

A Computational Approach on the position of Load Centre of a Slipper Bearing

Dr. Kishan Choudhuri¹

¹ National Institute of Technology, Agartala.

Received: 8 December 2011 Accepted: 3 January 2012 Published: 15 January 2012

Abstract

The slipper bearing is an integrated part of an axial piston pump. Proper lubrication is important for successful operation of the slipper bearing. This type of bearing is a type of hydrostatic thrust bearing. Many works have been done in the recent past on this type of bearing. This current work is based on the theoretical investigation of the locus of the load centre of this type of bearing. The leakage losses and the slipper drag are two important factors on which this type of bearing is designed. The load carrying capacity has a direct impact on those two factors. On the other hand the stability of the bearing depends upon the position of the load centre. There are many input factors on which the position of load centre is varied. These input factors thus play an important role on the smooth operation of such bearing. Such input variable are slipper tilt, applied pressure on the slipper, slipper speed, slipper non flatness angle or slipper land size. On the variation of these input variables, the nature of the position of load centre is plotted in this work. Based on the results the reasonable conclusions are made.

Index terms— Slipper, position of load centre, slipper tilt, lubrication.

1 Introduction

axial piston swash-plate type hydrostatic pumps are being used extensively in aircraft, industrial and agricultural systems since they can transmit large specific power and the flow rate from them can be varied. A basic difference in the design of various models of axial piston pumps is how the pistons contact the swash plate. Many design use a bronze slipper positioned between the piston and the swash plate. With this design, hydraulic fluid is fed through internal passages to the piston/slipper and slipper/swash plate interfaces to supply lubrication at these surfaces. Some axial do not use a slipper, but rather finish each piston with a case-hardened spherical dome. The spherical dome contacts the swash plate in such a fashion, much like the contact that occurs in ball bearings. Elimination of the slipper reduces costs and eliminates the disadvantages of the slipper design, but unfortunately, it creates other problems. One of these is wear at the spherical dome/swash plate interface. The fig. 1 shows a typical slipper-piston assembly. The slipper is pivoted on the ball at the end of the piston to allow it to adjust to the swash plate angle and to rotate relative to the piston. High pressure fluid from the piston is connected via the control orifices in the piston and slipper to the central slipper pool allowing covered the influence of the orifice size on the performance of the bearing. Koc and Hooke [1] examined the effect of the tilting couples on the behavior of the slippers experimentally. Wang and Yamaguchi [2], [3] clarified experimentally and theoretically the effects of nozzle and thermoplastic materials on the characteristics of hydrostatic bearing/seal parts in water hydraulic axial pumps and motors. Manring [4] investigated the effects of pressureinduced deformations on the characteristics of hydrostatic thrust bearing. Manring [5] investigated experimentally, the effect of different socket geometry in the performance of slipper bearing. They found the effect on the leakage flow, load carrying capacity and the film thickness of the slipper bearing. In the work of Nie S. L. [6], the characteristic equation of the hydrostatic slipper bearing with an annular orifice damper is formulated, where the effects of various geometric

103 It can be observed from figure 3 through figure 6, that the slipper center stays in the center line of motion
104 or X axis for small tilt angles. For small tilt angle the center of load moves firstly towards the negative X axis.
105 More tilt brings the center of load on the positive side of X axis. A little more tilt bring the centre of load to the
106 actual centre of slipper. But for more tilt angle the centre of tilt angle the load goes outside the area of slipper
107 which is not possible. Therefore a higher tilt is not possible to occur in actual slipper operation. Moreover the
108 variation of pressure, speed, slipper area and non flatness angle put different response to the position of load
109 centre. It can be observed that higher pressure in the slipper pocket tries to keep the slipper load centre in the
110 actual centre of slipper and lower pressure tries to deflect the load centre away from actual slipper centre. In the
111 same way higher speed of slipper tries to keep the slipper load centre in the actual centre of slipper and lower
112 speed tries to deflect the load centre away from actual slipper centre. From figure 5, it can be observed that a
particular amount of nonflatness angle tries to keep the load centre in the actual slipper centre.



Figure 1: Op??

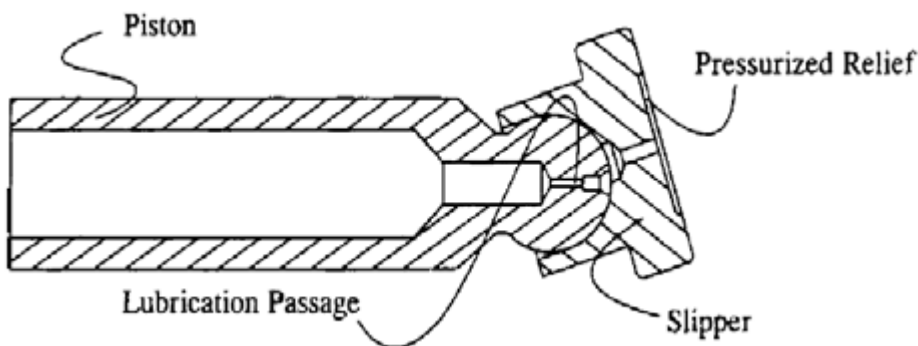
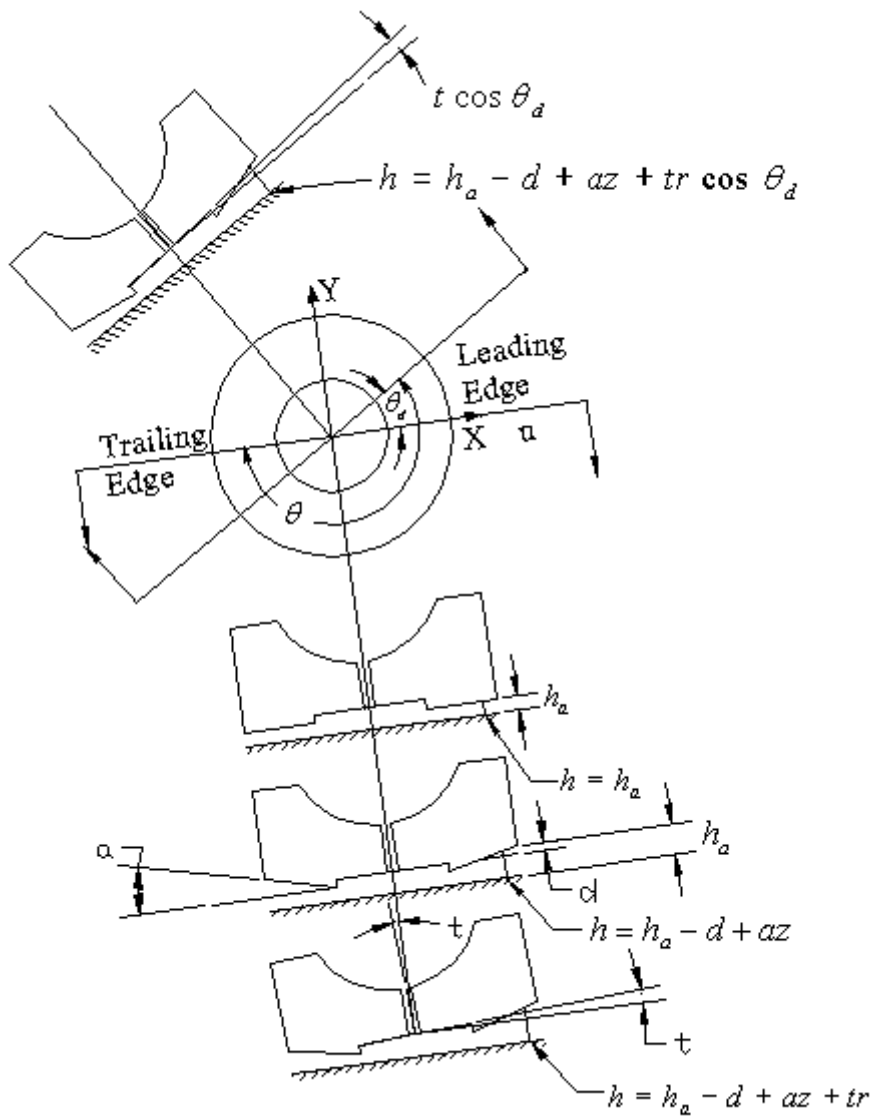
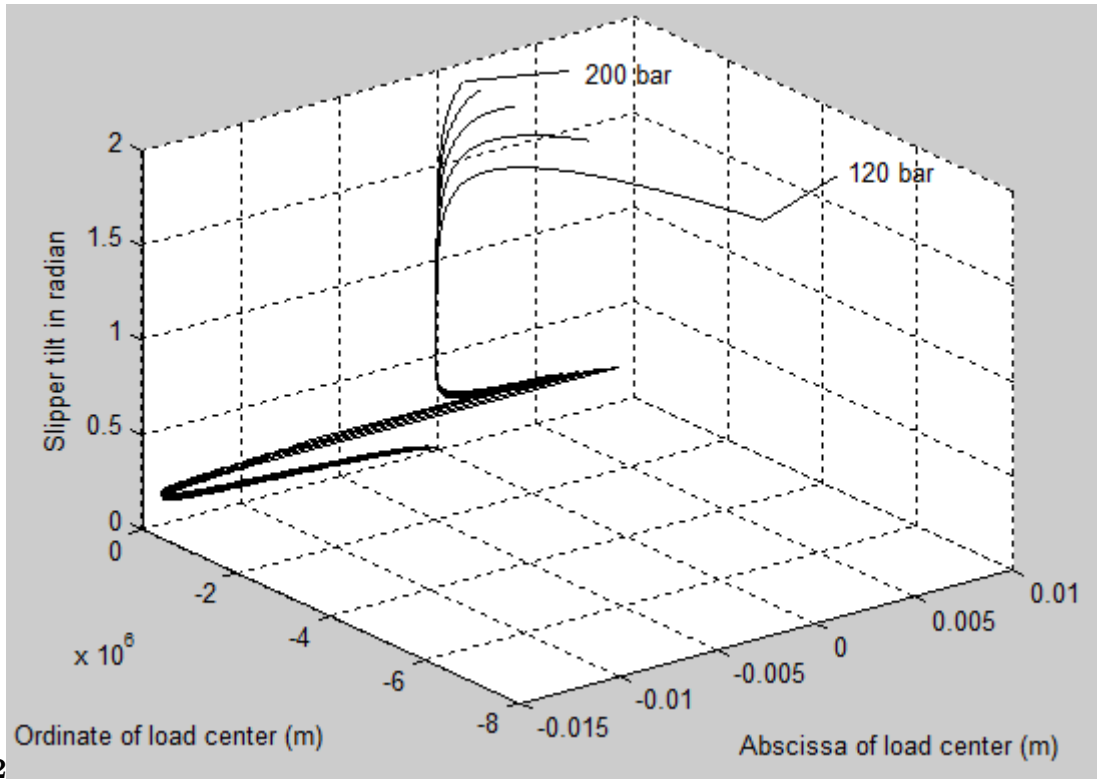


Figure 2: A



1

Figure 3: Figure 1 :



2

Figure 4: Figure 2 :

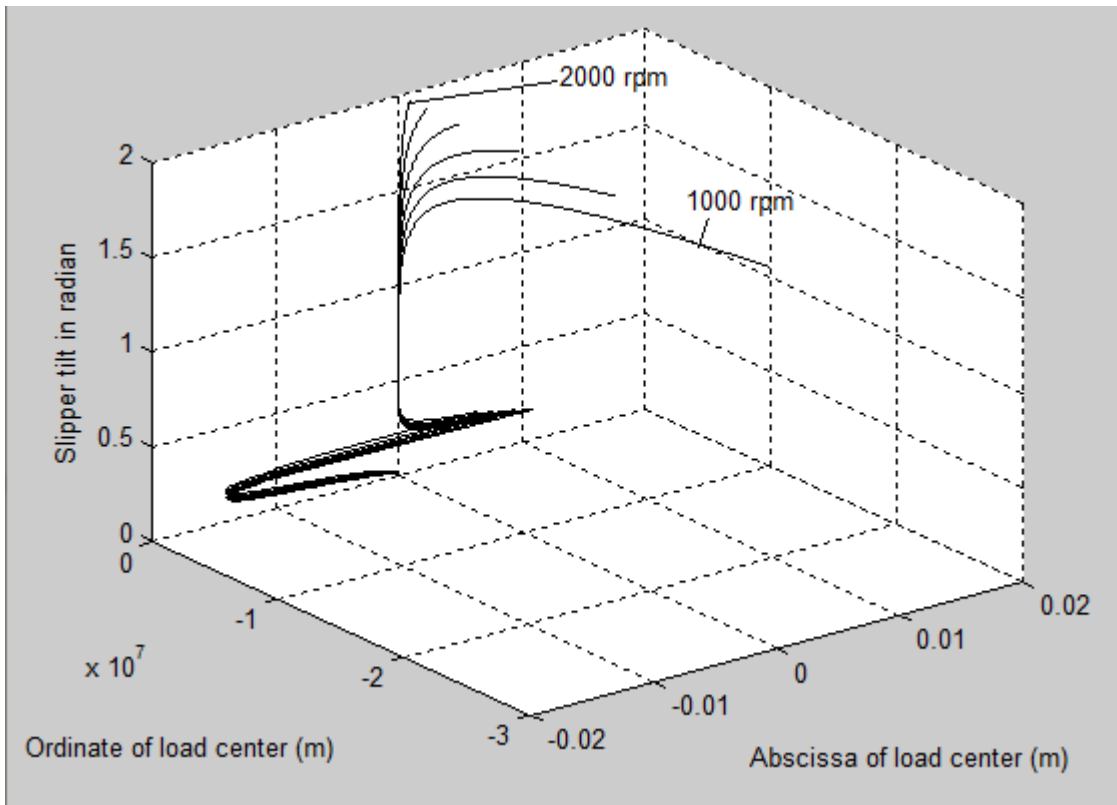
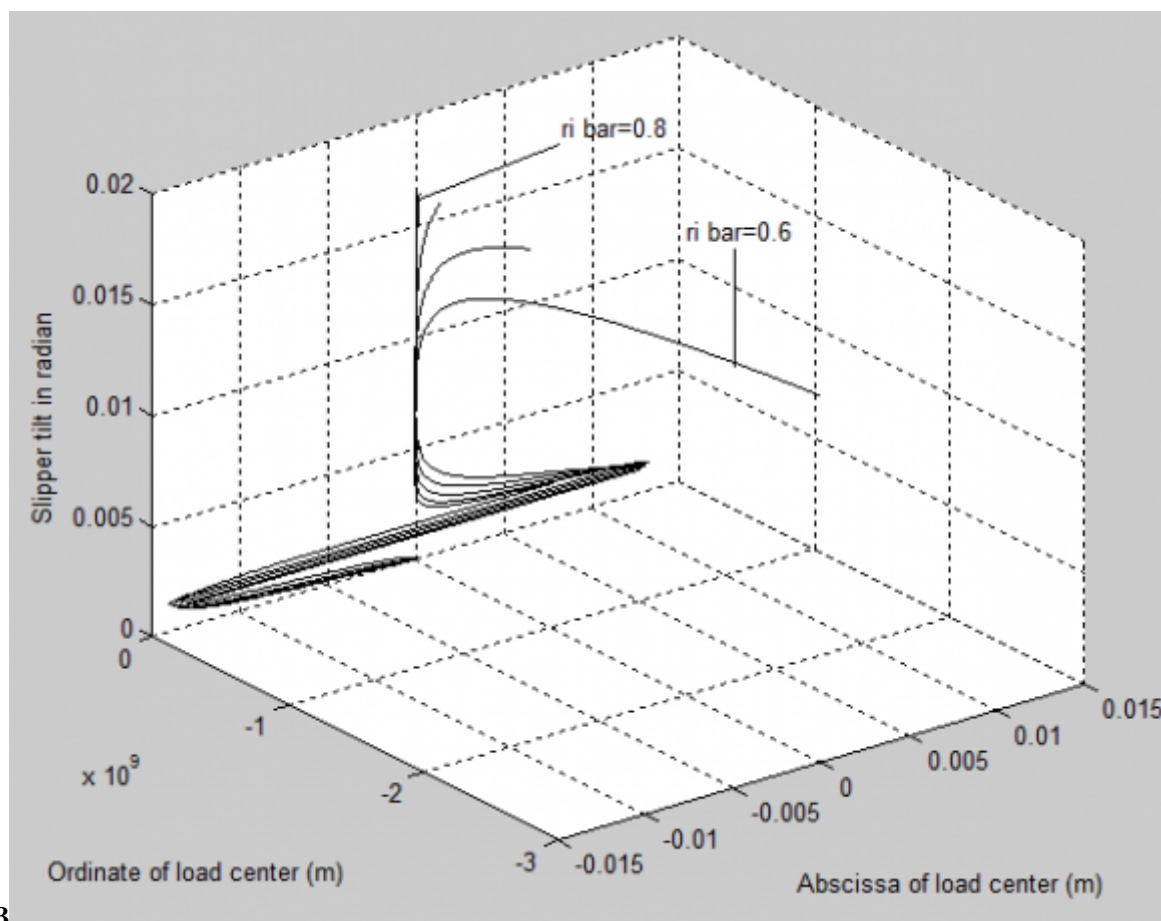
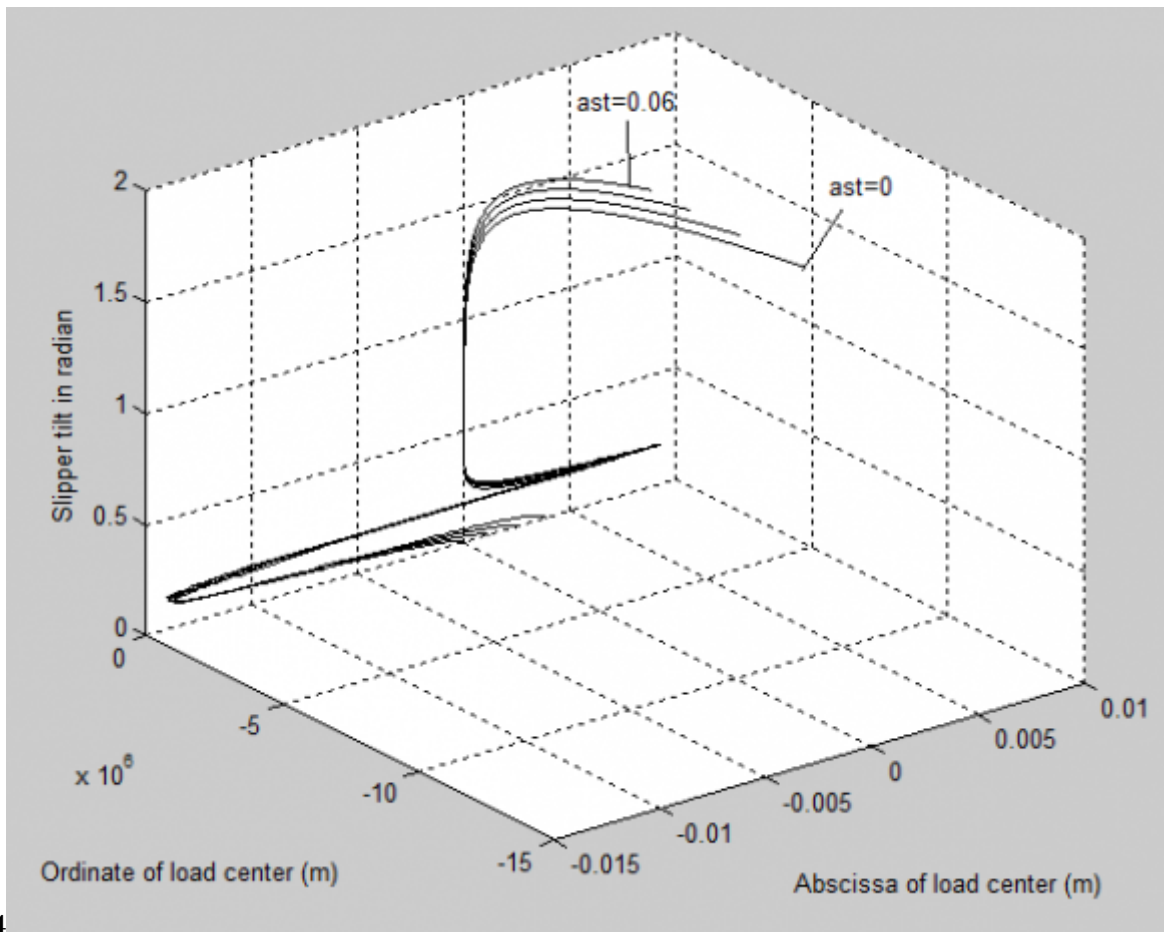


Figure 5: M



3

Figure 6: AFigure 3 :



4

Figure 7: Figure 4 :

114 From figure ??, it can be observed that a more amount of slipper are tries to keep the load centre in the actual
 115 slipper centre and hence stabilize the bearing fast.

116 IV.

117 .1 CONCLUSION

118 From the results of the computation of the position of load centre, it can observed that higher tilt angle gives
 119 irrelevant results. It may happen the fluid film breaks at higher tilt angles. The stability of slipper increases
 120 with increase of slipper pressure (vide figure ??). The stability increases with increase of slipper speed (vide
 121 figure ??). The stability of slipper increases with increase of slipper non flatness angle (vide figure ??). The
 122 stability increases with increase of slipper land size (vide figure ??). For more practicality of this tilt angle more
 123 computation is required.

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3 RESULTS

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