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OPTIMISATION OF THE LIFT CHARACTERISTICS OF AN AXIAL PISTON PUMP GROOVED SLIPPER.

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ABSTRACT

Grooves effects on slippers are understood from a qualitative point of view and analytically for slippers without grooves. This study addresses the effect of groove dimensions and position via expanding the equations of flow, pressure distribution and force developed for any number of lands. A unique advantage of having explicit equations is that they may then be used to design slippers in order to have maximum lift, via optimizing the groove length, depth and position. This cannot be achieved at present using previous knowledge apart from many simulation runs using a numerical simulation package.

KEY WORDS

axial piston slipper, effect of lands, direct solution

NOMENCLATURE

h_i slipper general height (m)
n number of flat plates, including the grooves
P_i general pressure (N / m^2).
Q_i generalized flow (m^3/ s)
r_i slipper general radius (m)
μ fluid dynamic viscosity. (kg / m s)

INTRODUCTION

In the majority of cases the effect of the different pressure balancing grooves cut on pistons and slippers has been neglected. Although the groove effect on the flow and the pressure distribution is not expected to give a radically different pattern from the single-land case, the extension developed in this paper is a step forward towards a better understanding of the effect. A mathematical approach was presented by Bergada and Watton [1,2] for a single grooved slipper and Fisher [3] studied the case of a flat slipper with single land on a rotating plate. Böinghoff, [4] studied theoretically the static and dynamic forces and torques acting on a single piston. Hooke [5] showed that a degree of non-flatness was essential to ensure the successful operation of the slipper, and the non-flatness must have a convex profile. He later[6] studied more carefully the couples created by the slipper ball. Iboshi and Yamaguchi [7,8], working with a single land slipper, determined the flow and the main moments acting on the slipper. Hooke [9] also studied more carefully the effect of non-flatness and the inlet orifice on the performance of the slipper. In [10,11] Hooke focused on the lubrication of overclamped slippers and considered the three different tilting couples acting on slipper, finding that the tilting couple due to friction at the slipper running face is much smaller than the ones created at the piston-cylinder, piston-slipper interfaces, and the centrifugal term. Takahashi et al [12] studied the unsteady laminar incompressible flow between two parallel disks with the fluid source at the centre of the disks. Li et al [13] studied the lubrication of composite slippers on water based fluids. Koc et al [14] focused their work on checking whether underclamped flat slippers could operate successfully or whether a convex surface was required. They took into account the work done by Kobayashi et al [15] on the measurements of the ball friction. They concluded that polishing of the running face of the slipper to a slightly convex form, appeared to be essential for successful operation under all conditions. Harris et al [16], created a mathematical dynamic model for slipper-pads, in which lift and tilt could be predicted, the model was able to handle the effect of the possible contact with the swash plate. Koc and Hooke[17,18] studied more carefully the effects of orifice size, finding that the underclamped slippers and slippers with larger orifice sizes run with relatively larger central clearances and tilt more than those of overclamped slippers with no orifice. Ivantysynova [19] developed a package called CASPAR which uses the bi-dimensional equation of lubrication and the energy equation in differential form. It was shown how transient cylinder pressure could be computed by knowing the leakage characteristics of the piston chamber. In addition, the dynamic clearance and tilt of the slipper was studied over one revolution of the pump and a single land slipper plate was used in the theoretical and experimental analysis.

ANALYSIS

Although a large amount of work has been done in order to better understand slipper behaviour, very little work has focused on flow characteristics for slippers with two or more lands, and no attempt to explain mathematically the two land slipper behaviour has been found. The main piston and slipper assembly used in this study is shown in figure 1, and is one of nine pistons from a pump having a maximum volumetric displacement is 0,031 dm^3 /rev.

It will be seen that the slipper design uses two full lands and the approach selected seems to be the personal design philosophy of the particular pump manufacturer. Considering previous work [1,2] together with the assumptions of laminar flow, no slipper tilt, steady state conditions, rotation effects are negligible, the flow is radial, the pressure drop is dominated by the radial flow, then the solution of Reynolds gives equation 1.

Such a generic equation can be used for a slipper with any number of lands, bearing in mind that the equation presented here does not regard the slipper pocket as one of the lands.

$$Q = \frac{\pi}{6\mu}\,\frac{p_{inlet} - p_{outlet}}{\sum_{i=1}^{i=n} \frac{1}{h_i^3} \ln\frac{r_{(i+1)}}{r_i}} \qquad (1)$$

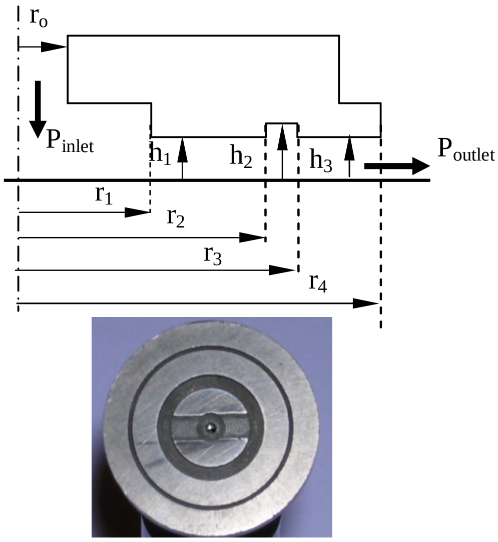

Figure 1. Slipper studied (Courtesy Oilgear Towler UK)

The dominant hydrostatic force will be given as the addition of the slipper pocket force plus the force due to the integration of the pressure distribution along the slipper two lands and groove. The force balance for the single-groove slipper design under study will be:

$$\begin{aligned} F_{lift} &= P_{inlet}\,\pi(r_4^2 - r_o^2) \\ &- C\pi\, r_4^2 \left(\frac{1}{h_1^3}\ln(\frac{r_2}{r_1}) + \frac{1}{h_2^3}\ln(\frac{r_3}{r_2}) + \frac{1}{h_3^3}\ln(\frac{r_4}{r_3}) \right) \\ &+ C\pi \left(\frac{r_2^2 - r_1^2}{2h_1^3} + \frac{r_3^2 - r_2^2}{2h_2^3} + \frac{r_4^2 - r_3^2}{2h_3^3} \right) \end{aligned} \qquad (2)$$

$$C = \frac{P_{inlet} - P_{outlet}}{\frac{1}{h_1^3}\ln(\frac{r_2}{r_1}) + \frac{1}{h_2^3}\ln(\frac{r_3}{r_2}) + \frac{1}{h_3^3}\ln(\frac{r_4}{r_3})} \qquad (3)$$

In this study the slipper is assumed to be parallel to the swash plate, hence $h_1 = h_3$.

MODIFIYING GROOVE GEOMETRY AND POSITION.

Data for the two slippers evaluated is shown in Table1.

Figure 2 shows the typical pressure distribution along the entire slipper face for a 0.4 mm width groove, when the groove is located near the inner radius or near the outer radius. It was noticed that for groove depths of 0.4 mm, pressure is maintained constant along the groove length and for depths of few microns there is a considerable pressure differential between groove borders. It must be pointed out that the pressure distribution curve for an inner groove with 15 microns depth falls below the curve at 0.4 mm depth, while the opposite happens for the outer groove. Integrating the pressure distribution shows that the grooves located at an inner radius will produce higher lift than when located in any other position.

Table 1. Dimensions of the two slippers considered

	Slipper 1 (1 groove)	Slipper 2 (1 groove)
r_0 (mm)	0.5	0.4
r_1 (mm)	5	7.7
r_2 (mm)	7.43	11.175
r_3 (mm)	7.83	11.725
r_4 (mm)	10.26	15.2
Pocket depth (mm)	$h_1 + 0.6$	$h_1 + 0.55$
Groove depth (mm)	$h_1 + 0.4$	$h_1 + 0.4$

Calculations in Table 2 show the leakage flow percentage increase compared with the single land slipper, when modifying groove depth and length, while maintaining the slipper clearance at 10 microns. Different groove positions such as groove centred, groove located at the centre of the inner land and grove located at the centre of the outer land are also evaluated. Notice that leakage increases with groove length and depth, on the other hand leakage increases much faster when the groove is located at the inner land. Due to its linearity, the leakage percentage increase is independent of the inlet pressure.

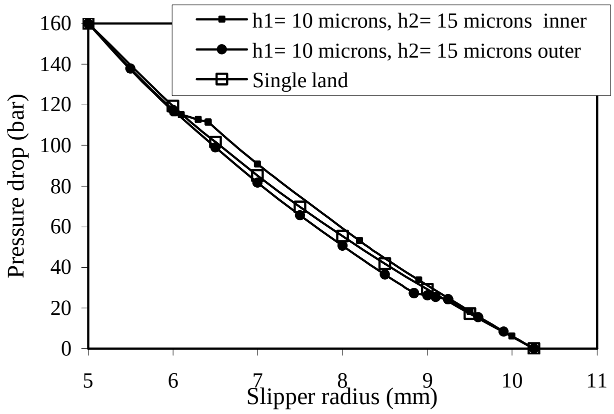

Figure 2 Pressure distribution for different groove positions, groove length 0.4 mm, inlet pressure 160 bar (slipper 1)

Table 2 Leakage increase for different designs (slipper 1)

Groove depth Groove length	Leakage increase %	Groove depth Groove length	Leakage increase %	Groove depth Groove position	Leakage increase %
h_2 = 0.4 mm length = 0,8mm	16.6	h_2 = 15 microns length = 0.8mm	11.1	h_2 = 0.4mm Inner	9.8
h_2 = 0.4 mm length = 0.4mm	7.9	h_2 = 15 microns length = 0.4mm	5.4	h_2 = 0.4mm Central	7.9
h_2 = 0.4 mm length = 0.2mm	3.8	h_2 = 15 microns length = 0.2mm	2.7	h_2 = 0.4mm Outer	6.6

Table 3 Lift force % change for different slot positions and depths (slipper 1).

Slot position	Groove length= 0.4 mm Slipper single land force =1746N@100bar					
	h_1 = 2.54 microns		h_1 = 10 microns		h_1 = 20 microns	
	h_2 microns	Force %	h_2 microns	Force %	h_2 microns	Force %
Inner	5	2.60	15	2.10	25	1.50
	400	3.00	400	3.00	400	3.00
Centred	5	-0.28	15	-0.23	25	-0.16
	400	-0.33	400	-0.33	400	-0.33
Outer	5	-2.60	15	-2.10	25	-1.40
	400	-3.06	400	-3.06	400	-3.06

In Table 3 the lift force has been evaluated when a groove of 0.4 mm length is located at a inner, centre and outer radius. It can be concluded that for an inner slot, lift force decreases as the gap slipper plate increases, and the force increases as the groove depth does. On the other hand when the groove is located at the outer radius the force will increase when increasing the slipper clearance, and it will decrease as the groove depth increases. It is clear that a groove located at an inner land will produce lift forces higher than for a single land slipper. From the equations presented, and based on (slipper 1) is shown that the effect of groove depth on pressure change across the groove is negligible for depths beyond 0.1 mm and absolutely beyond 0.3 mm.

A more comprehensive comparison of the effect of groove position and width on the lift force and leakage flow rate, for the same geometrical conditions, may be seen from the 3D plots shown as figure 3.

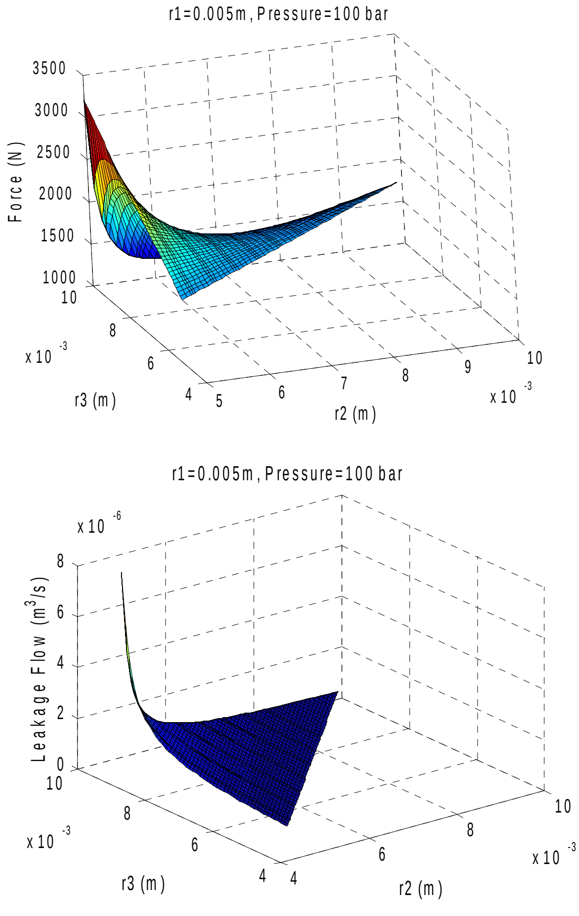

Figure 3. Variation of the force over the slipper face, and the leakage flow rate. Inlet pressure 100 bar (slipper 1).

The hydrostatic force provided at the piston side is 1678N@100bar piston pressure which has to be ostensibly balanced by the slipper hydrostatic lift by design. For no groove the hydrostatic lift would be 1746N@100bar piston pressure and the slipper would be underclamped. When operating with grooved slippers, it would be useful to know the groove dimensions to obtain maximum lift. This is now possible analytically for the first time using the previously set of equations. Since several parameters such as groove length, depth and position affect the lift force, some conditions must be established:

- dimensions r_1 and r_4 will be maintained constant.
- Given inner land outside radius, r_2, it is possible to find the value of r_3 to obtain maximum lift.
- this value can be obtained from the derivative of the total lift force, equation (2), versus r_3, and equating the resulting equation to zero. This process leads to the equation:

$$0 = r_3\left[\frac{1}{h_1^3}\ln\left(\frac{r_2}{r_1}\right)+\frac{1}{h_2^3}\ln\left(\frac{r_3}{r_2}\right)+\frac{1}{h_3^3}\ln\left(\frac{r_4}{r_3}\right)\right]- \left(\frac{1}{r_3}\right)\left[\frac{1}{h_1^3}\frac{r_2^2-r_1^2}{2}+\frac{1}{h_2^3}\frac{r_3^2-r_2^2}{2}+\frac{1}{h_3^3}\frac{r_4^2-r_3^2}{2}\right] \quad (4)$$

When substituting the values of r_1, r_2, r_4, h_1, h_2, and h_3 in equation (4), the optimum external groove radius r_3 to obtain maximum lift can be found.

Figure 4 shows the optimum groove external radius r_3 to obtain maximum lift for a given internal radius r_2, and for a set of inner land inside radius r_1 and the two different slippers. Notice that for a given slipper, the smaller the radius r_2 the bigger r_3 can be, and this will lead to bigger lift forces. The evaluation of slipper leakage for different inlet radius r_1, a set of inner groove radius r_2 and for each case the optimum outer groove radius r_3 shows that the leakage increases as r_2 decreases.

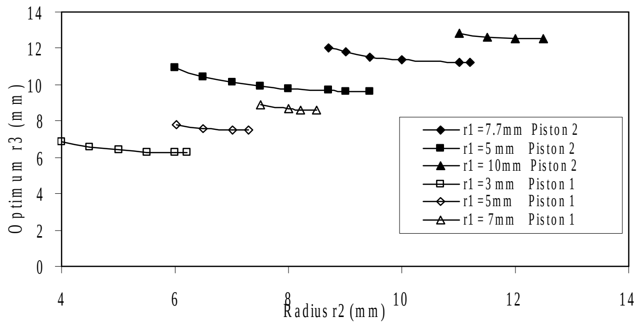

Figure 4 Optimum r_3 versus r_2 for a set of inner land inside radius, both slippers.

CONCLUSIONS

- Considering the boundary conditions for a multi-land slipper, it has been shown that a completely generalised solution for flow rate and radial pressure distribution can be obtained.
- It has been shown that the hydrostatic lift force can vary with groove placement and design and a localised optimum position exists.
- Lift in any single land slipper can be increased when creating a groove at an inner radius of the slipper land and maximum lift can be obtained when groove length is design according to equation (4).
- Grooves located at the slipper centre or towards the outer radius create lifts lower than a single land slipper, and therefore should be avoided unless necessary for stability.
- To maintain pressure constant along the slipper groove, it has been found thanks to the equations presented that a depth higher than 0.3 mm is needed for the groove and about fifty times the slipper clearance (about 0.5$\rightarrow$ 0.6 mm) for the pocket.

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