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Computer aided design of axial piston machines having a roller piston bearing

Mohamed Elashmawy[1, 2, *], Abdulaziz Alghonamy[1], Isam Elbadawi[1]

[1]Engineering College, University of Hail, Hail, Saudi Arabia
[2]Engineering Science Department, Faculty of Petroleum and Mining Engineering, Suez University, Suez, Egypt

Email address:
arafat_696@yahoo.com (M. Elashmawy), alghonamy@yahoo.com (A. Alghonamy), isam149@gmail.com (I. Elbadawi)

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Abstract: Swashplate axial piston machines are simple, compact and low price. This simplicity is at the expense of piston transverse forces which limits machine characteristics. The aim of this study is to propose more effective developed design using roller piston bearing. Ball bearing was proposed to reduce transverse forces acting on the piston end of the axial piston machines. The proposed roller bearing design will provide line contact bearing between roller and cam contour compared to point contact of ball bearing arrangement. The roller runs on a flat surface contour formed on the swashplate which is simpler in manufacturing process. The sliding friction between swashplate and slipper is replaced by a rolling friction between roller and runway of cam surface contour. Results show the feasibility of the developed design. The proposed design promises to increase the pressure limitation of the ball bearing arrangement. Parameters such as piston displacement, cam action angle are the same for both roller and ball piston bearing. A comparison analysis was also performed between two alternative cam contours, sinusoidal and linear piston displacement. The selection criterion was based on piston transverse torque. Results show that sinusoidal piston displacement is much better choice than the linear one.

Keywords: Roller Bearing, Ball Bearing, Swashplate, Axial Piston, Transverse Forces, Tribological Contact

1. Introduction

In 1600 Johannes Kepler invents the gear pump as first practical use of hydrostatic power, then the principal of hydraulic press was established (1663), the first practical application was in London (1749-1814) by Josef Bramah. After James Watt invention of steam engine (1736-1819), the water pump was used and operated by the steam engine. The first use of oil as a working fluid in hydraulic machines appears in 1910 by Hele Shaw then Hans Thoma starts developing the axial piston machines in 1930. Harry Vickers (1936) developed a pilot operated pressure valve. Jean Mercier (1950) starts for the first time the biggest hydro pneumatic accumulator [1]. Hydraulic machines that operated by oil as a working fluid is very important in many applications worldwide. The main reason is its compact design and very low weight with high torque ability compared with other non-hydraulic equipment. Nowadays axial piston machines are common type used worldwide in numerous fluid power applications. A swashplate axial piston machine design is compact and low price machine compared with other axial piston machine types. Many parameters affecting the performance of the swashplate axial piston machines were studied.

A theoretical study to predict axial piston machine performance is performed by developing a new coupled multi-domain computer simulation model. The model is composed of different modules, each model analyzing different operating machine problems. The major part of these models is analyzing the lubrication of the three main pump sliding tribological interfaces considering the impact of elasto-hydrodynamic, thermal, and micro-motion effects on fluid film thickness [2]. Other research efforts could be categorized according to the main three tribological contact friction pairs as follows:

1.1. Piston-Cylinder Pair

Piston-cylinder interface geometry was investigated by comparing test bench results, some geometry parameters such as gap width, cylinder length are to be optimized. Results show that contouring and geometry can be optimized for specific working points [3]. A one piston test bench with a PVD-coating piston was built and an experimental test was performed, results show that the coating pistons recorded limited wear for the first hour of operation and very significantly reduces the friction between piston and bushing [4].

1.2. Cylinder Block-Valve Plate Pair

The problem with cylinder block-valve plate pair is not only with the friction or wear, but also leakage from the lubricating gap that dissipates percentage of machine's energy. The results of a simulation study show that the power loss due to lubricating gap between cylinder block and valve plate could be reduced by 50% for the waved surface proposed [5].

Surface treatment with TiN plasma coating of the cylinder barrel of the axial piston pumps was tested to reduce friction and wear of the valve plate for low speed of rotation (100 rpm), results show that the friction between the TiN-coated valve plate and a cylinder barrel is reduced by 22% compared with the uncoated valve plate [6]. Another study of coating the barrel surface with a thin film of CrSiN was performed, results show that the friction coefficient could be reduced to 50% compared with those of the plasma nitrided, the wear parameter is also enhanced for the barrel surface, while no significant wear occurred on the bronze valve plate [7]. The effect of pressure and temperature on barrel film thickness was investigated; the results show that elastic metal to metal forces and damping coefficient are essential in studying the barrel dynamics. Results show that increasing of temperature decreases damping coefficient, giving the barrel more freedom of movement [8].

1.3. Swashplate-Slipper Pair

The swashplate-slipper pair is our focus in this study. This pair is the key element in a swashplate piston machines. It contains a slipper component which is moving face to face over the swashplate surface under a very high contacting force depending on pump operating pressure. Slipper is composed to high friction and wear that lead to a limitation due to swashplate tilt angle. Friction between slipper and swashplate impacts the friction force between piston and inner cylinder surface due to piston transverse forces. Increasing transverse forces will lead to an increase of piston and cylinder friction as well as a surfaces wearing.

Developing of an accurate slipper lubrication film simulation models is quite important because of practical impossibility to observe variations of fluid film thickness measurements. The new dynamic micro-motion simulation model opens a window and allow for more slipper robust and efficient designs [2].

Static and dynamic characteristics of a piston pump slipper with a groove have been studied using CFD with 3-D Navier-Stokes equation, results show that with moving the groove towards the inner pocket of the slipper leakage and slipper force increases. While with positioning the groove near the outer slipper boundary, decreasing the groove width increases the slipper force and consequently decreases the leakage [9]. Later on a CFD is developed and enabled to deal with any number of grooves [10].

Theoretical model was developed for swashplate-slipper friction pair, it includes the pressure distribution and leakage, according to results, the oil film could be very thin due to the slipper tilt angle which causes a partial abrasion on the sealing belt outer part. This abrasion intensifies the slipper tilting and accelerates this abrasion furthermore and consequently the slipper leakage will increase. In order to improve pump efficiency, optimization of the slope on the sealing belt inner edge is recommended [11].

A transient model with a novel numerical coupling scheme, of thermo-elastohydrodynamic lubrication within swashplate-slipper pair, has been developed. Results show the importance of accurate transient deformation squeeze pressure effects on the lubrication model of a thin lubrication film between the swashplate and the slipper surfaces [12].

Some patients were published concerning reducing piston transverse forces by using a ball piston bearing. A US patient was published in 1967 introducing an invention based on a mechanism that uses a piston having a spherical socket outer end within which a metal ball is rotatably mounted. The ball is rolling on a circumferential groove runway formed on the surface of the swashplate of the axial piston motor. No need for springs because this proposal was for motors and the applied pressure will force the pistons to move outwardly. In the case of pumps springs would be necessary [13]. Another US patient was published in 2012 concerning an improvement of the performance of swashplate axial piston machines. The radial piston forces was eliminated by using ball bearing piston end rolling on a circumferential runway cam groove [14]. Recently a US patient was published in 2014. In this invention a ball bearing piston end was also used for a micro compressor. The ball is rolling on a flate surface cam contour in such manner that eliminates the radial piston transverse forces. An outher spring action was used for suction stroke [15].

The present study aimed to develop a second alternative of ball piston bearing arrangement. The idea is to use a roller bearing instead of ball bearing. The proposed roller bearing will provide roller swashplate line contact interface.

2. Problem Statement

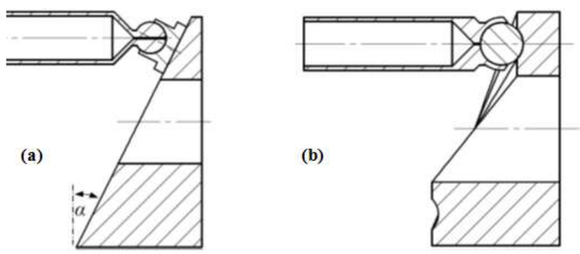

Figure 1. *Swashplate design (a) in comparison with ball piston bearing with cam contour (b) at ODC ($\theta=0^o$).*

Increasing swashplate tilt angle increases piston end transverse force. This causes the pump displacement as well as the pump working pressure to be limited. Fig. 1 shows ball piston bearing design in comparison with the conventional swashplate design. The ball bearing is normally of point tribological contact which limits the pump operating pressure (expected up to 250 bar). The present study proposes an alternative design of a roller bearing arrangement trying to increase pump pressure limitation.

3. Proposed Development Design

In order to overcome pressure limitation of the ball bearing arrangement, a further developed design is proposed. The idea is simply to use a roller bearing instead of ball bearing. The roller bearing will satisfy liner tribological contact between the roller and the cam runway which promises to increase pump pressure limitation.

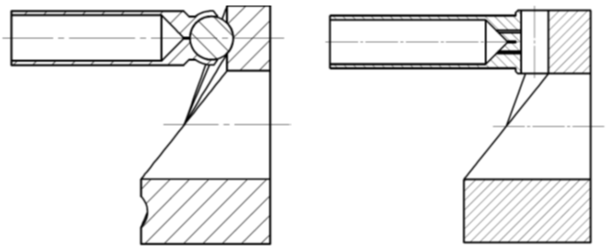

Figure 2. *2-D geometry comparison between ball and roller bearings at ODC ($\theta=0^o$).*

Fig. 2 shows a schematic 2-D geometry comparison between ball and roller bearing designs.

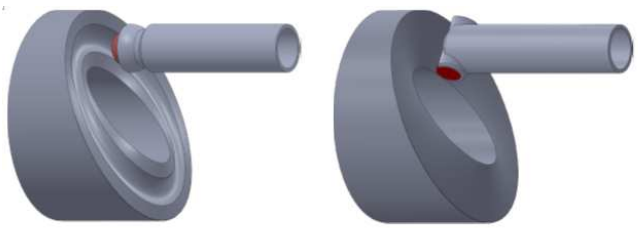

Figure 3. *3-D geometry comparison between ball and roller bearings at 30^o from ODC ($\theta=30^o$) .*

Fig. 3 shows a 3-D geometry comparison between ball and roller bearing designs. The geometry of roller bearing arrangement shows simpler cam contour for manufacturing.

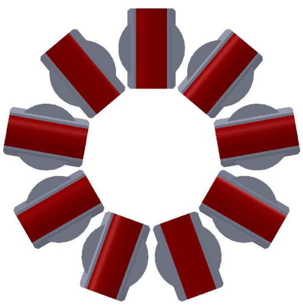

Figure 4. *Trajectory view showing the 9 piston geometry design of the roller piston bearing mechanism.*

Fig. 4 shows the geometry of the nine piston arrangement with the roller attached to each piston end. The geometry analysis shows a feasibility of roller bearing design with the nine axial pistons pump.

4. Results and Discussion

Geometry analysis of the roller piston bearing arrangement shows the same displacement distribution and the same cam action angle of the ball bearing design. The difference appears is the contact area which is point contact for ball bearing and line contact for roller bearing. Results show also no effect of diameter of neither ball nor roller on the cam action angle or on the piston displacement. The main player in this game is the cam contour. Two different contours were studied as follows:

4.1. Sinusoidal Piston Displacement and Its Cam Contour

The conventional swashplate piston displacement is of sinusoidal curve profile. The cam contour is the important parameter controlling the piston displacement. The developed roller piston bearing design is proposed to form the same sinusoidal piston displacement on a cam contour instead of the swashplate.

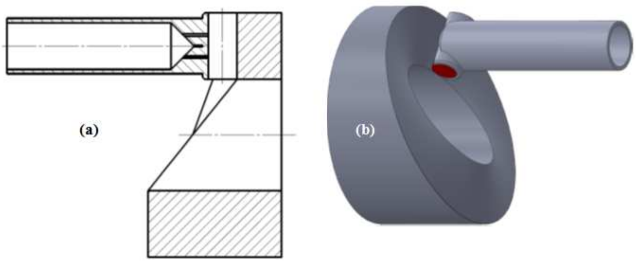

Figure 5. *Sinusoidal displacement cam contour, (a) cross sectional view at ODC ($\theta=0^o$) and (b) 3-D view at 30^o from ODC ($\theta=30^o$).*

Fig. 5 shows the roller piston bearing arrangement with the corresponding sinusoidal piston displacement cam contour.

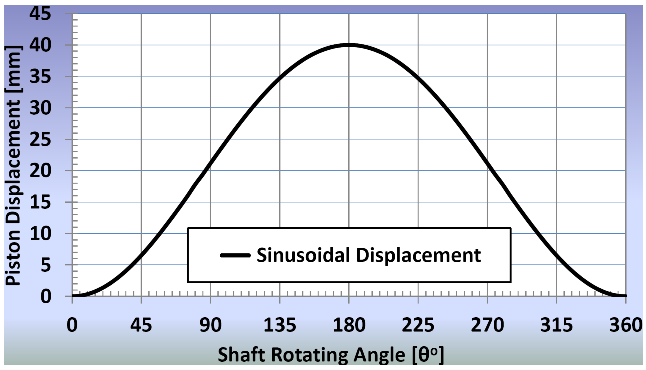

Figure 6. *Sinusoidal piston displacement.*

Fig. 6 shows the sinusoidal piston displacement of the roller piston bearing. The piston displacement of the roller and ball piston bearing are both the same and exactly as swashplate piston displacement.

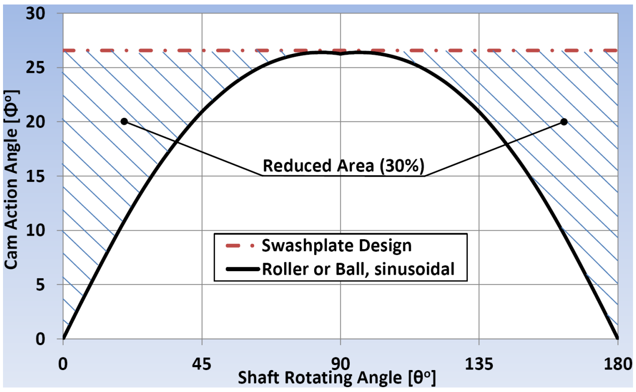

Figure 7. *Cam action angle (ϕ) versus shaft rotating angle (θ) for roller sinusoidal piston displacement and conventional swashplate designs.*

Fig. 7 shows the cam action angle (ϕ) for the roller with sinusoidal piston displacement and for the conventional swashplate. The comparison shows about 30% area reduction of the curve which indicating almost the same percentage of transverse force and power reduction.

$$F_{TC} = P\,A_P \tan\phi \qquad (1)$$

Equation (1) is the relation between transverse (F_{TC}) and cam action angle (ϕ), where P is the pump operating pressure and A_P is the piston cross sectional area.

4.2. Linear Piston Displacement and its Cam Contour

I order to satisfy linear piston displacement the cam contour should follow spiral curve, Fig. 8.

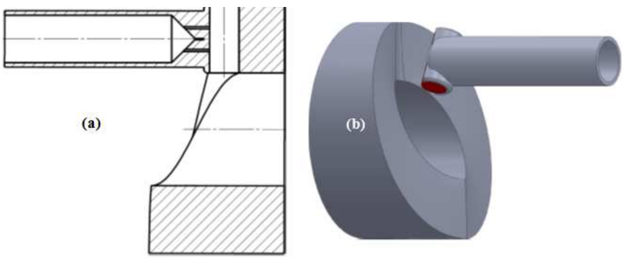

Figure 8. *Linear displacement cam contour, (a) cross sectional view at ODC (θ=0°) and (b) 3-D view at 30° from ODC (θ=30°).*

Fig. 8 shows the roller piston bearing arrangement with its corresponding linear piston displacement cam contour. Fig. 8 (a) shows clearly the spiral curve of cutting the cam contour.

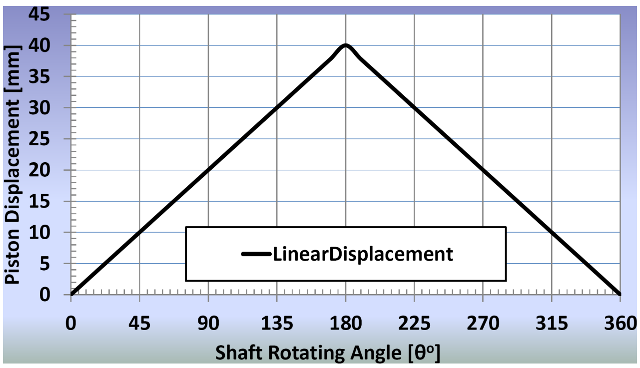

Figure 9. *Linear piston displacement.*

Fig. 9 shows the linear piston displacement of the roller piston bearing. The piston displacement of the roller and ball piston bearing are both the same and unlike the swashplate piston displacement.

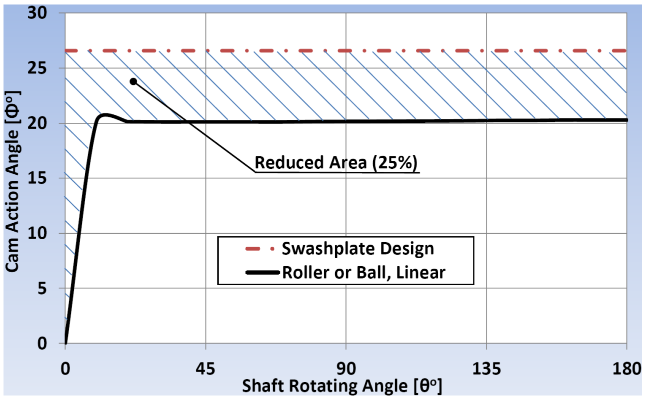

Figure 10. *Cam action angle (ϕ) versus shaft rotating angle (θ) for roller sinusoidal piston displacement and conventional swashplate designs.*

Fig. 10 shows the cam action angle (ϕ) for the roller with linear piston displacement and for the conventional swashplate. The comparison shows about 25% area reduction of the curve which indicating almost the same percentage of transverse force and pump power reduction.

4.3. Comparison between Sinusoidal and Linear Piston Displacements

The important question appears is Sinusoidal or Linear? Which one is better design? According to the area reduction of cam action angle curves of Figs. 7 and 10 sinusoidal piston displacements satisfies more reduced area of the curve, which indicates more transverse force reduction. This parameter alone is not enough to judge the benefits of each design. The most important parameter is the relation between cam action angle and the piston displacement. At the ODC (θ=0°) the piston is at its maximum length outside the cylinder, at such position cam action angle matters as well as at low values of cam action angles.

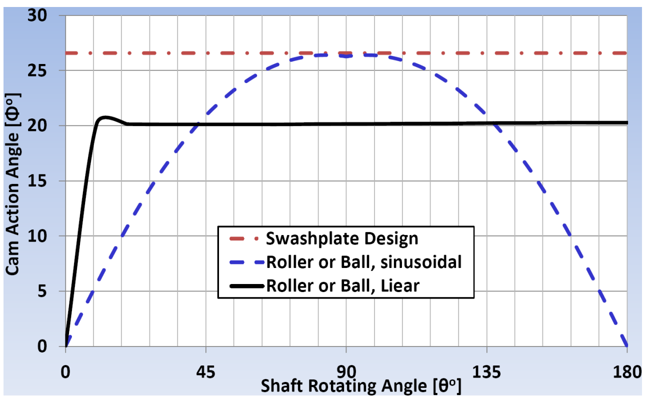

Figure 11. *Cam action angle (ϕ) versus shaft rotating angle (θ) for roller sinusoidal and linear piston displacement and swashplate designs.*

Fig. 11 shows that for linear piston displacement the cam action angle (ϕ) reaches its maximum value of almost 20° at shaft angle ($\theta=10^o$) and keeps constant after that at all delivery stroke up to 180^o. In comparison, sinusoidal piston displacement increases gradually from $\phi=0^o$ and reaches $\phi=5.63^o$ at shaft angle ($\theta=10^o$).

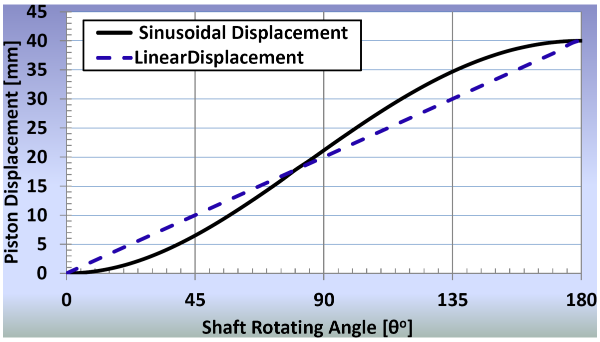

Figure 12. *Roller piston bearing with sinusoidal and linear displacement.*

Fig. 12 shows a comparison between sinusoidal and linear piston displacement. For linear piston displacement, at $\theta=10^o$ the piston moves only 4.46 mm of its all displacement of 40 mm. For sinusoidal piston displacement, at $\theta=10^o$ the piston moves only 0.35 mm.

The base parameter of comparison will be the torque exerted on the piston end due to transverse forces. The torque will be calculating according to piston displacement of Fig. 12 for linear and sinusoidal cases multiplied by the transverse force (1) based on cam action angles of Fig. 11. The results then plotted on Fig. 13.

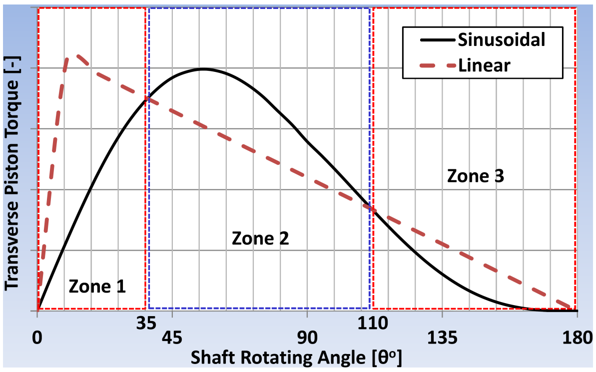

Figure 13. *Transverse piston torque distribution as a function of shaft angle (θ) for roller sinusoidal and linear piston displacement.*

Fig. 13 shows that transverse piston torque comparison curves could be divided into 3 zones: zone1, $\theta=0^o$-35^o, zone-2, $\theta=35^o$-110^o and zone-3, $\theta=110^o$-180^o. The most important zone is zone-1 at which the piston still has significant length outside the cylinder; this situation significantly increases the chances of piston cylinder pair high friction and wear. Based on results obtained from Fig. 13 linear piston displacement choice is not recommended even it will be better than sinusoidal for zone-2.

5. Conclusion

3-D geometry analysis shows a feasibility of the proposed development of roller piston bearing design. the design provides simpler cam contour which will simplify the manufacturing process. The roller piston bearing satisfies higher operating pump pressure compared with ball piston bearing. Parameters such as piston displacement, cam action angle (ϕ) are the same for both roller and ball piston bearings. A comparison analysis was also performed between two alternative cam contours. First cam-contour satisfies sinusoidal piston displacement and the second one satisfies linear piston displacement. The selection criterion was based on piston transverse torque due to each cam-contour. Results show that the cam-contour of sinusoidal piston displacement is much better choice than the linear one.

This study is based on geometry and theoretical analysis of the proposed roller arrangement and is promising to reduce piston transverse forces by almost 30%. Further experimental studies should be performed to predict the performance enhancement of such roller piston bearing arrangement.

Acknowledgments

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