

Numerical Analysis of the Dynamic Behaviour of Axial Piston Pumps and Motors Slipper Bearings

M. Borghi, E. Specchia and B. Zardin

University of Modena and Reggio Emilia

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ABSTRACT

This paper reports an analysis of the lubrication mechanism and the dynamic behaviour of axial piston pumps and motors slipper bearings. A numerical procedure is used to solve the Reynolds equation, written here with respect to the slipper-swash plate gap, whose height is considered variable in a two dimensional field and with time.

The contributions of forces and moments acting on the slipper are illustrated and discussed, then the numerical method is presented to solve the Reynolds equation. Taking into consideration the slipper surface that is facing the swash plate, different geometry profiles are considered and the subsequent dynamic behaviour of the slipper is investigated; in particular, it is shown that a flat profile cannot always guarantee the bearing capability of the slipper and the lubrication in the gap is compromised for some critical operating conditions.

Successively, different non-flat profiles of the surface are considered and their impact on the slipper bearing dynamic behaviour is investigated in terms of tilt of the slipper α and central clearance height h_0 . Then, focusing on one of these non-flat profiles, the results are discussed highlighting the influence of machine operating conditions. These include rotational speed and a high pressure level and involve introducing a friction coefficient in order to evaluate the magnitude of friction losses determined by the slipper behaviour. Finally, some considerations are expressed about differences between mode of operation of the pump and the motor related to slipper behaviour.

INTRODUCTION

Axial piston swash plate type pumps and motors have been the objective of studies of many researchers

worldwide for decades. In the last thirty years, the main interest has focused on energy dissipation and improvement in efficiency. This can also lead to lower weight to power ratio, a crucial design guideline, for instance, in mobile applications. An axial piston swash plate unit is shown in Figure 1. The total efficiency of the machine depends strongly on the lubricating gaps behaviour which characterizes the machine operation. These gaps can be identified as between the valve plate and the cylinder block, the slipper bearings and the swash plate, the piston and the cylinder and between the piston ball joint and the slipper socket. The role of the gaps is essentially to maintain an adequate lubrication oil film between the movable parts of the units, preventing contact between them during operation. This, therefore, ensures good mechanical efficiency and avoids eventual catastrophic ruptures during operation. On the other hand, oil flow through the gaps has to be reduced as much as possible because it negatively affects the volumetric efficiency of the unit.

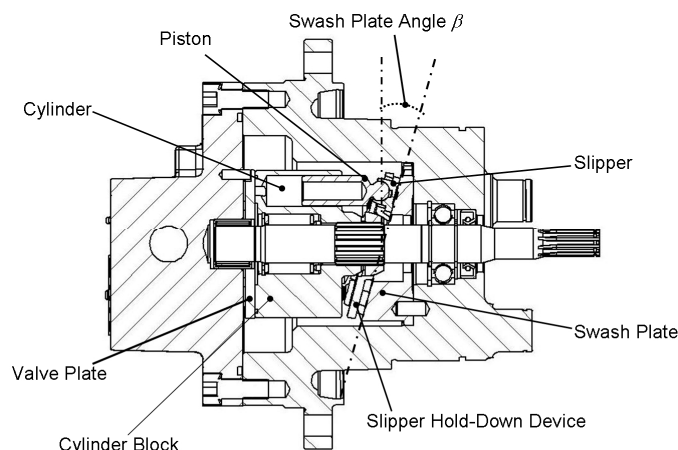


Figure 1: Swash plate type axial piston machine.

With the aim of evaluating the overall efficiency of swash plate type axial piston machines and to improve it, M. Ivantysynova, U. Wiecek, R. Lasar, Ch. Huang et al. ([1] to [18]) have made significant advances in modelling and simulating the machine behaviour through the development of a computer aided tool named CASPAR. In this, they also implement the variable geometry of the lubricating gaps corresponding to the valve plate-cylinder block, slipper bearing-swash plate and the piston-cylinder assembly. The approach they followed in these works was to determine the gaps height through a numerical procedure that involves the pressure distribution calculation within the gaps. They also considered influence of temperature on the fluid properties (non-isothermal flow), which can be a key parameter in the determination of the lubricating and bearing capability of the oil film. They are developing a model to determine the heat exchange between the oil and the components of the machines, to be integrated in CASPAR for each of the three gaps considered. Moreover, they demonstrated the influence of the elasto-hydrodynamic effect generated by surface deformation and of the macroscopic/microscopic surface geometry on the bearing capability of the gaps. They suggested how, by controlling the surface microscopic geometry, it is possible to improve the overall efficiency of the unit. This confirmed the fact that a certain amount of roughness has a positive impact on the gap bearing capability. The aim of all this work is to develop a multi-body simulation tool able to perform a fully coupled fluid-structure-thermal simulation of a swash-plate type axial piston machine. They verified the numerical results in stages by comparing them with instantaneous pressure into the cylinder, pressure ripple at the high pressure port and total leakage. They also developed a special test rig, named Tribo Test Rig, which allows the friction force measurement under real operating conditions of a swash-plate type axial piston unit.

Murrenhoff et al. ([19] to [21]) also developed a computer tool in order to evaluate the dynamic behaviour of an axial piston machine and study the influence of gaps geometry and their tribological characteristics on the total efficiency. To do this, they coupled for the first time two types of simulation tools, DSHplus and the multi body simulation environment ADAMS.

However, although the above authors have developed detailed models able to perform analyses of all the gaps mentioned, they did not presented full results about the their dynamic behaviour. Rather, they concentrate their attention on the machine overall efficiency and on the evaluation of friction losses and leakages.

In current work, the attention is focused on the slipper bearing lubrication mechanism, with the aim of investigating the dynamic behaviour of the slipper. The numerical procedure described may be successively applied also to the other gaps, in order to determine their dynamic behaviour and to evaluate, through friction losses and leakage flow calculation, the overall machine efficiency.

Figure 2 gives a diagram of the slipper bearing – piston assembly. The slipper function is to bear the load exerted by the piston, avoiding direct contact with the swash plate, which would cause high values of friction losses. Moreover, this function should be maintained for a wide range of rotational speed and high pressure values characterizing machine operation.

The slipper bearing capability during machine operation can derive partly from the hydrostatic lift generated by the pressure inside the slipper pocket (connected to the cylinder through an orifice) and partly from the hydrodynamic lift which depends on the pressure distribution beneath the land of the slipper. While the hydrostatic lift depends on the land mean radius, the hydrodynamic lift is related to the land width. Of the several authors who have addressed the analysis of such a problem, only the contributions which have been considered more significant for the purpose of this paper are briefly discussed as follows.

Hooke and Kakoullis used the ratio between the load exerted on the slipper by the piston to the load hydrostatically generated underneath the slipper (named the clamping ratio C_r) to identify different slipper designs. For C_r greater than 1, the slipper design is such that the remaining load to sustain the slipper has to be provided by the hydrodynamically generated pressure distribution within the gap between the land and the plate; in this case, the slipper is identified as overclamped. In the other case, the slipper is identified as underclamped. In [22], [24] Hooke and Kakoullis have made a theoretical analysis of the balance of an overclamped slipper of a pump. They concluded that a certain amount of non-flatness is necessary to have successful operation of the slipper in all the machine operating conditions. They also considered both the case of a load acting on the centre of the slipper and over a zone adjacent to the centre.

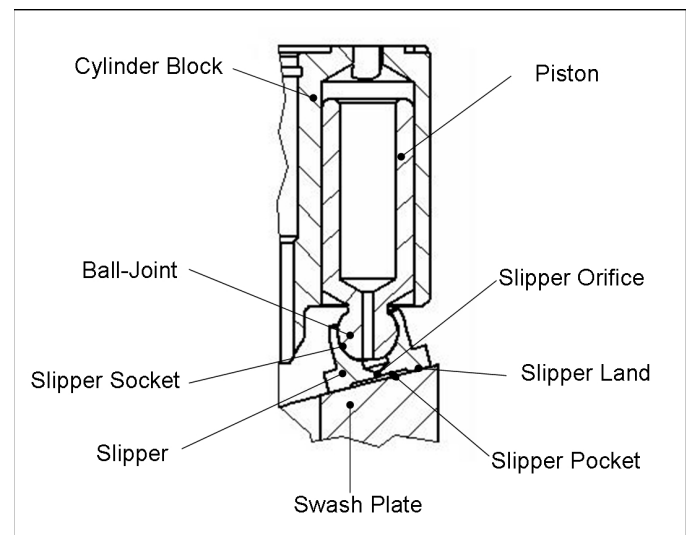


Figure 2: Piston – slipper assembly.

In [25], [26] Hooke and Li have shown that the minimum clearance height of an overclamped slipper – swash plate gap is not particularly dependent on the magnitude of the surface non-flatness but a convex profile is needed to assure bearing capability and adequate film lubrication. They well demonstrated and explained the

influence of tilting couples on the slipper. They obtained experimental data from a special bearing test rig, designed to overcome the difficulty of measuring the gap height on an actual pump during operation, these being compared with the theoretical results. It should be highlighted that in this special test rig the slipper – piston assembly is fixed and the plate rotates under the slipper with a swash angle β equal to zero, which is a quite far condition from the actual one during pump operation.

In [27] Koç, Hooke and Li and in [28], [29] Koç and Hooke presented the results obtained by both experiment (always in steady state conditions) and theoretical analysis on slipper balance considering three different slipper designs (both underclamped and overclamped) at high speed conditions. The results confirmed the necessity to have a slightly convex profile to ensure successful operation for both the slipper designs, with a larger magnitude of non flatness when speaking of underclamped design. The presence of an orifice which connects the slipper pocket with the cylinder seems to destabilise the slipper behaviour in terms of tilt direction of the slipper, in particular with the underclamped design. Generally, the slippers behaved at their best if the orifice is as little as possible or blanked off. This allows the slipper to have its maximum resistance to tilting moments and to run with higher minimum gap height. The clamping ratio, offset of the load applied on the slipper, orifice size and also high pressure value strongly affect the slipper behaviour. In all the works mentioned, however, the authors always performed their analyses for steady state conditions, for a given slipper tilt, imposed flow, load and moment equilibrium for a given sliding velocity and swash plate angle.

Manring et al., [30], analyzed the behaviour of a hydrostatic thrust bearing in a stationary setting. They based their considerations on the fact that during normal operation, the bearing always deforms with either a concave or convex profile being observed. They found that a linear concave profile increases the load carrying capacity while a linear convex profile decreases it. However, in the case of a concave profile the leakage through the bearing is increased and this means a higher power loss. Generally speaking, profile deformations have more impact on the flow rate than on the load carrying capacity. In [31] Manring et al. reported the experimental investigation of the behaviour of a three slipper bearings which differed for the socket geometry. They showed that, by determining the location of the contact point between the ball and socket joint with respect to the ball, it was possible to influence the deformation of the slipper and hence its load carrying capacity and flow leakage. In particular, if the contact point is close to the bottom of the socket, leakage and power losses are reduced but, at the same time, the load carrying capacity is lowered; the opposite situation establishes if the contact point is located to the top of the socket. Manring [32] further analyzed the case of a critical speed value for the swash plate axial piston pump for which the slipper bearing moves away from the swash plate thus generating catastrophic rupture of the hold down plate. He highlighted the importance of a good prediction of the slipper hold

down force to avoid this. Although a steady state approach has been followed in these works, interesting considerations about the slipper design were exposed and some of the critical aspects found will be the subject of future work for the authors.

Harris, Edge and Tilley [33] presented the results from an analysis of the behaviour of slipper pads developed with a dynamic model. This model includes possible contact with both the swash plate and slipper retaining plate, and it has been integrated in the Bathfp software package for the dynamic simulation of swash plate type piston pumps. They identified two different behaviours of the slipper: at low speeds, the slipper is heavily tilted radially inwards during the suction stroke but only slightly tilted during the delivery stroke. At higher speeds, the slipper runs heavily tilted radially inwards for both strokes. At very high speeds, the slipper runs in contact with both the swash plate and the retaining plate. The considerations they highlighted represent a reliable basis for comparison with the results obtained in the present work.

Iboshi and Yamaguchi [34], [35] developed a full dynamic analysis of a flat slipper and compared the results with experimental analysis. They suggested that even a flat slipper can run successfully. However, as evidenced also by other authors, the slipper they analyzed is characterized by a very wide land in comparison with the traditional designs and the friction at the ball joint is taken to be very low, so the results cannot be regarded as of universal validity.

Carbone and Soria [36] also analyzed hydrodynamic and hydrostatic slipper behaviour by means of a numerical procedure; they showed how hydrodynamic effect due to relative motion with respect to the swash plate can modify the hydrostatic pressure distribution. However, the authors considered a flat slipper sliding on a plane in a straight direction, which is quite different from the actual situation during machine operation.

In this paper, the dynamic behaviour of an overclamped slipper is analyzed using a numerical procedure. The Reynolds equation is written and solved with respect to the slipper-swash plate gap, whose height is considered variable in a two dimensional field and with time. Both the pressure distribution and the height of the clearance are obtained as results of the numerical procedure.

The contributions of forces and moments acting on the slipper are illustrated and discussed, then the numerical method adopted to solve the Reynolds equation and slipper equilibrium is presented. Then, the influences are studied of the profile geometry of the slipper surface and of the machine operating conditions on the slipper dynamic behaviour. The results are presented in terms of tilt of the slipper α , central clearance height h_0 and minimum clearance height angular position $\theta_{\min}$.

Moreover, a friction coefficient is introduced, in order to evaluate the magnitude of the friction losses determined by the slipper behaviour. Finally, detailed considerations about the slipper behaviour in the pump and in the modes of operation are made.

SLIPPER DESIGN

It can be observed in Figure 3 that the slipper is characterized by a pocket in the central zone connected to the piston chamber through an orifice of diameter d_0 . The land that defines the clearance amplitude between the slipper and the swash plate is positioned around the pocket, being confined between the inner and outer radii R_i and R_e respectively.

If the machine angular velocity is ω , the slipper also is pivoted in rotation by the piston with an angular velocity ω ; moreover, it can also rotate about its z-axis with angular velocity ω_s (spin). This depends upon the friction between the ball joint and the slipper socket, as well as on moments acting on it.

$$C_r = \frac{2 \cdot R_p^2 \cdot \ln\left(\frac{R_e}{R_i}\right)}{(R_e^2 - R_i^2) \cdot \cos \beta} \quad \text{Eq.1}$$

In Eq. 1 the clamping ratio expression as a function of the geometrical characteristics of the slipper is shown as presented by Hooke et al.; β is the swash plate angle and R_p is the piston radius.

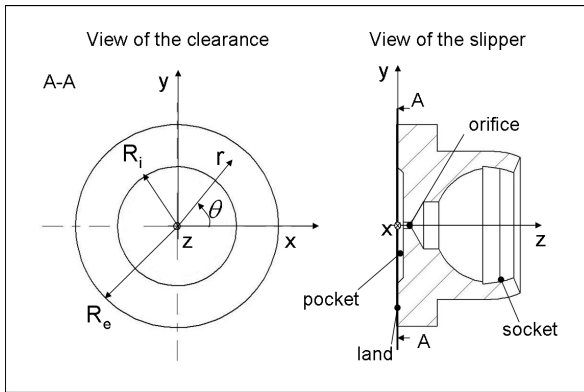


Figure 3: View of the slipper and frame of reference adopted.

NUMERICAL PROCEDURE

In order to study the lubrication mechanism of slipper bearings in axial piston machines, in the case of negligible inertial and buoyancy forces and with a reference gap height small compared with the other dimensions, the Reynolds equation has to be written here with reference to the slipper-swash plate gap. Solving the Reynolds equation allows the determination of the pressure distribution within the gap. Moreover, the total load F_f and the moment M_f acting on the slipper can be obtained from the integration of the pressure distribution in the domain considered. In Figure 3, both the Cartesian and polar frame of references adopted are sketched. During unit shaft rotation, the Cartesian frame of reference maintains the x axis tangential to the trajectory of the slipper land centre, while the y axis is directed radially outwards, as it is sketched in Figure 4.

It can be said that the total load components acting on the slipper along the x and y axes determine the viscous friction force. Moreover, the components of the moment

about x and y axes are responsible for the slipper rotation and hence for its tilted position with respect to the swash plate. The z-component contributes to the determination of the slipper rotation about z axis, also called the spin of the slipper ω_s . However, to find out ω_s it is also necessary to determine the viscous moment developed by the piston ball joint – slipper coupling but until now the authors haven't done this. For this reason, in a first approximation, the spin angular velocity ω_s has been set equal to zero.

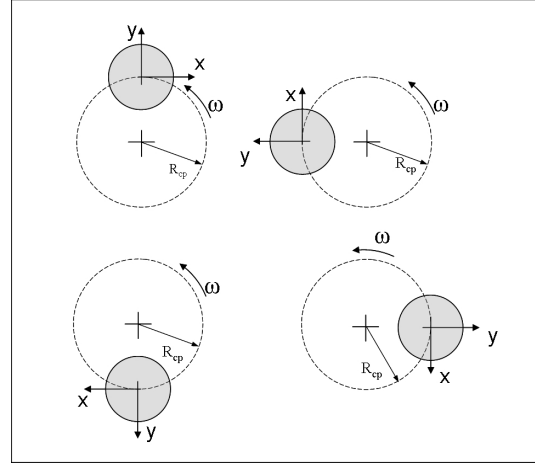


Figure 4: Frame of reference adopted for different angular position of the slipper land centre.

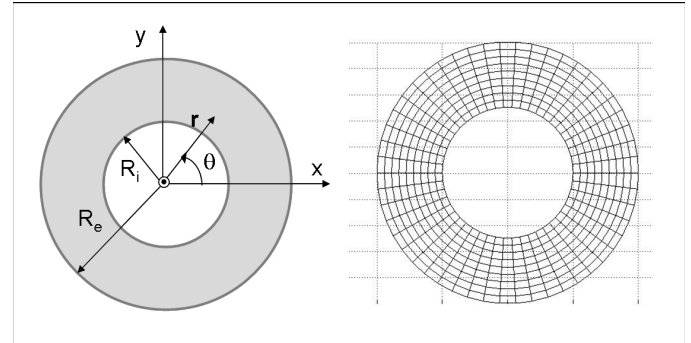


Figure 5: Example of a constant step grid applied to the slipper-swash plate clearance.

The Reynolds equation is formulated with respect to the frame of reference sketched in Figure 5, for polar coordinates (r, θ) :

$$r \cdot \frac{\partial}{\partial r} \left(r \cdot h^3 \cdot \frac{\partial p}{\partial r} \right) + \frac{\partial}{\partial \theta} \left(h^3 \cdot \frac{\partial p}{\partial \theta} \right) = 6 \cdot \mu \cdot r \cdot \left(\frac{\partial(u \cdot h \cdot r)}{\partial r} + \frac{\partial(v \cdot h)}{\partial \theta} + r \cdot \left(\frac{\partial h}{\partial \theta} \cdot \omega_s + 2 \cdot \frac{\partial h}{\partial t} \right) \right) \quad \text{Eq.2}$$

In Eq. 2 p is the pressure inside the gap, h the height of the gap, μ the dynamic viscosity of the fluid, supposed constant, and u and v are the swash plate velocity components along (r, θ) as expressed in Eq. 3 (radial and circumferential velocity components):

$$u = \omega \cdot R_{cp} \cdot \cos \theta$$

$$v = -\omega \cdot (R_{cp} \cdot \sin \theta + r) \quad \text{Eq.3}$$

In Eq.3 ω is the unit angular velocity and R_{cp} the slipper land centre trajectory radius.

Substituting Eq. 3 in Eq.2 the following can be obtained:

$$\left(\frac{\partial}{\partial r}\left(h^3 \cdot \frac{\partial p}{\partial r}\right) + \frac{h^3}{r} \cdot \frac{\partial p}{\partial \theta}\right) + \frac{1}{r^2} \cdot \frac{\partial}{\partial \theta}\left(h^3 \cdot \frac{\partial p}{\partial \theta}\right) =$$

$$= 6 \cdot \mu \cdot \left(u \cdot \frac{\partial h}{\partial r} + \frac{v}{r} \cdot \frac{\partial h}{\partial \theta} + \left(\frac{\partial h}{\partial \theta} \cdot \omega_s + 2 \cdot \frac{\partial h}{\partial t}\right)\right) \quad \text{Eq.4}$$

The instantaneous gap height, Eq. 5, is defined as a function of three points heights h_1 , h_2 , h_3 ; the points are chosen as in Figure 6. Once the height $h(r, \theta, t)$ is known, the terms $\partial h / \partial r$, $\partial h / \partial \theta$ are obtained by differentiating it with respect to the polar co-ordinates (r, θ) . For what concern the boundary conditions, it was chosen to put the pressure at the outer radius R_e equal to the drain pressure and the pressure at the inner radius R_i equal to the pocket pressure. This pocket pressure is determined by applying the continuity equation to its volume, considering the contribution of oil flow through the orifice and the inner boundary of the land.

$$h(r, \theta, t) = (2 \cdot h_1(t) - h_2(t) - h_3(t)) \cdot \frac{r \cdot \cos \theta}{3 \cdot R_e} +$$

$$+ (h_2(t) - h_3(t)) \cdot \frac{r \cdot \sin \theta}{\sqrt{3} \cdot R_e} + (h_1(t) + h_2(t) + h_3(t)) \cdot \frac{1}{3} \quad \text{Eq.5}$$

$$\frac{\partial h}{\partial r} = (2 \cdot h_1(t) - h_2(t) - h_3(t)) \cdot \frac{\cos \theta}{3 \cdot R_e} + (h_2(t) - h_3(t)) \cdot \frac{\sin \theta}{\sqrt{3} \cdot R_e}$$

$$\frac{\partial h}{\partial \theta} = -(2 \cdot h_1(t) - h_2(t) - h_3(t)) \cdot \frac{r \cdot \sin \theta}{3 \cdot R_e} + (h_2(t) - h_3(t)) \cdot \frac{r \cdot \cos \theta}{\sqrt{3} \cdot R_e}$$

$$\frac{\partial h}{\partial t} = (2 \cdot \dot{h}_1 - \dot{h}_2 - \dot{h}_3) \cdot \frac{r \cdot \cos \theta}{3 \cdot R_e} + (\dot{h}_2 - \dot{h}_3) \cdot \frac{r \cdot \sin \theta}{\sqrt{3} \cdot R_e} + (\dot{h}_1 + \dot{h}_2 + \dot{h}_3) \cdot \frac{1}{3}$$

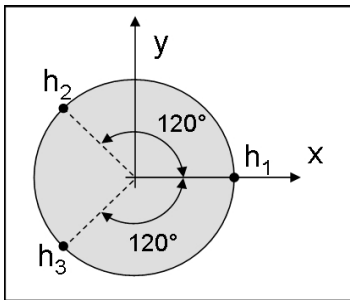


Figure 6: Bearing surface of the slipper and the three points whose heights define the height of the gap.

In this study, a finite difference method over a constant step mesh along the radial and circumferential directions is applied in order to solve the partial differential equation 4. To avoid the case of negative computed pressures within the gap, a negative value is set equal to zero when it occurs during iteration. Moreover, elastic contact between the slipper and the swash plate has been considered.

The height of the clearance is not known a priori so an iterative process has to be developed. At the first time step, the geometry of the gap, i.e. h , is assigned as an initial condition of the calculus; a first attempt value of $\dot{h}_1$, $\dot{h}_2$, $\dot{h}_3$ are used and the gap pressure distribution is obtained from the Reynolds equation. Equilibrium of forces and moments acting on the slipper is verified and the procedure is

stopped only if a certain tolerance is reached. If not, the terms $\dot{h}_1$, $\dot{h}_2$, $\dot{h}_3$ are updated and the procedure is repeated. At each step the height $h(r, \theta, t)$ is obtained from h_1 , h_2 , h_3 , which are determined by numerical integration of $\dot{h}_1$, $\dot{h}_2$, $\dot{h}_3$ at the previous time step.

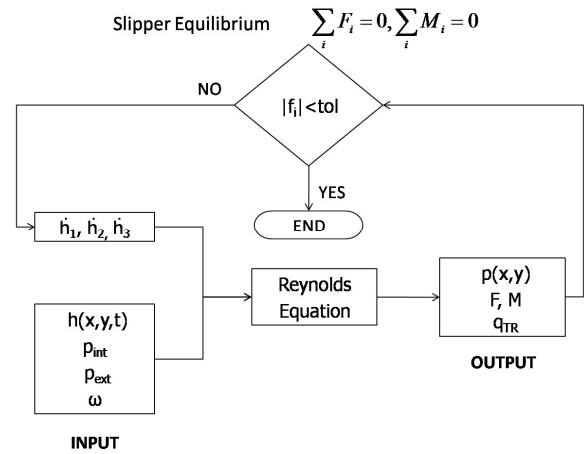


Figure 7: Diagram of the numerical procedure.

FORCES ACTING ON THE SLIPPER

It is postulated that the slipper can move along the z axis and rotate about both x and y axis. Three equilibrium equations can therefore be written:

$$I_x \cdot \ddot{\delta}_x = \sum_i M_{x,i}$$

$$I_y \cdot \ddot{\delta}_y = \sum_i M_{y,i} \quad \text{Eq.6}$$

$$m_{sl} \cdot \ddot{z} = \sum_i F_{z,i}$$

In Eq. 6 I_x and I_y are the moments of inertia about x and y axes respectively and m_{sl} the slipper mass.

Given the fact that inertial forces are negligible compared with the other forces considered, the contributions of forces and moments acting on the slipper are the following ones: the force and moment exerted by the fluid film in the gap, F_f and M_f respectively. These can be obtained by the integration of the pressure distribution determined through the Reynolds equation, the forces originated by the piston action on the slipper, F_{P-Sl} , and the centrifugal force F_C .

By imposing the condition of axial equilibrium of the piston, it is possible to determine the force exerted by the slipper on the piston F_{Sl-P} . This will be opposite to the force exerted by the piston on the slipper. This contribution is in the z direction and it is considered to be applied to the centre of the piston ball joint. The forces which contribute to the piston axial equilibrium, see Figure 8, are: the force generated by the fluid pressure acting on the piston F_P ; the inertial force F_i of the slipper and piston assembly; the viscous friction force $F_{CB-P,v}$ arising in the piston-cylinder gap characterized, for simplicity, by a constant height; the force exerted by the slipper on the piston F_{Sl-P} . The contribution of F_{P-Sl} is expressed by Eq.7, where β is the swash plate angle.

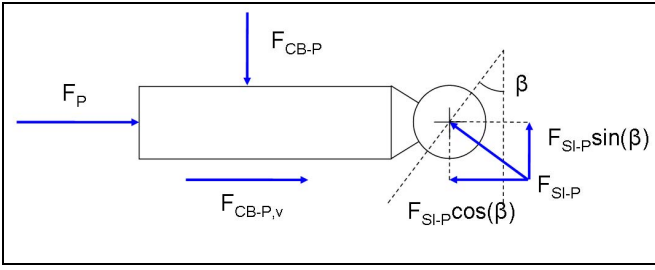


Figure 8: Forces acting on the piston

$$F_p + F_{Sl-P} \cos(\beta) + F_{CB-P,v} = m \cdot \ddot{x}_p = F_i$$

$$\Rightarrow F_{P-Sl} = -F_{Sl-P} = -\left(\frac{F_i - F_p - F_{CB-P,v}}{\cos(\beta)} \right) \quad \text{Eq.7}$$

In Eq.7 m is the piston-slipper assembly mass.

The centrifugal force F_c acting on the slipper can be expressed as follows:

$$F_c = m_{Sl} \cdot R_{cp} \cdot \omega^2 \cdot \cos \beta \quad \text{Eq.8}$$

It can be said that, the force exerted by the fluid on the slipper, directed along the z axis, is contrasted and balanced by the piston force F_{P-Sl} . On the other hand, the centrifugal force and the force components derived by the action of the fluid on the slipper and directed along x and y axes appear to be unbalanced. In this work, it is assumed that these components are balanced by the action of forces which are applied to the centre of the piston ball-joint and have the only function to generate moments on the slipper.

Equations 4 and 6 allow the determination both of the pressure distribution within the gap and of the gap shifting velocity $\vec{h} = [\dot{h}_1, \dot{h}_2, \dot{h}_3]^T$. It should be appreciated that equations 6 are not linear and some of their terms do not have an analytical expression. This is because they depend on the solution of a partial differential problem (Eq.4). In vectorial notation it can be written:

$$J \cdot \vec{h} = \vec{F} \quad \text{Eq.9}$$

In Eq.9 both the analytical expression of $\vec{F}(\vec{h}, t)$ and the Jacobian J are unknown; hence, to solve this system of equations it is necessary to perform an iterative process whereby the Jacobian J is numerically built up. In this work, the Broyden method is used for this procedure, coupling it with appropriate artifices in order both to avoid singularities in the Jacobian and to adapt at optimal stages the time step of the method.

$$\vec{h}_{n+1} = \vec{h}_n - J_n^{-1} \cdot \vec{F}(\vec{h}_n)$$

$$J_n = J_{n-1} + \frac{(\vec{F}_{n+1} - \vec{F}_n) - J_{n-1} \cdot (\vec{h}_{n+1} - \vec{h}_n)}{\left\| \vec{h}_{n+1} - \vec{h}_n \right\|^2} \cdot (\vec{h}_{n+1} - \vec{h}_n)^T \quad \text{Eq.10}$$

The iterative process ends only when:

$$\left| F_i(\vec{h}, \vec{h}, t) \right| < \varepsilon, \quad i=1...3 \quad \text{Eq.11}$$

where ε is a quantity of convenient smallness.

It may be worth mentioning that, in order to numerically solve Eq.4, the appropriate non-dimensioned quantities have been introduced to overcome the problem of greatly different orders of magnitude in the variables involved; the pressure and height of the gap are instances.

RESULTS

SLIPPER SURFACE PROFILE AND LOAD CARRYING CAPABILITY

In this section, the results obtained from the analysis of a slipper bearing used in axial piston motor are presented. They are illustrated in terms of height of the clearance corresponding to the slipper land centre, h_0 , and with the slipper tilt, α , valued for the section characterized by the minimum clearance height. The results are sketched as a function of the angular position of the motor shaft θ_r . The slipper bearing considered is characterized by the geometrical parameters reported in Table 1 and by a flat surface profile.

Parameter	Value	Unit
Clamping Ratio C_r	1.1	-
Orifice Diameter d_o	0.8	mm

Table 1: Geometrical parameters of the slipper

To focus on the load carrying mechanism only, two particular test conditions were chosen. The swash plate angle β was set to zero and the pressure within the cylinder was considered constant and equal to the high pressure value p_H . The analysis was performed for the operating conditions listed in Table 2.

Operating Parameter	Value	Unit
High Pressure Value p_H	1 - 5 - 10	MPa
Low Pressure Value p_L	0.1	MPa
Drain Pressure p_D	0.1	MPa
Rotational Speed n	1000 - 2000 - 3000	rpm

Table 2: Operating parameters adopted in the simulation

For the lower value of p_H the central height h_0 , plotted as a function of the unit shaft angular position θ_r , see Figure 9, shows an oscillatory trend which is emphasized at high rotational speed. In particular, for $n=3000$ rpm, the oscillation amplitude is so high to cause direct contact between the slipper and the swash plate at the external edge of the land. For this condition the simulation prematurely stops. This unstable behaviour is clearly due to the inability of the slipper to find an equilibrium position. This is as much evident as the hydrodynamic