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Fluid Flow in the Oil Pumping System of a Hermetic Compressor

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Abstract. This work deals with the investigation of the oil pumping system in hermetic compressors for refrigeration application. The oil pump which is used for this study consists of two pumping areas: the lower pumping area with a pick-up tube and an eccentric bore, and the upper pumping area with a helical groove. This study focuses on the helical groove in the upper pumping area. To analyse the fluid flow in the helical groove, a numerical approach is introduced. In this approach the Navier-Stokes equations are adapted to the problem and are solved by using the finite volume method. Compared to analytical models, this method is able to obtain the flow field in the cross section of the helical groove at higher resolution. The higher geometrical resolution also enables the analysis of the flow in the small gap between the rotating crankshaft and the stationary wall. The present method is used to quantify different operating parameters on the oil mass flow rate.

Nomenclature

Roman symbols

e	gap width
g_x	body force in *x*-direction
g_y	body force in *y*-direction
g_z	body force in *z*-direction
n	rotational speed
p	pressure
u	velocity component in *x*-direction
v	velocity component in *y*-direction
w	velocity component in *z*-direction
x	channel cross-section coordinate
y	channel cross-section coordinate
z	down channel coordinate

Greek symbols

α	helix angle
μ	dynamic viscosity
ρ	density

1. Introduction

The oil supply in hermetic reciprocating compressors for refrigeration application is usually provided by integrated pumping devices on the compressor crankshaft. Two different pumping systems are widely used in hermetic compressors: the helical pump and the centrifugal pump. These two pumping systems can be used separately or in combination. Based on the significant influence of the oil supply system on the reliability and the thermal behaviour of a compressor, a number of studies has been carried out and published which deals with the investigation of this system. In addition to experimental investigations, mathematical approaches are used to describe the fluid flow of oil pumping devices.

These approaches can be divided into analytical (or semi-analytical) methods and methods based on computational fluid dynamics (CFD).

Numerical studies using commercial software (FLUENT) were carried out in [1] and [2]. The two-phase flow was simulated in three dimensions with the Volume of Fluid technique. The authors investigated characteristic parameters for the operation behaviour like the oil climbing time, the time to reach steady-state conditions or the oil mass flow rate. The focus in these studies was on the whole oil supply system which resulted in a significant large number of grid cells and a high computational time. Therefore, CFD simulations of the whole oil path are used to investigate existing models and they are not appropriate for the use in the design phase of hermetic compressors. A simplified uncoupled simulation model was presented in [3]. The authors split up the oil path and simulated only the helical pump using commercial software (CFX). The results of the calculated oil mass flow rate depending on the compressor speed and the immersion depth showed good agreement with experimental data in addition to a relatively low computational time of two hours.

Several analytical or semi-analytical models to describe the oil flow in hermetic reciprocating compressors were published in the open literature. In [4], a direct analogy between compressor oil flow and an electric circuit has been utilized. The authors analogized the individual lubrication elements by equivalent electric elements and formed an electric circuit which was solved by numerical methods. The calculated results of the oil flow rate showed good agreement with measured data. An analytical solution of an isothermal, Newtonian flow in a single screw extruder was developed in [5]. The authors adapted the Navier-Stokes equation to a finite channel with specified boundary conditions. To solve the resulting Poisson equation they applied the finite sine-transformation method. A comparison with experimental data from published literature showed satisfying accuracy of the presented solution. Based on the models presented in [5], [6] used a generalized integral transform technique (GITT) to solve the Poisson equation. The authors investigated the effects of screw geometry parameters on the oil mass flow rate. In [7], the same analytical solution was used to couple a screw channel pump and the shaft channel region. In this study the analytical model was compared on the one hand with experimental data to verify the calculated oil mass flow rate, and on the other hand with CFD data to verify the oil climbing time.

The base for the model developed in this study is the oil supply system of a hermetic reciprocating compressor for refrigeration application shown in Figure 1. The compressor operates in ON/OFF mode with constant speed. The oil supply system consists of two pumping areas: the lower pump is partially immersed in the oil sump and consists of a centrifugal pump, and the upper pump consists of a helical groove machined on the surface of the crankshaft.

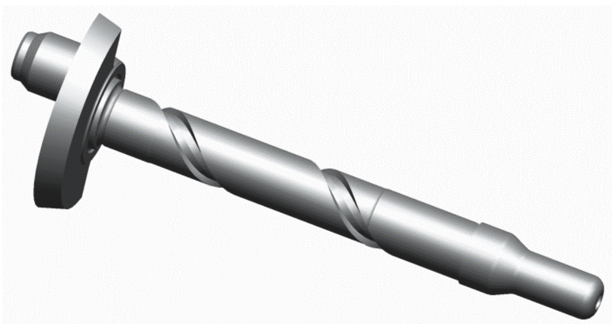

Figure 1. The crankshaft with the integrated oil pumping system.

The purpose of the present paper is the modeling of the flow in the helical groove of a hermetic compressor oil pump at higher resolution compared to analytical models. The relationship between the flow field in the cross section of the channel and the resulting oil mass flow rate is pointed out. Furthermore, the small gap between the crank shaft and the barrel wall is considered. The method is used to quantify the influence of different operating conditions (speed, oil viscosity) and geometric parameters (helix angle, gap width) on the mass flow rate.

2. Problem Formulation

The numerical modeling of the oil flow in the helical groove is based on the steady-state governing equations of motion in rectangular coordinates. Assuming that the fluid is isothermal, incompressible, Newtonian and has constant properties, the equations for laminar flow (the Reynolds-Number for this problem is approximately 250) are given by:

$$\rho\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}+w\frac{\partial u}{\partial z}\right)=-\frac{\partial p}{\partial x}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right)+\rho g_x \tag{1}$$

$$\rho\left(u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}+w\frac{\partial v}{\partial z}\right)=-\frac{\partial p}{\partial y}+\mu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right)+\rho g_y \tag{2}$$

$$\rho\left(u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}+w\frac{\partial w}{\partial z}\right)=-\frac{\partial p}{\partial z}+\mu\left(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right)+\rho g_z \tag{3}$$

In order to simplify the problem, the cross section in the middle of the helical groove is investigated. The channel is considered to be unwrapped and the helical channel is assumed to make one full turn on the crankshaft, so the area of the cross sections expands half distance until a virtual neighboring channel. Figure 2 shows the volume of the channel with the position of the cross section and the unwrapped geometry with the coordinate system.

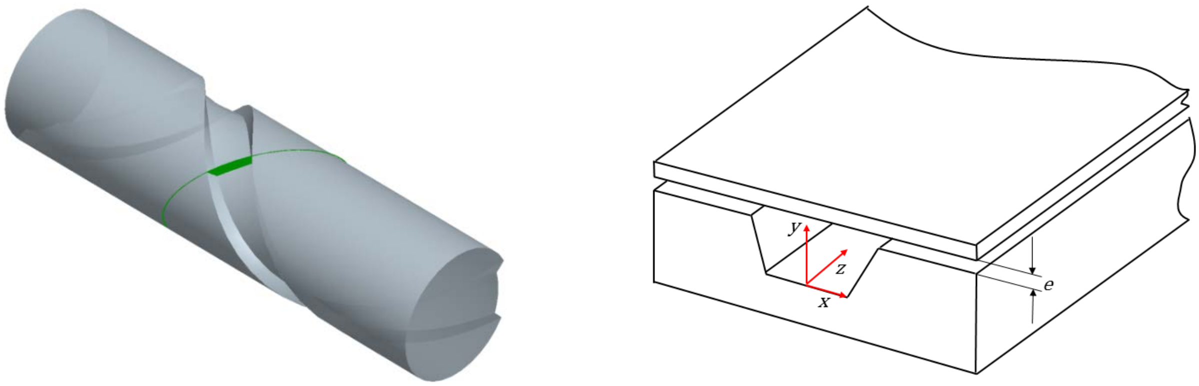

Figure 2. The position of the cross-section in the channel and the unwrapped geometry.

The following assumptions are made for the present problem: (i) the flow field is fully developed in z-direction and (ii) the body forces in y-direction are neglected. With these assumptions the above equations can be reduced to:

$$\rho\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}\right)=-\frac{\partial p}{\partial x}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right)+\rho g_x \tag{4}$$

$$\rho\left(u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}\right)=-\frac{\partial p}{\partial y}+\mu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right) \tag{5}$$

$$\rho\left(u\frac{\partial w}{\partial x}+v\frac{\partial w}{\partial y}\right)=-\frac{\partial p}{\partial z}+\mu\left(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}\right)+\rho g_z \tag{6}$$

Due to the assumption that the flow is fully developed in z-direction, the cross-section flow is independent of the transport velocity w and the equations (4) and (5) can be solved in a two-dimensional domain. The transport velocity w depends on the flow field in the domain and is treated like a passive scalar. The pressure term in the z-momentum equation can be neglected, if the pressure difference between the inlet and outlet of the channel is zero. The flow field in the cross-section is modeled with the finite volume method [8] using a grid with approximately 5000 cells. An evaluation concerning the mesh size has shown that there is no significant deviation between two grids with 5000 and 8000 cells, respectively. Below a grid size of 4000 cells significant deviations could be observed. The SIMPLE algorithm [9] applied on staggered grids in *x*- and *y*-direction is used for the pressure-velocity coupling. The discretization of the diffusion term is done by using the central differencing scheme and the discretization of the convective term is done by using the second order upwind scheme. Under-relaxation factors for pressure and velocity are implemented to get stable computations. The inclined walls in the cross-section are modeled with fine steps to get higher geometrical resolution. Figure 3 shows the computational domain and the set of boundary conditions which take the relative motion between the crankshaft and the barrel wall into account. The relative motion is modeled with the classical pumping model [10] where the screw channel walls are stationary and the barrel surface becomes a flat plate moving with constant velocity. The boundaries at both sides in the gap are modeled with periodic boundaries.

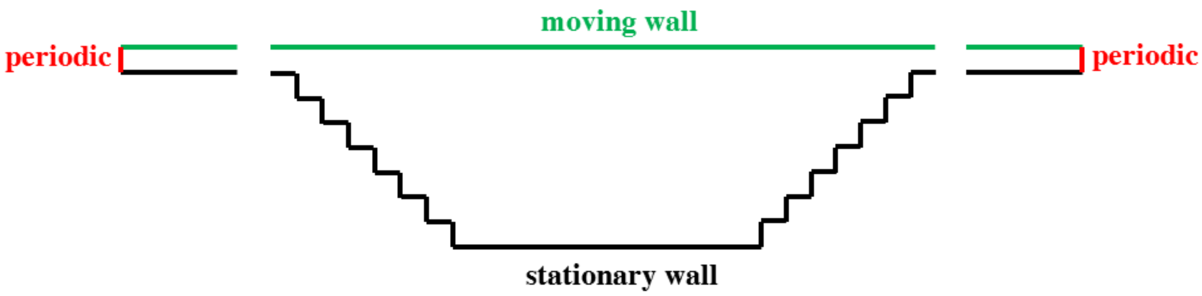

Figure 3. The computational domain with boundary conditions.

The net mass flow is calculated with the total mass flow in *z*-direction and the mass flow through the gap in *x*-direction.

3. Results

A fundamental set of input parameters is defined to verify the present model. The input parameters are related to the geometrical parameters of the investigated crankshaft and the operation parameters are adjusted to the conditions at an oil pump test bench. Table 1 shows the input parameters for the basic version. The test bench is able to measure the oil mass flow rate at different submersion depths of the crankshaft at constant rotational speed. An uncoupled investigation of the helical groove from the rest of the oil path is not possible, a fact which should be considered in the comparison between the calculated and the measured results. Due to the neglect of the pressure difference between inlet and outlet, the influence of submersion depth is not regarded in the numerical model and thus the validation with only one measuring point of the test bench is not meaningful. Two measuring points at different submersion depths are used to define a mass flow range. Table 2 shows the comparison between the calculated mass flow rate of the basic version and the measured mass flow rate. The considerable deviation between the calculation and the measurement can be explained by the neglect of the pressure term in *z*-direction. The pressure term would include the influence of the lower centrifugal pump which results in a higher pressure at the inlet compared to the outlet and an increase of the mass flow rate.

Table 1. Input parameters for the basic version.

Density [kg/m³]	832
Viscosity [cSt]	8
Rotational speed [rpm]	3000
Gap width [mm]	0.1
Helix angle [°]	49.3

Table 2. Comparison between calculated and measured mass flow rate (normalized on basic version).

Measured 7 mm submersion depth	1.51
Measured 2 mm submersion depth	1.23
Calculated basic version	1

The results of the simulated flow field are shown in Figure 4. The flow field in the gap between the crankshaft and the stationary wall appears as a simple Couette flow with gravitational influence. In the area of the channel the flow is diverted by the inclined walls and a vortex occurs. Due to the high dissipative impact of the oil on the flow the vortex is concentrated mostly on the top right side of the channel.

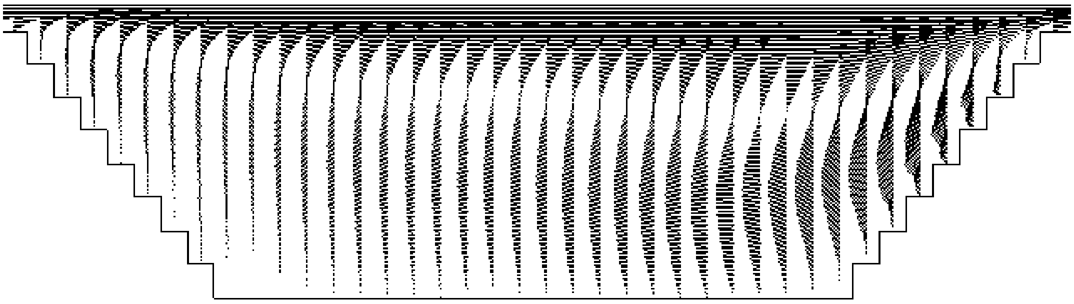

Figure 4. Velocity field in the channel cross-section.

After the simulation of the two-dimensional flow field, the calculation of the velocity in z-direction can be carried out. The result of the simulated w-field is shown in Figure 5. The classical analytical models ([5], [6] and [7]) solve only the Poisson equation of the velocity in channel direction which results in a symmetric solution over the channel cross-section. Compared to these models, the present model includes the influence of the two-dimensional flow field and a non-symmetric distribution of the w-velocity can be seen. The influence of the above mentioned vortex results in higher values of the w-velocity, concentrated at a small area at the top right of the domain. A comparison between the simulation with and without respect to the two-dimensional flow field and the gravitational forces results in a 3.15 times higher mass flow rate in the model without influence of the flow field. This shows that the cross-section flow field and the gravitational forces have a considerable influence on the results of the mass flow rate.

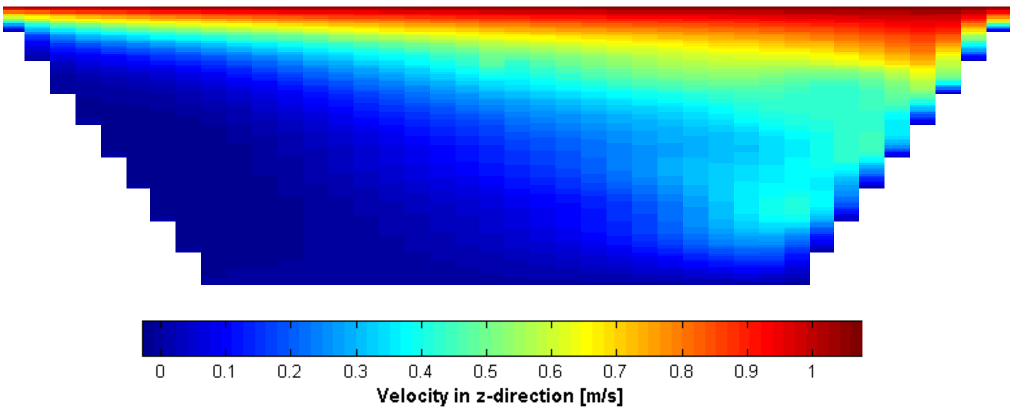

Figure 5. Distribution of the w-velocity.

The present model is used to consider the influence of the rotational speed, the oil viscosity, the gap width and the helix angle on the oil mass flow rate. The results are normalized to the oil mass flow rate of the basic version. The investigation of the influence of the several parameters on the friction behaviour of the oil pump is not included in the following variations.

3.1. Rotational speed

The influence of the rotational speed of the compressor on the operation behaviour of the oil pump is a very important detail in the design of variable speed compressors. Figure 6 shows the trend of the oil mass flow rate over the rotational speed. The variation of the speed shows an approximately linear dependency on the mass flow rate. Especially if the compressor runs at speeds below 2000 rpm for longer periods, the design of the oil pump should be improved to avoid lubrication failures.

3.2. Oil viscosity

As shown in Figure 7, the oil mass flow rate decreases significantly at lower viscosity. With oil viscosity over 8 cSt, which is used in the base calculation, an improvement in the efficiency of the oil pump regarding the mass flow rate could be observed. Due to the strong dependence on the temperature, the viscosity underlies significant variations during the compressor operation. Especially during so called ‘Pull-Down’ cycles in which the thermal load on the compressor is very high, the oil mass flow rate might decrease. The thermal behaviour of the compressor has a considerable impact on the choice of the oil.

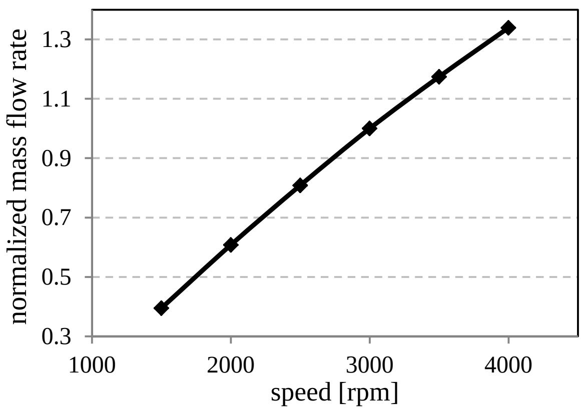

Figure 6. Influence of the rotational speed on the oil mass flow rate.

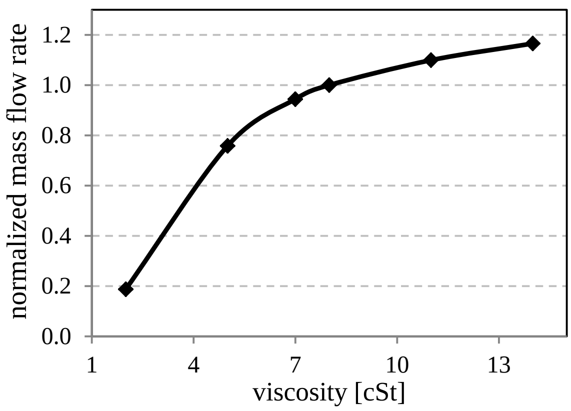

Figure 7. Influence of the oil viscosity on the oil mass flow rate.

3.3. Gap width

The width of the gap between the rotational crankshaft and the stationary barrel wall shows an approximately linear dependency on the oil mass flow rate (Figure 8). A smaller gap width results in a higher value of the mass flow rate due to a stronger influence of the velocity difference of the two parts on the velocity in channel direction. Also the higher mass flow rate in the gap at higher gap widths has a negative effect on the oil pump mass flow rate because as the mass flow rate in gap direction increases, the effective mass flow rate in channel direction decreases.

3.4. Helix angle

Another important influence factor on the pumping efficiency is the helix angle. The results of the helix angle variation are shown in Figure 9. Based on the vector components of the relative velocity between crankshaft and barrel wall, a smaller helix angle results in a higher velocity in channel direction which has a direct impact on the oil mass flow rate. An increase of the helix angle of 10° compared to the basic version has the same effect as a decrease of the rotational speed of about 2000 rpm.

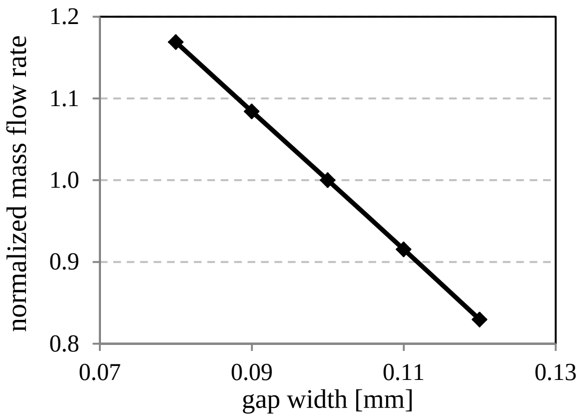

Figure 8. Influence of the gap width on the oil mass flow rate.

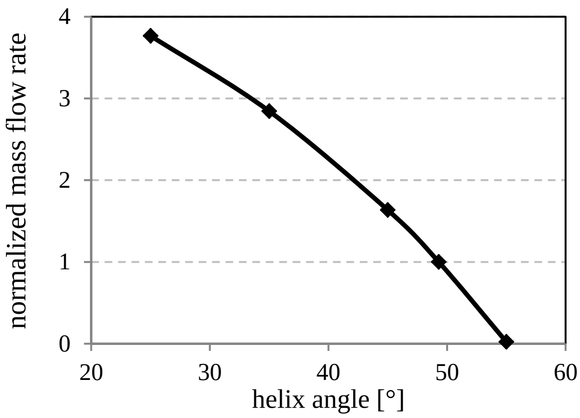

Figure 9. Influence of the helix angle on the oil mass flow rate.

3.5. Design variation
In order to show how to improve the pump efficiency for compressors which operate at low rotational speed, a calculation of a version with 1500 rpm, gap width of 0.08 mm and helix angle of 25° was carried out. Compared to the version without design variations, the mass flow rate can be increased significantly. The normalized mass flow rate with this configuration is 1.7. An additional investigation of the friction losses should be carried out to get a full evaluation of the design variation.

4. Conclusion
A numerical model for the investigation of the helical oil pump of a hermetic compressor for household appliances is presented in this paper. The model is based on the governing equations of motion which were adopted for the present problem. Compared to analytical models from published literature, the present model is able to calculate the flow field in the channel at higher geometrical resolution including the gap between the crank shaft and the barrel wall. The proposed model was used to carry out a parameter variation. The influence of the rotational speed, the oil viscosity, the gap width and the helix angle was investigated. An additional design variation was carried out to show the possibility to increase the mass flow rate at lower rotational speed with a decrease of the gap width and the helix angle.
The model can be used to rate design parameters of helical oil pumps at higher geometrical resolution compared to analytical models but less effort compared to three-dimensional CFD simulations with commercial tools as the time consuming meshing process is omitted. The influence of adjoining pumping devices on the boundary conditions of the model and a validation of the energy consumption of the pump will be explored in a future study.

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