A Lumped Parameter Approach for GEROTOR Pumps: Model Formulation and Experimental Validation

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Abstract

This paper describes a high fidelity simulation model for GEROTOR pumps. The simulation approach is based on the coupling of different models: a geometric model used to evaluate the instantaneous volumes and flow areas inside the unit, a lumped parameter fluid dynamic model for the evaluation of the displacing action inside the unit and mechanical models for the evaluation of the internal micro-motions of the rotors axes. This paper particularly details the geometrical approach, which takes into account the actual geometry of the rotors, given as input as CAD files. This model can take into account the actual location of the points of contact between the rotors as well for the actual clearances between the rotors. The potentials of the model are shown by considering a particular GEROTOR design. A specific test set-up was developed within this research for the model validation, and comparisons in terms of steady-state pressure versus flow curves and instantaneous pressure ripples are shown for the reference pump.

KEYWORDS: Gerotor, pump, lumped parameter model, pump model, geometrical model

1. Introduction

Gerotor pumps are fixed displacement units which find a wide range of application in low pressure fluid power systems (<50 bar) and in automotive (AWD systems, engine lubrication, fuel injection systems) thanks to their low cost, compactness, reliability, robustness under severe operating conditions and low noise level. Although the basic kinematic theory pertaining to the trochoidal profile of the rotors date back more than a century /1/, and several studies were done to find profiles suitable for pump operation, the development of Gerotor units has followed trial and error or very simplified numerical approaches for years. In recent years, several approaches have been presented to simulate the fluid dynamic aspects characterizing the displacing action of Gerotors. This past effort can be grouped in two categories: CFD based approaches, such as /2/ and fast lumped parameter approaches, such as /2-4/. Both these approaches showed high potentials for predicting aspects related to fluid compressibility, such as filling capability, effect of port timing, and so on. Simulation time can limit the use of CFD approaches for optimization studies; moreover, the need for a continuous fluid domain implies the impossibility of reproducing actual contacts between the rotors. Additionally, the position of the rotors axes has to be assumed as simulation input, and cannot be considered as a resultant of the actual loading on the rotors. On the contrary, lumped parameter approaches are fast, and can be easily coupled with other mechanical models to permit the study of micro-motions of the internal parts. This approach has been successfully implemented for on external gear units /5/ as well as axial piston units /6/. However, existing lumped parameter models for Gerotor units are limited by simplistic assumptions pertaining the location of the contact points (it is often assumed contacts at every tooth), the delimitations of the tooth space volume and the geometry leakages of the internal leakage paths resulting from geometrical tolerances.

This paper addresses the above mentioned limiting aspects of lumped parameter models for Gerotor pumps and presents a lumped parameter model based on a detailed evaluation of the internal geometric features. The main element of novelty respect to existing approaches is the numerical algorithm used to generate the inputs for the fluid dynamic simulation. Although not detailed in the paper, the model has also the capability of predicting the actual position of the rotors resulting from the pressure loading on the rotors and the behavior of the lubricated journal bearing.

The model potentials were investigated within this research through tests performed at Thomas Magnete GmbH, on a test rig specifically developed to measure both steady-state and transient performance of Gerotor units. Significant comparisons between simulation results and experimental data are shown in the results section of the paper.

2. The Gerotor Model

The structure of the proposed simulation model is shown in **Figure 1**. Different submodules compose the simulating tool: the geometrical module, the fluid-dynamic module and three modules dedicated to the evaluation of the micro-motions of the rotors. In particular, the forces module evaluates the loading of the rotors based on the fluid pressure; the journal-bearings model evaluates the behavior of the lubricated interface at the outer rotor interface; the rotors radial movement model evaluates the instantaneous position of the rotors, as pertains their axis of rotation and orientation. While the geometrical model as well as the journal-bearing model are developed in C++, the other modules are implemented as built in C models in Amesim simulation environment, to permit an easy simulation of a complete system that includes a Gerotor unit. The following section details the numerical geometric model, being one of the most important elements of novelty of the proposed model.

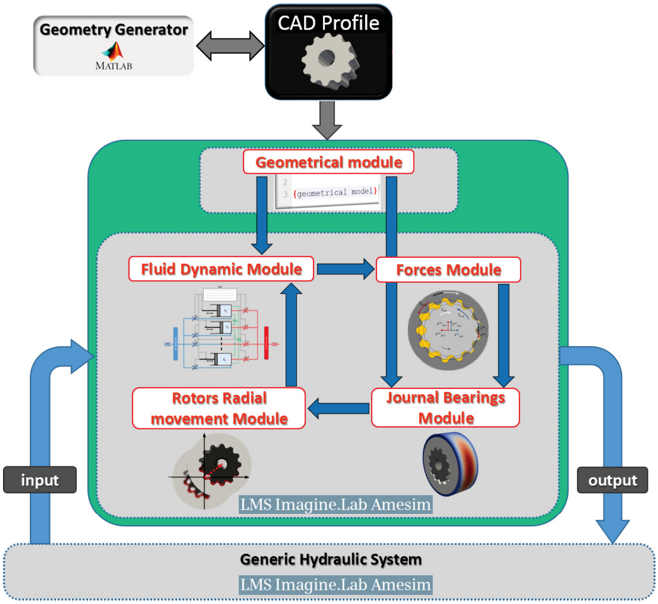

Figure 1: Structure of the Gerotor Simulation Tool.

2.1. The Geometrical Module

The geometrical model is an essential pre-processor for all the modules of the overall simulation tool. This module is not based on the analytical definition of the rotor profiles, but it is entirely based on numerical calculation performed on the CAD files (in TXT or STL format) of rotors and of the inlet and outlet ports. The module is implemented in C++ and takes advantage of open-source GSL libraries. The output of the model is given by TXT input files that can be used for stand-alone considerations on the pump geometry (using ParaView) and for the input of the Amesim fluid dynamic model of the unit. The placement of the rotor, as pertains rotation center and relative rotation, is given as input of the model (**Figure 2**). A set of different rotors position is created before the fluid dynamic simulations, in order to create a look-up table of different geometric configurations for the same unit. During the simulation, the mechanical model of the rotors determines the actual rotor configuration, interpolating the values from the pre-generated look-up table.

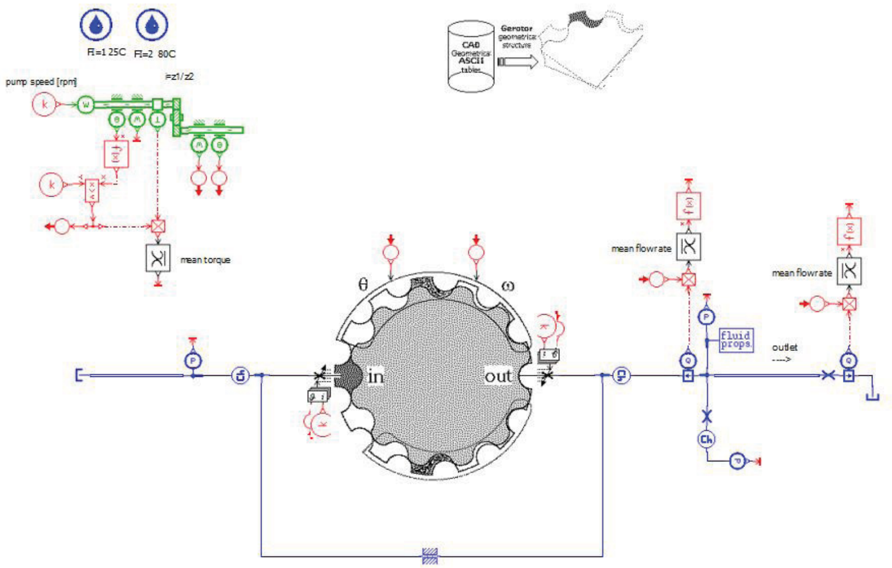

Figure 2: Gerotor Simulating Tool in Amesim environment.

This process allows for a non-nominal initial positioning of the rotor that can be representative of the error in the centering of the two gears in a real life scenario. A realistic operation of the unit is reproduced by rotating the drive rotor of an angle $\delta\theta$, as shown in **Figure 5**, that permits to establish a condition of contact between the rotor profiles. To minimize the calculations and promote simulation swiftness, the rotors are cut in slices of seven half-tooth profile, which is the minimum possible number in order to perform the evaluation of the geometrical features over an entire revolution of the outer rotor. Additionally seven different geometrical features required by the fluid dynamic module are evaluated through the geometrical module: displacement chamber volume (i), area and hydraulic diameter of the connection between the displacement chamber and suction and delivery ports (ii, iii, iv, v), height of the gap between the rotors (vi) and equivalent gap length (vii). For a Gerotor, a number n of control volumes (displacement chambers) equal to the number of teeth of the outer rotor can be defined. An entire revolution of the outer rotor (that corresponds to $\frac{n}{n-1} \cdot 360°$ rotation of the inner rotor) is then necessary for a complete evaluation of the geometrical features (**Figure 3**).

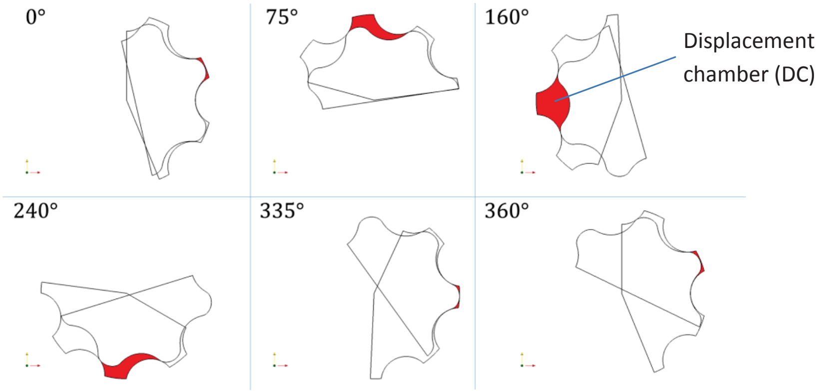

Figure 3: Shape and location of the DC volume for 360° of revolution of the outer rotor

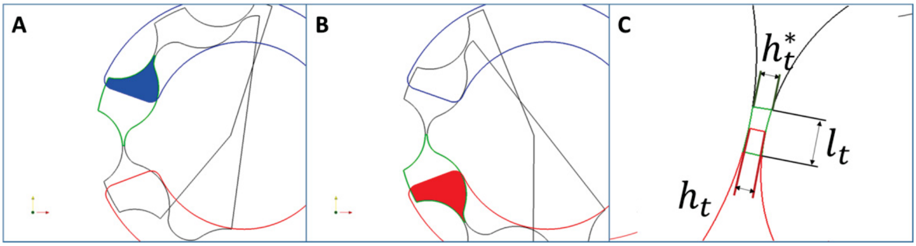

Figure 4: Connection between the DC volume and the suction (A) and delivery (B) ports; definition of minimum gap height h_t and equivalent gap length l_t (C)

The displacement chambers (DCs) are simply defined as the space between the points of minimum distance between two rotors, in this way the algorithm can work also if clearances are present. Since the law of variation of each DC is the same, with a proper angular lag, the definition of one DC is sufficient to define all the control volumes of the fluid dynamic model. It is of outmost importance to ensure that discontinuities are not present in the DC volume variations throughout the entire revolution. In fact, discontinuities would lead to unrealistic pressure variations in the solution of the pressure of the DC, as it will be clarified in section 2.2. Since the DC profile is not generated through an analytical formula but it is derived from CAD drawings, the profile is defined by a discrete number of segments. The volume per unit width of the DC is therefore calculated numerically using the formula below:

$$A_{DC} = \frac{1}{2} \left| \sum_{i=1}^{m-1} x_i y_{i+1} + x_m y_1 - \sum_{i=1}^{m-1} x_{i+1} y_i - x_1 y_m \right| \quad (1)$$

m being the number of segments constituting the DC profile equivalent polygon. The connection between the DC volume and the suction/delivery ports is calculated as the intersection area between the DC profile and the port. The hydraulic diameter of the connection is also evaluated as:

$$D_H = \frac{4A}{P_{DC}} \tag{2}$$

An important feature evaluated by the geometric model is the radial gap between adjacent DCs, which is one of the major contributor of leakages. An ideal positioning of the rotors would lead to constant gap height, however, a condition of actual contact between the rotors lead to variable gap height (**Figure 5**). Once the gap height is identified as minimum distance criteria, an equivalent gap length l_t is defined as the distance between two points positioned inside adjacent DCs where the gap height is $h_t^* = h_t \cdot (1 + t)$ (**Figure 4**). An example of the outputs of the geometrical model for a given positioning of the rotors for a reference unit is given in **Figure 6**.

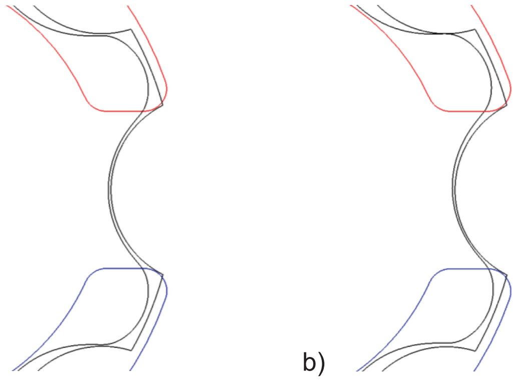

Figure 5: Effect of rotor positioning to the radial gap: a) ideal positioning (constant gap); b) actual positioning.

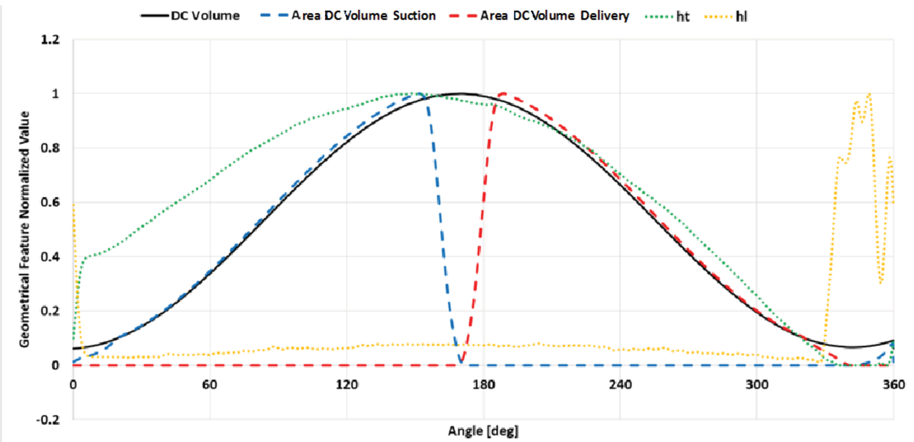

Figure 6: Trend of geometrical features predicted by the geometrical module.

2.2. The Fluid Dynamic Module

The lumped parameter fluid dynamic module permits the calculation of the pressure inside each DC as well as the flow between each DC and the ports. The fluid domain of the pump is discretized with a finite number of control volumes (CVs) interconnected among each other as shown in **Figure 7**.

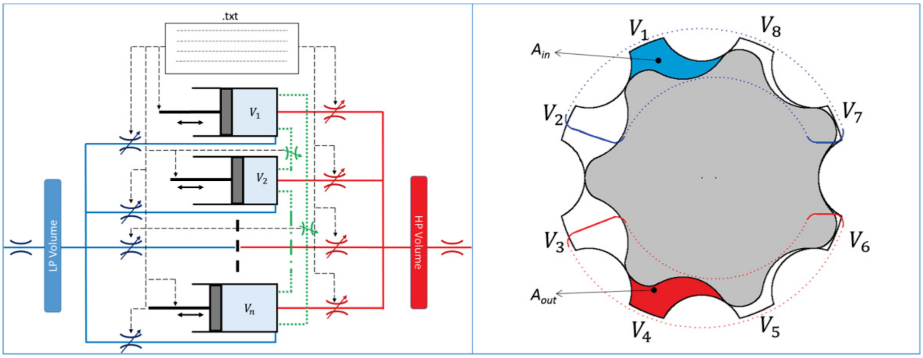

Figure 7: Connection between control volumes.

Each CV has uniform fluid properties that depend only on time. The CVs are treated as variable DCs while the connections between the different CVs as well as between CVs and inlet/outlet ports are treated as variable orifices. Based on the flow between adjacent CVs, mass conservation law and fluid state equations the variation of the pressure p_i inside each CV with respect to time is determined through the pressure build-up equation

$$\frac{dp_i}{dt} = \frac{1}{V_i}\frac{dp}{d\rho}\bigg|_{p=p_i} \cdot \left[\sum \dot{m}_{in,i} - \sum \dot{m}_{out,i} - \rho|_{p=p_i}\frac{dV_i}{dt}\right] \quad (3)$$

The mass entering and leaving each CV is evaluated using two different approaches: for the connection between a CV and the inlet/outlet port the orifice equation has been used assuming a dependence of the discharge coefficient on the Reynolds number

$$\dot{m} = \frac{p_i - p_P}{|p_i - p_P|} \cdot \rho|_{p=\overline{p_{i,P}}} \cdot \alpha \cdot A_{i,P} \cdot \sqrt{\frac{2 \cdot (p_i - p_P)}{\rho|_{p=\overline{p_{i,P}}}}} \quad (4)$$

For the connection between the different CVs, a modified Poiseuille equation (including the Couette flow) is used instead, assuming laminar flow conditions:

$$\dot{m} = \rho\left[-\frac{h_t^3}{12\mu}\frac{p_i - p_j}{l_t} + \omega_{outer} \cdot 2\pi \cdot r_{outer} \cdot \frac{h_t}{2} - \omega_{inner} \cdot 2\pi \cdot r_{inner} \cdot \frac{h_t}{2}\right] \cdot b \quad (5)$$

For this research, the flow through the lateral gap of the rotors is modeled as an equivalent laminar orifice connecting the suction and delivery environments. This simplified approach permits to evaluate the overall volumetric losses of the unit.

3. Results

An extensive validation of the proposed model has been performed. This section reports results generated for a 7/8 teeth Gerotor pump at different operating conditions in terms of speed of the working fluid: the simulation results have been then compared with experimental data.

3.1. Test-rig Setup

A test rig was specifically designed (**Figure 8**) within this research to measure steady state operation of Gerotor units as well as the pressure oscillation at the delivery port. Similarly to what described in /7/, a delivery load system based on a sharp orifice restriction and a line that avoids geometrical discontinuities was implemented in order to have a delivery line easy reproducible in simulation. In this way, an accurate comparison between simulated and measured outlet pressure oscillations could be performed considering the behaviour of the delivery line.

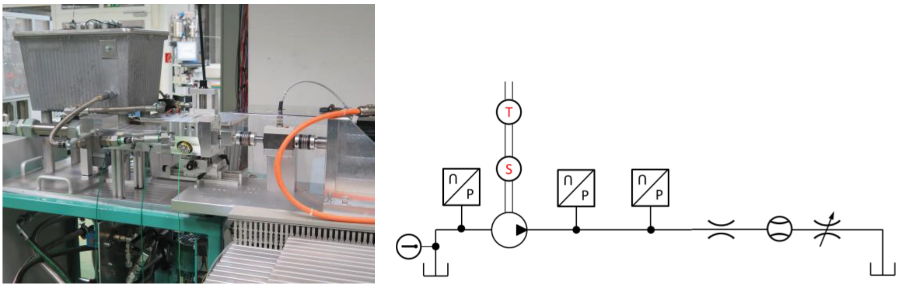

Figure 8: Picture and ISO schematic of the circuit used for the measurements of the pressure ripple and characterization of the pump.

3.2. Model Results and Validation

The comparison between measured and simulated pressure-flow curves in steady state conditions for the reference pump are shown in **Figure 9**. From this figure, it appears evidently how the model is capable of reproducing the experimental data with a good accuracy, particularly at low speed operating conditions. Similar agreements were obtained at different oil temperatures. The reason of the mismatch at higher speeds are due to the approximations pertaining the lateral gap leakage flow. In fact, Couette flow dragging effects, particularly visible at low operating pressure (when pressure-driven leakages are less important), are not properly considered by the equivalent laminar orifice used in the model. Although the comparisons are satisfactory, to improve the accuracy the implementation of a 2D gap model for the lateral gap flow is currently under development.

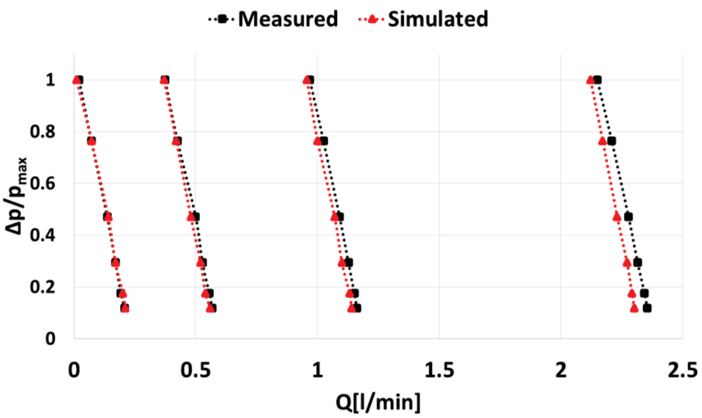

Figure 9: Measured and predicted steady state performance.

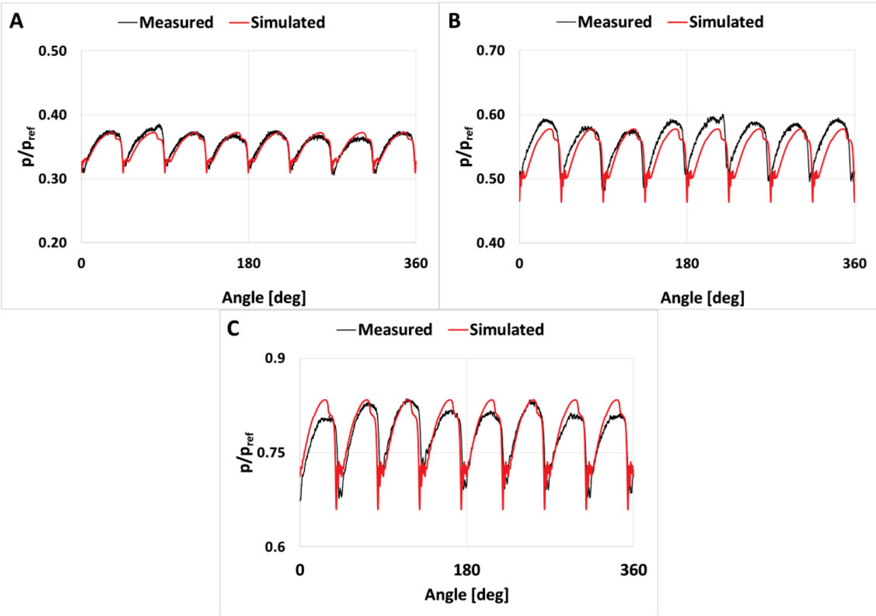

Figure 10: Comparison between simulated and experimental pressure ripple at the delivery for 300(A), 400(B), 500(C) rpm.

Further validation of the model was obtained through the evaluation of pressure pulsation at the delivery with the system shown in **Figure 10**. The simulation was obtained by coupling the pump model with a 1-D finite element method for the modelling of the pressure waves in the line /7/. Figure 10 shows the comparisons between simulation results and experimental data for three different operating conditions for the reference pump. The same sharp orifice (diaphragm) was used for all tests, and only the speed of the pump was varied. The comparison shows a very good agreement for the three different operating conditions especially at low speed **Figure 10A**; the difference at higher speed is due mainly to the fluctuation of the flow related to the shaft irregularity.

Once the accuracy of the model is confirmed by the experimental validation, useful considerations on pump operation can be performed by looking at the detailed results of the model. For example, the model can provide detailed analysis of the displacing action realized by the internal DCs as well as the effect of the inlet and outlet port timing. **Figure** 11 shows the pressure profile of one DC plotted against the DC volume: regions of pressure drop below the saturation pressure, associated with cavitation, and pressure peaks can be spotted in the pressure trend. These effects might not be visible at the delivery but can affect the pump performance in terms of efficiency, noise and durability. These phenomena are associated to the geometry of the ports, that play a timing function, and to the tolerances between the rotors.

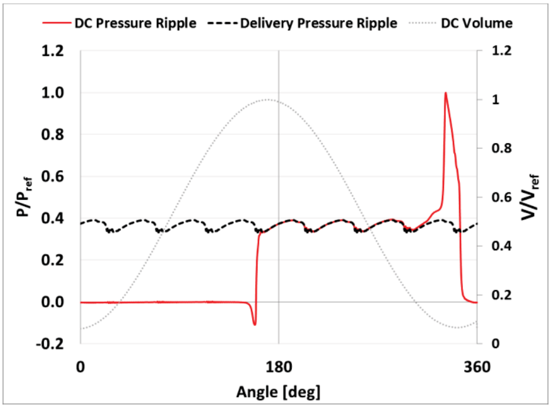

Figure 11: Pressure profile inside a DC for one revolution of the outer rotor plotted against DC volume.

4. Conclusions

A simulation model for Gerotor unit has been presented in this work. The model couples different submodules. The particular focus of this work was the geometrical module, which uses a numerical approach capable of taking into consideration the actual rotor geometry, the clearances between the rotors and the realistic positioning of the gears. The geometric model provides the data to a lumped parameter fluid dynamic model, which is able to provide an accurate evaluation of the flow through the unit. An experimental activity was performed at Thomas Magnete GmbH to validate the model predictions, and a high fidelity of the simulation results was found at different operating conditions. Although not detailed in this work, the fast lumped parameter approach permits the coupling of mechanical sub-models for rotors. These sub-models permit the evaluation of the force balance at the rotors including the behaviour of the lubricating journal bearing established by the outer rotor. In this way, the actual position of the rotors and their micro-motions resulting from the pressure fluctuation can be evaluated.

The proposed model can be considered a valuable tool for simulation of different designs of Gerotor pumps and thanks to the low computational cost can be considered ideal for optimization studies.

5. References

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6. Nomenclature

A = Area, m²

$\dot{m}$ = Mass flow rate, kg/s

(x_i, y_i) = i^{th} vertex or corner of the polygon, m

t = Time, s

P = Perimeter, m

α = Coefficient of discharge, -

D_H = Hydraulic Diameter

ω = Angular velocity, rad/s

h_t = Gap Height

r = Radius, m

h_l = Equivalent Gap length

μ = Dynamic Viscosity, Pa/s

V = Volume, m^3

b = Rotors' Width, m

p = Pressure, Pa

o = Order of Magnitude, -

ρ = Density, kg/ m^3

t = Tolerance, -

Subscripts:

DC = Displacement chamber

$\mathrm{in/out}$ = Entering/Leaving

P = Delivery/Suction Port environment

ref = Reference Value