
Ultra Compact High-Speed Axial Piston Pumps for Electrified Mobile Machinery: Optimizing and Testing Tribological Gaps for 10,000 rpm

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Abstract.

High-speed electric drives are being adopted in mobile machinery, with automotive e-motors commonly operating up to 10,000 rpm and beyond. To exploit this speed range in electro-hydraulic powertrains, hydrostatic units must be qualified and redesigned for ultra-high rotational speed, where tribological limits are often the limiting factor, when boosting the inlet pressure. In the present work, a wobble-plate type axial piston pump platform was selected due to advantage of mass independence in the rotating kit. This rotating kit can be operated at 61.1 cm³/rev (fixed 12° wobble angle) or 94.6 cm³/rev (18°). The baseline pump is rated at 2,400 rpm in open-circuit operation, where the self-priming speed limit was confirmed by measurements as the primary limit.

To isolate the tribological speed limit, the inlet was pressurized and a temperature and particle monitoring instrumentation was implemented. Temperatures were monitored at bearings, piston–cylinder gap, and slipper leakage locations. Together with online particle monitoring these metrics were used as an early warning indicator. With this methodology, stable overspeed operation up to 4,000 rpm was achieved at various pressures. At 3,800 rpm and 300 bar, a rapid thermal rise at the slipper interface and increased particle activity indicated imminent failure, and the test was terminated intentionally before catastrophic damage occurred. Post-test inspection confirmed that the pump remained intact, enabling failure mode analysis without obscured failure chains.

The paper presents the test methodology, the sensor-based stop criteria, and a simulation-guided redesign workflow for high-speed operation. Improvements of the slipper design are demonstrated by simulation and measurement, followed by a thermally adaptive, meso-structured slipper concept targeted at 10,000 rpm. In parallel, the requirements for a next-generation high-speed test rig are derived to enable testing at approximately 360 kW (61.1 cm³/rev, 10,000 rpm, 350 bar), which would upgrade this mobile pump to a power density of 17 kW/kg, outperforming all existing pumps on the market.

Keywords. Digital Twin, Thermal Modeling, Tribology, Efficiency.

1. INTRODUCTION

The electrification of mobile working machines is past its exploration stage, and increasing requirements are being imposed on hydrostatic powertrains. Electric motors have proven themselves as superior prime movers to diesel engines in terms of efficient, robustness and controllability. The technological progress both in battery and electric machine design has far from peaked, which has been primarily driven by high-volume automotive production. Power densities above 5 kW/kg have been reported for current traction motor technologies, with examples such as BYD or Tesla stating up to 5.5 kW/kg in mass production vehicles [1], [2]. In motorsport-oriented and aerospace-oriented machines, substantially higher peak power densities have been published; Lucid reported 11 kW/kg for an electric drive unit [3], and Evolito reported 18.5 kW/kg for a 240 kW peak motor at 13 kg. In parallel, prototype axial-flux machines have been demonstrated with short-term peak values far above today's series-production levels (e.g., YASA reporting up to 59 kW/kg peak in a prototype [4]). These values are not continuous ratings, but the trend indicates that further increases in electric motor power density are expected rather than stagnation.

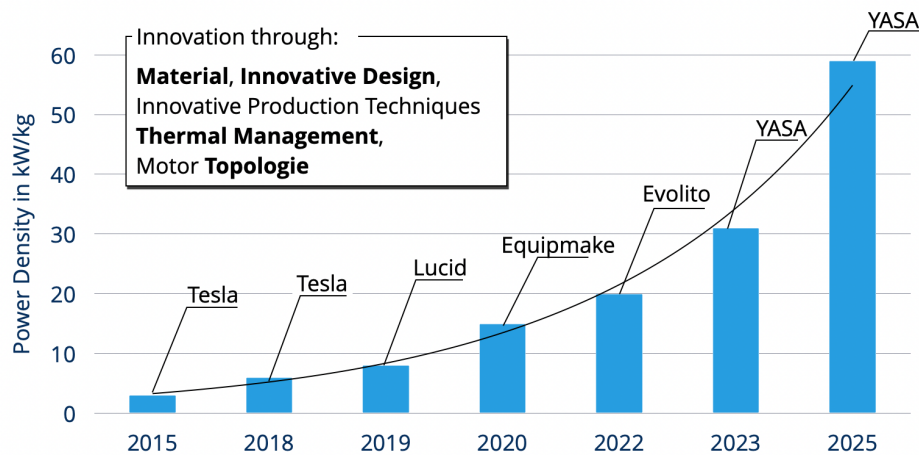


Figure 1.1. Power density trend in modern electrical motors

The main drivers for this development are mass and packaging constraints in mobile machinery and, even more strongly, in aviation. Because battery systems are heavier and larger than fuel tanks for comparable usable energy, weight and volume savings in all other subsystems are being prioritized. Across the reported high power density electric machines, one shared enabler is high rotational speed. In series-production vehicles, high efficiency regions are typically located at high machine speeds (e.g., 12,000–16,000 rpm reported for a Tesla Model 3 power unit efficiency map), and similar or higher speed levels are commonly used in modern traction and propulsion concepts. Similar high-speed motors are currently being designed for mobile construction machines [5], [6].

On the hydraulic side, several levers are available to increase power density, but progress has been limited in widely deployed products. In axial piston machines further increasing the displacement angle has been used successfully, but geometric limits and secondary

effects (longer piston stroke, increased sliding speeds, and stability constraints) restrict further gains. Higher operating pressure can raise power density, and high-pressure pumps exist; for example, Danfoss specifies closed-circuit pump pressure capability in the ~400 to 500+ bar class [7], while specialist solutions for significantly higher pressure levels (1000bar+) are available [8], [9]. However, system-level implementation at very high pressure is constrained by fittings, hoses, manifolds, and long-life requirements, particularly for medium to high flow rates.

Consequently, increasing rotational speed is considered a direct and scalable pathway to higher hydraulic power density, and it aligns with the E-motors provide. The main challenges of higher speed are concentrated inside the pump or motor itself, while the rest of the hydraulic system primarily experiences increased flow, which is generally easier to scale than pressure.

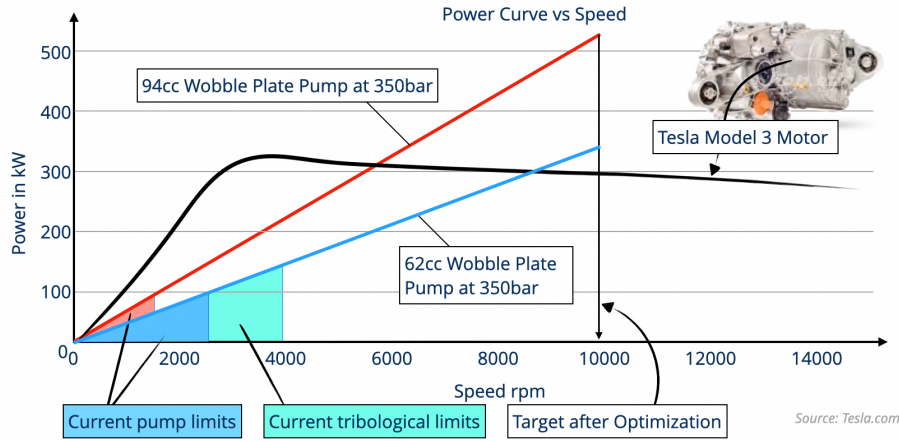


Figure 1.2. Power curves of current and targeted operation limits of a 62cc and 94cc pump, compared with a Tesla Model 3 motor [1].

In this work, the challenges associated with high-speed operation are addressed with emphasis on the tribological speed limit of fluid film interfaces for a 62cc wobble plate type axial piston pump. Methods for identifying this limit experimentally, supported by simulation-based redesign of critical components for maximum speed up to 10000 rpm, are presented along with first validations.

2. PRIOR WORK AND RESEARCH GOAL

The maximum rotational speed of hydrostatic machines can be grouped into three primary limitation classes:

1. **Fluid-dynamic limits** – e.g., self-priming capability and cavitation during the suction stroke.
2. **Body-force limits** – e.g., inertial and centrifugal forces acting on the rotating group, including cylinder-block tilt and piston stick phenomenon.

3. **Tribological limits** – e.g., fluid-film stability, hydrodynamic lift-off, and loss of contact control in lubricated interfaces.

Most hydrostatic pumps in mobile machinery are open-circuit pumps, where the inlet is supplied directly from a tank. In this configuration, the inlet pressure is usually only slightly above atmospheric and is strongly influenced by reservoir head and inlet line design. During the suction stroke, the pressure at pump inlet is reduced relative to the tank due to line losses (bends, restrictions) and viscous losses in the pipes and hoses. Because pressure and velocity are coupled, increasing flow velocity increases the local pressure drop for a given geometry, which may reduce the local pressure at the kidney region below the cavitation pressure. This mechanism has been discussed in the literature [10], [11], and it explains the characteristic volumetric efficiency drop above a certain speed in self-priming operation.

2.1. Fluid-dynamic speed limit: cylinder-block filling and cavitation

Manring expressed the limiting speed for incomplete filling using a Bernoulli-based formulation (Eq. 22 in [10]) as:

$$\omega^2 < \frac{2(P_i - P_{cav})}{\rho r_p^2 \left(1 + \left[\left(\frac{A_p}{A_k} \right)^2 - \left(\frac{N A_p}{\pi A_i} \right)^2 \right] \tan^2(\beta) \right)} \quad \text{Eq. 1}$$

This expression highlights that the filling limit is dominated by fluid density ρ , piston pitch radius r_p , piston area A_p , kidney area A_k , intake area A_i , number of pistons N , displacement angle β , and inlet pressure P_i . For open-circuit pumps, the most practical method to increase the limit is typically an increase of inlet pressure via charge pumps, often implemented using impellers.

In addition to inlet pressurization, mitigation approaches reported in the literature include increasing kidney areas A_k [11], reducing displacement angle, and reducing pitch radius r_p (e.g. by inclining the pistons), because these measures reduce characteristic velocities and thus reduce the associated pressure drop.

2.2. Body-force speed limit: cylinder-block tipping and rotating-group inertia effects

The body-force limitation for axial piston pumps is mainly governed by mass properties and pitch radius. The cylinder-block tipping mechanism is driven by inertial forces of the rotating and reciprocating components, producing a tipping couple that must be counteracted by the cylinder-block spring and pressure-induced clamping forces. Manring derived the corresponding limiting speed (Eq. 5 in [10]):

$$\omega^2 < \frac{2F_{Sp}R_{Block}}{N(M_p + M_s)r_p^2 \tan(\beta)} \quad \text{Eq. 2}$$

The relationship shows that higher speed capability is obtained by increasing spring force F_{Sp} or by reducing piston/slipper mass (M_p, M_s), pitch radius, and displacement angle. The detailed physical contributors to block tipping and the design criteria to prevent tipping were discussed in detail by [12], [13], including the interaction of pressure forces and inertial forces. Zhang has studied impact of piston mass on the maximum speed [14], while Achten

has proposed a counter force using a lever [13], however this type of solution is only suitable for the floating cup principle.

It should be noted that block tipping is not the only body-force-dependent limitation. At high speed, piston and slipper tilting tendencies increase, which may promote local metal-to-metal contact. In the slipper interface this typically manifests as accelerated wear, while at the piston–bore interface it may result in sticking or seizure, potentially leading to catastrophic failure called “piston stick”. These failure modes are also driven by inertial loading acting on the rotating group.

2.3. Tribological speed limit: slipper lift-off, film stability, and separation risk

At high speed, the tribological operating point may be destabilized by changes in hydrodynamic lift, local heating, and altered micro-motion of the slipper and piston. Excessive slipper lift-off may increase leakage or cause adverse force states at the ball joint. If a hold-down mechanism is present, opposing forces may be generated across the joint, potentially increasing play or causing partial separation. For the cylinder block / valve plate interface, increased hydrodynamics can lead to block lift, which results in a catastrophic failure due to instabilities in the displacement chamber.

Manring summarized the slipper tipping tendency in a limiting expression (Eq. 30 in Manring 2014), where the combined effects of tilting moments, spring force distribution, and pitch radius appear explicitly:

$$\omega^2 < \frac{F_{SP}}{Nr_p \left(M_s \frac{X_s}{R_s} + (M_p + M_s) \tan(\beta) \right)} \quad \text{Eq. 3}$$

According to Eq. 3 the tribological speed can hence be improved by increasing the spring force acting on the slipper F_{SP} , or decreasing the pitch radius r_p , number of pistons N , swash plate angle β , or the masses of the components. To counteract block lift at high speeds, it the main counter measure is reducing the pitch radius or increasing the spring force acting on the block. Zhang summarized in detail some measures that were taken in the past to improve tribological stability, but no single approach can be mentioned that would significantly improve this speed limit, other than again reducing mass [15].

2.4. Wobble plate type pump principle

Instead of increasing speed capacity primarily by mass reduction, an alternative approach is pursued in this work: the mass dependence is reduced structurally by changing the kinematic principle of the axial piston unit. In a wobble-plate arrangement, the inclined plate rotates with the shaft while the pistons and slippers undergo primarily translational motion, and the cylinder block remains stationary. As a consequence, the sealing gap between cylinder block and valve plate is eliminated and replaced by a check-valve commutation to the low-pressure and high-pressure ports.

Figure 2.3 illustrates the working principle of the check-valve-based wobble-plate pump used in this work. The investigated unit has a nominal displacement of 60 cm³/rev and uses a 12° inclined plane with up to nine pistons that are spring-loaded to maintain slipper contact with the wobble plate. The shaft is supported by two radial bearings and one axial bearing. In this design the wobble plate is the only rotating part, and the piston and slipper motion is

purely translatory. There is no valve plate, reducing the number of sealing gaps to two, and removing the need for speed depended grooves designs.

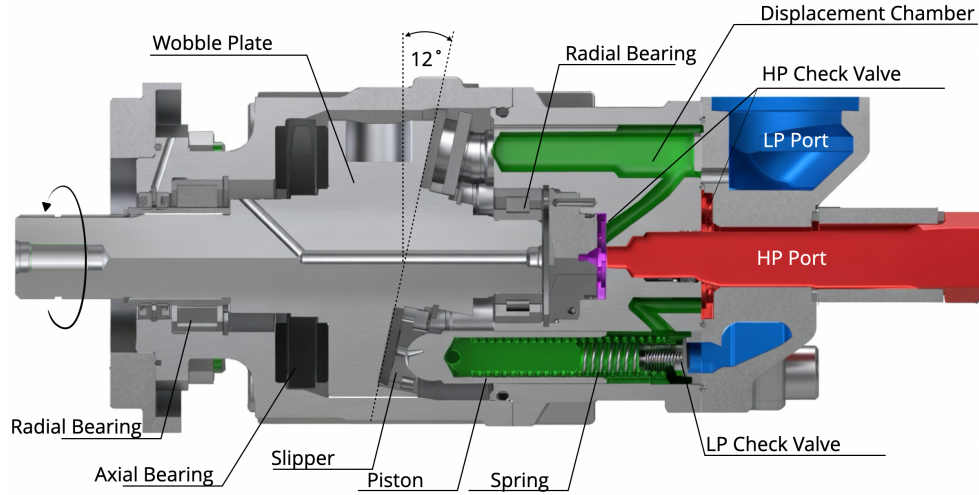


Figure 2.3. Working principle of check valve based wobble plate pump

Each displacement chamber is hydraulically isolated and connected to high and low pressure via dedicated check valves. This concept offers two key functional advantages:

1. **Pressure equalization during commutation** is governed by the check valves and therefore is less sensitive to geometric port timing, which reduces the tendency for pressure spikes at high speed.
2. **Modular piston count** is enabled because each bore is isolated; the unit may be operated with reduced piston count (e.g., six pistons) to reduce displacement while maintaining the same kinematic hardware.

The typical disadvantages of wobble plate pumps are fixed displacements and the inability to run in motoring mode. However, modern fast-switching-valves could overcome these limitations, as has been shown in digital displacement pumps [7]. With the dawn of speed variable displacement control, there is a new market for highly compact fixed displacement pumps.

The pump was tested in the configuration where all pistons delivered into a single high-pressure port.

2.5. Modified speed-limit expressions for the wobble-plate principle

By removing cylinder-block rotation, the tangential velocity component at the kidney is removed from the Bernoulli-based filling limit. Under this assumption, Eq. 1 simplifies to:

$$\omega^2 < \frac{2(P_i - P_{cav})}{\rho r^2 \tan^2(\beta) \left(\left(\frac{A_p}{A_k} \right)^2 - \left(\frac{NA_p}{\pi A_i} \right)^2 \right)} \quad \text{Eq.4}$$

Where the denominator is reduced when compared to Eq.1. For the same geometry and inlet pressure, this modification increases the theoretical filling speed limit because the tangential component of the kidney velocity is not present. In addition, the kidney opening A_k can be selected larger because it is no longer required to contribute to cylinder-block clamping as in a conventional valve-plate design. Combined, these effects indicate that the wobble-plate architecture should achieve higher self-priming speeds than a conventional rotating-block axial piston pump.

Because the cylinder block is stationary, the centrifugal tipping mechanism described by Eq. 2 is not present, and the associated speed limitation disappears for this architecture. The wobble plate itself remains a rotating body, and it can be easily designed to be rotationally balanced.

For the slipper lift-off / hold-down limitation, the centrifugal contributions are removed under the same assumption, leaving a simplified form where the primary limiting term is the slipper hold-down spring force:

$$\omega^2 < F_{SP} \quad \text{Eq.5}$$

2.6. Surface structures as a complementary path for conventional machines

Although the wobble-plate principle enables higher speed capability in theory, most industrial axial piston units are not of wobble-plate type. Therefore, surface structures on pistons and slippers are considered as an additional pathway to extend tribological stability for traditional architectures.

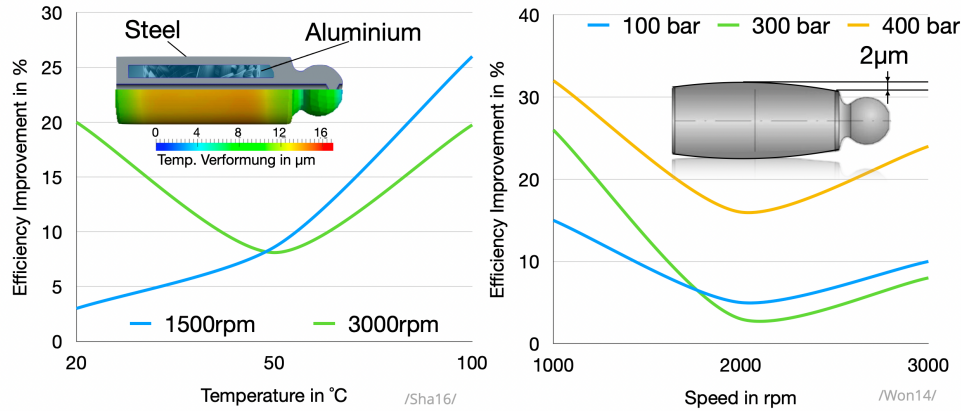


Figure 2.4. Piston surface structures improve efficiency also at high speeds

Figure 2.4 summarizes published examples where piston surface structures improved efficiency under high pressure and high speed conditions. [16], [17] reported that defined piston surface shaping can increase efficiency at elevated pressure and speed, with improvements exceeding 20% depending on operating point (right graph in Figure 2.4). The effect was attributed to improved piston micro-motion and the ability to reduce clearance while still permitting necessary piston tilt. Such surface shapes are typically only a few micrometers in height and therefore require precise manufacturing.

[18] proposed a thermally adaptive piston concept in which a hollow piston is filled with a material of higher thermal expansion (e.g., aluminum), producing a temperature-dependent convex shape. Efficiency improvements near 20% at high speed (left graph in Figure 2.4) were reported at elevated temperature, which is relevant because critical speed conditions are typically accompanied by increased local heating.

Slipper surface structures have primarily been reported for low-speed robustness and torque reduction, as [19] published a measured 80% reduction in torque. However, the same stabilization of the fluid film is expected to be beneficial at high speed. [20] showcased simulated a reduction of 60% of the slipper power losses at the corner power of the pump.

Combining surface structures on both piston and slipper with the wobble plate principle should lead to even further speed gains, as will be studied in this research.

2.7. Research goal

The objective of the present research is to establish the tribological maximum speed of the 60 cm³/rev wobble-plate pump platform and to demonstrate design and test methods to increase this limit. In the longer term, surface structures on both piston and slipper are intended to be combined with the wobble-plate architecture to extend the tribological speed capability toward 10,000 rpm, exceeding typical pre-charged axial piston pump limits by more than 100% at comparable displacement (see Figure 2.5). The present paper reports the first step in this endeavor by (i) establishing a methodology to identify the tribological speed limit and (ii) demonstrating initial improvement measures to the slipper.

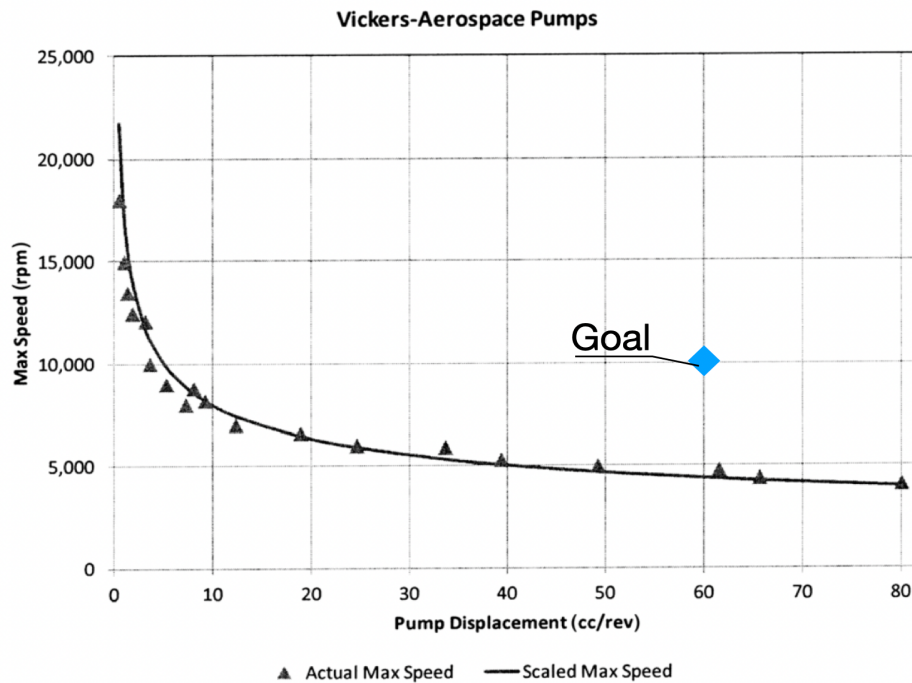


Figure 2.5. Typical Speed limits of pre-charged axial piston pumps [10].

3. MEASUREMENTS

This chapter presents the measurement setup, highlights key measurement results, and establishes the current tribological speed limit of the unmodified wobble-plate pump. Measurement results for the improved slipper design are reported in Chapter 4.

3.1. *Measurement Set-Up*

To define a baseline for subsequent design improvements, the tribological maximum speed of the unmodified wobble-plate pump was experimentally determined. The investigated unit was a Sunfab SLPD 40/20 wobble-plate pump with a total displacement of 61.1 cm³/rev. The manufacturer-rated maximum speed is 2200 rpm, and the maximum working pressure is 350 bar. The wobble-plate angle was chosen to be fixed 12° for the presented tests. A maximum angle of 18.2° is available on the same platform, which increases the displacement to 94.6 cm³/rev while maintaining the same rotating kit.

The pump mass is 22 kg, resulting in a power density of 3.6 kW/kg at 12° and 5.5 kW/kg at 18.2°. These values exceed typical swash-plate axial piston units of comparable displacement, and they indicate that the pump platform is well suited for high power density investigations.

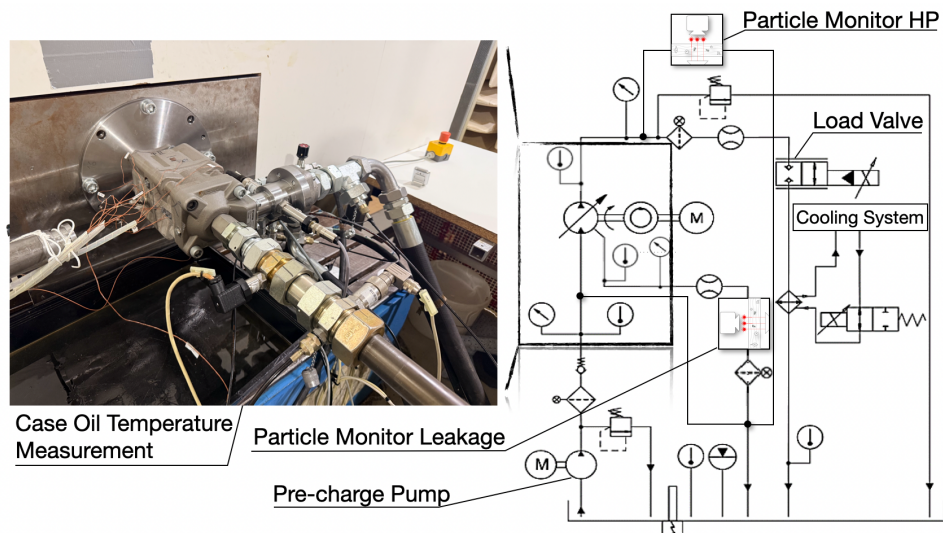


Figure 3.6. Measurement Set-up of the 61.1 cc wobble plate pump with thermal and particle monitoring.

Figure 3.6 shows the overall test rig configuration including photo and hydraulic schematic. A combined thermal and particle monitoring approach was selected to enable a controlled approach to the tribological limit, where imminent damage could be detected before catastrophic failure occurred. The test rig includes a pre-charge pump capable of up to 20 bar boost pressure, a load valve for pressure control, and a 160 kW cooling system. The cooling system is capable of controlling the inlet temperature within ± 0.1 °C. Unless stated otherwise, all measurements were recorded at thermal steady state with an inlet temperature of 40 °C.

Particles were monitored in both the high-pressure line and the leakage line to provide early warning of increasing wear activity. Temperature was measured at the ports (high-pressure, low-pressure, leakage) and at multiple internal locations inside the pump, as shown in **Figure 3.7**. Thermocouples were installed near both radial bearings and the axial bearing, at multiple internal case oil locations, and in the piston/bushing region.

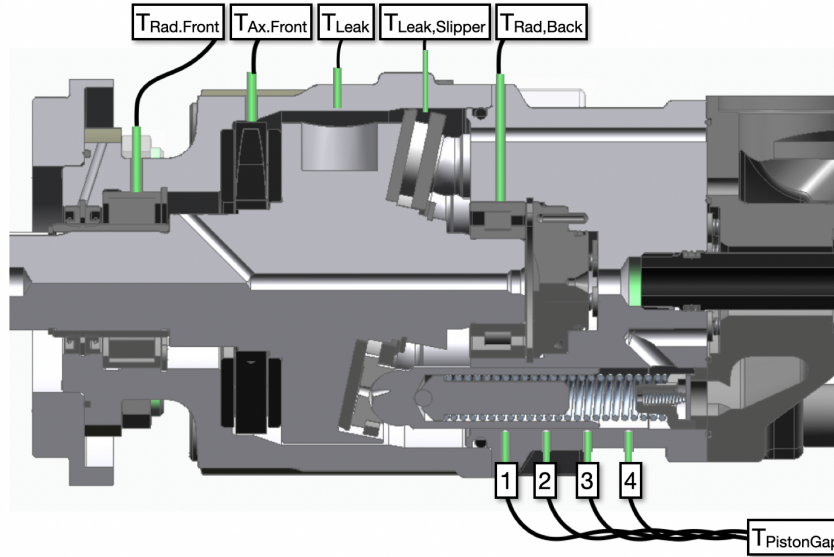


Figure 3.7. Thermal monitoring of the wobble plate pump at key areas.

A key advantage of the wobble-plate architecture is that the cylinder block is stationary, which simplifies the instrumentation of the piston–bushing gap. Four temperature probes were placed along the piston/bushing gap as illustrated in Figure 3.7, enabling a local temperature gradient to be captured from the displacement chamber region toward the outer bore section.

3.2. *Measurement of the tribological speed limit*

All operating points were considered to be at thermal steady state when the rate of change of all monitored temperatures was below 0.1 K/min. An example steady-state operating point is shown in **Figure 3.8**. The pump was operated at 1500 rpm and 300 bar, with an inlet temperature of 40.6 °C and an outlet temperature of 42.4 °C. The leakage temperature measured at the external leakage port was 67.3 °C. However, the internal measurements showed substantially higher local temperatures: Leakage temperatures inside the housing approached 120 °C near the slippers and approximately 108 °C near the wobble plate. The axial bearing exhibited the highest temperature, exceeding 125 °C. The piston gap temperatures ranged from approximately 51 °C to 91 °C, with lower values toward the displacement chamber side.

This operating point illustrates two critical findings:

1. **The axial bearing and slipper region represent the dominant thermal hot spots** in the investigated pump configuration.
2. **External leakage temperature measurements do not represent internal gap temperatures**, even when sensors are placed close to the pump (5 cm in this case). This deviation becomes particularly pronounced at low leakage flow. For the shown operating point, the leakage flow was 0.6 l/min, such that the external leakage line did not provide sufficient thermal coupling to reflect internal film temperatures.

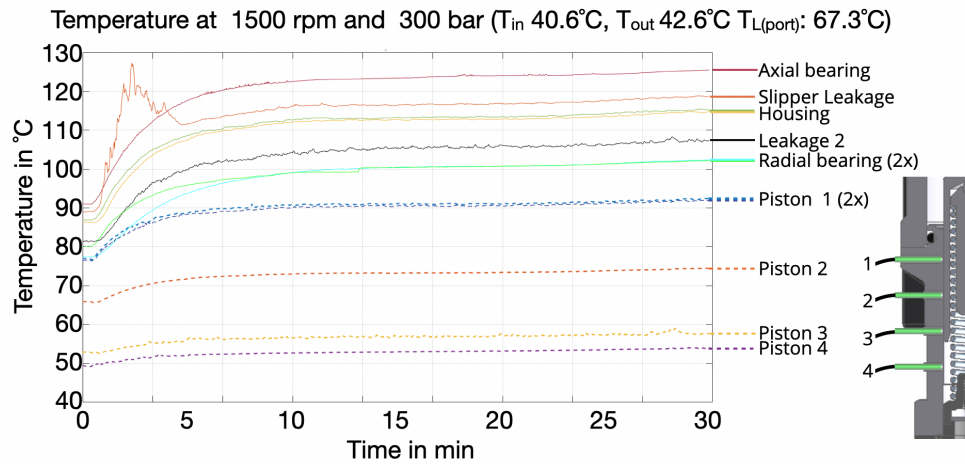


Figure 3.8. Internal temperatures at 1500 rpm and 300 bar.

The presence of steady-state internal gap temperatures above 100 °C at 1500 rpm indicates a limited thermal margin already at moderate speed. This condition differs from typical swash-plate axial piston pumps, where internal leakage flow and oil distribution often provide higher convective cooling of loaded interfaces.

The unmodified pump was tested up to 4000 rpm, exceeding the rated speed by approximately 80%. The tribological speed limit for the unmodified configuration was empirically identified at **300 bar and 3700 rpm**. At this operating point, the slipper-related temperature continued to rise beyond 150 °C without stabilizing, and a sustained increase in particle counts ($\geq 6 \mu\text{m}$) was detected in both leakage and high-pressure monitoring. The test was terminated based on these stop criteria to preserve component integrity for failure mode evaluation.

After disassembly, all parts remained functional; however, pronounced wear marks were observed on the slippers and on the wobble plate running surface, as shown in **Figure 3.9**. The pistons showed wear, but it remained within acceptable limits. All bearings were still operational after the approximately 40 h test campaign. However, the axial bearing exhibited the beginning of pitting.

In addition to the qualitative wear images, topographical measurements of the slipper profile were performed at for two slippers in three conditions: (i) new condition, (ii) after 16 h of

operation (up to approximately 2000 rpm), and the second slipper (iii) after 40 h at the end of the full overspeed campaign. The profile comparison in Figure 3.9 shows a continuous and substantial loss of material across a large contact region. This trend indicates that the original slipper was not designed for continuous operation under the combined conditions of high speed and high pressure, and it supports the temperature and particle signals that defined the tribological limit.

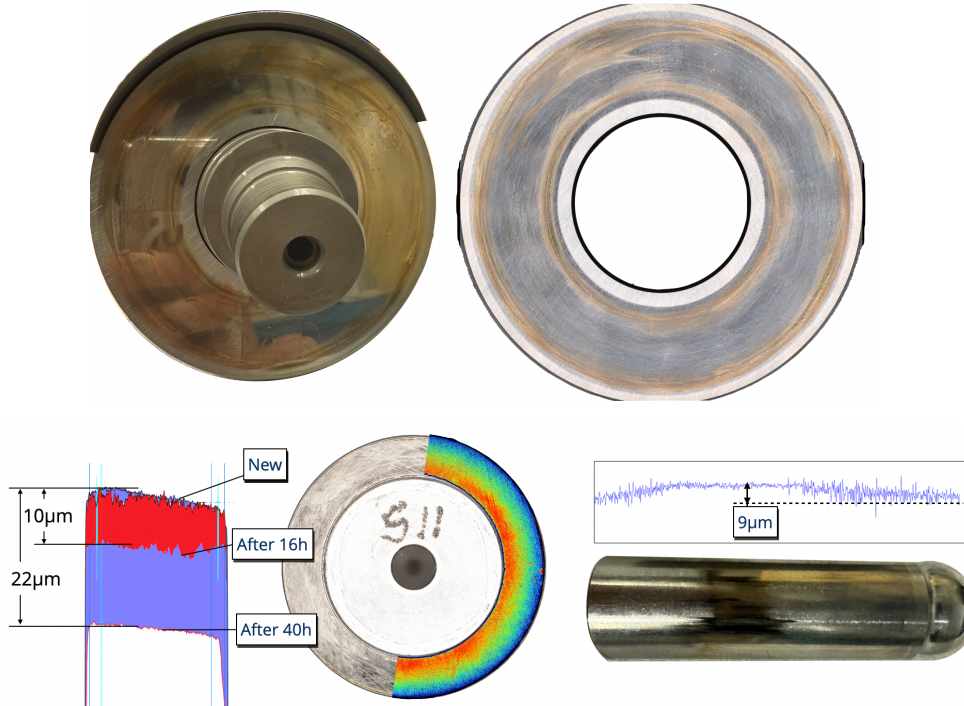


Figure 3.9. Worn components after the test campaign, including topological measurements.

Measured volumetric efficiency and hydromechanical efficiency at three pressure levels are shown in Figure 3.10. The rated maximum speed (2200 rpm) is marked by the dashed line. Volumetric efficiency remained high up to the rated speed and decreased beyond that point. Above 2500 rpm, boost pressure was applied and increased gradually, reaching 4.5 bar inlet pressure at 3700 rpm at 300 bar Δp . With boost pressure applied, volumetric efficiency remained above 90% for most operating points. Without boost pressure, volumetric efficiency dropped below 90% at 200 bar shortly after approximately 2600 rpm. The remaining drop at high speed despite boost pressure is likely caused by limitations in inlet check valve opening dynamics and available effective inlet flow area.

Hydromechanical efficiency is the more relevant metric for identifying tribological limitations because it represents the fraction of mechanical power converted into hydraulic power and therefore captures frictional losses. In the present measurements, the lowest hydromechanical efficiencies were observed at 100 bar, where efficiency decreased approximately linearly with speed. At 300 bar, hydromechanical efficiency remained above

90% over a large portion of the operating map. Importantly, the tribological limit at 300 bar and 3700 rpm could not be identified from efficiency trends alone. The limit was detected only by local temperature escalation and continuous particle generation.

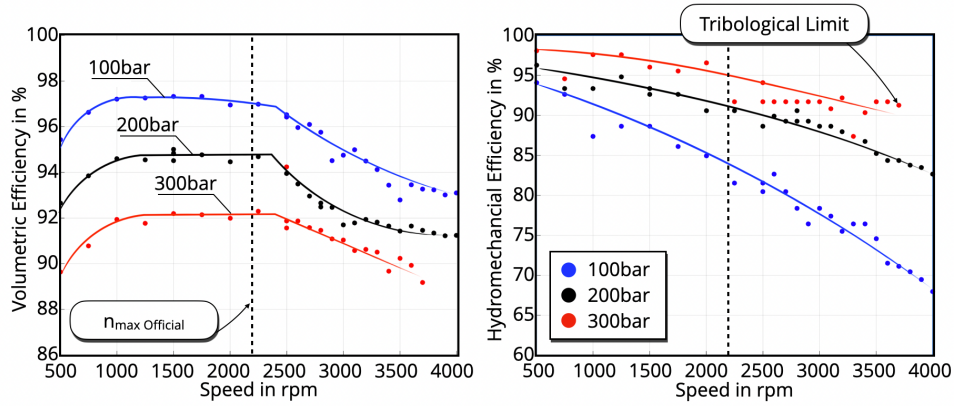


Figure 3.10. Volumetric and hydromechanical efficiency for three pressure levels

Figure 3.11 summarizes the temperature fields over speed and pressure for the slipper leakage, piston gap, and axial bearing. Across the investigated operating regime, the axial bearing consistently exhibited the highest temperatures. The dashed line indicates the region where the temperature is above 100°C. An improved design should stay below this temperature to ensure a long component and oil life-time. The bearing is a roller-type design specific to this pump platform. Due to the low leakage flow of the pump, insufficient oil flow appears to be available for convective cooling of this bearing at elevated pressure and speed. The resulting overheating increases the global case oil temperature and raises the temperature level in other lubricated interfaces.

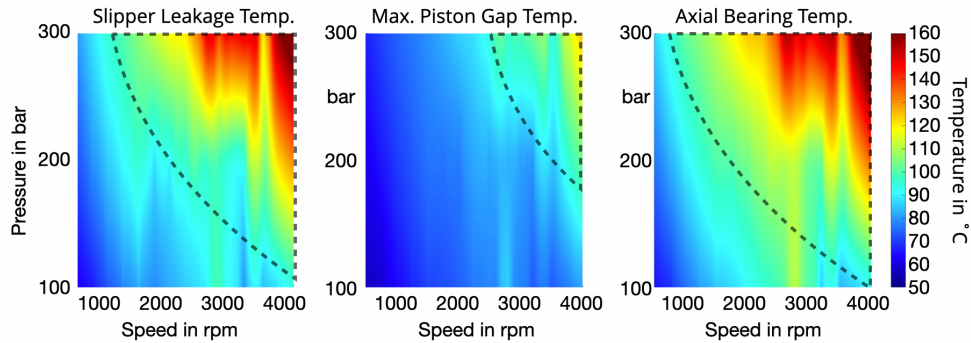


Figure 3.11. Captured slipper, piston and axial bearing temperature until 4000 rpm

The leakage temperature measured in the vicinity of the slipper gap increased strongly with speed and pressure, reaching approximately 150 °C at 3700 rpm and 300 bar. The maximum piston gap temperature (measured at the top bore position P1) exceeded 100 °C at high speed. This increase is attributed primarily to elevated case temperatures and thermal conduction through the stationary cylinder block. In contrast, at 3700 rpm and 300 bar the piston gap temperature near the displacement chamber (P4) remained comparatively low at

58.9 °C, while the external leakage port temperature was only 50.6 °C at a leakage flow of 0.7 l/min.

Overall, the pump survived the overspeed campaign, and a baseline tribological speed limit of 3700 rpm was established for the unmodified configuration. The central outcome of the baseline measurements is that **(i) the axial bearing is thermally under-supplied with oil**, which elevates the entire thermal level of the unit, and **(ii) the slipper interface operates at high temperature and exhibits continuous wear** (see Figure 3.9). To establish a more generally applicable tribological speed limit for wobble-plate type pumps, both improved axial-bearing cooling and improved slipper performance are required.

4. THERMAL IMPROVEMENT OF SLIPPER

The pump was modelled using Caspar FSTI, a numerical simulation environment used to predict power losses and operating conditions in the lubricating gaps of axial piston machines. The tool was developed specifically for piston pump tribology, enabling efficient gap-level simulation without requiring extensive user effort in numerical discretization, meshing strategies, or sensitivity studies. The relevant micro-motion of components and elastic deformation due to both pressure and temperature are captured, which has been shown to strongly influence the operating conditions in tribological interfaces [21]. The tool was validated for the individual lubricating gaps by comparison with measured temperatures and gap heights, as reported in [22], [23], [24], [25], [26], [27], [28], [29].

Using this validated toolchain, both the piston–bushing gap and the slipper–swashplate gap were predicted to approach wear-critical conditions at elevated pressure and at speeds exceeding 3000 rpm. However, a critical thermal behavior was predicted for the slipper already at approximately 1500 rpm. In this regime, the slipper leakage was predicted to be too low, causing the oil to remain trapped in the film region. As a result, heat generated by viscous shear was not removed sufficiently by convective transport, leading to a rapid increase of local film temperature. The temperature rise was predicted to deform the swashplate, thereby worsening the gap geometry and accelerating wear progression.

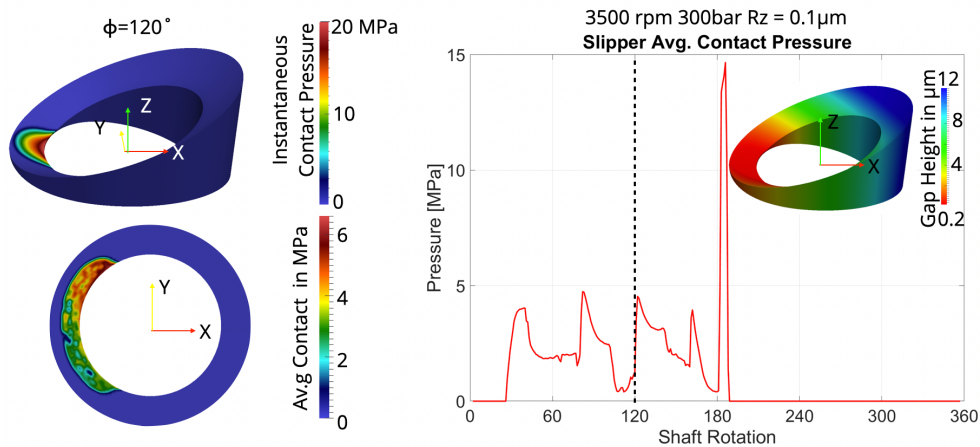


Figure 4.12. Predicted slipper contact at high speed and high pressure

Figure 4.12 shows the predicted contact behavior of the slipper film at 3500 rpm and 300 bar. The instantaneous contact pressure distribution is shown at a rotational angle of $\phi = 120^\circ$, corresponding to the middle of the high-pressure stroke. Local contact pressures up to 20 MPa were predicted during this instance. The right-hand plot shows the averaged contact pressure per rotational degree, obtained by averaging the local contact pressure across all computational cells of the gap and evaluating this average for each shaft angle. In addition, the top-right inset shows the predicted film thickness distribution at $\phi = 120^\circ$, where the positive X-direction corresponds to the direction of motion. The film is predicted to form a positive wedge, building hydrodynamic pressure toward the trailing end. At this low gap height this inclination causes contact at local regions of the slipper.

The accumulated contact pressure over one shaft revolution is shown in the bottom-left of Figure 4.12, where the average instantaneous contact pressure is integrated (or accumulated) across the revolution and visualized as the spatial distribution of cumulative contact exposure. This representation indicates that contact initiates at the inner edge of the sealing land and then progresses outward. This predicted wear initiation location is consistent with the measured wear pattern presented in Figure 3.9, where the inner sealing-land edge exhibits the largest material loss.

To reduce slipper wear and to decrease excessive leakage oil temperatures, the slipper dimensions were varied using the validated simulation model. The study was conducted across four operating conditions (OC1–OC4), shown in **Figure 4.13**, spanning the regular operating regime of the pump. The purpose of this first slipper iteration was to shift the thermal behavior toward a more typical piston-pump baseline and to evaluate whether a conventional improvement of hydrostatic balance can increase speed capability before surface structures are introduced.

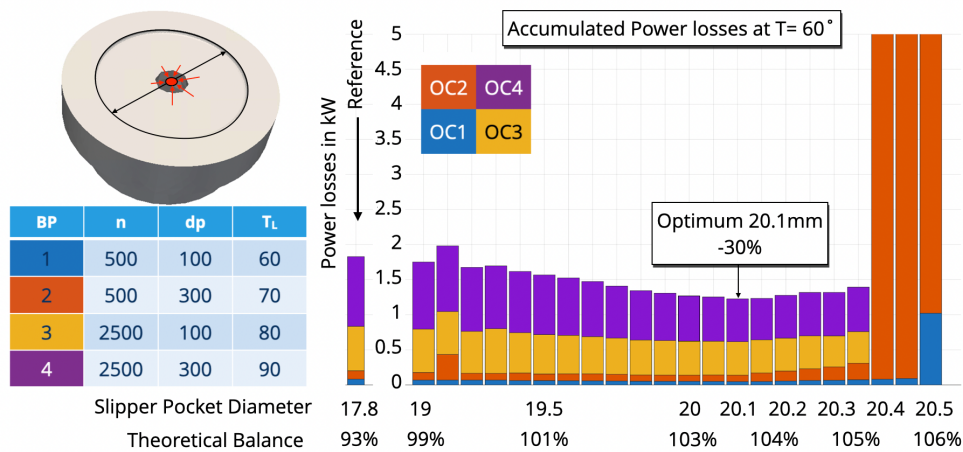


Figure 4.13. Slipper pocket diameter study vs accumulated power losses across four operating conditions.

When varying the inner pocket diameter of a single-land slipper (i.e., a slipper with one active sealing surface), the hydrostatic balance is altered. The balance is defined as the hydrostatic force generated in the pocket and across the sealing land divided by the piston force acting on the slipper. The ratio is normally designed to be lower than unity because

hydrodynamic pressure generation adds to the hydrostatic support. A balance higher than 1 would imply more lifting force than clamping force, which may cause excessive lift-off and loss of sealing ability. However, these simplified balance estimates rely on strong assumptions, in particular that the slipper remains parallel to the running surface and that elastic deformation due to pressure and temperature does not alter the film geometry. In practice, slipper motion is dynamic and the film is almost never parallel. Furthermore, pressure and thermal loads deform the components on the micrometer scale, which is comparable to the film thickness itself. Consequently, balance estimates are not fully reliable as absolute design rules, but they remain useful as comparative metrics between designs and platforms. The reference slipper has an inner pocket diameter $\text{dinG} = 17.8$ mm, corresponding to approximately 93% balance. This is within typical industrial design practice, where balances between 90–95% are common. As shown in Figure 4.13, an increase of pocket size to a balance well above 100% was predicted to be beneficial for the investigated wobble-plate pump, with the minimum accumulated power loss occurring at a pocket diameter of 20.1 mm ($\approx 103.8\%$ balance). At this point, the simulation predicts approximately 30% lower slipper losses compared to the reference.

From a traditional design perspective, a balance above 100% is typically avoided because high leakage would be expected. The simulation results show that this concern becomes dominant only beyond approximately 20.4 mm pocket diameter ($\approx 105.5\%$ balance), where leakage is predicted to increase sharply with pressure and the pump would effectively cease to operate as intended. This risk is the reason why conservative designs are often selected. In the present case, the simulation predicts that the $>100\%$ balance region can still be feasible because the slipper deforms under pressure and temperature, altering the sealing land shape and effectively reducing active hydrostatic area. This reduces the net hydrostatic lifting force and leaves margin for hydrodynamic effects without leading to catastrophic lift-off. It is therefore emphasized that this is **not** a universal recommendation to design above 100% balance. The central point is that without a power-loss simulation tool that captures elastic and thermal effects, it is difficult to determine the optimum dimensions reliably, and design “recipes” cannot be transferred directly from one pump to another because material, thickness, and thermal boundary conditions change the deformation and therefore the tribological outcome.

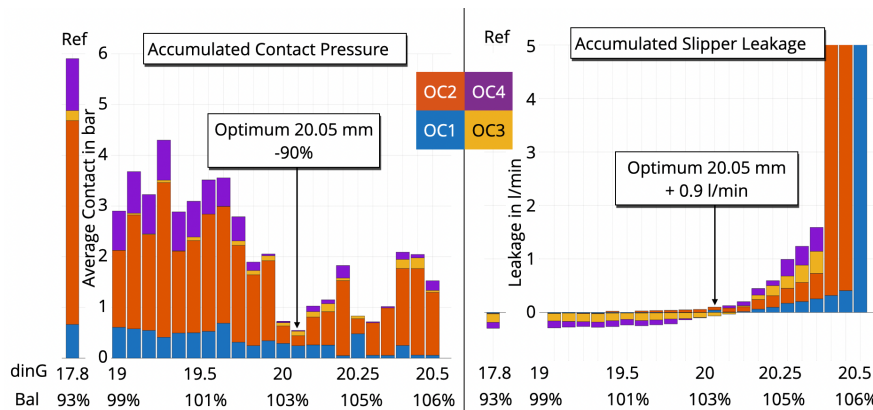


Figure 4.14. Accumulated average contact over 4 operating conditions (left) and accumulated leakage vs slipper pocket diameter (dinG) and balance.

As shown in **Figure 4.14**, increasing the inner pocket diameter for this wobble-plate slipper reduces accumulated contact pressure (particularly at OC2: 500 rpm / 300 bar) while keeping accumulated leakage nearly constant over most of the tested range. Only beyond approximately 20 mm does leakage start to increase noticeably. Some leakage is beneficial because it cools the film and reduces the risk of thermal runaway. In practice, the target is not minimum leakage, but minimum total power loss, which implies a compromise between frictional losses and leakage losses.

Based on the combined objective of minimal total power loss, minimal contact, and controlled leakage, an inner pocket diameter of **20.05 mm** was selected. While some low-speed contact was still predicted, it was substantially reduced relative to the reference design. The remaining contact was predicted to primarily smooth the relatively high initial surface roughness and then diminish as the running surfaces adapt.

The improved slipper design was predicted to reduce viscous shear losses in the slipper film and thereby reduce the thermal loading of the wobble plate. As shown in **Figure 4.15**, the peak swash/wobble plate temperature at corner power was predicted to decrease from 155°C to 82°C. This 73 K reduction of the local hot spot was predicted to reduce leakage temperature by approximately 40 K. It is noted that the axial bearing was not included in the simulation model; therefore, thermal boundary conditions had to be estimated, which may influence the absolute values. Nevertheless, the modified slipper is expected to improve the thermal behavior of the pump while only marginally increasing leakage.

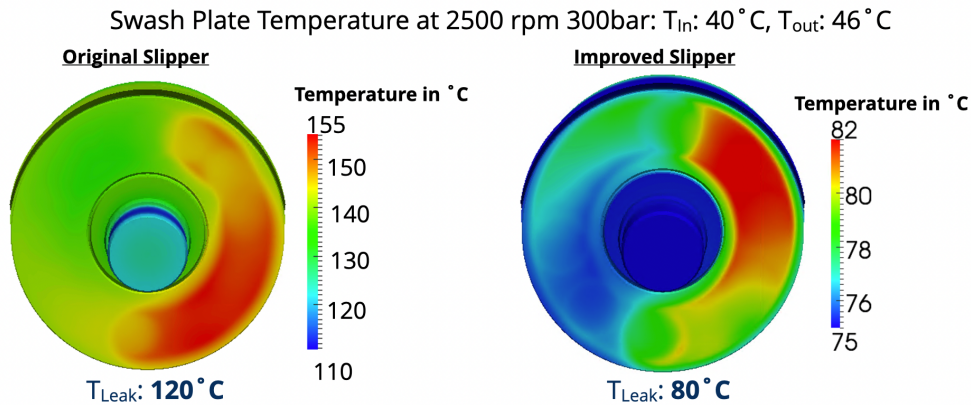


Figure 4.15. Simulated swash plate temperature at 2500 rpm and 300 bar for two different slipper designs: Left: reference slipper; Right: new slipper.

4.1. Measurement results of the improved slipper

The improved slippers were manufactured and implemented into a new rotating kit, and the test campaign was continued. Although measurements are still ongoing, most of the regular pump operating envelope has already been covered. The observed improvements are summarized in **Figure 4.16**. The leakage temperature near the slipper gap decreased substantially with the new slipper design, particularly at high pressure. At 300 bar and 2000 rpm, a temperature reduction of 28 K was measured. While the corner power condition has not yet been measured, the current trend indicates a temperature reduction exceeding 30 K at that operating point.

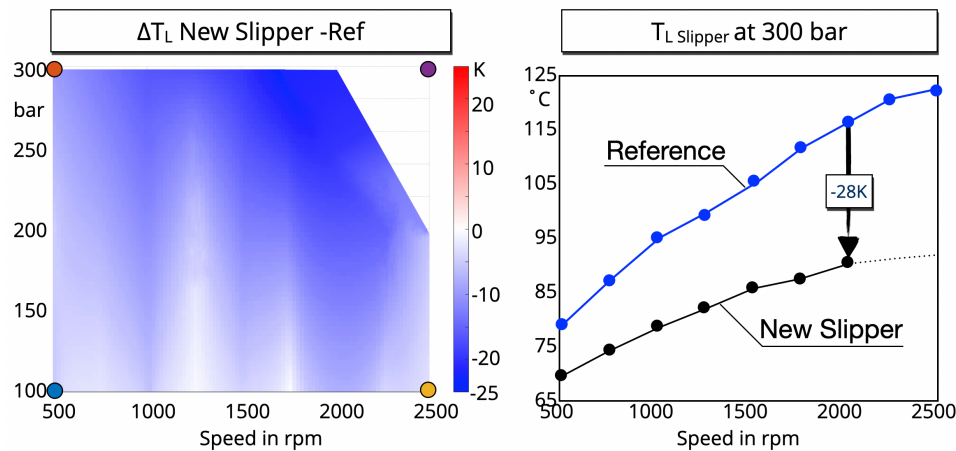


Figure 4.16. Reduced slipper leakage temperature (left) and slipper leakage temperature trend with speed at 300 bar (right)

The primary objective—reduction of slipper temperature—was achieved with the modified slipper. While the measurement campaign is still ongoing, the observed trend indicates that the design represents a successful improvement. The achieved temperature reduction at 2500 rpm and 300 bar is not expected to match the simulated magnitude exactly, but the trend is captured well. The reduction is attributed primarily to reduced viscous losses in the slipper film rather than a large increase in leakage.

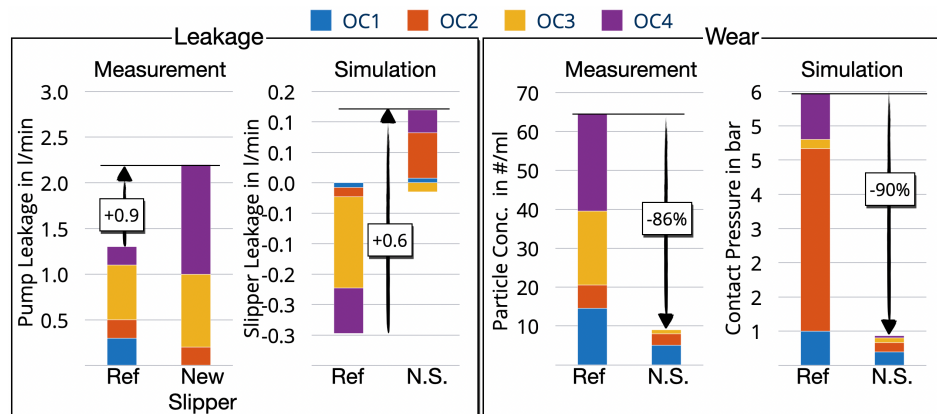


Figure 4.17. Predicted and measured accumulated leakage (left) and wear (right) for the reference and the new slipper (N.S.).

This is supported by the leakage evaluation shown in **Figure 4.17** (left), where accumulated pump leakage is compared across the four operating conditions. Although the OC4 leakage value is currently a prediction, the data indicate that the additional leakage is below 1 l/min when accumulated across all four operating conditions. The measured trend matches the simulation, although not in absolute magnitude. Such deviations are expected because measured total pump leakage includes contributions from multiple leakage paths, whereas the simulation focuses on an individual gap. Nevertheless, no significant increase in pump

leakage was observed, confirming that the selected ~104% balance did not introduce excessive leakage.

The right-hand side of Figure 4.17 shows that the predicted reduction in wear tendency was confirmed by measurement. The simulation predicted approximately 90% lower accumulated contact pressure across the four operating conditions, and the measured particle concentration of particles $\geq 6 \mu\text{m}$ decreased in a similar manner. This result may change slightly once OC4 is fully measured; however, the current trend does not indicate significant wear at corner power for the new slipper.

Overall, while the measurement campaign is still ongoing, it can be concluded that the new slipper design reduces operating temperature at high speed and high pressure and provides a more representative baseline for the next research step: the addition of surface structures on both slipper and piston and subsequent testing toward higher rotational speeds.

5. CONCLUSION

This paper reported the first step toward an ambitious goal: operating a ~60 cm³/rev hydrostatic unit at rotational speeds up to 10,000 rpm for electrified mobile machinery. This speed increase would push the power density of this already highly compact pump from 3.7 kW/kg to 16.2 kW/kg. Within this paper the pump was already operated at 6.4 kW/kg, a 60% increase. It was shown that the wobble-plate principle removes several dominant speed-limiting mechanisms of conventional swash-plate axial piston pumps. In particular, cylinder-block rotation is eliminated, such that centrifugal-force-driven effects and the valve-plate sealing interface no longer define the operating boundary. Although self-priming and inlet filling remain relevant, the tribological speed limit is not constrained by an inherent physical barrier in the same manner as in rotating-block architectures, and it can be shifted by targeted design measures.

A validated Caspar FSTI-based simulation workflow was used to predict gap behavior, thermal loading, and contact risk, and it was applied to redesign the slipper for high-speed/high-pressure operation. By increasing the hydrostatic balance of the slipper from 93% to approximately 104% (i.e., +11% absolute), the excessive heat generation of the slipper interface was reduced in simulation and subsequently confirmed experimentally.

Baseline testing of the unmodified pump demonstrated that, with inlet boosting, the operating speed could be increased substantially beyond the rated value. A tribological speed limit was empirically identified at 300 bar and 3700 rpm, triggered by continuously rising slipper-related temperatures and increased particle activity, while the hardware remained intact for post-test inspection. The measurements showed that internal gap temperatures were significantly higher than external leakage temperatures and that the axial bearing and slipper interface dominated the thermal loading. This finding established a clear improvement path: thermal management of the axial bearing and reduction of slipper viscous losses were identified as primary levers to extend speed capability.

The redesigned slipper was manufactured and tested, and a clear improvement was measured. The leakage temperature in the slipper region was reduced by more than 30 K at high pressure and speed, while the total pump leakage increased only marginally (below

1 l/min accumulated across the evaluated operating conditions). In parallel, particle generation was reduced by more than 80%, matching the predicted reduction of accumulated contact and indicating substantially improved tribological stability.

Overall, the presented methodology—instrumented overspeed testing with early-warning criteria combined with simulation-guided redesign—was shown to be effective for identifying and shifting the tribological speed limit of a wobble-plate pump platform. In the next step, thermally adaptive and surface-structured concepts will be engineered on both slipper and piston interfaces to further stabilize the fluid film at ultra-high speed. These designs will be validated on a dedicated high-speed test rig capable of testing operation up to 10,000 rpm.

6. NOMENCLATURE

Latin symbols

- A_i – inlet flow area of intake manifold / suction port [m^2]
- A_k – kidney area (effective port area per displacement chamber) [m^2]
- A_p – piston cross-sectional area [m^2]
- F_{SP} – assembled spring force (cylinder-block / hold-down spring force) [N]
- M_p – mass of one piston [kg]
- M_s – mass of one slipper [kg]
- N – number of pistons [-]
- N.S. – New slipper design
- P_i – inlet pressure at intake manifold / suction port [bar]
- P_{cav} – cavitation pressure (threshold pressure for vapor formation) [bar]
- R_{Block} – outer radius of the cylinder block [m]
- R_s – maximum lever arm of swash reaction on slipper [m]
- r_p – piston pitch radius [m]
- X_s – distance from piston–slipper ball joint to slipper mass center [m]

Greek symbols

- β – displacement angle (swash/wobble plate inclination angle) [$^\circ$]
- ω – angular speed of shaft [rad/s]
- ρ – fluid density [kg/m^3]
- ϕ – shaft rotational angle [$^\circ$]

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