RESEARCH INTO THE INFLUENCE OF CRITICAL HYDRAULIC TANK PARAMETERS WHEN USING TWO PUMPS SIMULTANEOUSLY

DOI
https://doi.org/10.18690/um.fs.7.2025.15

ISBN
978-961-299-049-7

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Simulation of positive displacement machines requires highly skilled engineers, use of advanced simulation tools and advanced simulation approach. The paper presents recent activities and progress on simulation of positive displacement machines – in particular, the axial piston pump and the radial hydraulic motor. Despite that these machines have been designed and produced for decades, there are still (design) features and phenomena not being investigated in detail or never being simulated. The simulation advancements mainly refer to the application of complicated kinematic motion, fluid properties, physics to consider as well as mesh and numerical algorithm techniques. In this paper, the focus is on modelling of advanced fluid material properties. Numerical approach has been performed by means of CFD within the environment of Siemens Simcenter Star-CCM+. The anticipation of cavitation has been possible by implementation of existing "full cavitation model".

Keywords:
hydraulic tank,
two pumps,
interaction,
uneven flow distribution,
cavitation

University of Maribor Press

1 Introduction

Hydraulic devices are very common in many fields, in various industries, aviation, shipping, agriculture, construction and other fields. Hydraulic pumps are an indispensable component of hydraulic systems, as they ensure the flow of hydraulic fluid and thus perform work on the actuators side. Depending on their design and mode of operation, there are several types, the most common of which are gear pumps with external and internal gearing, vane pumps, axial piston pumps with swash plate or swash roller, radial piston pumps, etc. In closed circuits, the pumps delivered the liquid through pipes to the actuators. When the fluid has completed the circuit, it returns to the pump, from where a new circuit begins through the consumers. In open circuits, the hydraulic fluid is stored in tanks, from where it is pumped through pipes and to the consumers, and the path ends back in the tank. In this problem definition, we are focussing on an open circuit. In presented case, a hydraulic system with two gear pumps with internal gearing and two separate suction lines was considered. The larger pump has a displacement of 125 cm^3/rev, the smaller one 63 cm^3/rev. Under certain operating conditions, such a hydraulic system is likely to lead to interactions between the fluid flows at the inlet to the suction lines of the two pumps, and in extreme cases even to burning of the fluid between the pumps.

The lack of liquid or the presence of air pockets in the suction line will consequently lead to a reduced pumping capacity and an increase in the speed at the inlet to the suction line of this pump, as well as a corresponding drop in pressure due to the lack of liquid at the inlet. The air pockets are then transported with the flow into the pump, where the expansion and compression of the liquid causes cavitation and cavitation erosion of the material between the gears and the housing, which in extreme cases can also lead to pump failure.

The long-term presence of cavitation or the conditions under which cavitation occurs, lead to erosion of the surfaces exposed to the collapse of the vapour bubbles. This manifests itself as pitting on the surfaces where the bubbles have collapsed. Figure 1 shows an example of a pump with internal gearing that has failed due to cavitation erosion. The valve plate is shown, and the pitting is visible on the part of the plate where the opening for the discharge line begins. The reason for the occurrence of cavitation and the failure of the pump was too low a level in the otherwise large-volume but too low tank [1]. This led to surface turbulence, with which the pump sucked in air, and a vacuum was created in the suction line. This created vapour bubbles, which were transported forwards by the rotation of the pump until the static pressure of the liquid rose above the vapour pressure of the bubbles again, causing them to collapse on the pressure side and erode the surface over time.

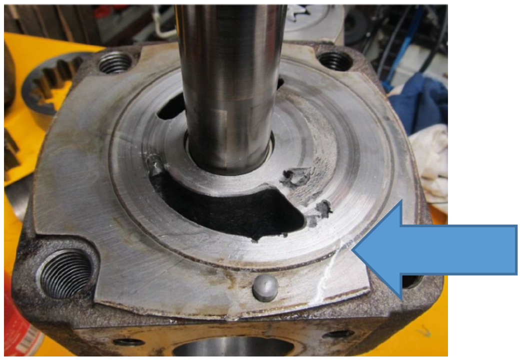

Figure 1: Internal gear pump with damaged valve plate due to cavitation erosion [1].

2 Materials and methods

In the following, the parameters of the tested hydraulic pump and the procedure with parameters for numerical calculations will be presented.

2.1 Tested internal gear type of hydraulic pumps

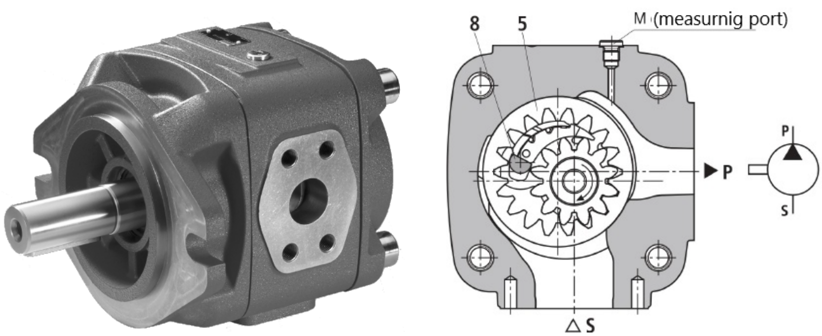

Figure 2: Tested hydraulic pump type PGH (Rexroth) [2].

Rexroth hydraulic pumps of the PGH type were used in the study (Figure 2) [2]. These are hydraulic pumps with internal gearing and constant displacement. This is 125 cm^3/rev for the larger pump and 63 cm^3/rev for the smaller pump.

2.2 Hydraulic reservoir with different suction ports for two pumps

In the experimental research we had to construct a suitable tank to investigate the influencing parameters. The model of the tank was created using the SolidWorks programme. Given the space available in the laboratory and the recommendations for the dimensions of the tank, we first established rough initial dimensions – 950 mm wide, 700 mm deep and 500 mm high – which we then used as the basis for the modelling. As we will be observing the inside of the tank, particularly the area around the suction pipes, during pump operation, the front of the tank is made of glass. The other sides and the bottom of the tank are made of sheet metal, material ST37 or S235, which is bent and then welded into sub-assemblies and into the overall assembly (Figure 3).

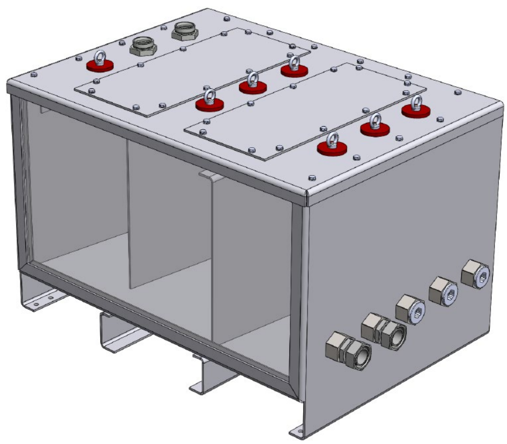

Figure 3: Special hydraulic tank for analysing the influence of the suction lines of two or more pumps (1050 mm x 750 mm x 670 mm).

2.3 Numerical calculation of the fluid dynamics of two pumps

In the following, we present the parameters of the simulations carried out in the Ansys Fluent software environment. This is a software package for computational fluid dynamics, hereinafter referred to as CFD. This is a discipline that deals with analysing flow systems, heat transfer and associated phenomena such as chemical reactions. It uses numerical methods and algorithms (e.g. finite difference methods, finite volume methods, etc.) to search for solutions based on input parameters.

The basic (guiding) equations that describe the motion of fluids in CFD are the Navier-Stokes equations. These are a series of partial differential equations that describe the conservation of mass and momentum of the fluid.

One of the most important steps before the actual numerical calculation of the mesh is the definition of turbulence models. Most of the flows we encounter in technical practise are turbulent. Such a flow is characterised by sudden fluctuations in velocity and a disordered and rapid flow movement due to the formation of vortices. These represent an additional mechanism of energy and momentum transfer which, in contrast to molecular diffusion, occurs much faster in laminar flow, which consequently leads to higher values for friction, mass and heat transfer.

The geometric model for the simulation studies was created using the SolidWorks programme. Several models with individual differences were defined according to the simulated influencing parameters. We started from the dimensions shown in Figure 4. It is a section of a part of the suction pipes of the tank. The geometric model consists of two bodies to enable an efficient network.

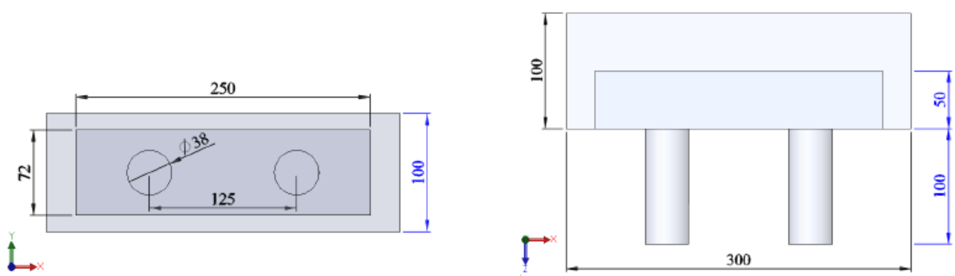

Figure 4: Dimensions of the geometric model of the simulation research (all dimensions are in mm).

The mass flow rate at the outlet surfaces of both pipes was determined (shown in red in Figure 3) as the outlet boundary condition. Here, the same volume flow was assumed as specified in the manufacturer's catalogue [2] for pumps, measured at

n = 1450 min^{-1}. Using the known volume flows of both pumps and the known density of the oil, the mass flow rate for each pump was calculated, according to the expression $\dot{m} = q_V\,\varrho$. This amounts 2.583 kg/s for the pump with a displacement of 125 cm^3/rev, and 1.2915 kg/s for the pump with a displacement of 63 cm^3/rev. The mass flows are shown in Figure 5 as red arrows pointing out of the range.

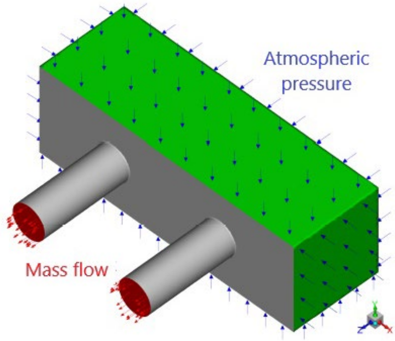

Figure 5: Boundary conditions of the simulations of two suction lines.

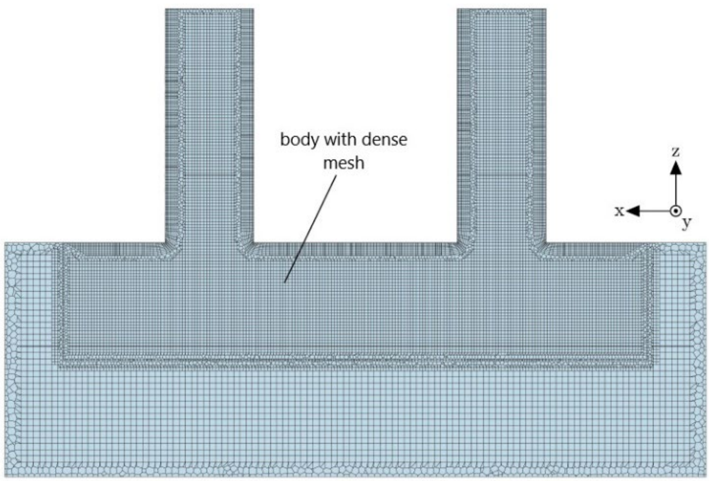

Figure 6: Meshed geometric model in the xz cross-sectional plane [9].

The geometric models were meshed with the CFD tool Fluent (With Fluent Meshing) of the Ansys software. The poly-hexcore meshing was used, which is one of the mosaic meshing technologies. It is a combination of polyhedral and hexahedral control volumes, where polyhedral volumes fill the outer region near the wall (boundary layer) and allow the description of more complex parts of the geometry, while hexahedral volumes fill the inner region of the volume [3]. In this case, each hexahedron at each mesh density is subdivided into eight smaller control volumes called octrees (from the Greek word octo – eight and the English word tree) in a tree structure. Numerous studies and articles [4 to 8] have shown that the use of polyhexkernel meshing leads to a lower number of control volumes, higher mesh quality, shorter computation time and higher solution accuracy (Figure 6) [9].

3 Results

Here, the results of a study on the effects of different distances between two suction pipes and the influence of oil level will be presented.

3.1 Numerical results

Figure 7 shows the distribution of velocity fields for case of two suction lines at a distance of 125 mm. The left suction line belongs to a pump with a displacement of 63 cm^3/rev, the right one to a pump with a displacement of 125 cm^3/rev. It can be seen that the velocity fields of both suction lines curve or tilt in the direction of the neighbouring line. The pressure curves shown in Figure 8 show the same, whereby each is shown with its own scale due to the large differences in the pressure values of the two suction lines. The curvature or tilting of the velocity lines would mean that the velocity and pressure gradients are no longer symmetrical to the longitudinal axis (z) due to the mutual influence of the suction lines, but are greater on the side that is closer to the other suction line. This can also be seen from the velocity profiles in Figure 9.

The calculation of the difference in the absolute values of the gradients of the velocity profiles along the walls of one suction pipe and the other also proves the asymmetry.

Figure 9 shows the flow lines at the inlet of the suction lines. Due to their proximity, the interaction of the flow lines creates a "virtual liquid wall" between the delivery areas of the two pumps. This prevents liquid from being pumped through from the delivery area of the other pump.

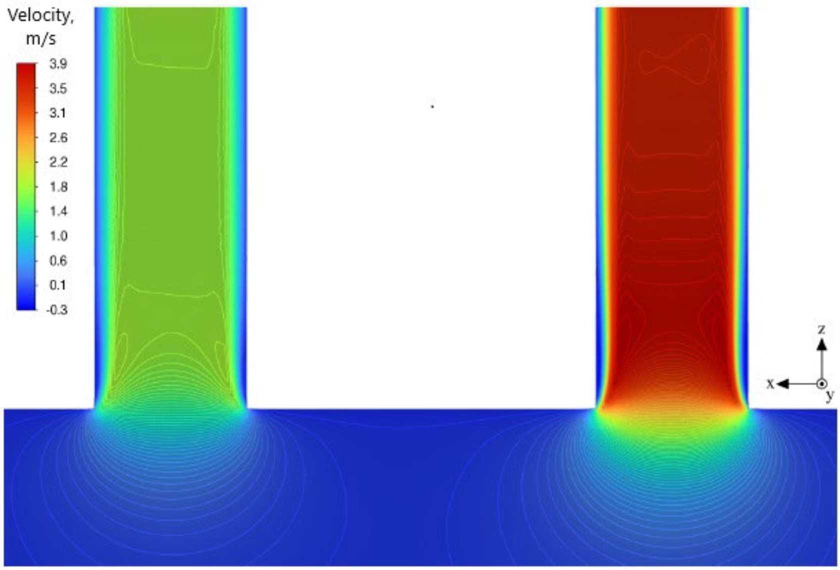

Figure 7: Velocity fields of two suction lines at a distance of 125 mm

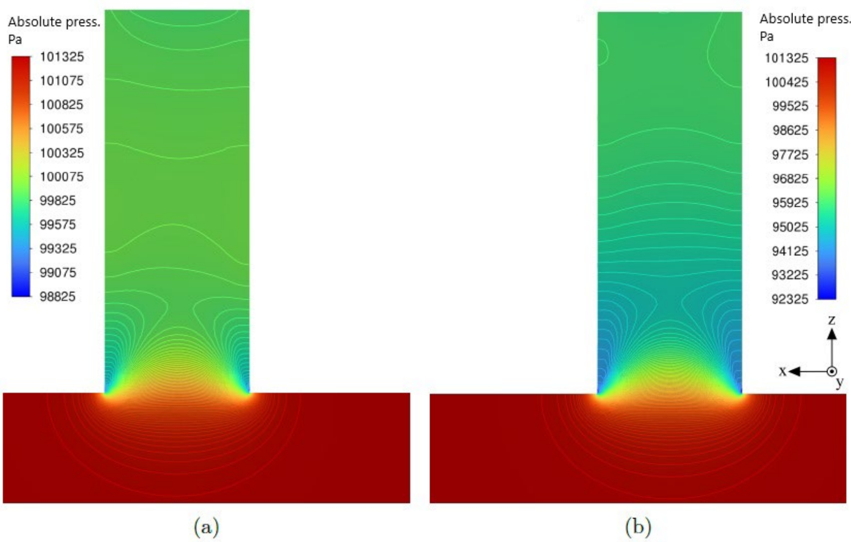

Figure 8: Absolute pressure curves of a pump with a displacement of 63 cm³/rev (a) and 125 cm³/rev (b) with suction lines at a distance of 125 mm.

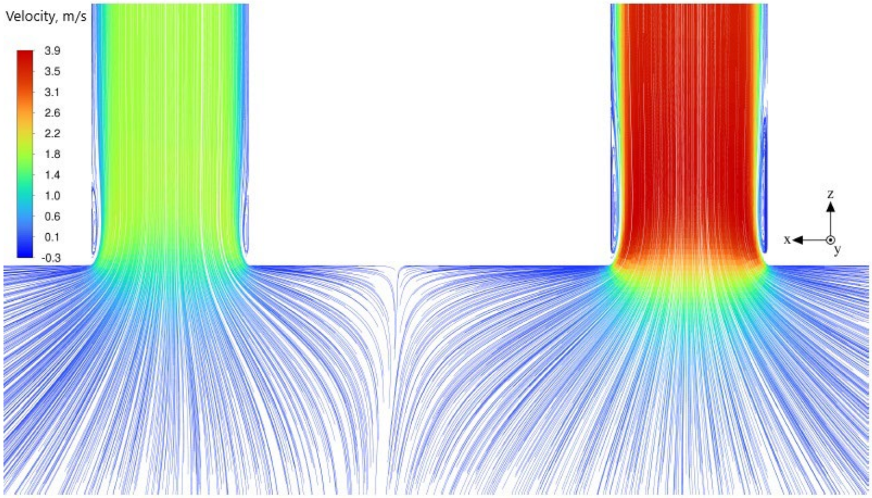

Figure 9: Streamlines at the inlet of the suction lines at a distance of 125 mm.

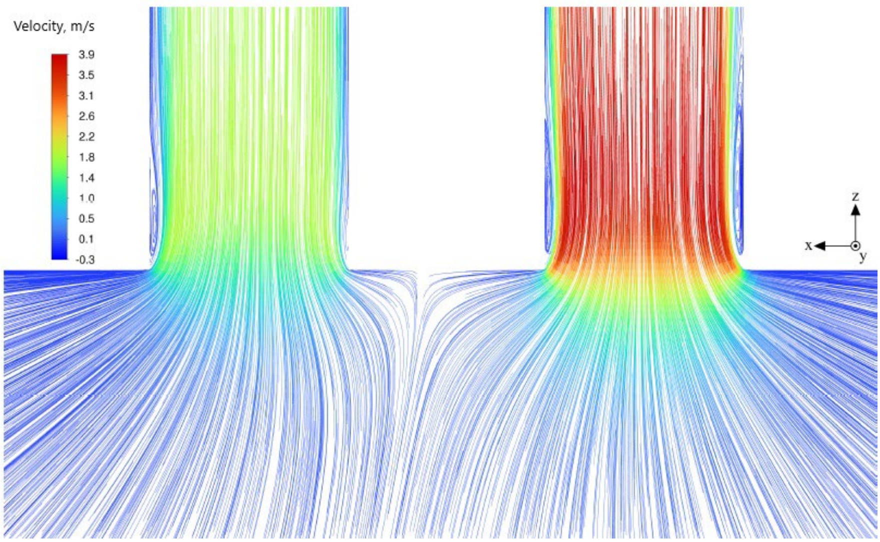

Figure 10: Streamlines at the inlet of the suction lines at a distance of 75 mm.

Due to the uneven power distribution between the suction lines, the pump with a displacement of 125 cm^3/rev has a greater influence on the liquid particles that would otherwise follow the nozzles into the suction line of the pump with a displacement of 63 cm^3/rev. The larger pump therefore interferes with the delivery area of the smaller pump and prevents a normal flow of liquid. The apparent wall of the nozzles is therefore not straight but curved in the direction of the pumping area of the smaller pump. This leads to an asymmetrical flow of liquid into the suction lines. The result is a thinning of the stagnant recirculation area along the wall, as can be seen in Figure 10 and Figure 11 - the recirculation area of the nozzles in the suction line of the smaller pump almost disappears in this part, while it starts to decrease in the case of the larger pump.

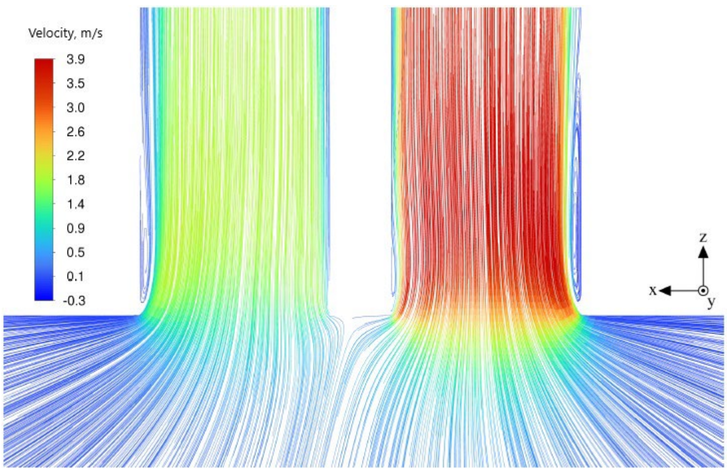

Figure 11: Streamlines at the inlet of the suction lines at a distance of 50 mm.

3.2 Experimental results

Table 1 summarises the average values of the measured absolute pressure of both sensors with the associated expanded uncertainties of type A as well as the maximum and minimum pressure values for all 4 distance measurements.

The maximum pressure drop or the lowest absolute pressure due to the change in suction conditions, as a result of the mutual influence of the suction lines is given at the smallest distance of 125 mm. Compared to the next largest distance of 250 mm, the pressure drop in the suction lines of both pumps is 800 Pa greater and the absolute pressure is correspondingly lower. The maximum absolute pressures of both pumps are also 1400 Pa lower, and the minimum absolute pressures are also lower, namely 1300 Pa for the larger pump and 1000 Pa for the smaller pump. Above a distance of 125 mm, the pressure changes in the suction lines of both pumps are no longer uniform and begin to fluctuate around stable values. From this it can be concluded that above a distance of 125 mm the mutual influence of the suction lines is no longer present or is negligible.

Table 1: Results of the absolute pressure influence of the suction line distance and the associated type A measurement uncertainties.

	125 mm	250 mm	375 mm	500 mm
p_1, Pa	64200 ± 11	65400 ± 9	65800 ± 13	65300 ± 11
p_2, Pa	80300 ± 13	81100 ± 8	81000 ± 6	81100 ± 8
$p_{1,\mathrm{maks}}$, Pa	65800	67200	67200	67200
$p_{1,\mathrm{min}}$, Pa	62200	63500	63500	63500
$p_{2,\mathrm{maks}}$, Pa	81600	83000	82500	83000
$p_{2,\mathrm{min}}$, Pa	79300	80300	80300	80300

A reduction in the oil volume in the tank or a lowering of the level has a negative effect on the suction conditions. The already low oil volume at a given flow rate is reduced even further, resulting in turbulent oil and a turbulent level. In addition, with high flow rates and oil turbulence and a lower fill level in the tank, it often happens that the suction power of the pumps creates a localised vacuum in the oil level. This creates a vortex that is sucked into the suction line. This vortex forms and disappears within 1 to 2 seconds, as shown in Figure 12.

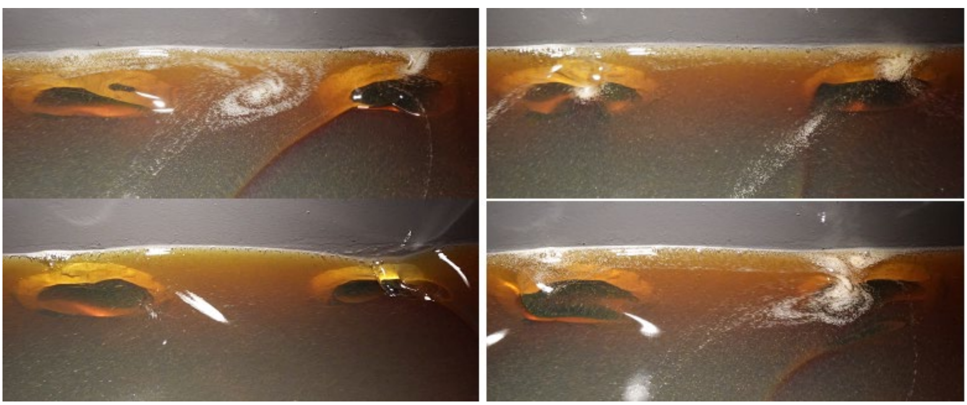

Figure 12: Pumping air from the surface into the suction lines – view of the tank from above.

4 Conclusions

In the present study, we determined numerically and experimentally the conditions for the interaction of two pumps mounted outside the tank with two separate suction lines and flow rates of 125 cm^3/rev and 63 cm^3/rev. During the experiments, we used a glass front to provide a real-time view of what was happening inside the tank, which contributed to more efficient and high-quality results.

The main results and findings are:

- The phenomenon of interaction of the suction lines of two hydraulic pumps was confirmed by numerical situations.
- For the experimental studies, an experimental test rig with an open circuit was designed, as well as a tank, suitable for measuring different mutual distances and other variable parameters.
- In the case of using two hydraulic pumps with internal gears and a corresponding displacement of 125 cm^3/rev and 63 cm^3/rev and two separate suction lines, we carried out pressure measurements in the suction lines at the pump inlet.
- It was found out that the mutual distance has a great influence on the flow field near the suction lines and on the non-uniform flow distribution. In our particular case, using these two pumps, the mutual distance between the suction pipes over 125 mm no longer plays a major role.
- It was experimentally confirmed, that very undesirable conditions occur when the volume is reduced and especially when the liquid level in the tank is reduced. If the volume is too low, the oil swirls strongly, and too low a level also leads to the absorption of air from the surface in the form of funnels. Air in the oil consequently leads to conditions for cavitation to occur, which can have negative consequences for efficiency and damage to pumps in the event of long-term operation without monitoring or troubleshooting.

Acknowledgments

We would like to thank the programme group P2-0231 and the agency ARIS for the financial support of the projects L2-4474, L2-2618 and L2-50083. We would also like to thank Mr Ludvik Karmuzel from the company Nord Fluid for the motivation for this work and also for the rented pumps.

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