

Article

Virtual Prototyping of Axial Piston Machines: Numerical Method and Experimental Validation

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Abstract: This article presents a novel methodology to design swash plate type axial piston machines based on computationally based approach. The methodology focuses on the design of the main lubricating interfaces present in a swash plate type unit: the cylinder block/valve plate, the piston/cylinder, and the slipper/swash plate interface. These interfaces determine the behavior of the machine in term of energy efficiency and durability. The proposed method couples for the first time the numerical models developed at the authors' research center for each separated tribological interface in a single optimization framework. The paper details the optimization procedure, the geometry, and material considered for each part. A physical prototype was also built and tested from the optimal results found from the numerical model. Tests were performed at the authors' lab, confirming the validity of the proposed method.

Keywords: fluid power; numerical model; lubrication; virtual prototyping; axial piston machine

1. Introduction

Axial piston machines' current state-of-the-art design methodology of the lubricating interfaces is based on a trial-and-error process and relies on empirically derived formulas. This antiquated methodology of designing hydraulic components is very expensive and time-consuming. The design of these machines is considered like black box design where the designers change different parameters until the pump passes laboratory tests successfully and finally works under all the desired operating conditions. This design methodology often results in a non-optimal design for performance, compactness, reliability, or cost. Increase in demand for highly efficient, robust, reliable, and cost-effective components have motivated the use of virtual prototyping for the design of axial piston machines. The focus of this paper is the methodology of designing the most critical geometrical features impacting the behavior of the lubricating interfaces.

The complexity of designing these machines lies in the rotating group: the cylinder block, and piston and slipper assemblies. This complexity is due to the physical phenomena that take place in the lubricating interfaces: the cylinder block/valve plate, piston/cylinder and slipper/swash plate interface shown in Figure 1. These three lubricating interfaces perform two main functions simultaneously by bearing the external loads imposed on the rotating group and sealing of the fluid in the ports and displacement chambers. The main external load imposed on these parts of the machine originating from the periodically changing pressure in the displacement chamber. Additionally, the lubricating interfaces are the main sources of energy dissipation of the machine. The source of energy dissipation is divided into two categories the leakage flow through the interface towards the case volume and the viscous friction in the fluid. To minimize the sources of energy dissipation, the two areas outlined have opposite requirements. A low fluid film is required to reduce leakage flow whereas to minimize viscous friction a large fluid film is ideal.

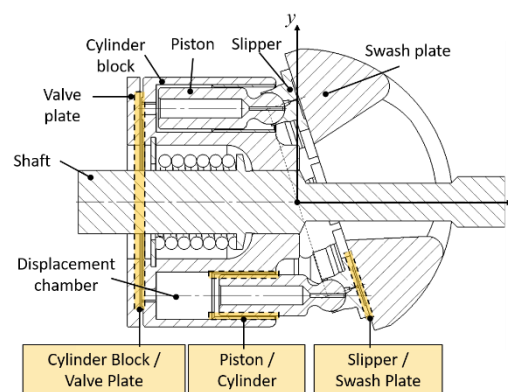


Figure 1. Cross-section of swash plate type axial piston machine with lubricating interfaces.

Advanced numerical models enabled by an increased computational power from the last five decades coupled with the development of numerical models capable of capturing the physical phenomena in mechanical systems has enabled virtual prototyping methodologies for mechanical components in a diverse range of fields. Previous research on axial piston machines has focused on valve plate design features to reduce noise emissions of axial piston machines. The valve plate timing influences the pressure build-up in the displacement chambers. Becker [1] studied the impact of valve plate timing on the pressure in the displacement chamber and ports. Helgestad et al. [2] investigated the effect of pressure rise in the displacement chambers in noise generation axial piston machines. Many numerical models to simulate the kinematics were developed by, to name some, Edge and Darling [3], Palmberg [4], Harrison and Edge [5], Ivantysynova [6,7], and Wieczorek and Ivantysynova [8]. These models were part of research towards reducing noise emission from axial piston machines by introducing design changes, some of these were pre-compression filter volumes (Pettersson et al. [9]; Ivantysynova et al. [10]; Johansson [11]), cross-angle of the swash plate (Johansson et al., [12]; Manring and Dong, [13]), and others. More recently (Seeniraj and Ivantysynova [14]; Kim [15]; Kalbfleisch [16]) developed VpOptim, a genetic algorithm optimization tool for the valve plate design. Kalbfleisch [16] coupled the information from the pump dynamics model predicting the pressure build-up in the displacement chambers and ports with the algorithm non-dominated sorting genetic algorithm (NSGA) II (Deb, et al. [17]), that optimizes the valve plate design.

Richardson et al. [18] used a customized test rig for a swash plate type axial piston machine with a floating valve plate. Using this test rig, they measured the fluid film thickness and used these measurements to develop a model for the valve plate/cylinder block interface. Hashemi et al. [19] developed a multi-dynamics model for the slipper/swash plate interface which also includes a mixed-lubrication model, using the authors' software Tribo-x. The model was validated through the creation of a special test rig in which the friction between the slipper and swash plate was measured [20]. Similarly, thermo-elastohydrodynamic (TEHD) models have been developed that are able to predict the behavior of the fluid film for the interfaces between the following components: piston/cylinder (Pelosi [21], Mizell [22] and Shang [23]), cylinder block/valve plate (Zecchi [24]), and slipper/swash plate (Schenk [25]). These models capture various physical phenomena including the micro-motion of the cylinder block, non-isothermal flow in the lubricating gap, and pressure and thermal elastic deformation on the solid bodies. These TEHD models have been used to investigate micro-surfacing to increase the load carrying ability of the lubricating interfaces [26–29] on axial piston machines.

Ivantysyn [30] utilized an extended version of calculation of swash plate type axial piston pump and motor (CASPAR) [31] and the VpOptim developed by Seeniraj and Ivantysynova [32] where an analysis of the sensitivity to the balance factor in the cylinder block/valve plate and slipper/swash plate interface was performed. The models at the time were less advanced and did not allow for a very reliable prediction of fluid film behavior and machine performance. Schenk and Ivantysynova [33]

implemented an optimization for the slipper/swash plate interface using a response surface method diminishing the computational effort. A simulation study was performed, but the design was not tested.

Additionally, Shang and Ivantysynova [34] developed a model that predicts the case and port flow temperature based on thermodynamics and heat transfer. Also, Shang and Ivantysynova [35] investigated filling the inside of the piston with aluminum material, which adds an adaptive temperature capability for the piston through the thermal deflection at different operating conditions. The researchers found they could reduce overall energy dissipation for the piston/cylinder interface. Additionally, these authors also published an analytical study of the scaling of the lubricating interfaces in axial piston machines [36]. These models haven't been utilized for designing a complete pump design from the start.

Pump designers strive to achieve the best pump design by delivering the highest energy efficiency, power density, and reliability while maintaining low noise emissions for a range of operating conditions. These are all equally important, and some have adverse effects on each other. One example of such a scenario is high efficiency may result in higher manufacturing costs due to more precise manufacturing tolerance requirements or a higher cost material selection.

These numerical models have enabled and motivated the work in this publication where the intention is to propose a new computational design approach and to demonstrate the feasibility of this novel approach. The models provide a unique insight into the behavior of the lubricating interfaces and allow predictions of fluid film thickness, load carrying ability, leakage flow, energy dissipation and the temperature fields of those fluid films separating high dynamic loaded moving parts of the axial piston machine.

Section 1 describes the objective, state-of-the-art of axial piston machine modelling and virtual prototyping, external forces, and a brief description of the numerical models used in the methodology. Section 2 describes the design methodology proposed in this paper in detail. Later, in Section 3, the valve plate optimization is described, and a case study is presented. Afterward, Section 4 illustrates the process of designing the solid bodies that have an impact on the interfaces due to pressure and thermal elastic deformations. Section 5 describes in detail the process of the design of the lubricating interfaces and its parameters that play a role in the design of the rotating group. Next, Section 6 shows the experimental results and validation against simulation. Section 7 is a brief discussion of the findings in this paper. Section 8 is the conclusion.

1.1. External Loads on the Rotating Group

The external forces are derived from the pressure in the displacement chamber and inertial loads. The loads exerted on the rotating group are of extreme importance to the calculation of the fluid film geometry in the lubricating interfaces. Figure 2 presents the external forces applied to the piston/slipper assembly. The pressure force F_{DB} depends on the displacement chamber pressure p_{Di} and surface area A_D . Second, the spring force F_{FB} pushes the cylinder block in the direction of the valve plate, on the z-axis. The spring force prevents the block from tipping when running at high speed and very low pressure. Third, the force due to the friction F_{TB} between the piston and the cylinder bore in the piston/cylinder interface. Furthermore, the force due to the centripetal acceleration of the piston/slipper assembly, $F_{\omega K}$, acts in the radial direction of the cylinder block. The main force is due to the pressurized fluid in the displacement chamber, F_{DK} pushes the bottom of the piston in the direction of the swashplate. The force due to the inertia of the piston/slipper assembly acts on the z-axis, F_{aK} ; and finally, the force due to the friction between the piston and the cylinder bore also acts in the z-axis, F_{TK} . The total sum of these forces, F_{DK} , F_{aK} , and F_{TK} , must be reacted by the swashplate. The forces related to the piston/slipper assembly are all transmitted to the cylinder block summed into a resultant side force, F_{RK} , which is the same force represented for a single piston on as F_{RBi} (not represented in the figure). The external forces and moments need to be balanced by the forces and moments generated by the fluid film pressure field. Refer to Ivantysyn and Ivantysynova [37].

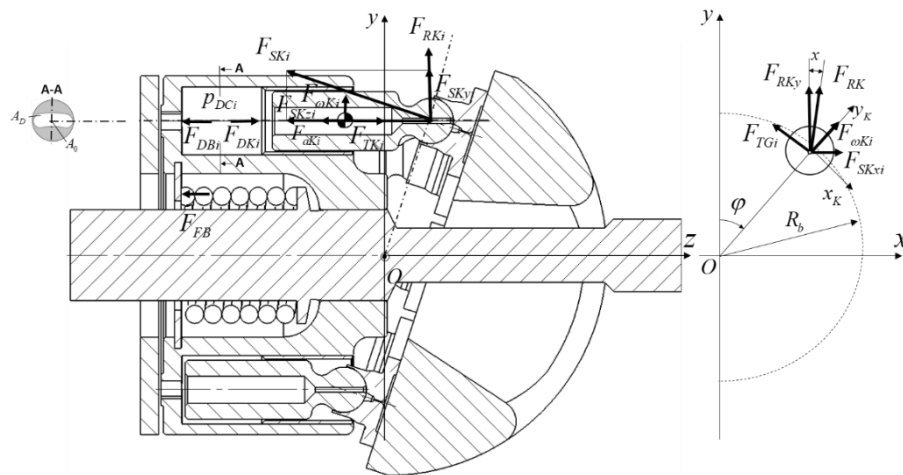


Figure 2. Free body diagram of the forces acting on the piston/slipper assembly.

1.2. Pressure Module

The instantaneous pressure in the displacement chambers is calculated using the pressure module [10]. This numerical model is essential for virtual prototyping since it calculates the kinematics of the machine. The pressure in the displacement chamber is assumed to be uniform. Therefore the model is a lumped parameter approach which utilizes the pressure build-up Equation (1) and the orifice Equation (2). Figure 3 shows the components encompassing the control volume and the leakage flows through the piston/cylinder (Q_{SKi}), slipper/swash plate (Q_{SGi}), and cylinder block/valve plate (Q_{SBi}) interfaces.

$$\frac{dp_{DCi}}{dt} = \frac{K}{V_i} \left(Q_{ri} + Q_{SKi} + Q_{SBi} + Q_{SGi} - \frac{dV}{dt} \right). \quad (1)$$

$$Q_{ri} = \alpha \cdot A_r \cdot \sqrt{\frac{2 \cdot \Delta p}{\rho}}. \quad (2)$$

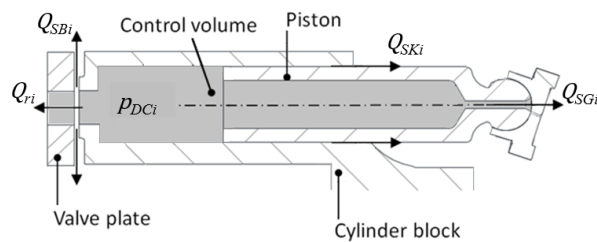


Figure 3. Instantaneous pressure calculation control volume.

The instantaneous pressure in each displacement chamber allows the calculation of the following:

- Swash plate moments
- Flow and pressure ripple
- Volumetric efficiency (due to internal leakage and compressibility)

1.3. Thermo-Elastohydrodynamic Model for the Lubricating Interfaces

As mentioned previously a thermo-elastohydrodynamic model was developed in the author's research center. The TEHD model assumes a full fluid film lubrication regime. The model is composed of four main modules, shown in Figure 4. The first module is the non-isothermal fluid flow module where the Reynolds Equation (3) and Energy Equation (4). is solved using a finite volume approach

yielding the pressure field, temperature, and heat fluxes from the gap. The oil properties are updated dynamically corresponding to the oscillating external loads. The update of the oil properties permits the precise calculation of the load carrying ability, leakage flow, torque loss due to viscous friction, and total energy dissipation. The second module is the finite element method (FEM) module where the elastic deformation of the solid bodies is calculated using an influence matrix method. The third is the thermal module where the solid body temperature is calculated, and fourth, the corresponding FEM module calculates for thermal deflections. The model yields the pressure field distribution, the temperature field distribution, surface elastic deformations, leakage flow and the energy dissipation due to viscous flow in all three lubricating interfaces. For more details on the TEHD model refer to [21,24,38]:

$$\frac{\partial}{\partial x} \left(\frac{\rho h^3}{12\mu} \frac{\partial p}{\partial x} \right) + \frac{\partial}{\partial y} \left(\frac{\rho h^3}{12\mu} \frac{\partial p}{\partial y} \right) = \frac{\partial}{\partial x} \left(\frac{\rho h(u_a + u_b)}{2} \right) + \frac{\partial}{\partial y} \left(\frac{\rho h(v_a + v_b)}{2} \right) + \rho(w_a + w_b - u_a \frac{\partial h}{\partial x} - v_a \frac{\partial h}{\partial y}) - h \frac{\partial \rho}{\partial t}. \quad (3)$$

$$\Delta \cdot \left(\rho VT - \frac{\lambda}{c_p} \Delta T \right) = \frac{\mu}{c_p} \Phi_D. \quad (4)$$

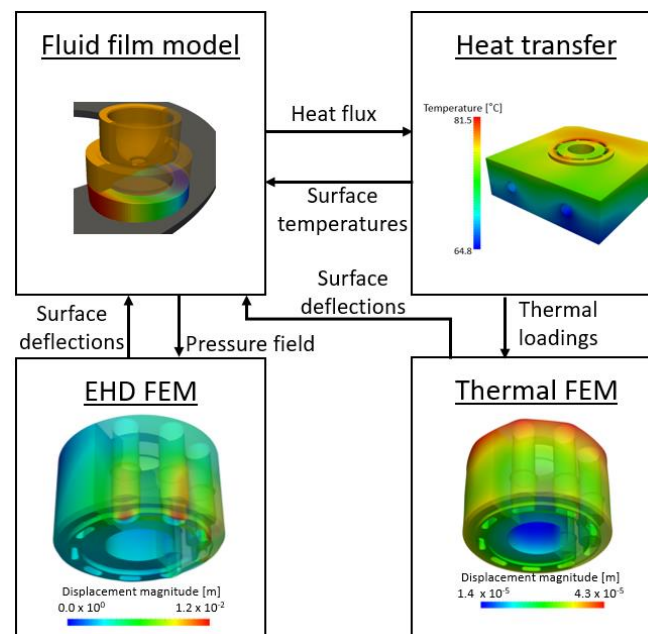


Figure 4. Lubricating interfaces thermo-elastohydrodynamic model structure.

2. Virtual Prototyping Axial Piston Machines Methodology

The novel methodology proposed in this article for the virtual prototyping of swash plate type axial piston machines is described. The axial piston machines virtual prototyping design methodology for valve plate porting and lubricating interfaces is shown in the Figure 5 flow chart. The first step is to identify the required performance data from the positive displacement machine. The main performance data of the machine are maximum displacement volume, maximum operating pressure, and maximum and minimum speed. This first step is crucial because the rest of the machine design derives from the performance requirements. The second step is the preliminary design, where all the main parameters of the machine are derived based on analytical calculations.

Next, the first step within virtual prototyping is the analysis of the preliminary design in terms of calculating the following parameters using the mathematical models described in the state-of-the-art section. The instantaneous pressure calculation in the displacement chamber using a lumped parameter approach provides effective flow rate, flow ripple, the pressure in the displacement chamber, pulsating swash plate moments M_x , M_y , and M_z , as well as, internal leakage (cross-porting). The thermal

model will be used to predict fluid temperatures of the machine's discharge port and the case volume developed in [34]. The TEHD models for the three lubricating interfaces are used to calculate the following parameters:

- Fluid film thickness between the piston and the cylinder bore, the cylinder block and the valve plate, and the slipper and the swash plate.
- Pressure fields in the fluid film in the lubricating interfaces.
- Leakage flows in all three lubricating interfaces.
- Energy dissipation due to viscous flow in all three lubricating interfaces.
- Temperature distributions in the fluid film and main pump parts (cylinder block, piston, valve plate, slipper, swash plate, and end case).
- Surface deformations of the cylinder block, piston, valve plate, slipper, and swash plate due to pressure and thermal loading of these parts.

The parameters will provide the machine designer with the necessary insight to make the decision of which parts require design improvement through optimization.

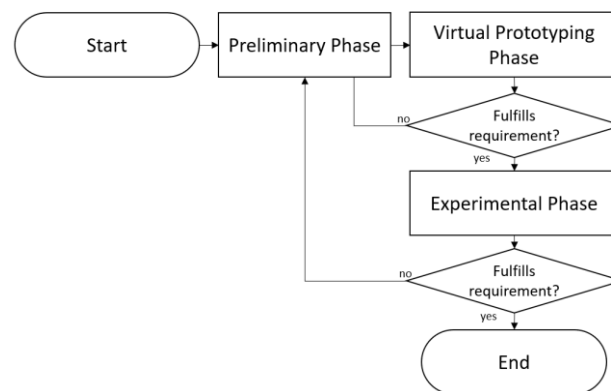


Figure 5. Virtual prototyping flowchart.

2.1. Preliminary Design

The main dimensions of the components for the axial piston machine are calculated in the preliminary design phase. Figure 6 details the steps within the preliminary phase at a high level. First, the number of pistons must be selected together with the minimum shaft diameter required to transmit the desired maximum torque. The base geometrical dimensions are shown in Figure 7. These two parameters constrain all the base geometrical dimensions of the machine parts; i.e., maximum displacement angle β , piston length l_K , pitch diameter d_B , piston diameter d_K , etc. The combination of these parameters defines the geometrical displacement of the machine. The shaft diameter calculation must be done first since it will constrain the size of all the other remaining dimensions. An odd number of pistons is most commonly selected because it results in a smaller flow and torque ripple as Ivantysyn and Ivantysynova [29] have shown. The remaining variables are calculated in an iterative process. The outer and inner cylinder block diameters depend on the shaft, the cylinder block spring, and the pitch diameter which depends on the maximum piston diameter. The piston diameter should be in a range where no physical limitations could be violated due to side forces on the piston resulting in large stresses on the solid body. The displacement angle β is the angle formed between the swash plate running surface plane and the valve plate running surface plane. The swash plate angle is a critical parameter influencing the dimensions of the machine and its performance.

An advantage of axial piston machines of swash plate type with large displacement angles is their compactness. The disadvantages are resulting in large side forces and moments on the piston/cylinder interface. These large side forces and moments generate issues when balancing the external loads and fluid forces in the piston/cylinder interface and generate additional stresses on the cylinder block

solid. One way to reduce the moments on the piston/cylinder interface is to change the overhang length l_o , shown in Figure 7. One of the existing proposed solutions is to change the geometry of the slipper by extending the distance from the center of the slipper to the running surface [39]. The starting geometrical dimensions for the lubricating interfaces (cylinder block/valve plate and slipper/swash plate and piston/cylinder) also need to be generated in this step within the preliminary design. Within this process, the three lubricating interfaces are assumed to have parallel gaps resulting in fixed fluid film heights. This simplification allows finding the first main dimensions for the lubricating interfaces by neglecting hydrodynamic and elastohydrodynamic effects, i.e., considering only hydrostatic forces created in the fluid film. Because this is a simplified assumption, which on one side allows the calculation of main dimensions of these interfaces using analytical expressions, correction factors, so-called balancing factors are introduced to make up for the missing hydrodynamic and elastohydrodynamic (EHD) effects being present in a real machine. The balance factors were found in an experimental trial and error process by manufacturers over the last five decades. Similarly, for the piston/cylinder interface the required clearance is defined assuming a centered position of the piston in the cylinder bore. More details about the preliminary design and the range of balance factors can be found in Ivantysyn and Ivantysynova [37].

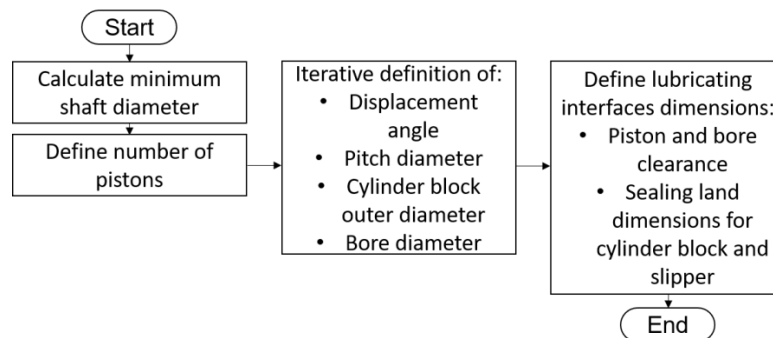


Figure 6. Preliminary phase flow diagram.

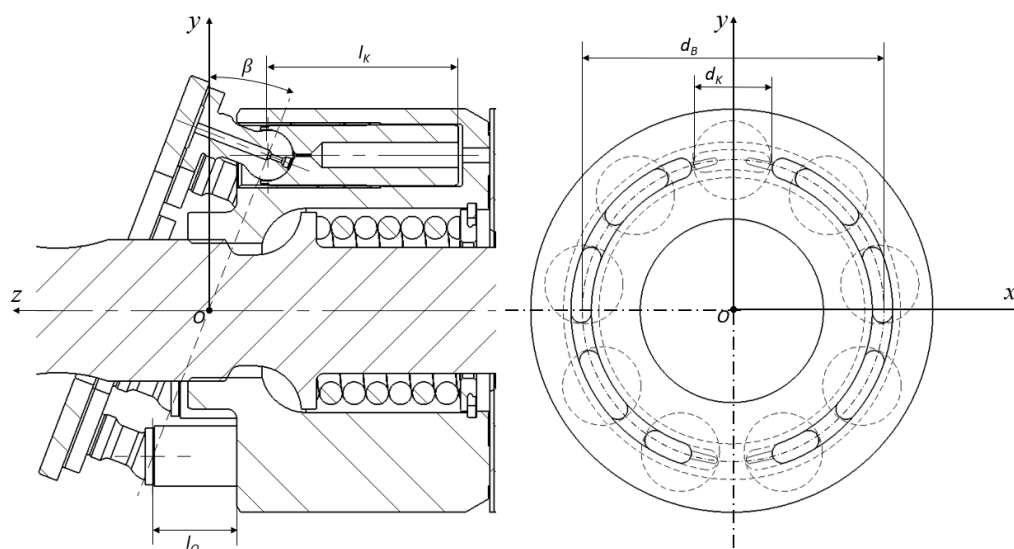


Figure 7. Main geometrical dimension of axial piston machines of swash plate type.

2.2. Virtual Prototyping

The virtual prototyping methodology for swash plate type axial piston machines is summarized in the Figure 8 flow chart. The goal is to use the sophisticated TEHD models, which can consider all the features of the machine; part shape and dimensions, along with material and fluid properties to

determine optimal final dimensions for a given preliminary design. The main parameters are derived from the previous section the preliminary design phase. These design parameters are the inputs for the various numerical models utilized in the virtual prototyping. The valve plate design is the first step since it will determine the displacement chamber pressure profile. The displacement chamber pressure will impact the design of all three lubricating interfaces. The next step is the creation of three-dimensional computer-aided designs (CAD) models. The three-dimensional CAD models are to be compliant with the base design parameters from the preliminary design phase, and they need to be stiff enough to bear the external loads imposed on them while maintaining minimal surface elastic deformations. The latter performance requirement is evaluated utilizing a FEM model to quantify stress distributions and elastic deformations. The three-dimensional CAD models need to be discretized into a three-dimensional mesh and have the boundary conditions defined to analyze it in the FEM model. The maximum surface deformation allowed is defined by the pump designer, and this is dependent on the lubricating interface sensibility to the same. The three-dimensional CAD models need to be modified accordingly to meet all the design requirements. The leakage flows through all three lubricating interfaces is used as an input for the pressure module, which updates the calculation of the instantaneous pressure in the displacement chamber, the effective flow, the pressure and flow ripples, the internal leakage and swash plate moments. The energy dissipation and updated case flow are used as inputs for the thermal model to update the discharge port and the case volume temperatures. This process becomes an iterative process until the discharge flow, and case volume temperatures, the case flow, and the energy dissipation in the lubricating interfaces do not change from one iteration to the next (within a given tolerance).

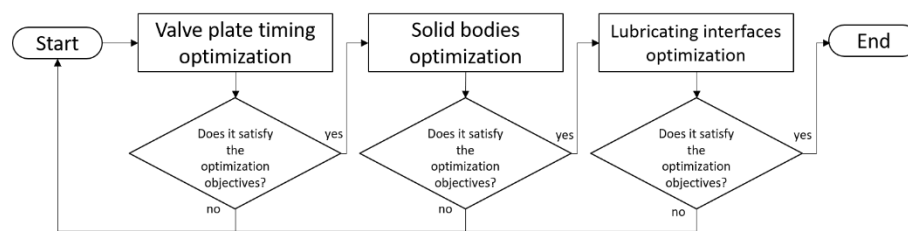


Figure 8. Virtual prototyping flowchart.

The objectives of the virtual prototyping phase are to achieve a design which fulfills the following:

- Lowest energy dissipation in the given range of operating conditions.
- A stable fluid film with sufficient load carrying ability.
- Low flow and torque ripple.
- No cavitation or incomplete filling

The parts shapes and material selection can be varied until an optimum design is found. Finally, the lubricating interfaces design block use the following as inputs: the base parameters collected from the preliminary design phase, the valve plate design, and the solid bodies design are utilized as inputs for the TEHD. The thermal model is coupled with the TEHD model. The thermal model predicts the discharge port and case volume temperatures and feeds them into the TEHD model. The TEHD model uses the updated temperatures from the discharge ports, and case volume calculated with the thermal model and the displacement chamber pressure in the displacement chamber, and the three-dimensional meshes for the cylinder block, the valve plate/end case assembly, the slipper, the swash plate, and the piston.

The TEHD model will calculate, as mentioned previously, the fluid film thickness, the pressure field distribution, the temperature field distribution, surface elastic deformations, leakage flow and the energy dissipation due to viscous flow in all three lubricating interfaces. The leakage flows through all three lubricating interfaces is used as an input for the pressure module, which updates the calculation

of the instantaneous pressure in the displacement chamber, the effective flow, the pressure and flow ripples, the internal leakage and swash plate moments. The energy dissipation and updated case flow are used as inputs for the thermal model to update the discharge port and the case volume temperatures. This process becomes an iterative process until the discharge flow, and case volume temperatures, the case flow, and the energy dissipation in the lubricating interfaces do not change from one iteration to the next (within a given tolerance). The lubricating interfaces design process shown in the flowchart in Figure 8 is a general overview of the process. The designer sets the minimal energy dissipation requirements for the lubricating interfaces. This design process will maintain as a constraint not to allow extreme minimum fluid film thickness within the range of operating conditions.

3. Valve Plate Timing Optimization

The valve plate timing optimization refers to the optimization of the connection from the port to the displacement chamber as a function of the shaft angle. These connections are critical to the pressure build-up in the displacement chamber. The main objectives in valve plate optimization are the following:

- Optimize pressure orifice in the displacement chamber to avoid cavitation.
- High volumetric efficiency.
- Low flow ripple.
- Low control effort on the swash plate.
- Low moment pulsation (ΔM_x and ΔM_y).

The cost functions are defined in Table 1. The cost functions are to minimize the internal leakage, the flow ripple, ΔM_x , ΔM_y , and the mean M_x .

Table 1. Cost functions for the valve plate optimization.

Cost Functions
$f_1(\bar{x}) = \text{Leakage (\%)}$
$f_2(\bar{x}) = \Delta Q_{hp} \text{ (L/min)}$
$f_3(\bar{x}) = \Delta M_x \text{ (Nm)}$
$f_4(\bar{x}) = \Delta M_y \text{ (Nm)}$
$f_5(\bar{x}) = \overline{M_x} \text{ (Nm)}$

The optimization algorithm utilized within the virtual prototyping process changes precompression and decompression grooves to optimize the objective functions described in the previous section. The valve plate design process relies on the pressure module and the VpOptim models coupled with multi-objective genetic algorithm NSGA II. It is a non-dominated sorting genetic algorithm, which offers many advantages over other optimization schemes. It can solve large globalization optimizations and generates a distributed Pareto front [17].

Second, the NSGA II generates a design of experiments (DOE) then this is transferred to VpOptim. VpOptim translates the DOE into inputs for the pressure module. The pressure module evaluates the entire population of designs. VpOptim parses the output from the pressure module into the function evaluations necessary for the NSGA II optimization algorithm. Once the Pareto front has been generated, the designer needs to choose a design based on a weighted cost function. The optimization methodology was modified based on previous work done which considered only one quadrant of operation [16].

The valve plate optimization scheme proposed in this paper is shown in Figure 9. First, the required performance parameters are defined. The population size of the initial and following generations is determined by the number of design variables and quadrants of operation being optimized for the axial piston machine operation. Second, the population is generated randomly. Third, is the function evaluation block. The correct set of function evaluations must be determined with the

adequate objective functions and constraints to accommodate for the operating condition for which is being optimized.

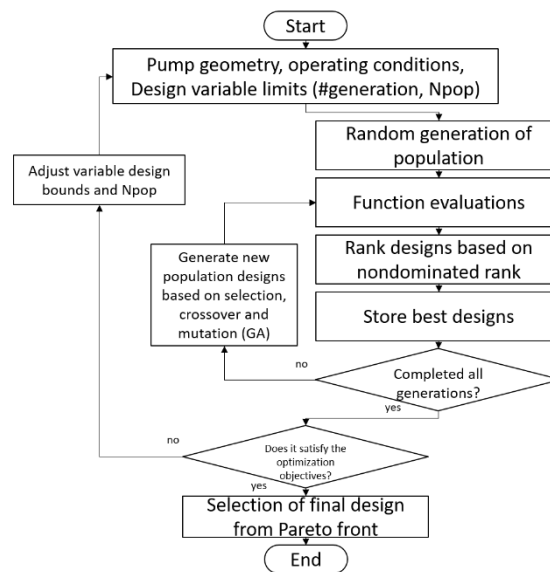


Figure 9. Valve plate optimization flowchart.

3.1. Valve Plate Optimization Design Parameters

The main design parameters are taken from the preliminary phase design. A common design feature is to add precompression and decompression grooves to the valve plate openings which help to control the amount of flow to and from the chamber, see Figure 10 on the left. Figure 10 shows the top view of a valve plate example. The precompression grooves are labeled one and two in the figure. The decompression grooves are labeled three and four. The valve plate design objectives require the use of an optimization algorithm. The valve plate design analysis and optimization is the first step in the virtual prototyping process. Figure 10 on the right-hand side shows the variables that define the effective orifice area opening from the displacement chamber to the port. The endpoint variable (E) defines the end of the groove. The length (L) to the end of the ellipse defining the non-linear start of the groove. The parameter R is the semi-minor axis of the ellipse and $R \cdot x$ define the semi-major axis of the ellipse.

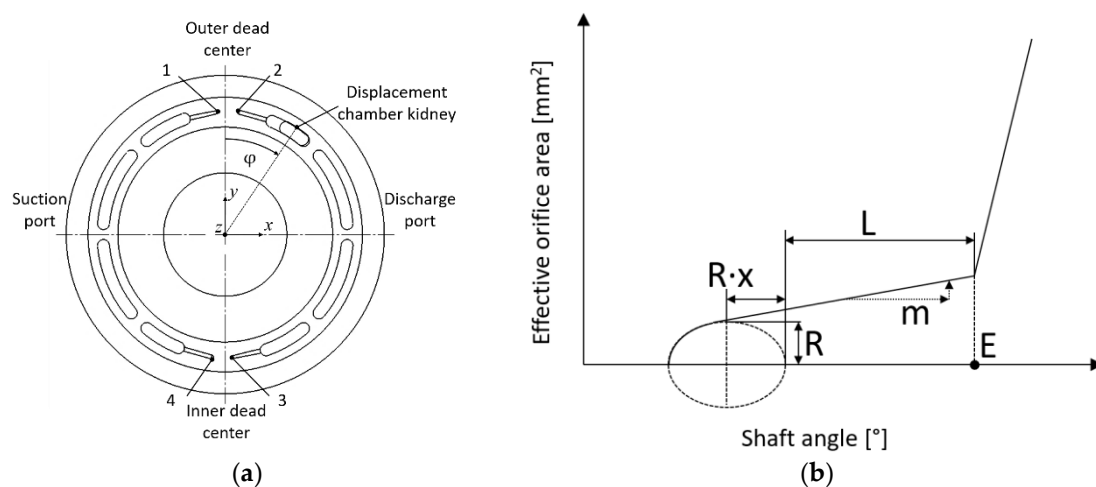


Figure 10. Valve plate top view (a) and nonlinear groove area opening profile (b).

3.2. Optimization Results Example

This subsection shows a sample of the outcome of a valve plate optimization for a 24cc axial piston machine. In this example, one of the design goals was to achieve low control effort (low mean M_x), and the value was required to be negative such that the swash plate de-strokes in case of a failure in the control system. The mean M_x objective allows for the design space to be reduced since it is largely dependent on the location where the groove finishes. Figure 11 on the left shows the volumetric efficiency plotted against the ΔQ_{hp} with the color representing the ΔM_x . It is shown that ΔQ_{hp} is all over the plot concerning the volumetric efficiency; there is not a high correlation between the two. Figure 11 show a high correlation between the volumetric efficiency and the ΔM_x . The lowest values for ΔM_x are only possible by compromising the volumetric efficiency of the axial piston machine at the volumetric efficiency operating condition. Both these figures also depict the recommended design out of this optimization with a red filled circle. It can be observed that it only has about 10% volumetric efficiency at the volumetric efficiency operating condition, but it performs well for both the ΔM_x and ΔQ_{HP} . A different publication will be issued in all the intricacies of valve plate optimization since it is highly critical for the overall performance of axial piston machines. Compressibility and internal leakage losses which impact volumetric efficiency can make up to 30% or more of the overall power losses in an axial piston machine.

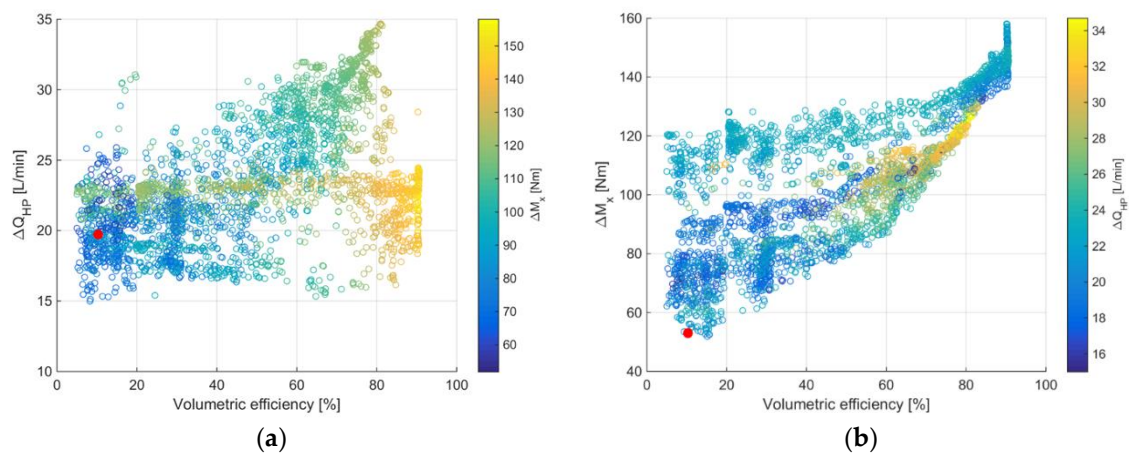


Figure 11. Valve plate optimization results Volumetric Efficiency vs ΔQ_{hp} (a) and Volumetric Efficiency vs ΔM_x (b).

4. Virtual Prototyping for Solid Bodies

Also, the impact of the elastic deformation of the end case/port block on the cylinder block/valve plate interface has been taken into account [40]. An example of virtual prototyping for a single lubricating interface was previously done by changing the sealing land dimensions and material combinations of the valve plate [41]. These researches demonstrated how crucial surface elastic deformations are to the pressure build-up in the thin fluid film due to hydrodynamic effects.

This section expands on the design of the components impacting the three lubricating interfaces. Previous research [40] show the impact the lubricating interfaces. Also, the axial piston machine in virtual prototyping is designed to be as compact as possible, without compromising performance due to the lack of structural stiffness. Therefore, the machine parts are designed while considering the impact they have on the lubricating interfaces and size.

Figure 12 shows the flowchart for the proposed solid bodies design methodology. The process starts with the initial three-dimensional CAD models generated using the preliminary design base. In a second step, the parts are to be discretized into three-dimensional meshes using commercial software. As the third step, the mesh is assigned the corresponding boundary conditions on the solid part surfaces. The boundary conditions are set in similarly as they are extracted from the lubricating interfaces

non-isothermal gap module by considering rigid bodies and a fixed parallel fluid film. An example of such a pressure field for the slipper/swash plate interface is shown in Figure 13. The example shown in these figures has an inlet pressure condition of 20 bar and an outlet pressure condition of 450 bar. This pressure field is then applied to the sliding surface of the component. Additionally, the solid bodies are constrained to avoid rigid motion, shown in Figure 14. These constraints are set by utilizing either a zero Dirichlet constraint condition or the inertia relief constraint method. Figure 13 shows the FEM analysis set up for the swash plate using the pressure calculated in the TEHD model and using zero Dirichlet constraints on the rolling bearing surfaces. The swash plate and valve plate/end case assembly utilize the zero Dirichlet constraint method since in the physical world they are mechanically constrained by the housing.

Once all the boundary conditions and constraints have been set to mimic the real mechanical system. The FEM analysis is conducted using commercial software (in the example shown Altair RADIOSS was utilized). The maximum relative deformation in the lubricating sliding surface is evaluated. The mechanical properties of the part need to be considered such as keeping the maximum stress below the material's yielding point. The relative surface deflection is the cost function because it is what impacts the lubricating interface, not the total deformation. The algorithm then compares the maximum relative deformation against the maximum relative deformation allowed defined by the designer.

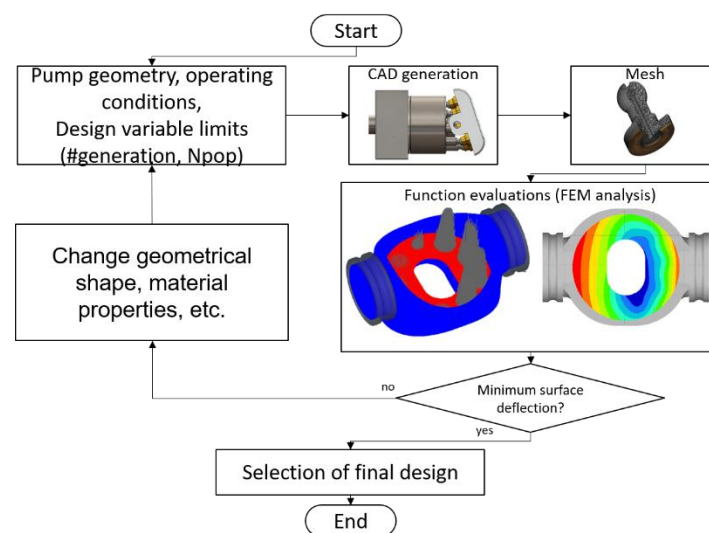


Figure 12. Area file as a function of the shaft angle.

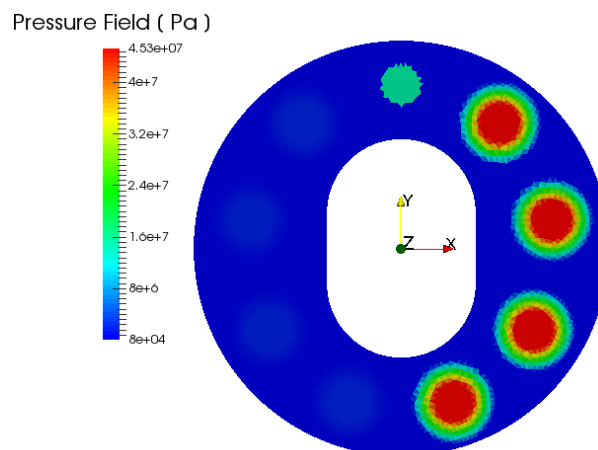


Figure 13. Example of a swash plate pressure field calculated in the TEHD model.

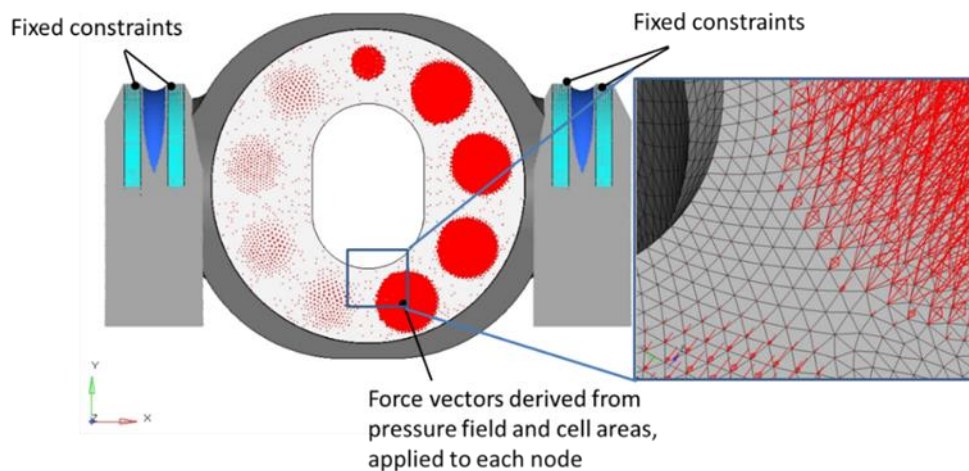


Figure 14. Example of the swash plate pressure boundaries applied to the three-dimensional mesh.

Figure 15 shows the von Mises stress distribution obtained from the FEM analysis in commercial software Altair. Note it is important to maintain the stress magnitude below the yield strength of the selected material. Figure 15 shows the swash plate's sliding surface deflection in the z-direction due to the pressure loading imposed by the fluid film between the slipper and the swash plate. The maximum surface deflection is compared against the previous design and the designer's requirements. If the design doesn't satisfy the conditions of being the smallest deflection and being below a specific value, the geometry or material selection are modified accordingly. An example of a geometrical design is adding geometrical features which add stiffness to the component. Once the design has achieved the desired relative surface deflection, the design can then be used in the following steps of virtual prototyping. This process might need to be repeated if in the following step the maximum relative deformation is found to be not sufficiently low.

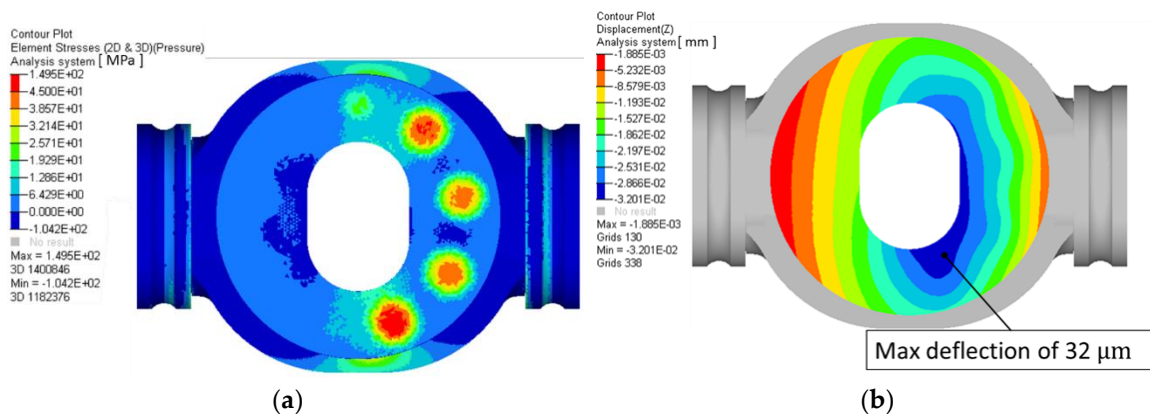


Figure 15. Example of swash plate von Mises stress distribution on loaded swash plate (a) and surface deformation of the sliding surface in the normal direction to the surface (b).

5. Virtual Prototyping for Lubricating Interfaces

The objective of the virtual prototyping for the lubricating interfaces is to minimize the energy dissipation while maintaining full fluid film lubrication over the entire range of operating conditions. The first step is to simulate the performance of the lubricating interfaces resulting from the parameters defined in the preliminary design phase. The following parameters will be calculated:

- Fluid film thickness between the piston and the cylinder bore, the cylinder block and the valve plate, and the slipper and the swash plate.
- Pressure fields in the fluid film in the lubricating interfaces.

- Leakage flows in all three lubricating interfaces.
- Energy dissipation due to viscous flow in all three lubricating interfaces.
- Temperature distributions in the fluid film and main pump parts (cylinder block, piston, valve plate, slipper, swash plate, and end case).
- Surface deformations of the cylinder block, piston, valve plate, slipper, and swash plate due to pressure and thermal loading of these parts.

The second step, after the preliminary design analysis of all three lubricating interfaces, the lubricating interface design process shown in Figure 16 can be started. The lubricating interface design process starts with the selection of material for the cylinder block, the valve plate and end case assembly, the slipper, the swash plate, and the piston. The operating conditions which will be used for the function evaluations of the objectives and constraints; the fluid properties as a function of pressure and temperature; and the variable bounds for the design of experiments.

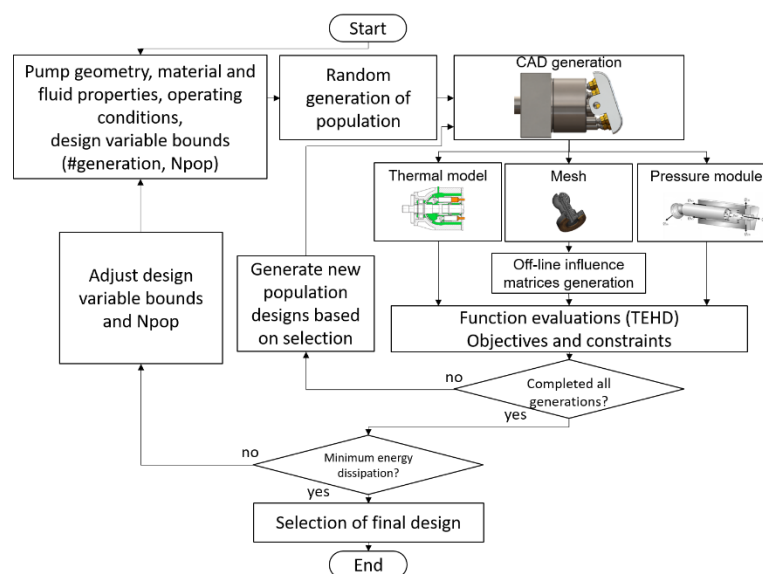


Figure 16. Lubricating interfaces general design methodology.

The next step is to generate a DOE. The design parameters to be varied are different for each of the lubricating interfaces and will be discussed in more detail later. Once, the design parameters have been selected the DOE is generated. The DOE method is selected based on the number of design variables and operating conditions considered for the optimization. The following step is to generate the new CAD models based on the design parameters populated by the DOE.

Next, the displacement chamber instantaneous pressure calculation is performed in the pressure module. In parallel, the CAD models are meshed to be utilized in the pressure and thermal deformations, as well as for the calculation of the temperature distribution in the solid bodies. The three-dimensional CAD models are discretized into three-dimensional mesh utilizing tetrahedral elements in commercial software. Once the mesh has been generated the corresponding influence matrix needs to be calculated off-line to be able to predict elastic surface deformations due to pressure in the fluid film. Also, in parallel, the thermal model is utilized to predict the discharge port and the case volume temperatures, utilizing the information from the operating conditions, the pressure module and the TEHD if it is not the first iteration.

Afterwards, the DOE for the lubricating interfaces are evaluated by running simulations using the TEHD. The simulation results yield the fluid film thickness, pressure field distributions, temperature field distributions, leakage flows, surface deformations due to pressure and thermal effects, and the energy dissipation due to viscous flow for all three lubricating interfaces. The energy dissipation is the cost function to be minimized, and the constraint is to maintain functionality by limiting the allowed

areas of minimum fluid film thickness which can result in metal-to-metal contact, therefore failure of the machine.

The selected design parameters can be optimized by a variety of optimization schemes on a case by case scenario. A response surface or surrogate model method is recommended since it reduces the count of function evaluations significantly. The function evaluations of the lubricating interfaces are of high computational cost. Therefore, a response surface approach coupled with a genetic algorithm reduces the time spent on the virtual prototyping. A sequential approximate method can be coupled with the surrogate model approach. This method will refine the area near the optimum design space. For this design refinement, the lower and upper bounds for the DOE are adjusted, and the function evaluations are analyzed using the TEHD model and a second surrogate model is generated. Once the response surface method has finalized the function evaluation for the optimal design, the method should be rerun using the TEHD to validate the surrogate model simulation results. Also, if the number of design variables is small, a full factorial design approach is feasible, and it doesn't require any additional optimization schemes. The minimum energy dissipation design is compared against the desired minimum energy dissipation desired for the axial piston machine. If the design achieves the desired performance, the design process is completed successfully.

5.1. Material Selection

The three lubricating interfaces' performance is strongly influenced by the material selection of the cylinder block, the valve plate and end case, the slipper, the swash plate, and the piston. One key step within the virtual prototyping process is the selection and analysis of the interface performance dependent on different material properties. The distinctive pressure and thermal deflections of the solid bodies will result in a different fluid film thickness distribution and energy dissipation. The TEHD model considers all the material properties described in Table 2. The virtual prototyping methodology allows the exploration of design possibilities which have not been explored in traditional pump design of trial and error. The material selection coupled with the FEM analysis described in previous sections permits the introduction of optimal materials to be used for the solid parts. The material properties influencing the lubricating interface behavior are:

Table 2. Material properties influencing the lubricating interfaces' behavior.

Description	Symbol	Unit
Young's modulus	E	(Pa)
Poisson's ratio	ν	(-)
Density	ρ	(kg/m ³)
Thermal conductivity	λ	(W/mK)
Coefficient of linear thermal expansion	α	(-)

5.2. Operating Conditions

Axial piston machines usually operate in a wide range of operating conditions. For the virtual prototyping process, the selection of these operating conditions is of paramount importance to minimize the computational effort while considering the crucial operating condition points for all three lubricating interfaces. The computational effort for the virtual prototyping has a linear relation with the number of operating conditions to be considered. The cylinder block/valve plate, slipper/swash plate and piston/cylinder interface behave differently at different operating conditions. Each interface has some point of higher and lower difficulty to carry the loads. The cylinder block/valve plate interface performs poorly at low pressure operating conditions since it is designed to work with a large hydrostatic component. The piston/cylinder interface has the largest difficulties at conditions of low relative sliding velocities at high pressures, which correspond to low displacements and low rotating speeds. The slipper/swash plate interface also relies heavily on the hydrostatic component, so it performs poorly at conditions of low-pressure conditions. The corner operating conditions must be evaluated to approve

a final design. All three lubricating interfaces are optimized at the highest power corner operating condition (max operating pressure, max speed, and max displacement) and the operating conditions at which the lubricating interfaces can encounter performance issues. The max power operating condition represents the highest energy dissipation from the three lubricating interfaces. Nominal operating conditions at which the axial piston machine operates most frequently should be added into the design process. An example of a possible set of operating conditions to be utilized for the function evaluations in the virtual prototyping process is described in Table 3. This table includes the eight corner operating conditions, and in additions, there are two extra operating conditions at more moderate conditions. The last ones can be varied depending on the application of the axial piston machine.

Table 3. Operating conditions.

Operating Condition	Speed (rpm)	Δp (bar)	Displacement (%)
1	max	max	max
2	max	max	min
3	max	min	min
4	max	min	max
5	min	max	max
6	min	max	min
7	min	min	min
8	min	min	max
9	min	max	moderate
10	moderate	max	max

5.3. Cylinder Block/Valve Plate Interface Design Variables within Virtual Prototyping Variables

The cylinder block/valve plate interface design is done utilizing the methodology described previously and depicted in Figure 16. This section describes specific details of the cylinder block/valve plate interface, shown in Figure 17. The cylinder block rotates on top of the valve plate; both are concentric to the shaft. The lubricating interface is defined by the intersection of boundary surfaces from the cylinder block and valve plate.

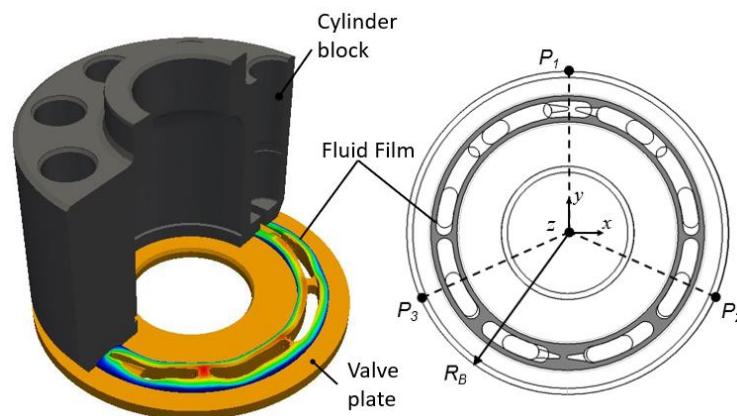


Figure 17. Cylinder block/valve plate interface representation.

The design parameters that are considered for the cylinder block/valve plate interface are the sealing land dimensions defined by the inner gap diameter d_{gi} , inner port opening diameter d_{oi} , pitch diameter d_B , outer port opening diameter d_{oo} , outer gap diameter d_{go} , and kidney length l_{kD} as shown in Figure 18a. Other design parameters are the cylinder block length (l_B) and cylinder block channel length (l_{CanalB}) which are shown in Figure 18b. The elastic deformation influences the performance of the interface therefore it might be necessary to return to the solid bodies design represented in Section 4. Additionally to the parameters shown above, the authors have done previous research [42] where the influence of micro-surfacing at the cylinder block/valve plate interface is detailed.

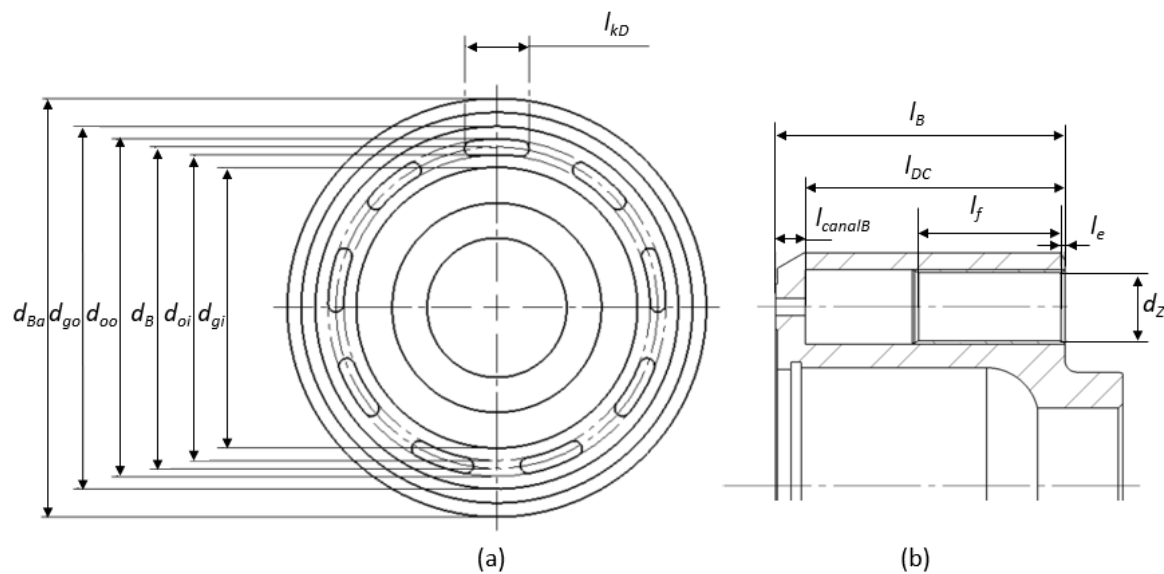


Figure 18. Main dimensions impacting the cylinder block/valve plate interface. Cylinder block's bottom view with sealing land dimensions (a) and lateral cross section with main dimensions (b).

5.4. Slipper/Swash Plate Interface Design Variables within Virtual Prototyping Variables

The slipper swash plate interface is represented in Figure 19 which shows a slipper sliding on top of the swash plate's running surface. The design parameters for the slipper/swash plate interface are shown in Figure 20. The slipper sealing land dimensions are the inner and outer sealing land diameters, d_{inG} and d_{outG} shown in as well. Similarly, as in the previous interface, the elastic deformation of the slipper and swash plate will have an impact on the lubricating interface. Therefore, the slipper and swash plate geometrical shape and material selection may need to be modified in a design iteration going through the process detailed in Section 4.

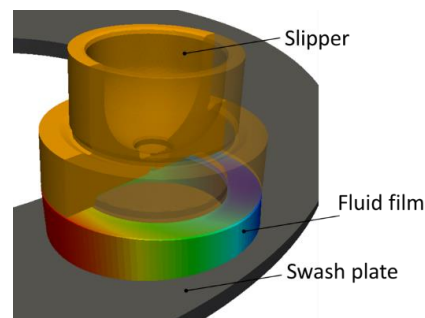


Figure 19. Slipper/swash plate interface schematic.

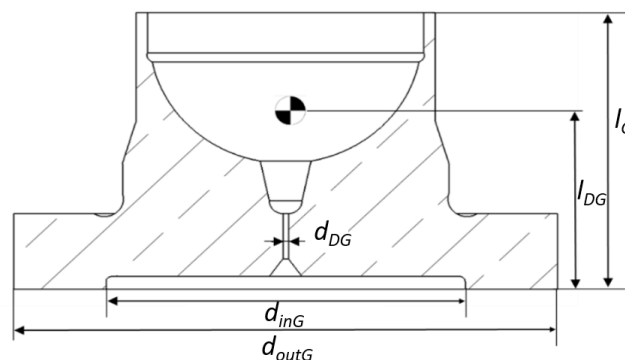


Figure 20. Slipper main dimensions.

5.5. Piston/Cylinder Interface Design Variables within Virtual Prototyping Variables

The piston/cylinder interface is shown in Figure 21; the lubricating interface is represented using a color scheme which indicates the pressure field. The design parameters for the piston/cylinder interface are shown in Figure 22. The diameter of the piston d_K and the diameter of the bore d_Z are the most critical dimensions since these together define the clearance of the lubricating interface which impacts the fluid flow through the gap and viscous friction. The length of the piston l_K and the length of the bore l_f are important as well as they define the rest of the lubricating interface geometry. The piston/cylinder interfaces performance is influenced by but not limited to the variables described in this section. Therefore, the solid bodies may need to be redesigned as shown in Section 4.

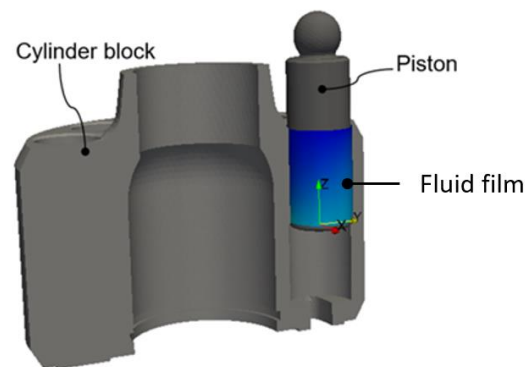


Figure 21. Piston/cylinder interface schematic.

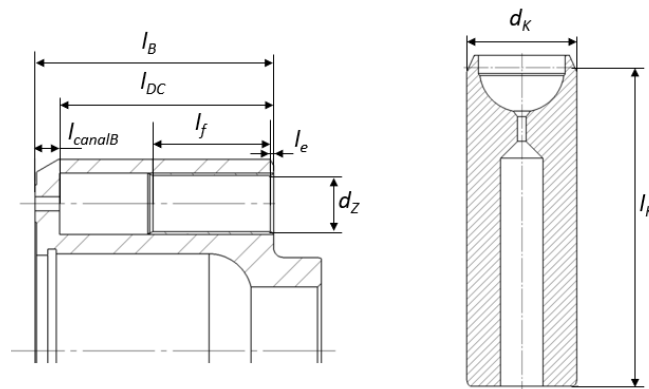


Figure 22. Main dimensions impacting the piston/cylinder interface.

6. Experimental Results and Comparison Against Simulations

In this section, the previously described methodology was utilized to design virtually a closed-circuit fixed displacement axial piston machine, the prototype was built, and tested. The prototype was selected to have a displacement volume of 24 cc per revolution. The machine has a swash plate angle of 21° . The chosen number of pistons was nine. The housing and end case were machined from solid parts of steel due to monetary and time cost limitations from castings. Figure 23 shows the cross-sectional views of the computational model of the axial piston machine. Figure 24 shows the physical prototype pictures of the disassembled components and the partially assembled machine. The experimental tests covered the main performance parameters of the pump such as volumetric and mechanical efficiencies. Other performance parameters such as noise emission and flow and pressure ripple were not part of this paper in the interest of maintaining a reasonable length.

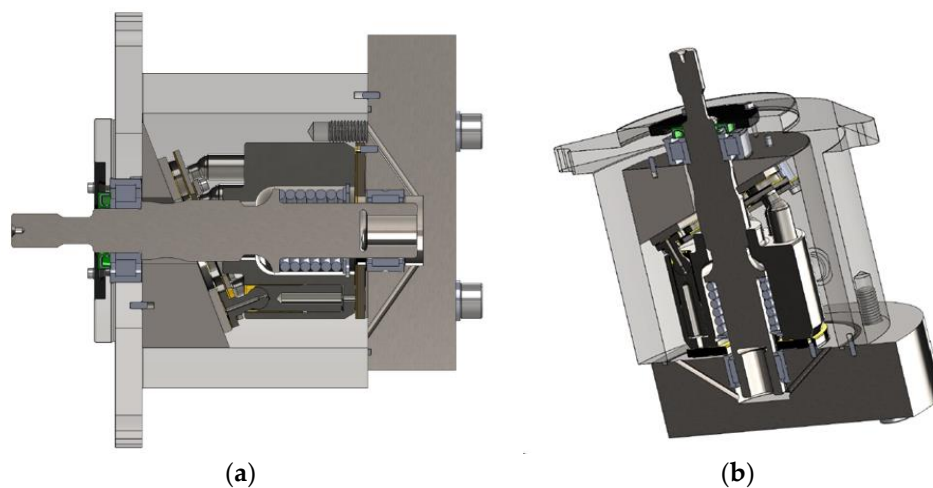


Figure 23. Closed circuit prototype computational model. Cross-section lateral view (a) and isometric view (b) are shown.

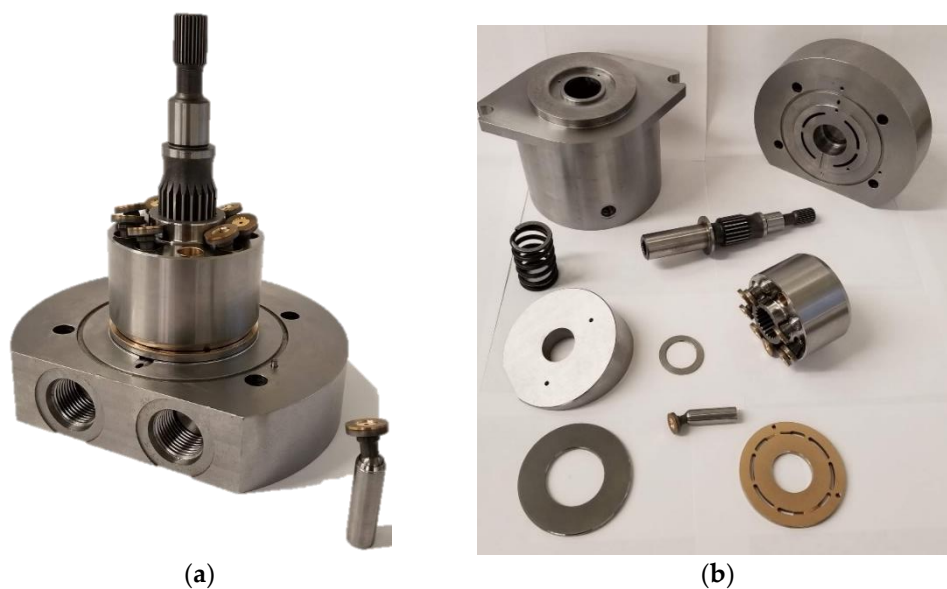


Figure 24. Closed circuit physical prototype.

The swash plate type axial piston machine prototype was specified to be able in a wide range of operating conditions. The pressure differential ranged from 0–400 bar, with max operating pressure of 450 bar. The prototype's rotating group components were designed to range from 0–21°. The temperature could vary from 0 °C to 80 °C. The closed-circuit pump prototype was tested on a wide range of operating conditions to prove that this pump could operate in the complete range that it was intended to. The operating conditions are described in Table 4.

Table 4. Operating conditions range.

Description	Specification
Pressure differential	50, 100, 200, 300, 400 (bar)
Speed	1000, 2000, 3000 (rpm)
Displacement	100 (%)
Temperature	42, 52, 72 (°C)

6.1. Test Rig Configuration

The 24 cc closed circuit pump was mounted on a 225 kW test rig at the Maha Fluid Power Research Center, and the test rig was appropriately instrumentalized to measure accurately within reason the performance of the newly prototyped closed-circuit pump. Figure 25 shows a picture of the test rig set up.



Figure 25. Steady state test rig with 24 cc prototype mounted.

Figure 26 shows the ISO hydraulic circuit of the test rig. The test rig setup is quite simple. The drive is an electric motor (1) connected to the pump via couplings and a torque cell (2,3). The inlet of the pump (4) is supplied by a power supply at a constant pressure of 20 bar. The outlet of the pump goes to a flowmeter (9) and then the pressure relief valve (13) which is acting as the load in this test rig setup. The test rig was instrumentalized with some transducers to record critical performance data from the positive displacement machine such as thermocouples on the inlet, outlet, and drain lines. Pressure transducers were also present in the inlet, outlet, and drain lines. Table 5 describes all the components utilized in this experimental setup. It includes the data acquisition (DAQ) cards from National Instruments (Austin, TX) that were utilized to record the signals coming from all the transducers. An ISO 32 fluid was utilized for these experiments.

Table 5. Test rig set up components.

ID	Description	Specification
1	Electric drive	Max power: 225 Kw, Max torque 615 Nm @3500 rpm
2, 3	Staiger Mohilo torque cell	0–500 Nm range, error $\pm 0.2\%$ of full scale
4	Closed circuit pump	24cc, fixed displacement, max torque 160 Nm @ $\Delta p = 400$ bar
5	Pressure transducer	WIKA S-10, 0–100 bar, 0.125% BFS
6, 8, 11	Thermocouple	Omega K-type Thermocouple, 2.2 °C error limit
7	Pressure transducer	HYDAC HAD 4445, 0.5% BFS
9	Flowmeter	VSE VS 10 Gear type, 1.2–250 L/min, 0.3% accuracy
10	Pressure transducer	WIKA S-10, 0–25 bar, 0.125% BFS
12	Flowmeter	VSE VS 0.2 Gear type, 0.02–18 L/min, 0.3% accuracy
13	Pressure relief	Max flow 350 L/min
14	DAQ	NI cDAQ, NI 9213
15	DAQ	NI cDAQ, NI 9201

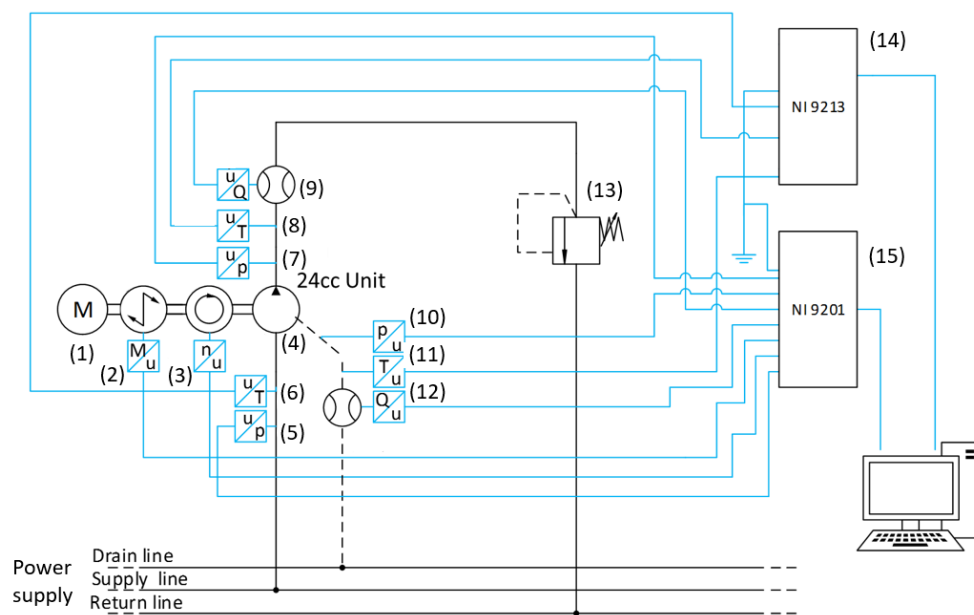


Figure 26. ISO schematic of the steady-state test rig.

6.2. Experimental Results

The test rig experimental results are shown in this section. The pump was run through total of 45 operating conditions under steady state conditions. The pump ran for a total of 30 h. The experimental results shown here were recorded after the initial break-in. Figures 27–29 shows the performance data of the steady state measurements taken of the 24 cc axial piston machine prototype. The measurements are shown for 15 operating conditions only since the measurements follow very similar across the two additional temperature of 42 °C and 72 °C. The total efficiency of the machine fluctuates from 60 to 82%. The results vary with speed and pressure. It is clear from these measurement results that the axial piston machine has lower efficiencies at the 1000 rpm condition. The low efficiency can be explained due to high internal leakage due to the valve plate design. Previous research by Kim et al. [43] shows similar results in simulation.

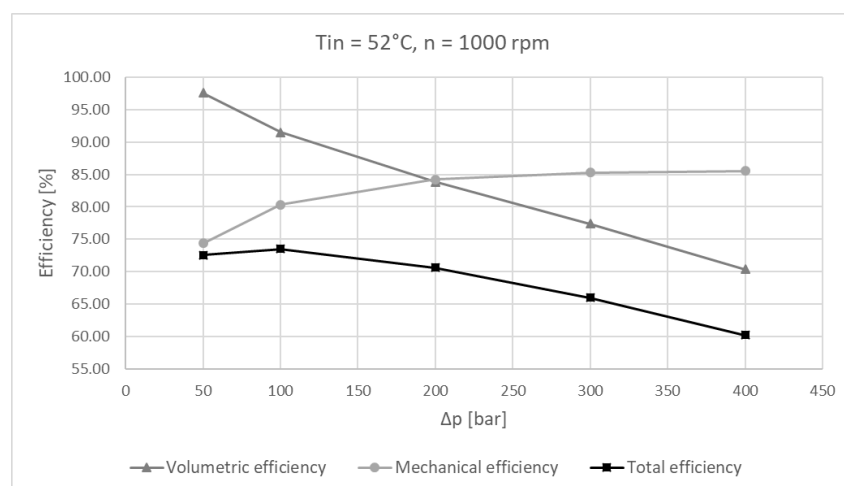


Figure 27. Prototype efficiencies at $n = 1000$ rpm, $T = 52$ °C, and $\Delta p = 50$ –400 bar.

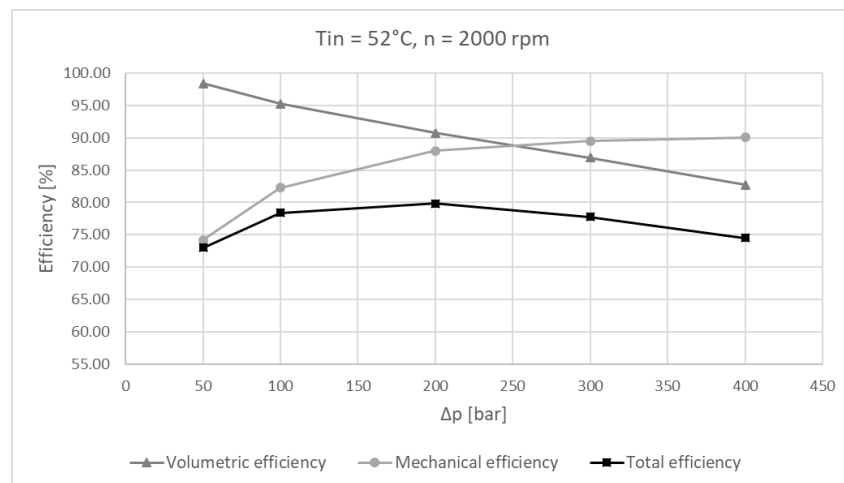


Figure 28. Prototype efficiencies at $n = 2000$ rpm, $T = 52$ °C, and $\Delta p = 50$ –400 bar.

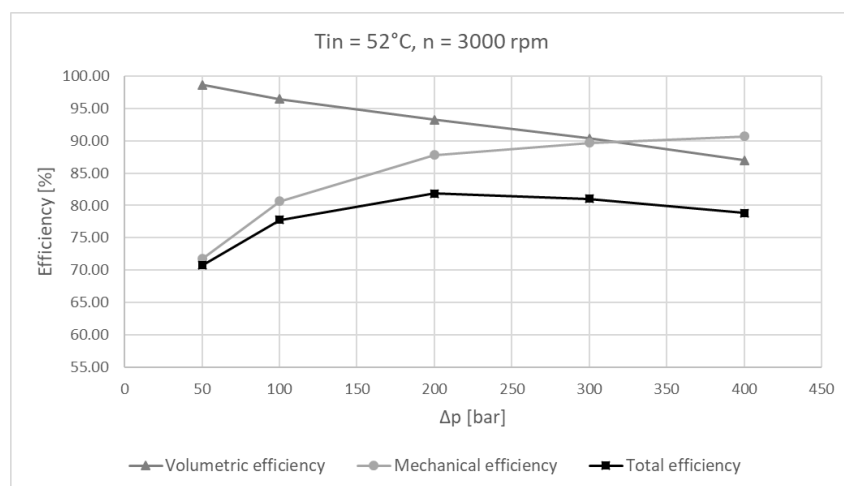


Figure 29. Prototype efficiencies at $n = 3000$ rpm, $T = 52$ °C, and $\Delta p = 50$ –400 bar.

6.3. Experimental Results and Simulation Comparison

Experimental results are compared to simulation results utilizing the same operating conditions, geometry, fluid properties, etc. Figures 30–32 show the volumetric efficiency measured compared against the simulation results calculated utilizing the pressure module described in Section 1.2. The pressure module took as input the leakage flow rate measured and the operating conditions, geometry, fluid properties, etc. The lumped parameter approach requires the use of a discharge coefficient α shown in Equation (2). The simulation model utilized a discharge coefficient of 0.8 in the simulation results. The model results for the volumetric efficiency follow those from the measurements closely.

The mechanical efficiency comparison for three operating conditions is shown in Figures 33–35. The TEHD model could replicate the trends but not the magnitudes. More details on the mechanical efficiency will be discussed in Section 7.

The drain flow measurements are shown in Figures 36–38. The drain flow comes from the leakage flows through the three lubricating interfaces from the fluid in the ports and displacement chambers to the case volume. The drain flow is considered a necessary energy loss since this flow is also responsible for rejecting the heat generated from the pump due to viscous friction. The TEHD models follow the measured drain flows closely.

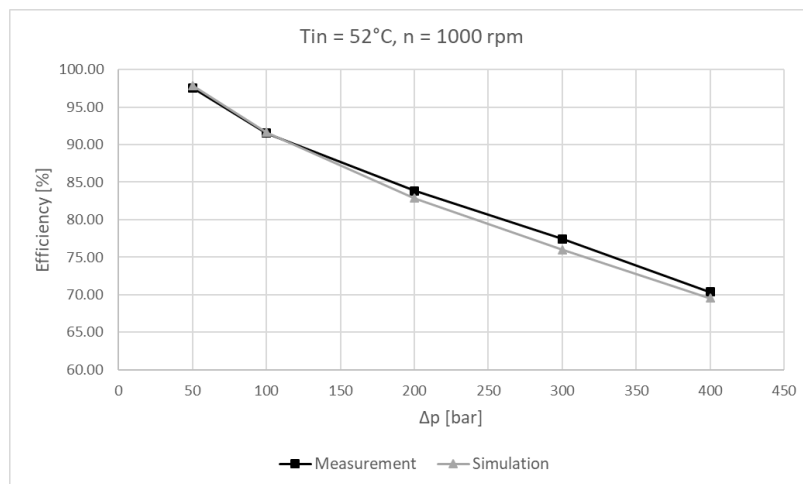


Figure 30. Volumetric efficiency measured compared against simulated at $n = 1000$ rpm, $T = 52$ °C, and $\Delta p = 50$ –400 bar.

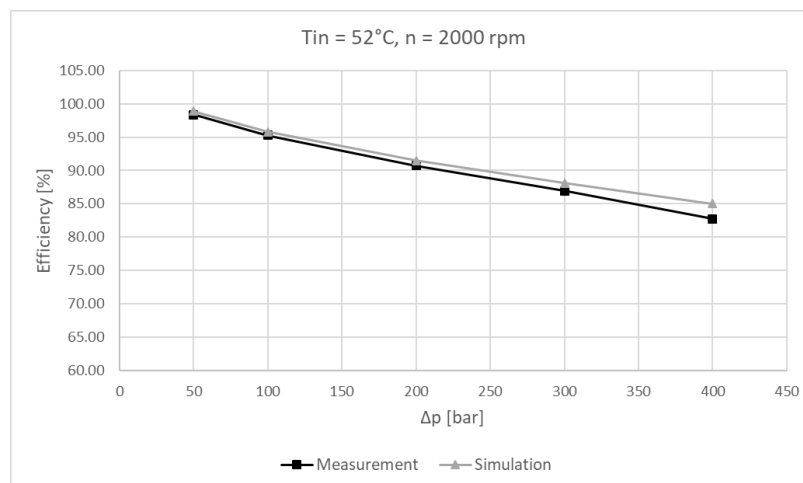


Figure 31. Volumetric efficiency measured compared against simulated at $n = 2000$ rpm, $T = 52$ °C, and $\Delta p = 50$ –400 bar.

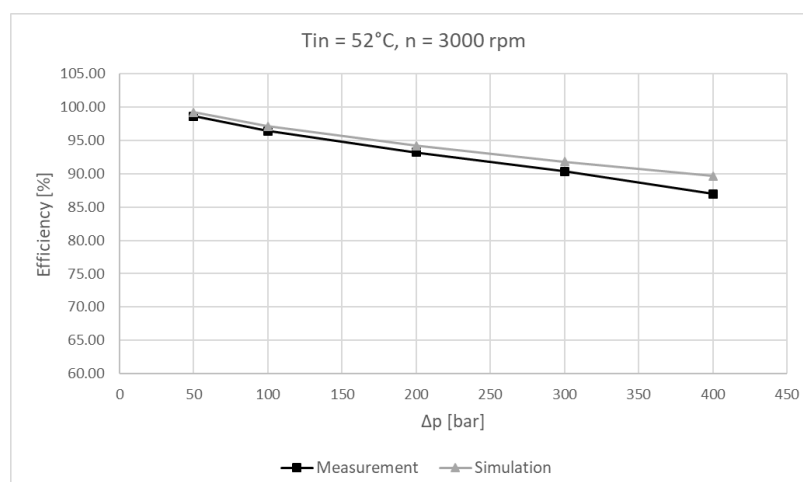


Figure 32. Volumetric efficiency measured compared against simulated at $n = 3000$ rpm, $T = 52$ °C, and $\Delta p = 50$ –400 bar.

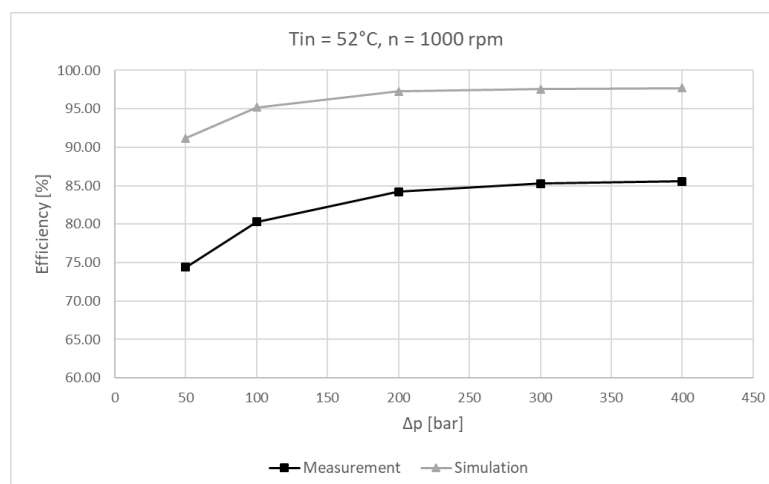


Figure 33. Mechanical efficiency measured compared against simulated at $n = 1000$ rpm, $T = 52^\circ\text{C}$, and $\Delta p = 50$ –400 bar.

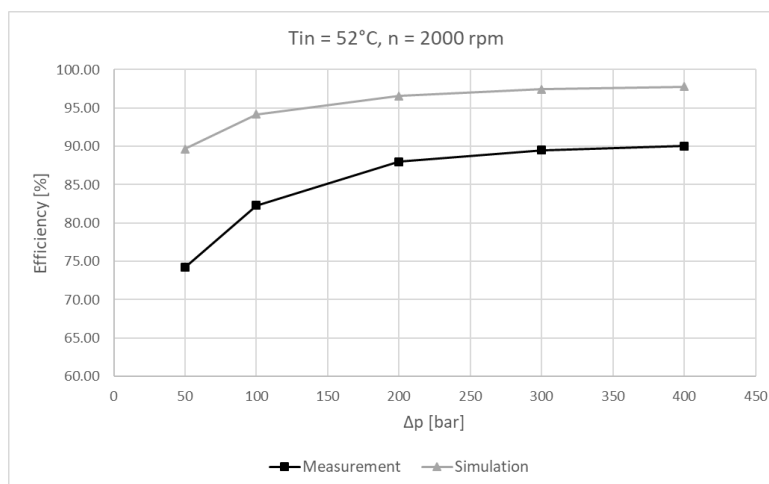


Figure 34. Mechanical efficiency measured compared against simulated at $n = 2000$ rpm, $T = 52^\circ\text{C}$, and $\Delta p = 50$ –400 bar.

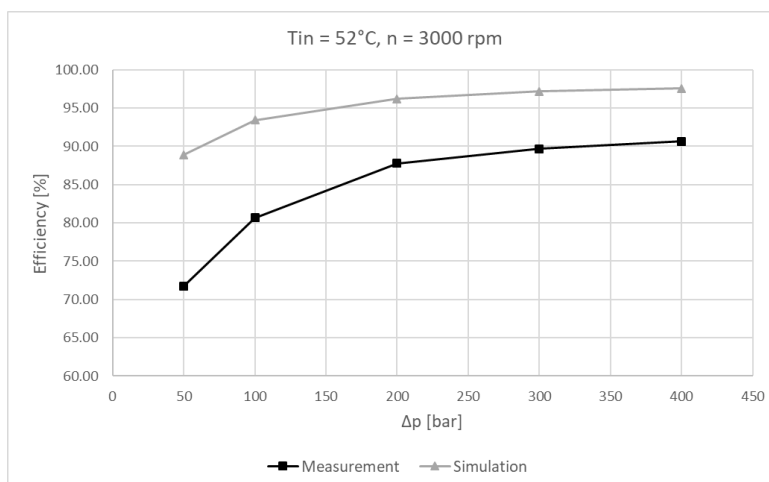


Figure 35. Mechanical efficiency measured compared against simulated at $n = 3000$ rpm, $T = 52^\circ\text{C}$, and $\Delta p = 50$ –400 bar.

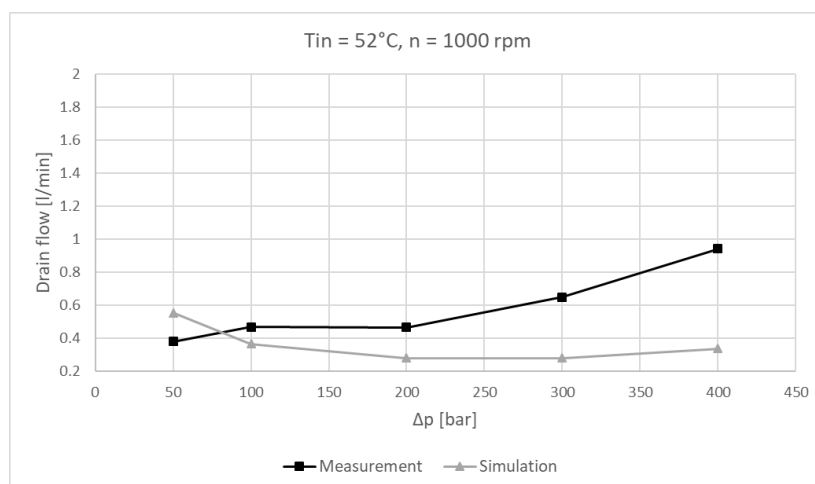


Figure 36. Drain flow measured compared against simulated at $n = 1000$ rpm, $T = 52$ °C, and $\Delta p = 50$ –400 bar.

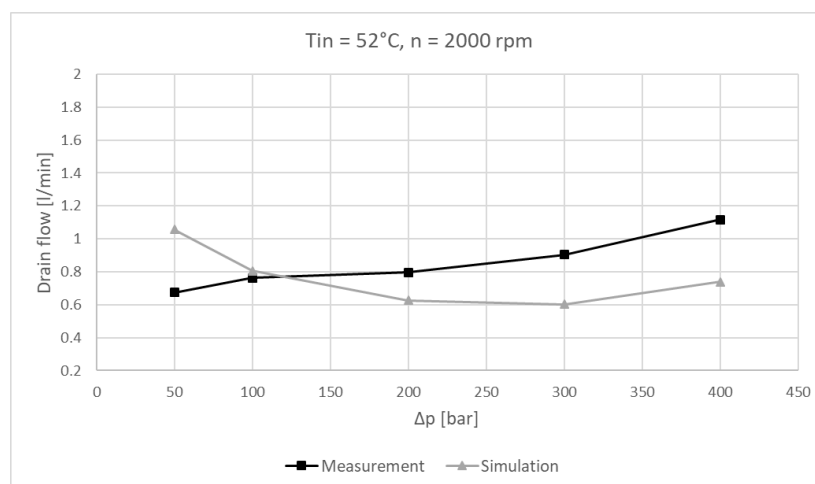


Figure 37. Drain flow measured compared against simulated at $n = 2000$ rpm, $T = 52$ °C, and $\Delta p = 50$ –400 bar.

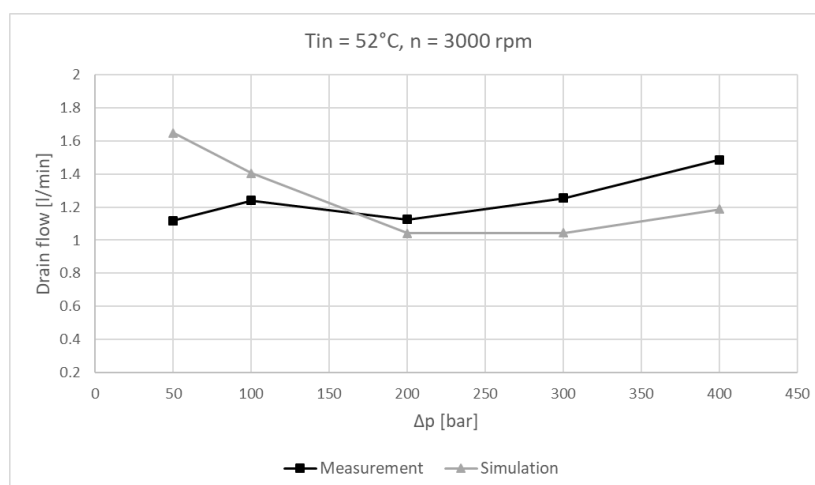


Figure 38. Drain flow measured compared against simulated at $n = 3000$ rpm, $T = 52$ °C, and $\Delta p = 50$ –400 bar.

7. Discussion

The efficiencies measured from the prototype were not as high as those from the simulation due to external factors other than the lubricating interfaces. One of the main contributors to the unanticipated torque loss is the friction between the slipper retainer plate sliding and the fixed hold-down mechanism. In the prototype wear was discovered after disassembly as shown in Figure 39. Other sources of friction are also present such as shaft bearings and churning losses. This paper focuses on the design of the components directly impacting the performance of the lubricating interface only. The design of the slipper retaining plate is out of the scope of this work. For future research and further refinement of this model, this should be taken into consideration. The author believes the hydromechanical comparison done in Section 6.3 is reliable because the TEHD models closely match the drain flows, this is only possible if the fluid film thickness is accurately predicted in the models and the drain flow is derived from the fluid film's fluid velocity field and fluid film's gap thickness.



Figure 39. Worn slipper retainer plate after measurements.

8. Conclusions

This article presented a novel methodology to design swash plate type axial piston machines based on computationally based approach. The methodology uses advanced numerical models and optimization schemes for the optimization of the valve plate and the lubricating interfaces. The valve plate optimization objective functions are to maximize the volumetric efficiency, minimize structure-borne noise sources, swash plate moment amplitudes ΔM_x and ΔM_y , fluid-borne noise source flow ripple ΔQ_{HP} , and control effort of the axial piston machine's swash plate (mean M_x).

The lubricating interfaces' virtual prototyping methodology utilizes a multi-domain numerical model which includes the calculation of the instantaneous pressure in the displacement chamber, the prediction of the delivery port and case volume temperatures and a novel thermo-elastohydrodynamic model (TEHD). The main design parameters modifying the sealing land dimensions and clearance ratios are varied to minimize the energy dissipation to achieve the highest efficiency design. The design simultaneously needs to maintain a full fluid film between the cylinder block and valve plate, piston and cylinder bore, and the slipper and swash plate by balancing the external loads imposed on the rotating parts due to the kinematics of the machine.

The methodology's feasibility was validated via the design and build of the first ever axial piston pump within a virtual prototyping framework. The prototype is a 24 cc closed-circuit pump prototype. Compared against simulation results, the measurement results showed a total efficiency ranging from 60 to 82 % which is lower from what the simulations showed. The prototype has the potential to be competitive to machines in the market. The mechanical efficiency could have been higher, but other problems external to the lubricating interfaces arise. The volumetric efficiency can be improved lower operating speeds by modifying the cost functions that were formulated in the valve plate optimization phase. The prototype was the result of a single iteration of the virtual prototyping methodology.

Author Contributions: The work in this paper was performed by R.C. as a Ph.D. student at the Maha Fluid Power Research Center under the supervision of M.I. The preparation of this manuscript was done mainly by R.C. due to the tragic loss of M.I. on the year of 2018. None of this work would've been possible without the unwavering support from M.I.

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Conflicts of Interest: The authors declare no conflict of interest.

Nomenclature

Symbols	Description	Units
h	Fluid film height	m
u	Velocity on x-axis	m/s
v	Velocity on y-axis	m/s
p	Pressure	Bar
t	Time	S
w	Velocity on z-axis	m/s
K	Bulk modulus	GPa
V	Volume	m ³
Q	Volumetric flow	m ³ /s
α	Discharge coefficient	-
A_r	Minimum cross-section area	m ²
μ	Dynamic viscosity	Pa·s
φ	Shaft angle	°
ρ	Density	Kg/ m ³
Subscripts	Description	
i	Individual displacement chamber -	
a	Sliding surface	-
b	Fixed surface	-
DC	Displacement chamber	-
SK	Piston/cylinder interface	-
SB	Cylinder block/valve plate interface	-
SG	Slipper/swash plate interface	-

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