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Discussion on the Reynolds equation for the slipper bearing modeling in axial piston pumps

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ABSTRACT

This work presents a new Reynolds equation for the slipper bearing in axial piston pumps which considers the slipper spin. As an extension of this new Reynolds equation, some typical cases are derived for different velocity boundary conditions. It is found that the present Reynolds equation is identical in form to that in the previous literature, but they are entirely different from each other in the velocity boundary condition. The reasons for the incorrect velocity boundary condition in the previous studies are analyzed, followed by a calculation of the relative velocity error. It is concluded that the Reynolds equation in the present work is more capable of representing the lubrication model for the slipper bearing.

1. Introduction

Axial piston pumps are widely used in many applications because of their advantages such as high working pressure, great power density, convenient flow regulation, and long service life [1]. A typical axial piston pump is shown in Fig. 1. The cylinder block containing nine pistons rotates together with the shaft by a spline mechanism. The piston connects itself with the slipper through a ball-and-socket joint. All the pistons reciprocate within the cylinder bores and the slippers slide on the inclined swash plate during the cylinder block rotation. A reasonable contact between the slipper and swash plate is maintained using the retainer. The displacement chambers in the cylinder block are communicated with the suction or discharge port by the kidney-shaped ports in the valve plate. When the piston passes over the suction side, it is pulled out of the cylinder bore and the low-pressure fluid flows into the cylinder bore. When the piston passes over the discharge side, it is pushed into the cylinder bore and the high-pressure fluid flows out of the cylinder bore. The above reciprocating motion repeats itself for each revolution of the shaft, accomplishing the basic task of converting the low-pressure fluid into the high-pressure fluid.

The pump performance strongly depends on the lubricating interfaces where the fluid film forms to separate heavily loaded relatively movable parts from each other. There are three main lubricating interfaces within an axial piston pump, the cylinder block/valve plate interface, the piston/cylinder block interface, and the slipper/swash plate interface. These lubricating interfaces serve as sealing and bearing functions, which are one of the critical design issues for axial piston pumps. As for the slipper/swash plate interface, it prevents the pressurized fluid in the displacement chamber from leaking into the pump case through the slipper land. Additionally, the fluid film within it fulfills the function of carrying the pressure load exerted by the piston. Both the sealing and bearing functions of the slipper/swash plate interface are determined by the lubrication characteristics of the slipper bearing, which depend on the slipper's motion on the swash plate. In addition to the macro motion governed by the pump kinematics, the slipper also undergoes some micro motions including tilting motion, squeezing motion, and spinning motion due to the additional degrees of freedom on the micro scale. The performance of the slipper bearing is often represented by its leakage flow, load-carrying capacity, and power losses etc. These critical performance parameters are related to the fluid film thickness and pressure across the slipper bearing which require to be calculated by the Reynolds equation for the slipper bearing.

Iboshi and Yamaguchi [2] derived the Reynolds equation for the slipper bearing from the Navier-Stokes equation to evaluate the fluid film parameters of slipper, including tilting angle, maximum tilting angle azimuth and mean gap height. For mathematical expediency, they simplified the pressure distribution across the slipper bearing as a power series of the slipper's tilting angle. Hooke et al. [3–7] investigated the tilting behavior and lubrication characteristics of the slipper bearing with different slipper running surface profiles theoretically and experimentally. They used the Reynolds equation to solve for the pressure distribution under the slipper land and thus the fluid force and moment

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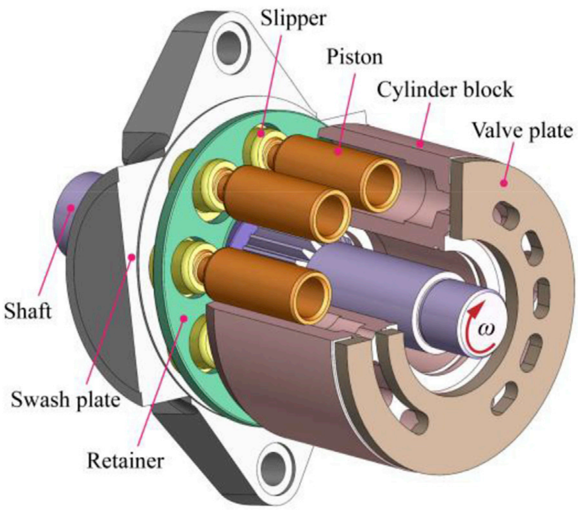

Fig. 1. General configuration of an axial piston pump.

acting on the slipper. The fluid film thickness under the slipper land could be predicted with a reasonable accuracy through solving the force and moment equilibrium equations of the slipper. Wieczorek and Ivantysynova [8,9] performed the pioneering work in the development of the sophisticated simulation tool CASPAR (Calculation of Swash Plate Type Axial Piston Pump/Motor) which was used to simulate the gap flows of the three main lubricating interfaces. As an example, the Reynolds equation for the piston/cylinder block interface was derived from the Navier-Stokes equation and was integrated in the simulation program to determine the gap pressure distribution. However, all the pump components were assumed to be rigid bodies in the simulation program and thus the elastohydrodynamic effects were not considered. Huang [10] extended the CASPAR software and proposed a modified simulation model for the slipper/swash plate interface, which considered the pressure deformation of the slipper and swash plate for the first time. He presented the Reynolds equation for the slipper bearing in cylindrical coordinate systems, in which the velocity on the slipper sliding surface at every grid point was given as the velocity boundary condition. However, we would like to humbly point out that the expression for the velocity on the slipper sliding surface was incorrect due to his misunderstanding of the slipper motion on the swash plate. Specifically, on a macro scale, the slippers do experience curvilinear translation rather than plane rotation on the swash plate. Similarly, Pelosi and Ivantysynova [11] integrated the Reynolds equation into a fluid-structure interaction model inside the simulation tool CASPAR to predict the gap height between the slipper and swash plate. However, they did not provide the velocity boundary condition of the Reynolds equation in detail. Schenk and Ivantysynova [12,13] calculated the power losses generated within the slipper/swash plate interface using the above fully coupled fluid-structure model. Again, the velocity boundary condition of the Reynolds equation was not presented although the same Reynolds equation for the slipper bearing was adopted. Xu et al. [14] conducted numerical and experimental studies on the slipper's partial abrasion phenomenon. In their numerical simulation model for the slipper bearing, they adopted the velocity boundary condition of the Reynolds equation developed by Huang [10]. The slipper spin was not considered in the Reynolds equation for the reason of insufficient theoretical and experimental studies on it. However, in their subsequent work [15] dealing with the effect of case drain pressure on the slipper performance, the slipper spinning speed was included in the Reynolds equation in the modified lubrication model for the slipper bearing. Borghi et al. [16] conducted a numerical study on the dynamic behavior of the slipper bearing, in which the slipper spinning motion was included in the Reynolds equation and its influence on the slipper performance was numerically analyzed. A weakness of this study, however, can be seen in the velocity boundary condition of the Reynolds equation which also failed to describe the slipper motion on the swash plate correctly. Tang et al. [17] developed a mathematical model for the slipper bearing which considered the oil thermal effect to predict the fluid film thickness and temperature within the slipper/swash plate interface under different operating conditions. In their mathematical model they adopted the Reynolds equation for the slipper bearing developed by Borghi et al. [16], therefore their conclusions may remain uncertain. Lin and Hu [18] employed a niche genetic algorithm method to solve their proposed tribo-dynamic model of slipper in which the slipper spinning speed was included in the Reynolds equation. Although the velocity boundary condition of the Reynolds equation was not presented in their paper, it seemed from their Reynolds equation for the slipper bearing that the slipper might also be mistakenly considered to rotate on the swash plate. Unfortunately, Tang et al. [19] used this Reynolds equation for the slipper bearing developed by Lin and Hu [18] in their latest prediction model for the thermoelastohydrodynamic lubrication characteristics of slipper bearing. Bergada et al. [20–22] derived a detailed set of analytical equations for the pressure distribution, leakage flow, and fluid force and moment across the slipper bearing from the Reynolds equation of lubrication. The slipper spinning speed was considered in their analytical model, but further information about how to integrate the slipper spinning speed in the Reynolds equation for the slipper bearing was not provided.

From a thorough review of the literature, it appears that the Reynolds equation for the slipper bearing plays a critical role in predicting the slipper behavior since the performance parameters such as load-carrying capacity, leakage flow, and fluid film thickness require to be calculated by solving it. However, it seems that all the previously mentioned Reynolds equations for the slipper bearing suffer from some serious drawbacks. Firstly, few researchers have derived the Reynolds equation for the slipper bearing in much detail, which leads to an insufficient understanding of it from the physical point of view. Secondly, the velocity boundary condition of the Reynolds equation for the slipper bearing is incorrect in many previous studies. The macro motion of slipper on the swash plate was wrongly considered as plane rotation, but actually the slipper purely translates on the swash plate on a macro scale. Thirdly, many Reynolds equations for the slipper bearing do not take into account the slipper spin for the consideration of insufficient knowledge of the slipper spin. However, the slipper spinning motion does exist within axial piston pumps [23] and has a significant effect on the slipper performance [16]. Therefore, the slipper spin should be included in the Reynolds equations for the slipper bearing.

The goal of this paper is to derive the Reynolds equation for the slipper bearing in cylindrical coordinate systems using the differential analysis of fluid flow. The velocity boundary condition of the Reynolds equation is evaluated based on the slipper kinematics on the swash plate, which accounts for the influence of the slipper spin. Also, the difference of the Reynolds equation for the slipper bearing between this work and previous studies is examined to explain why the new Reynolds equation is more able to represent the slipper bearing lubrication than those in the previous studies.

2. Kinematics of the slipper

To determine the velocity boundary condition of the Reynolds equation for the slipper bearing, an accurate description of the slipper kinematics on the swash plate is required. Fig. 2 shows the schematic of the slipper's macro motion on the swash plate, where two coordinate systems are defined. The (X, Y, Z) system is a global coordinate system which has its origin at the intersection between the centerline of the cylinder block and the special plane passing through all piston head centers. The Y-axis is chosen to be positive in the upward direction and the Z-axis is consistent with the centerline of the cylinder block. The (X_S, Y_S, Z_S) system is defined with respect to the swash plate and shares the