

Multibody dynamics of pivot slipper pad thrust bearing in axial piston machines incorporating thermal elasto-hydrodynamics and mixed lubrication model



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ABSTRACT

A model for the calculation of multibody dynamics incorporating a transient, three-dimensional, thermal elasto-hydrodynamic pivot pad contact in swash plate axial piston pumps is presented. The simulation is based on the numerical solution of equations of motion, generalized Reynolds, energy and Fourier heat equations. The equation of motion of the hybrid pivot bearing is derived by Lagrangian formalism and solved in due consideration of friction in the spherical joint and piston-barrel. Mass conserving cavitation, non-Newtonian flow and a mixed friction model with consideration of real-measured surface topologies are implemented. Results for a slipper pad with retain plate are presented. The impact of the retain device on slipper pad friction and temperature are discussed.

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1. Introduction

The swash plate type axial piston pump connects high power density with low installation space. A displacement volume is generated by rotation of the drive shaft and the angled swash plate (see Fig. 1). The cylinder block and piston-slipper assembly rotate with the drive shaft. The angled swash plate enforces the stroke movement of the piston and this pressurizes the fluid to be pumped into from the displacement chamber through the valve plate. High piston axial loads arise from the pressurized fluid and push the slipper pad against the swash plate. The friction in the slipper pad-swash plate contact and in the spherical joint is reduced by pressurized fluid. The pressurized fluid in the piston chamber is guided through the piston head and the slipper pad into both contacts by boreholes. According to this the spherical joint and the slipper pad are both hydrostatically supported. To minimize the lifting of the slipper pad the retain plate is mounted by a spring pressing the slipper pad on the swash plate.

The slipper pad and swash plate contact is present multiple times in the axial piston pump and affects the efficiency of the machine. The dynamic transition from delivery to suction stroke

can lead to a lift of the slipper pad. A higher film thickness between slipper pad and swash plate leads to higher leakage and a decrease in the hydraulic efficiency. Transition from suction to delivery stroke can lead to an impact of the slipper pad on the swash plate, mixed friction and damage of the slipper pad. On the one hand inappropriate design of the slipper pad can lead to operation of the slipper pad under high mixed friction and can lead to the breakdown of the pump. On the other hand the market demands pumps with higher pressure, displacement volume and lower installation space. The design engineer faces the challenge to design the slipper pad to run in delivery stroke under minimum mixed friction and in suction stroke with minimum leakage.

The slipper pad in axial piston pumps is a hybrid pivot pad thrust bearing under transient conditions. It is pivot-mounted by a hydrostatic spherical joint on the piston. The slipper pad is hydrostatically supported but works hydrodynamically as well. Depending on the location of the lowest friction torque, the slipper pad adapts the film thickness by rotations in the spherical joint or rotation of the piston in the cylinder bore. The calculation of the slipper pad – swash plate contact requires the consideration of the hydrodynamics in the fluid film as well as the thermal effects on the lubricant rheology and solid bodies. Since the slipper pad works in mixed friction area, microhydrodynamic effects have to be taken into account. High loads, hydrostatic pressure and thermal conditions presuppose the calculation of elastic deformations of the lubricated gap. The influence of the tribological contacts,

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Nomenclature

C	spring rate (N/m)
F	force (N)
F_p	pressure force in piston chamber (N)
I	inertia tensor
K	stiffness matrix
L	Lagrangian function
M	momentum (Nm)
Q^*	generalized forces and momentum
R	pitch circle (m)
T	kinematic energy (J) or temperature (K)
V	potential energy (J)
a	acceleration (m/s^2)
c_{hd}	hold down force partition coefficient (–)
d	diameter (m)
h	gap height (m)
l	length (m)
m	mass (kg)
p	pressure (Pa)
p_c	contact pressure (Pa)
$p_{c,lim}$	plastic flow pressure (Pa)
p_h	hydrodynamic pressure (Pa)
p_{hs}	hydrostatic pressure (Pa)
q	generalized coordinates
r	position vector (m)
t	time (s)
u, v, w	velocity in φ, r, z direction (rad/s), (m/s), (m/s)
x, y, z	Cartesian coordinates (m)
Ω	computation domain
Φ^p	pressure flow factor
Φ^s	shear flow factor
β	swash plate angle (rad)
β_{th}	coefficient of thermal expansion ($1/K$)
η^*	effective dynamic viscosity (Pa s)
λ	thermal conductivity (W/m K)
μ	friction coefficient
ω	angular velocity (rad/s)
ρ	density (kg/m^3)

τ	shear stress (Pa)
θ	gap fill ratio
ϑ	temperature (K)
φ, r, z	cylindrical coordinates (rad), (m), (m)

Frequently used indices

'	piston-coordinate
'	slipper pad-coordinate
A, B	contact points
CG	center of gravity
D	damper
G	global-coordinate
c	centrifugal
cav	cavitation
con	contour
cr	critical
cyl	cylindrical
d	drive shaft
def	deformed
el	elastic
f	friction
$grav$	gravitational
hd	hold down/retain plate
i	inertia
j	joint
liq	liquid
m	master node
mix	mixed
p	piston
pos	positive
red	reduced
res	resulting
rig	rigid
s	slave node or slipper pad
sph	spherical
sw	swash plate

piston bore and spherical joint, and the retain plate require the analysis of body dynamics.

The calculation of thrust bearings has been one of the priorities of tribology research in recent decades. The complexity of the thrust bearing model has risen from isothermal hydrodynamics to thermal elastohydrodynamic models with consideration of mixed lubrication condition. Zienkiewicz [1] indicates the importance of the variation of viscosity across the fluid film. Dowson [2] presents the generalized Reynolds equation in consideration of variation of viscosity both along and across the fluid film by implementation of integrals of viscosity across the film into the Reynolds equation. The thermal distortion of the pad influences the load-carrying capacity and is taken into account by Sternlicht et al. [3]. The effect of hot oil transfer between pads is analysed by Ettles and Avandi [4] and Vohr [5]. Illner et al. [6] present an elastohydrodynamic pivot pad thrust bearing model under consideration of real-measured surface topologies, mass conserving cavitation model and non-Newtonian fluid model.

In the area of axial piston machine design, the focus of the last decades has been on the prediction of the slipper pad efficiency. Much work has been done to determine the slipper pad leakage and friction force experimentally and numerically. First analytic model for the analysis of power loss of the slipper pad are

presented by Shute and Turnbull [7]. Hooke and Li [8] indicate the importance of the friction momentum in the spherical joint and between piston and barrel. Harris et al. [9] present equations of motion of the slipper pad incorporating with an analytic solution of the Reynolds equation and spring elements as contact model.

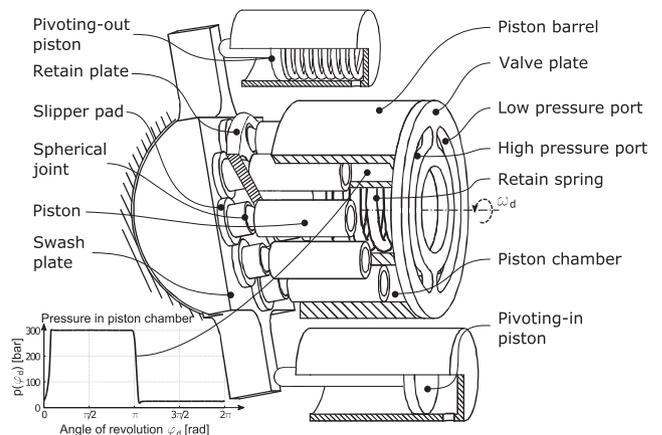


Fig. 1. Axial piston machine with retain plate.

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