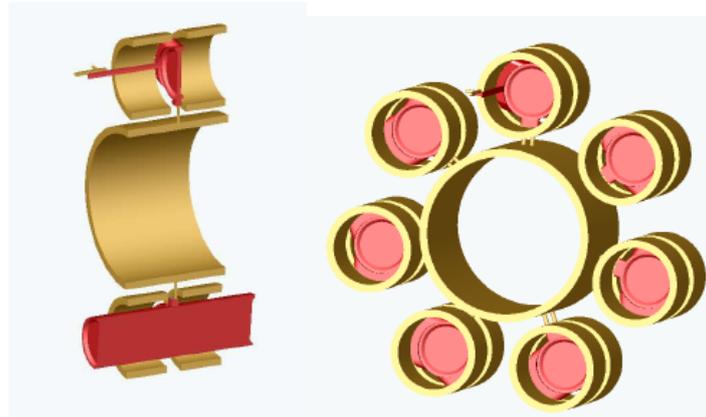


Figure 7 CAD models of a pump with two chambers (left) and seven chambers (right). The left model is a symmetric half model. All movable parts are shown in red.

and volume and their effect on the pressure peak values are to be investigated. CFD models of the pump and FEA models of the elastic compensation chamber are developed.

3 Coupled CFD and FEA Simulation

At first a CFD model of the PWK pump – including the compensation chamber – was developed. For most simulations a symmetric half model of a pump with two cylinder chambers and the compensation chamber was used to keep the model relatively small and simple. A full model of a pump with seven cylinder chambers was also developed. The CAD geometry that was used to generate the models is depicted in Figure 7.

Fluent (version 12.1) was used for the CFD simulations. The movement of the pistons and the commutating bushing (connecting the cylinder chamber

with the intake and outtake channels as well as the compensation chamber) was modeled with user defined function.

The wall of the compensation chamber is subject to pressure exerted by the hydraulic oil. To catch the deformation of the elastic wall an Abaqus/Explicit (version 6.11) model – only consisting of the wall of the compensation chamber – was created. Abaqus/Explicit simulates the deformation of the elastic wall using the fluid pressure calculated with Fluent as a boundary condition. Fluent receives the deformation of the wall and updates the fluid domain mesh. This is a classic example of fluid-structure interaction. To pass the quantities between Abaqus/Explicit and Fluent Fraunhofer SCAI's code coupling tool MpCCI was used.

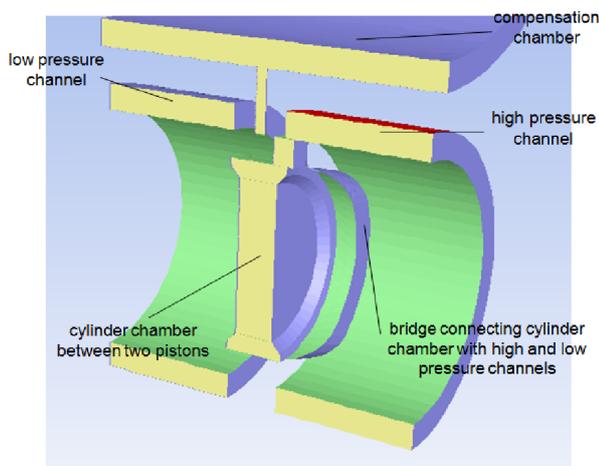


Figure 8 One cylinder chamber in Fluent. Walls are colored blue, interfaces green and symmetry planes yellow.

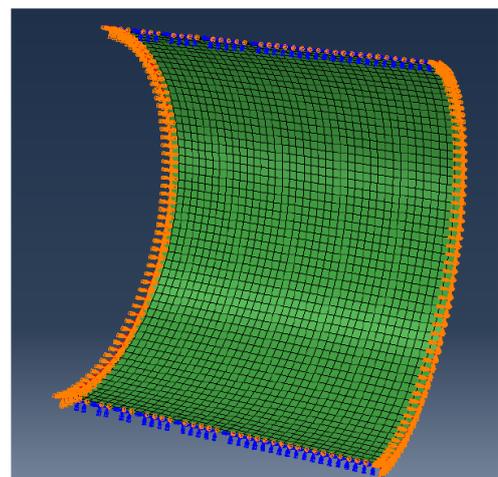


Figure 9 Abaqus model of the wall of the compensation chamber with boundary conditions.

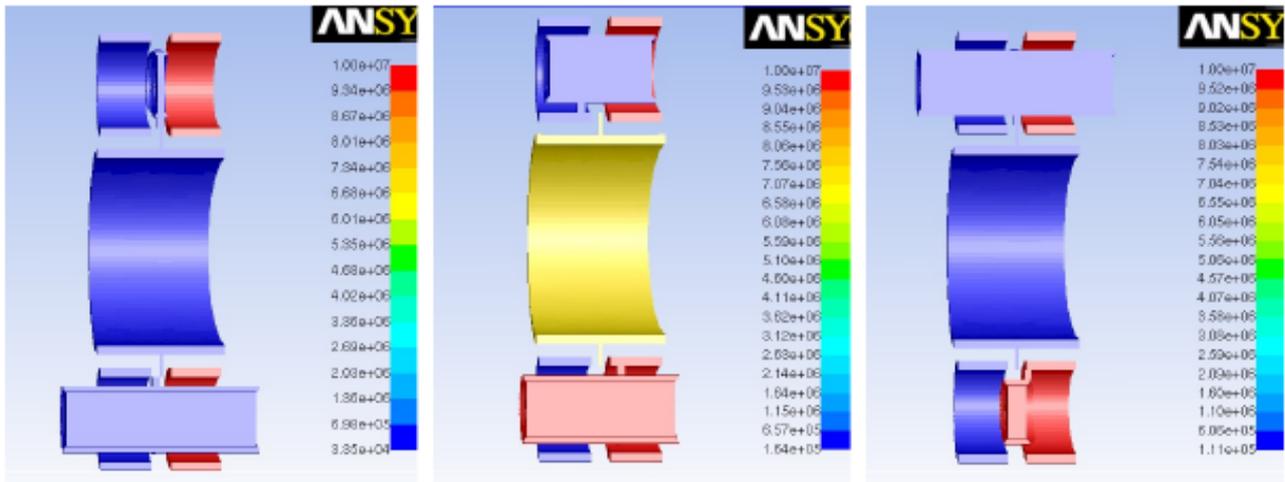


Figure 10 Pressure contours for the PWK-pump at different time values: on the left $t = 0.0002$, in the middle $t = 0.01$ and on the right $t = 0.02$

3.1 Simulation Models

This description of the simulation models will only refer to the symmetric half model with two cylinder chambers that can be seen on the left of Figure 7.

The mesh in Fluent consists of hexahedral elements. The increasing and decreasing of the cylinder chamber volume is realized by the dynamic layering method. The motion of the pistons and the bridge connecting the cylinder chamber to intake and outtake channels and the compensation chamber are implemented with user defined functions. Figure 8 shows one of the two cylinder chambers in Fluent in more detail. In the picture the two pistons are quite close to each other leading to a small volume of the cylinder chamber. The bridge is about to disconnect the cylinder chamber from the high pressure channel and moving towards the compensation chamber.

The hydraulic oil is assumed to be slightly compressible – the density is defined with a user-defined function. The transient problem is solved using the Spalart-Allmaras turbulence model and a coupled solver for pressure and velocity. Two cycles of the pump (depending on the configuration) take 0.08 s. A time step of $1e-05$ s or $5e-05$ s is used. The high pressure in the outtake channel is 10 MPa and the low pressure is set to 0.2 MPa.

The geometry of the Abaqus/Explicit model can be seen in Figure 9. It consists of linear quadrilateral shell elements of type S4. Boundary conditions are employed to keep the top and bottom fixed. The left and right sides of the chamber wall are also fixed with boundary conditions. The wall of the compensation chamber is slightly larger than the chamber itself. The overlapping parts on the left and right are kept in a fixed position due to pump design. The thickness of the elastic wall is 1.5 mm.

The Young's modulus of the isotropic elastic material is $2.1e11$ Pa, the Poisson ratio 0.3. The density is

7850 kg/m^3 . The problem is solved with a geometric nonlinear solver in Abaqus/Explicit. The time increment in Abaqus/Explicit is fixed and the same as in Fluent.

In MpCCI the quantities “relative wall force” and “nPosition” (the nodal position of the deformed wall) are selected for coupling. Relative wall force values (for the wall of the compensation chamber) are transferred from Fluent to Abaqus. Using the wall forces Abaqus/Explicit calculates the deformation of the wall which is then transferred back to the Fluent model. Fluent updates the mesh and performs the next time step. The coupling time step equals the time step from Fluent and Abaqus/Explicit.

Under-relaxation is used in MpCCI to stabilize the coupled simulations. Both quantities – the relative wall force and the displacements of the coordinates – are under-relaxed with a value of 0.8. The under-relaxed displacement is then added to the old coordinates to get the new position of the nodes.

Additionally, in Abaqus/Explicit the applied load is ramped linearly over the time step.

3.2 Simulation Results

The simulation results show the necessity of a coupled simulation for this application. The simulation including the fluid-structure interaction at the elastic wall agrees significantly better with the experimental findings than the stand-alone CFD simulation.

For an easy comparison of simulation results and experimental data during the simulation the pressure at three discrete points of the pump is monitored: one point located in the center of the upper chamber, one in the lower chamber and the last point lies in the center of the compensation chamber. The pressure at these points is monitored during the simulations and can be compared for different pump parameters or simulation variations.

The pressure contours of the fluid on the moving geometry can also be visualized to get a general idea

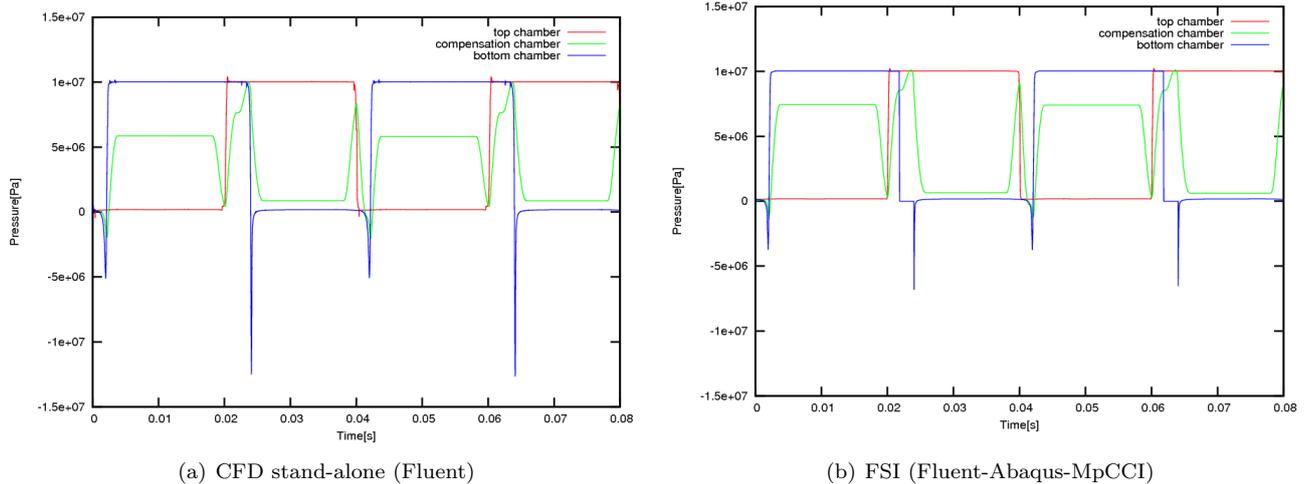


Figure 11 Pressure plots for three discrete points in the pump: a) shows the CFD and b) the FSI simulation. The differences in the pressure peaks and concerning the pressure in the compensation chamber are quite obvious.

of the working pump and the quality of the simulation – but the slight differences between the stand-alone CFD solution and the Fluid-Structure solution cannot be perceived visually on these contour plots.

In Figure 10 the pressure contour for three different time values is shown. During the first part of the pump cycle – which is depicted on the three contour plots – the upper chamber volume is increasing and the fluid in the lower chamber is compressed. The upper chamber was initialized with low pressure – although the pistons are close to each other – which can still be seen in the contour plot on the left.

The big negative pressure peaks that were observed for all simulations (CFD stand-alone and FSI) are probably due to cavitation occurring at the small passages between the different chambers. The oil reaches high velocities when flowing from a cylinder chamber to the compensation chamber. High velocity, narrow passages and high pressure point towards cavitation.

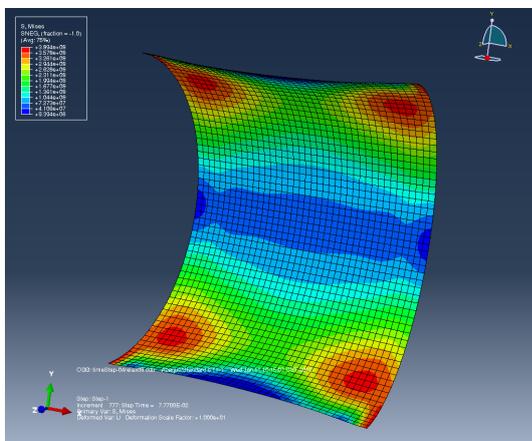


Figure 12 Von-Mises-stress on the (slightly) deformed elastic wall of the compensation chamber.

The results – the Von-Mises stress on the deformed elastic wall – of the geometrically nonlinear Abaqus analysis can be seen in Figure 12. The displacements are quite small and reach values around 0.04 mm.

Figure 11 shows pressure plots for the three discrete points. On the left side the CFD stand-alone plots are displayed, on the right the FSI pressure plots. The negative pressure peaks are much less pronounced in the FSI solution. Furthermore the pressure in the compensation chamber shows a different behavior in the two cases. The values during the high pressure phase are higher for the FSI solution. Also, a qualitative difference can be observed when investigating how the pressure rises or falls in the compensation chamber.

All in all the FSI simulations showed a very good agreement with the experiments that were conducted. The CFD model on its own is not capable of catching the pressure behavior – especially in the compensation chamber – in a satisfactory way. The elastic behavior of the wall cannot be integrated into a CFD model without coupling with a structural mechanics solver like Abaqus/Explicit.

4 Summary and Outlook

The FSI simulation for the simplified symmetric PWK pump model with two chambers produced results which coincide with experimental data in a very satisfactory way. Several configurations and settings for the simple pump model can now be checked with the help of numerical fluid-structure interaction simulation with the aim of finding an optimal configuration.

Simultaneously the full model of the pump should be investigated with a FSI simulation. This model is quite complex – for seven chambers the motion of the pistons and the motion of the bridges has to be defined – and

therefore will only be simulated for relevant cases that were identified using the symmetric half models.

Furthermore, the cavitation that supposedly occurs could be integrated into the computational fluid dynamics model in the future. This might reduce the negative pressure peaks further and lead to a more physical behavior of the fluid.

Another interesting aspect of the pump development is the leakage of hydraulic oil through small gaps. To capture such a phenomenon in a CFD simulation is a quite challenging task but would clearly lead to more realistic simulations. Especially high pressure leading to more leakage and thereby reducing the amount of fluid (and the pressure) is an important aspect.

Finally, for further research the pumps fatigue life might be investigated using Abaqus/Explicit. In [Osi2010] the fatigue life of the pump was calculated using the pressure that was obtained by CFD simulations.

References

- [Osi2009] Osiecki L., Piechna J., Zawistowski T. (2009) 'Computer analysis of dynamic phenomena in PWK-type variable pumps', *Proceedings of International Scientific-Technical Conference, Hydraulics and Pneumatics, Wrocław, Poland*.
- [Osi2010] Osiecki L., Piechna J., Zawistowski T., Zylinski B. (2010) 'Multidisciplinary Approach to the Component Design of a New Axial Hydraulic Pump', *Proceedings of the 1st Conference on Multiphysics Simulation, June 2010, Bonn*.
- [Osi2012] Osiecki L., Patrosz P., Zawistowski T., Landvogt B., Piechna J., Zylinski B. (2012) 'Compensation of Pressure Peaks in PWK-type hydraulic pumps', *Key Engineering Materials*, Vol. 490, pp. 33–44
- [Patent 0742 870] Osiecki A., Osiecki L. 'European Patent Nr 0742 870: Hydrostatic axial piston machine'
- [Wie2002] Wieczorek U., Ivantysynova M. (2002) 'Computer Aided Optimization of Bearing and Sealing Gaps in Hydrostatic Machines – The Simulation Tool', *International Journal of Fluid Power*, Vol. 3, No. 1