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Stefan Posch
Institute for Internal Combustion Engines and Thermodynamics, Austria, posch@ivt.tugraz.at

Johann Hopfgartner
Institute for Internal Combustion Engines and Thermodynamics, Austria, hopfgartner@ivt.tugraz.at

Martin Heimel
Institute for Internal Combustion Engines and Thermodynamics, Austria, heimel@ivt.tugraz.at

Erwin Berger
Institute for Internal Combustion Engines and Thermodynamics, Austria, berger@ivt.tugraz.at

Raimund Almbauer
Institute for Internal Combustion Engines and Thermodynamics, Austria, almbauer@ivt.tugraz.at

See next page for additional authors

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Authors
Stefan Posch, Johann Hopfgartner, Martin Heimel, Erwin Berger, Raimund Almbauer, and Peter Schöllauf

A Numerical Investigation of the Oil Pump Suction Behaviour in a Hermetic Reciprocating Compressor

Stefan POSCH[1]*, Johann HOPFGARTNER[1], Martin HEIMEL[1], Erwin BERGER[1], Raimund ALMBAUER[1], Peter SCHÖLLAUF[2]

[1] Institute of Internal Combustion Engines and Thermodynamics, Graz University of Technology, Inffeldgasse 19, 8010 Graz, Austria

posch@ivt.tugraz.at, +43 316 873 30233
hopfgartner@ivt.tugraz.at, +43 316 873 30240
heimel@ivt.tugraz.at, +43 316 873 30235
berger@ivt.tugraz.at, +43 316 873 30234
almbauer@ivt.tugraz.at, +43 316 873 30230

[2] Secop Austria GmbH, Jahnstraße 30, 8280 Fürstenfeld, Austria
p.schoellauf@secop.com, +43 3382 5010 984

* Corresponding Author

ABSTRACT

In addition to the adequate lubrication of the moving parts, the oil flow in a hermetic reciprocating compressor has a significant influence on the thermal characteristics of a compressor. The present work is concerned with the investigation of the oil pump system of a reciprocating hermetic compressor used in household refrigeration appliances. The considered oil pump system consists of a centrifugal pump immersed in the oil sump of the hermetic compressor and a helical groove machined on the crankshaft. The focus of this work lies on the immersed part of the centrifugal oil pump and its interaction with the oil in the oil sump. To analyse the flow in the immersed area of the oil pump, the commercial computational fluid dynamics (CFD) software ANSYS Fluent is used. The free surface of the oil flow is modelled with the Volume of Fluid (VOF) method. A numerical investigation is used to study the influence of the immersion depth and the oil pump design on oil mass flow rate and flow field at the oil pump intake. To evaluate the oil pump regarding the applicability in variable speed compressors, the influence of the rotational speed on the oil mass flow is also explored.

1. INTRODUCTION

The most common compressor type used in domestic refrigeration appliance is the hermetic reciprocating compressor. This kind of compressor uses a reciprocating piston driven by a slider-crank mechanism for suction, compression and discharge of the refrigerant. Small-scale hermetic reciprocating compressors have a crankshaft which is directly driven by the electric motor. In addition to the energy transfer from the motor to the piston, the crankshaft acts also as oil pump device. Depending on the compressor type and the required amount of oil, different oil pump designs are used which differ in the arrangement of centrifugal and helical pumps. Several studies are available which deal with the investigation of single pumping parts or the entire oil pump, respectively. A numerical study of a compressor oil pump using commercial CFD software can be found in Lückmann *et al.* (2009) and Kerpicci *et al.* (2013). The authors used the Volume of Fluid (VOF) technique to model the two-phase flow in three dimensions. Besides the calculation of the oil mass flow rate, specific focus was laid on the determination of the oil climbing time. To keep the number of cells and therefore the computational time under a tolerable limit, several simplifications like the neglect of the gap between shaft and crankcase in the helical grove have to be made. Kim *et al.* (2002) used the analogy between the oil supply system of a reciprocating compressor and an electric circuit. The authors described the pumping parts by equivalent electric elements to get a mathematical model of the system. A validation of the simulation results by experimental data showed good agreement in terms of the oil flow rate. The fluid flow in helical pumps using analytical models is analysed in the works of Alves *et al.* (2009 and 2010). Thereby, the Navier-Stokes equations were adapted to a finite channel with specific boundary conditions. The gap between shaft and crankcase was neglected in these studies. Different techniques like generalized integral transform technique have been presented to solve the resulting equation. A more detailed investigation of helical groove oil pumps was presented by Posch *et al.* (2015). In this study, the oil flow in channel direction is assumed to be fully developed and the 2d flow in the channel cross section has been calculated considering the gap between shaft and crankcase. The pressure gradient in channel direction can be considered to get the ability to couple the helical groove with other pumping parts. In Tada *et al.* (2014) an uncoupled simulation of the helical pump of a reciprocating compressor using commercial CFD software is shown. The method has been utilized to analyse the applicability of an oil pump for the use in variable speed compressors. The authors calculated the volumetric oil flow depending on the rotational speed and immersion depth of the shaft in the compressor oil sump. Another example of the use of commercial CFD software to analyse the performance of a compressor lubrication system can be found in Ozsipah *et al.* (2014). In addition to the investigation of the influence of geometric parameters on the pumped oil mass flow, the suction behaviour of the oil pump was analysed. Therefore, the oil sump of the compressor was also modelled, assuming that the oil expelled from the oil pump does not flow back into the oil sump. A calculation of the oil mass flow rate, transient oil distribution and pressure fluctuations at the pump inlet, depending on geometrical parameters, was carried out.

The present paper focuses on the investigation of the suction behaviour of a hermetic reciprocating compressor oil pump. Therefore, an uncoupled simulation of the immersed centrifugal pump interacting with the oil sump using commercial CFD software ANSYS Fluent is carried out. To increase the accuracy of the simulation, the expelled oil is assumed to be transported back into the oil sump. The approach is utilized to get an insight in the suction behaviour of the oil pump in dependence on the immersion depth, pump design and rotational speed. Additionally, an assessment according to the oil mass flow rate of the different cases is presented.

2. MODELLING

2.1 Theoretical background

Computational fluid dynamics (CFD) simulation is based on the solution of the flow governing equations using numerical methods. The governing equations for mass and momentum are as follows (Moukalled *et al.*, 2015):

$$\frac{\partial \rho}{\partial t} + \nabla \cdot [\rho \boldsymbol{v}] = 0 \tag{1}$$

$$\frac{\partial}{\partial t}[\rho \boldsymbol{v}] + \nabla \cdot \{\rho \boldsymbol{v}\boldsymbol{v}\} = -\nabla p + \mu \nabla^2 \boldsymbol{v} + \boldsymbol{f}_b \tag{2}$$

The simulation domain is divided into a finite number of cells, the so called control volumes. The governing equations are spatial discretized on the control volumes using the finite volume method to get a linear system of equations which is solved by mathematical methods. Commercial CFD software like ANSYS Fluent, which is used in the present study, executes these steps automatically.

Two phase flows with sharp interface are calculated with the volume of fluid approach (VOF) by Scardovelli and Zaleski (1999) which is implemented in ANSYS Fluent. The VOF approach models two or more immiscible fluids by tracking the volume fraction of each fluid in the domain, solving the volume fraction equation (3). Fluid properties like density or viscosity can be determined by summing up the volume fraction weighted values (4 and 5).

$$\frac{\partial \chi}{\partial t} + \frac{\partial (u_i \chi)}{\partial x_i} = 0 \tag{3}$$

$$\rho = \rho_g \chi + \rho_l (1 - \chi) \tag{4}$$

$$\mu = \mu_g \chi + \mu_l (1 - \chi) \tag{5}$$

Further information about the VOF method can be found in the work of Scardovelli and Zaleski (1999) and the Fluent User Guide (2011).

2.2 Simulation domain

The investigated compressor oil pump consists of three single pump parts: a helical pump and two centrifugal pumps at the inlet and outlet of the crankshaft, respectively. To increase the oil mass flow rate, the considered oil pump is designed with a spiral sheet (catcher) at the pump inlet. The aim of the present study is to investigate the suction behaviour of the oil pump and the flow conditions at the pump inlet. Therefore, an uncoupled simulation of the centrifugal pump at the crankshaft inlet in combination with a simplified oil sump is carried out. The computational mesh is created with about 250.000 cells for the pump and about 400.000 cells for the sump, respectively. The mesh is refined at interfaces, inlet and outlet to increase simulation accuracy. Figure 1 shows the position of the centrifugal pump in the crankshaft (a) and the computational mesh (b).

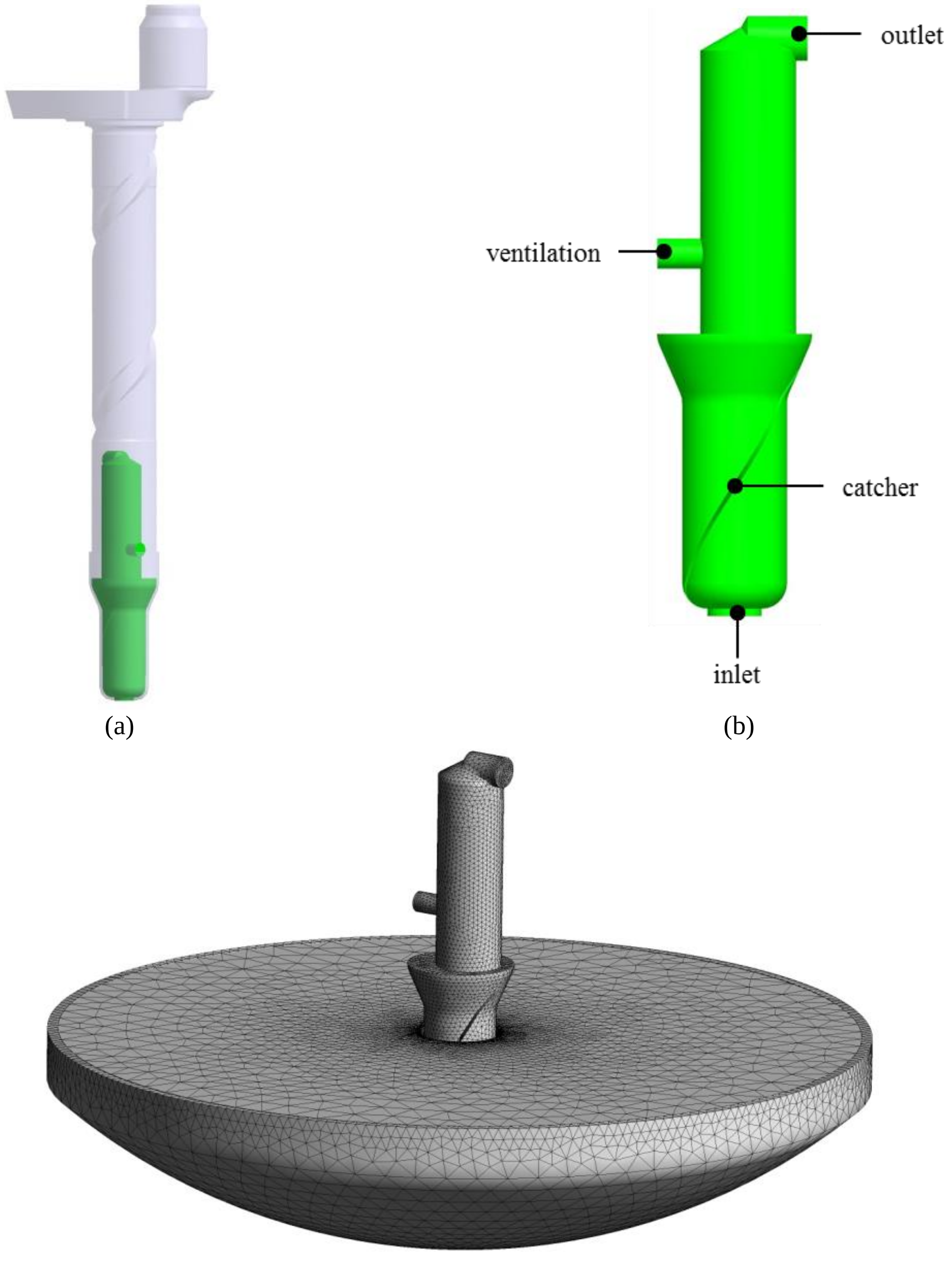


Figure 1: Position of the centrifugal pump (a), geometrical domain (b) and computational mesh (c)

To fulfil the overall mass balance, the oil at the outlet of the pump is filled back into the oil sump at the outer area of the oil sump surface to reproduce the wall oil flow. A mass flow boundary condition which is linked to the pump outlet via user defined function (UDF) is applied on the outer area of the oil sump surface. On the rest of the oil sump surface, a pressure inlet boundary condition with zero relative pressure is applied. At pump outlet and ventilation hole, a pressure outlet boundary condition is used with also zero relative pressure. To decrease computational time, the pump is filled with oil according to the current immersion depth at the initialization process. The simulation is carried out until pressure at the pump/sump interface and oil mass flow at the pump outlet show a steady-state behaviour.

2.3 Assumptions and Settings

The following assumptions are made for the simulation of the centrifugal pump and the compressor oil sump:

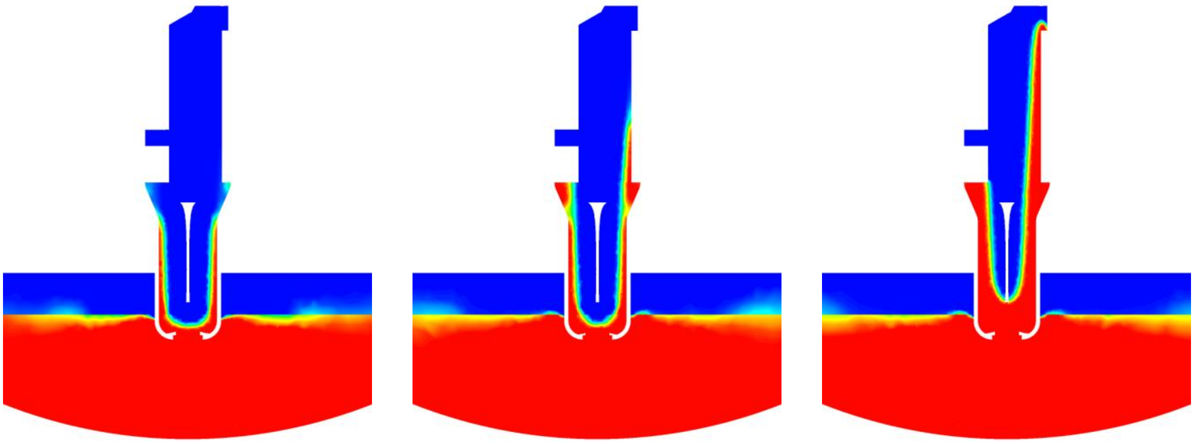
- Isothermal flow.
- Acceleration at the compressor start-up is infinite.
- Air is used for the gaseous phase.
- Dissolving of gaseous phase in liquid phase is neglected.
- Physical properties of the fluids are constant.
- Laminar flow.

The oil density is set to 832 kg m^{-3} and the oil viscosity is set to 6.7 x 10^{-3} Pa s. Surface tension is considered and is set to 0.028 N m^{-1}.

The simulation is carried out using implicit VOF method in combination with bounded second order implicit time formulation. The spatial discretization of the momentum equation is utilized with a second order upwind scheme (Moukalled *et al*., 2015) and, for the volume fraction equation, the compressive scheme is used. The pressure-velocity coupling is executed with the PISO (pressure-implicit with splitting of operators) algorithm (Isaa, 1986). Time step is set to 10^{-4} s and double precision mode is utilized.

3. RESULTS

The important properties to value the functionality of oil pumps for the usage in hermetic compressors are the oil mass flow rate, the oil climbing time and the oil distribution. To quantify the influence of the immersion depth, the rotational speed and the pump design on the mentioned parameters, a basic configuration with 7 mm immersion depth and 3000 rpm rotational speed in combination with the original geometry is used.
Figure 2 shows the oil volume fraction depending on the immersion depth at different time steps during the compressor start-up until and after reaching stead-state conditions. The figures show the development of the oil parabola due to the centrifugal forces. Although the variant with the highest immersion depth gives the highest static pressure at the pump inlet due to the hydrostatic pressure, the oil parabola sinks nearly to the pump inlet at the beginning of compressor start-up. This can be explained by the increasing friction forces on the outer surface which induce centrifugal forces. The centrifugal forces reduce the static pressure at the pump inlet. If the oil parabola would reach the pump inlet, the oil flow would break down immediately.



0.12s (a) 0.24s (b) 1s (c)

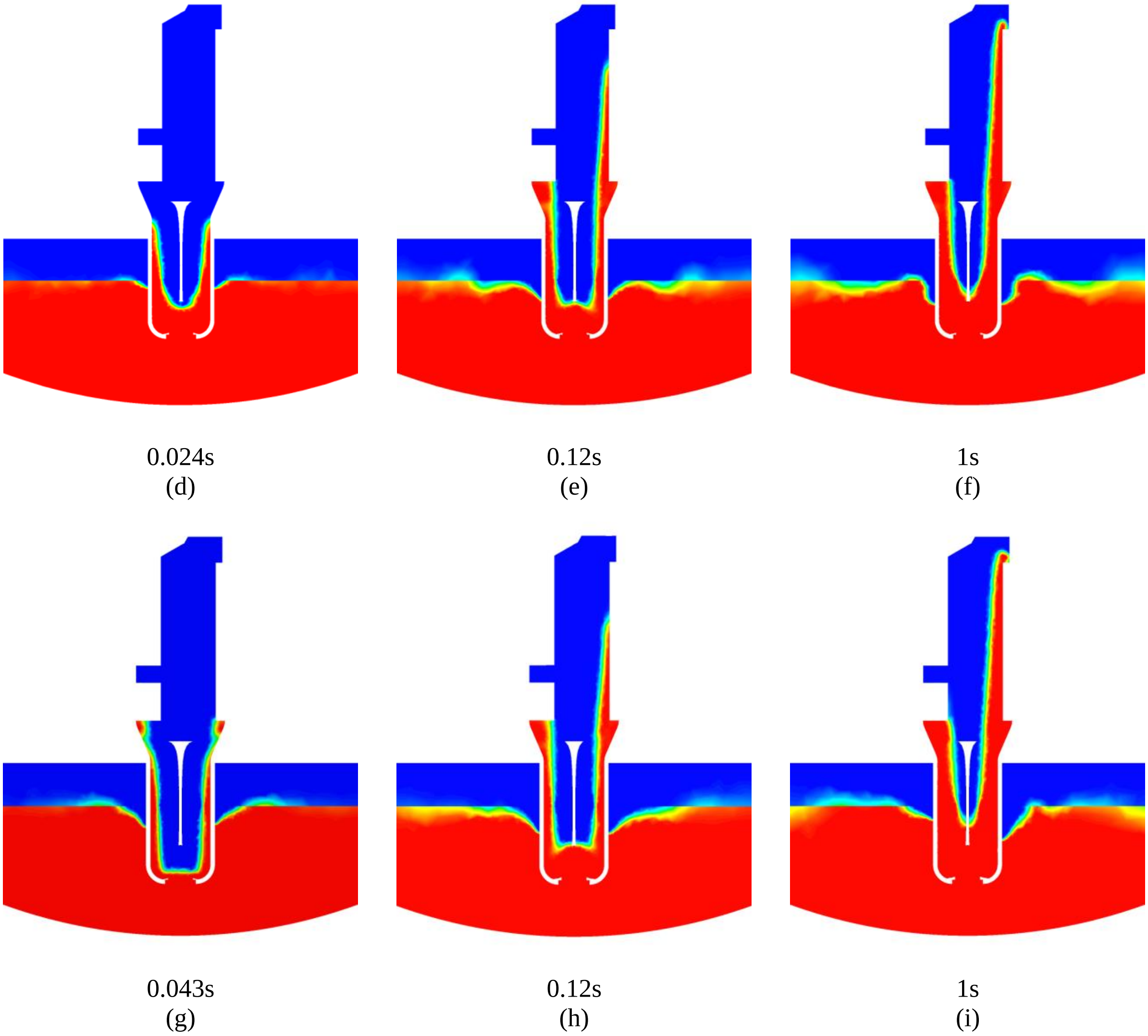
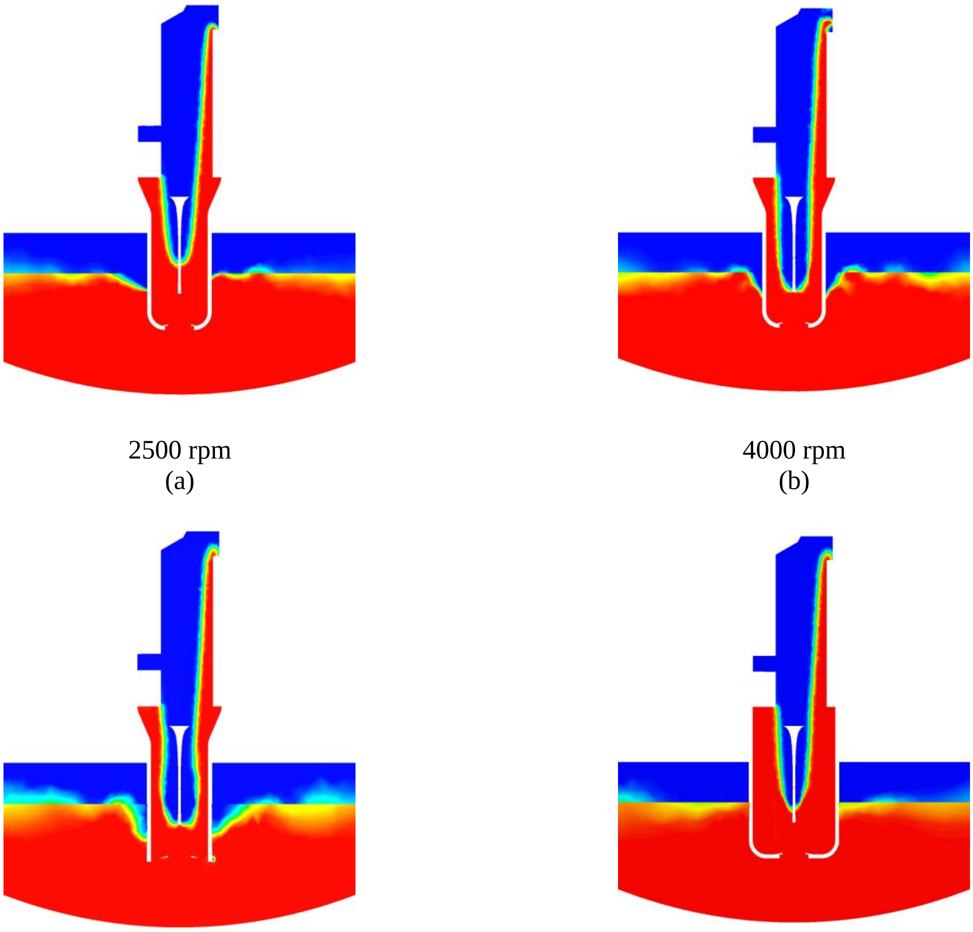


Figure 2: Oil volume fraction of 3 mm (a) - (c), 7 mm (d) - (e) and 9 mm (g) - (i) immersion depth at different flow time

The oil volume fraction at steady-state conditions for rotational speed variations and two different design variants can be seen in Figure 3. The results of the simulation show the increasing steepness of the oil parabola at higher rotational speed which leads to a lower oil level inside the pump. As already mentioned, if the oil parabola reaches the pump inlet, the oil mass flow would break down. Figure 3 (c) and Figure 3 (d) show two different design variants. Design 1 has a larger pump inlet diameter compared to the original geometry. Design 2 has the same diameter at the pump inlet, but compared to the original geometry, the conical shape is replaced by a cylindrical shape, thus the volume in the lower region of the pump is larger. The oil volume fraction of pump design 1 shows an uneven shape of the oil parabola due to the influence of the spiral sheet which interacts with the oil in the sump. A comparison between design 2 and the original geometry shows similar shape of the oil parabola and an increased amount of oil in the catcher due to the higher volume of design 2.



2500 rpm
(a)

4000 rpm
(b)

design 1
(c)

design 2
(d)

Figure 3: Steady-state oil volume fraction

Figure 4 shows the pressure distribution and the flow field of the basic configuration in the vicinity of the pump inlet. It can be seen that the main flow of the oil is situated around the centreline of the pump until the flow develops the oil parabola. Apart from the centreline, hardly oil is transported, thus the oil in that region undergoes only a rotational movement. The tangential velocity at the outer surface of the pump is a function of the distance from the centreline and increases with growing radius. Due to the higher velocities, the pressure decreases in the vicinity of the surface and oil is transported to low pressure regions causing vortexes. The vortexes spread out in the oil sump and can cause acoustic problems. High velocities at the inlet of the pump result in a decrease of the static pressure. Table 1 gives an overview of the static pressure reduction for the observed variants compared to the analytically determined geodetic pressure. It can be pointed out, that the major part of the static pressure reduction occurs due to the flow in axial direction. The influence of tangential and radial flow on the reduction of the static pressure can be considered as efficiency loss. Especially higher rotational speed induces higher tangential and radial velocities at the pump inlet due to higher frictional forces.

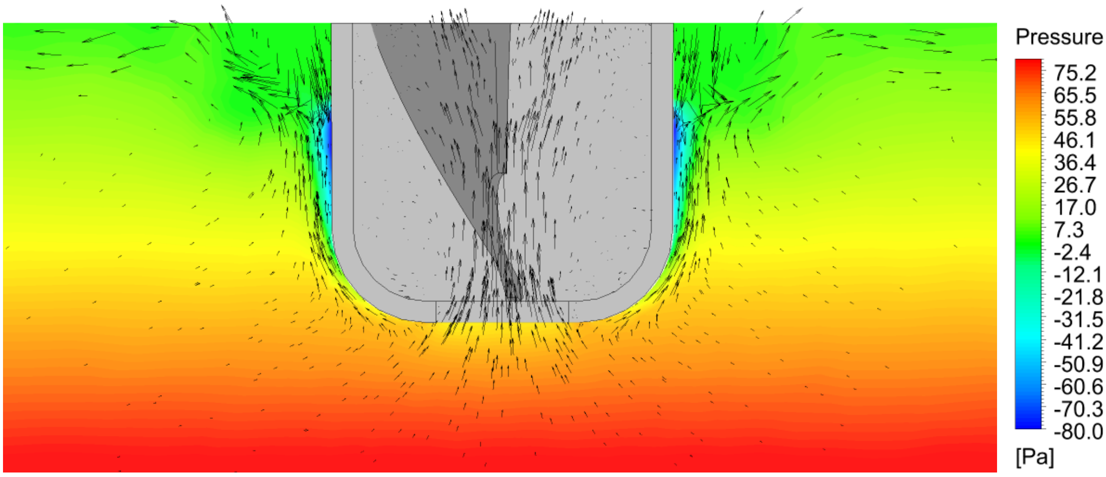


Figure 4: Pressure distribution and flow field in the vicinity of the suction hole

The Figures 5, 6 and 7 illustrate the mass flow rate at pump outlet as a function of time of the investigated variants. Simulated mass flow rates are normalized to the basic configuration. Variations of the immersed depth of the oil pump show the dependence of the oil mass flow rate on the immersion depth and therefore on the static pressure. Reduction of the immersion depth of ~50 % yields a decrease of the mass flow rate of 20 % and an increase of the oil climbing time of 0.4 s. The results also show that higher immersion depths do not affect the oil climbing time significantly but yield higher oil mass flow rates as expected. The comparison between the original geometry, design 1 and design 2 shows deviations of the oil mass flow rate of about 10 %. A shorter oil climbing time could be achieved with design 1 compared to the original geometry. Design 2 requires significant longer time for the oil reaching the pump outlet. This can be explained due to the higher volume in design 2 which has to be filled with oil until it is furtherly transported.

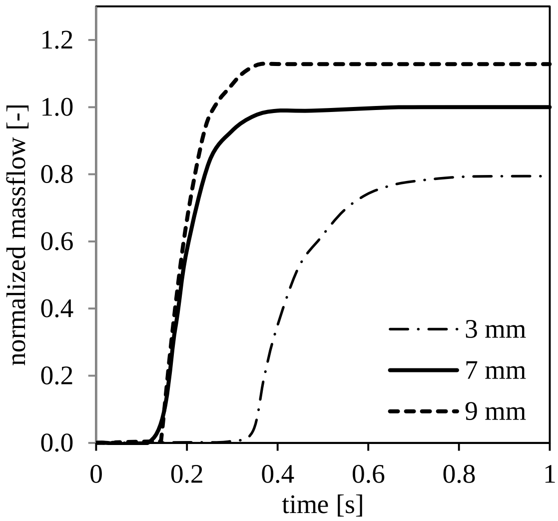


Figure 5: Normalized oil mass flow as a function of immersion depth

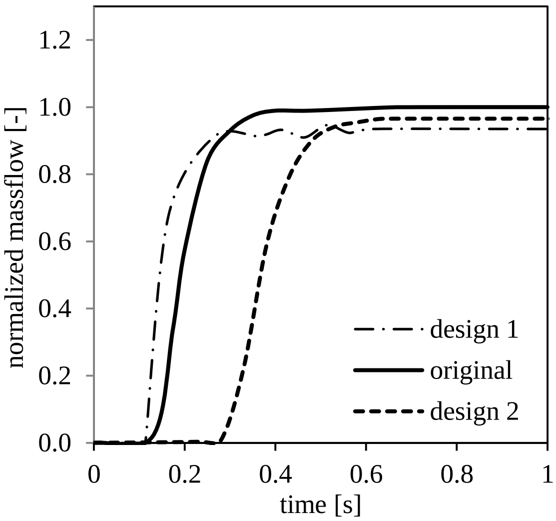


Figure 6: Normalized oil mass flow as a function of pump design

Rotational speed variations influence the oil mass flow rate and the oil climbing time in the same intent. Increasing the rotational speed by a third yields an increase of the oil mass flow rate by a factor of 2.2. The results also show the behaviour of the oil pump at low rotational speeds. A reduction of 500 rpm compared to the basic configuration results in an oil mass flow decrease of 60%. Simulations with reduced rotational speed show that no oil is transported under 2000 rpm, thus the use of the investigated pump design in variable speed compressors is unsuitable.

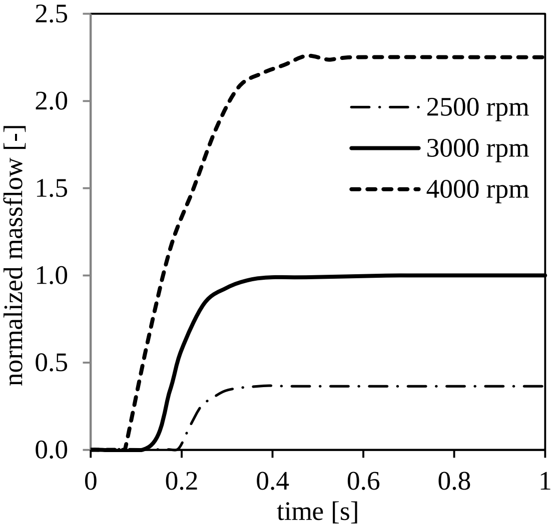


Figure 7: Normalized oil mass flow as a function of rotational speed

Table 1: Reduction of the static pressure between analytic (geodetic pressure) and numerical (CFD) calculation

Variant	Value [%]
basic model	21.0
3 mm immersion depth	45.5
9 mm immersion depth	24.1
2500 rpm	9.2
4000 rpm	71.2
design 1	55.3
design 2	52.5

4. CONCLUSION

A numerical investigation of the suction behaviour of the oil pump of a hermetic compressor is shown. The simulations are carried out using the commercial software ANSYS Fluent and they concentrate on the immersed centrifugal pump and its interaction with the oil sump. Special focus lies on the oil volume fraction in the pump, the oil mass flow rate, the oil climbing time and the static pressure at pump inlet. After analysing the simulation results the following conclusions can be drawn:

- Especially in the start-up phase of the compressor, the oil parabola can sink significantly which has to be taken into account in the pump design.
- Frictional forces reduce the static pressure at the pump inlet due to high radial and tangential velocities which occurs at high rotational speed.
- Vortexes are developed on the immersed pump surface which are spread out in the oil sump and can lead to acoustic problems.
- Volumes in the lower area of the pump should be kept small to decrease the oil climbing time considering the development of the oil parabola.
- The investigated oil pump is not suitable for the use in variable speed compressors because the oil transport fails below 2000 rpm.

NOMENCLATURE

$\boldsymbol{f}_b$	body force vector	(N/m³)
p	pressure	(N/m²)
t	time	(s)
u_i	velocity component	(m/s)
$\boldsymbol{v}_i$	velocity vector	(m/s)
x_i	coordinate component	(m)

Subscripts

g	gaseous phase	
l	liquid phase	

Greek symbols

μ	dynamic viscosity	(N s/m²)
ρ	density	(kg/m³)
χ	gas volume fraction	(-)

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