INTERNATIONAL JOURNAL OF FLUID POWER, 2017
VOL. 18, NO. 3, 181–187
https://doi.org/10.1080/14399776.2017.1337440

Taylor & Francis
Taylor & Francis Group

Experimental investigation of the effect of grooves cut over the piston surface on the volumetric efficiency of a radial hydraulic piston pump

Mohammad Javad Saheban Alahadi[a,b], Alireza Shirneshan[a,b] and Mojtaba Kolahdoozan[a,b]

[a]Department of Mechanical Engineering, Najafabad Branch, Islamic Azad University, Najafabad, Iran; [b]Modern Manufacturing Technologies Research Center, Najafabad Branch, Islamic Azad University, Najafabad, Iran

ABSTRACT

A radial piston pump often used in high pressure applications. The geometry of the pistons used affects the mechanical and volumetric efficiency of the pump and its long term performance. In the present study, the effect of grooves cut on the piston surface, the number of the grooves and their location over the piston surface on the volumetric efficiency of the hydraulic radial piston pumps was investigated. In order to perform the research tests, the three-piston fixed displacement pump with seven configurations was firstly designed and fabricated. The output flow rate of the pump was determined at pressure outlets of 350 and 500 bar and then the volumetric efficiency was calculated as the mean values of measured data. The results showed that the grooves cut on the piston surface produces a higher piston–cylinder overall leakage and then tend to reduce the volumetric efficiency. Also as the number of grooves being cut on piston surface increases, an increase in leakage will happen. In addition, by increasing the outlet pressure from 350 to 500 bar, the volumetric efficiency decreased significantly for the pumps with higher number of grooves cut. Among the configurations studied, pump with one groove at the inner edge at second groove position is the one bringing the best performance because of the minimum reduction in the volumetric efficiency between grooved piston pumps and producing higher restoring torque at all pump configurations according to previous literature.

ARTICLE HISTORY
Received 3 November 2016
Accepted 30 May 2017

KEYWORDS
Radial hydraulic piston pump; volumetric efficiency; groove; performance

1. Introduction

A radial piston pump is a form of hydraulic pump. The working pistons extend in a radial direction symmetrically around the drive shaft, in contrast to the axial piston pump. A piston pump often used in high pressure applications. The grooves along the piston surface depending on the manufacturer may or may not be used. Grooves are meant to make stable the piston but the amount of grooves needed for a specific application and where should they be located along the piston length is at the moment very much linked with the designers' expertise (Kumar and Bergada 2013). Piston dynamics has a fundamental role in two critical processes related to fluid flow in the pumps. The first is the flow leakage which may cause considerable reduction in the pump efficiency. The second process is the viscous friction associated, with the lubricant film in the radial clearance, eventually friction metal to metal might appear. Therefore, the geometry of the pistons used affects the mechanical and volumetric efficiency of the pump and its long term performance (Kumar and Bergada 2013). The present paper explains the effect of the grooves being cut on piston surface and study the requirement of their use.

Sweeney (1951), experimentally conducted the first studies about the groove balancing effect and examined the pressure distribution in the piston–cylinder clearance. This author established a relationship between the leakage flow and the geometry of the clearance. The pressure distribution in the clearance piston–cylinder was experimentally inspected by Sadashivappa *et al.* (2001). They concluded that the eccentricity of the piston affected the performance of the piston by influencing the frictional and leakage aspects. Some researchers have been pursued to find the flow and pressure distribution theoretically taking into account the effect of the grooves, Milani (2001) applied the continuity equation to link the Poiseulle equation in each land, and considered a constant pressure in each groove. The same method was used by Borghi *et al.* (1998, Borghi 2001), although they applied it to a single groove tapered spool. In both cases relative movement between piston and cylinder was not considered, yet eccentricity was taken into account. An analytical formulation for the pressure distribution and forces in narrow clearances was established by Black-burn *et al.* (1960) and Merrit (1967). They assumed that the pressure distribution in narrow gaps was not affected by peripheral flow rates; they made an

CONTACT Alireza Shirneshan arshirneshan@yahoo.com

easy estimation of the sticking phenomena effects. Such work although when grooves were not considered was undertaken by Ivantysynova (Ivantysynova and Huang 2002, Ivantysynova and Lasaar 2004) which found the dynamic pressure distribution and leakage between piston and barrel considering piston tilt, piston displacement and heat transfer. Fang and Shirakashi (1995) carried out a numerical of analysis in order to obtain the metal contact force between the piston and cylinder and he concluded that exist mixed lubrication between the piston and cylinder, being independent on pump operation conditions such as supply pressure or the rotation speed. Prata *et al.* (2000) performed a numerical simulation for a piston without grooves, by using the Finite Volume Method, considering both the axial and the radial piston motion and explained the effect of the operating conditions on the stability of the piston. On the other hand, the study of the machine element surfaces with grooves and narrow gaps is more generic and has a mature foundation in literature. Berger *et al.* (1996) investigated the effect of the surface roughness and grooves on permeable wet clutches. They concluded that friction and groove width significantly influence the engagement characteristics as torque, pressure and film thickness, on the other hand groove depth did not have a significant effect on engagement characteristics. Kumar *et al.* (2009) investigated the effect of the groove in slipper-swash plate clearance. They demonstrated that the presence of the groove stabilised the pressure distribution in the clearance slipper swash plate. The grooves position was having a considerable effect on the force acting over the slipper. An interesting amount of work has been undertaken until now, considering the geometric shape of the groves, friction parameters and its effect on the operating conditions in order to improve the piston performance, but despite all the work taken on by previous authors, there has never been studied, the effect of the number of grooves cut and specially the effect of modifying their position on the leakage flow and volumetric flow in radial piston pumps. So in this paper, the leakage flow and volumetric flow of the pump by changing the number of grooves and their position investigated will be discussed.

2. Methodology and experimental tests

One of the main parameter in design and manufacturing of the piston pumps is volumetric efficiency. Volumetric efficiency is the most significant factor in overall efficiency of the pumps which affects precision and manufacturing tolerances as well. Pump volumetric efficiency is defined as the ratio of actual output flow rate to theoretical flow rate of the pump:

$$\eta_v = \frac{Q_a}{D_P \times n_p} \tag{1}$$

where η_v: Volumetric efficiency of the pump, Q_a: Actual output flow rate (cm^3/min), D_P: The displacement or amount of fluid pumped per revolution of the pump (cm^3/rev), n_p: The revolution of per minute.

The overall efficiency of is defined as:

$$\eta_O = \eta_v \times \eta_M \tag{2}$$

where $\eta_{Overall}$: Overall efficiency, η_M: Mechanical efficiency.

2.1. Fabrication the pump and the pistons

In the present study to evaluate effects of the grooves along the piston surface of a hydraulic radial piston pump on volumetric efficiency, an outside impinged fixed displacement pump was firstly designed and fabricated based on the specification of the similar pump which had been made in Atos Company (Italy). The pump including three pistons with a diameter of 16 mm, 5 litres per minute theoretical flow rate and the maximum working pressure of 500 bar and the rotation speed of 1440 RPM. The pump is shown in Figure 1.

In order to perform the research tests, the pistons with configurations which are given by the following Figures 2–4 was designed and fabricated. The number

Figure 1. The designed and fabricated hydraulic piston pump.

Figure 2. The real manufactured piston.

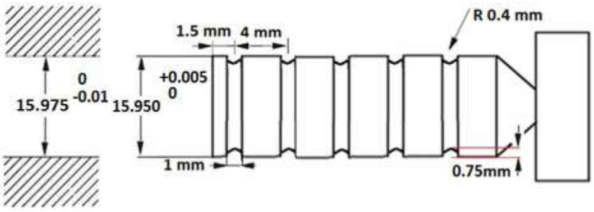

Figure 3. Dimensional schematic diagram of the position of the grooves along the piston surface.

of grooves and their position on the piston was chosen according to Kumar and Bergada (2013) research.

Figures 3 and 4 represent pictures of a dimensional schematic diagram of the position of the grooves along the piston surface and the initial configuration of the piston considered in this paper, respectively.

2.2. Determination the output flow rate and volumetric efficiency of the pump

In this study, the experimental test set-up was prepared according to the Figures 5 and 6. At first the pistons in the hydraulic pump was assembled and output flow rate of the pump was determined at pressure outlets of 350 and 500 bar with 6 measurement repeatability for different configurations of pistons. Then the volumetric efficiency was calculated as the mean values of measured data.

According to the Figures 5 and 6, a three-phase electromotor (1) with 7.5 hp output power and 1440 rpm was coupled with the pump (2). There is a pressure relief valve (3) in the testing set up to supply the working pressure of 350 and 500 bar. Moreover the system has two 2/2 directional valves (4) and (5) (solenoid operated) which are normal open and normal close, respectively. The set-up was also equipped with a relay system to switch the position of the directional valves to close and open position, respectively. Besides, it causes to start the timer to operate in a specified time. In addition, there is a low pressure safety relief valve (6) to control the pressure in the system to avoid failure of solenoid valves if the relay system does not work correctly and both of the valves stay in close position. When the valve (5) is opened for the specified time the hydraulic oil discharges to a 1000 ml graduated cylinder (7) and then the actual flow rate of the pump was measured by this method. After the mentioned time, the position of valve (4) and (5) changes to normal positions and then hydraulic fluid discharges to the tank.

2.3. Calculation of the theoretical output flow rate of the pump

The theoretical output flow rate of the pump was calculated according following equation:

$$Q_t = V \times Y \times N \tag{3}$$

where $V(\text{cm}^3)$: The volume displaced by a piston in the pump in a revolution, Y: Number of pistons of the pump, N(RPM): The rotation speed of the pump, $Q_t\,(\text{cm}^3/\text{min})$: theoretical output flow rate of the pump.

According to the pump specification the theoretical output flow rate of the pump was calculated as follows:

$$Q_t = \frac{d_p^2 \pi}{4} \times L \times Y \times N \Rightarrow 2 \times 0.58 \times 3 \times 1440 \approx 5011 \frac{\text{cm}^3}{\text{min}}$$

3. Results and discussion

3.1. Comparisons between the volumetric efficiency of the pump for non-groove and grooved piston pumps at the pressure of 350 and 500 bar

The mean values of volumetric efficiency of grooved and non-grooved piston pumps at outlet pressure of 350 and 500 bar have been indicated in Figures 7 and 8, respectively.

3.2. Effect of the existence of groove along the piston surface of the pump on volumetric efficiency at the outlet pressure of 350 and 500 bar

The results showed that as the groove being cut on the piston surface of the pumps, the volumetric efficiency of the hydraulic pump decreases from 94.8% (for non-grooved piston pump) to 78.2% (for a hydraulic piston pump with 5 grooves) at pressure outlet of 350 bar. Also the use of grooves cut on the piston surface brings a reduction in the volumetric efficiency (93.5–70.5%) at pressure outlet of 500 bar. This is due to an increase in leakage and consequently a reduction in the volumetric efficiency which causes by the grooves being cut on the piston surface of the hydraulic pumps. So that the volumetric efficiency compared to non-grooved piston pumps decreases at least (about 2%) for pumps with one groove cut on the piston. Also the maximum reduction in volumetric efficiency (about 17%) happens for pumps with five grooves along the piston surface compared to non-grooved piston configuration pumps at 350 bar. Besides, the volumetric efficiency decreases (about 3%), for the pumps with one groove cut on the piston and the

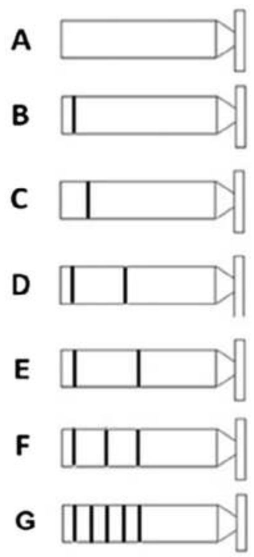

Figure 4. Seven different configurations of pistons: (A): Piston with no groove, (B): Piston with one inner groove, (C): Piston with one groove at second groove position from the inner edge, (D): Piston with two grooves at first and fourth groove position from the inner edge, (E): Piston with two grooves at first and fifth groove position from the inner edge, (F): Piston with three grooves at first, third and fifth groove position from the inner edge, (G): Piston with five grooves.

maximum reduction in the volumetric efficiency (23%) was obtained for the pumps with five grooves on the piston in comparison with the non-grooved piston pumps at the pressure of 500 bar. Although as established in previous literature, the use of grooves cut on the piston surface will tent to prevent cavitation from appearing and brings stability to the piston, since it increases piston stiffness (Kumar and Bergada 2013).

Based on the results, there is just 1.29% reduction in volumetric efficiency for non-grooved piston pumps by increasing the outlet pressure from 350 to 500 bar while there is a significant difference (7.72%) for pumps with five grooved pistons at higher outlet pressure condition. Thus, it seems that the higher outlet pressure has more effects on leakage and reduction in the volumetric efficiency when the number of grooves increases.

3.3. The effects of number of grooves over the piston surface on the volumetric efficiency at the outlet pressure of 350 and 500 bar

In recent work, to investigate the effects of number of grooves (1 to 5) over the piston surface on the volumetric efficiency, different configurations of piston have been modified. It is clear from the results (Figure 7) by increasing the number of grooves the piston–cylinder overall leakage tends to be higher and consequently the volumetric efficiency is decreased at the outlet pressure of 350 bar. Similar results were found at the outlet pressure of 500 bar (Figure 8).

Looking at the behaviour of hydraulic distributors, one of the aspects governing their stationary and dynamic behaviour is the locking force, with a corresponding pressure drop between boundaries. From a theoretical point of view, the overall leakage flow is affected by the locking phenomena which are in the presence of clearances with parallel cylinder walls (Figure 9). Moreover, the positive effect of hydraulic locking force decrease owing to the presence of a circumferential groove is well known.

As already established in previous literature (Sweeney 1951, Manhajm and Sweeney 1955, Milani 2001, Kumar and Bergada 2013), the tendency towards locking force reduction with an increasing number of grooves appears to be confirmed. An increase in groove number causes a strong decrease in locking force and a simultaneous and obvious increase in the total leakage flow. In addition, it can be explained by the Equation (4). If the flow can be considered as laminar and stationary, the axial flow rate across the clearance between pump piston and cylinder must be assumed to be linearly dependent on pressure drop:

$$Q = K \times \Delta p \tag{4}$$

where Q: Axial flow rate across the clearance between pump piston and cylinder, Δp: Pressure drop.

Figure 5. The test set up.

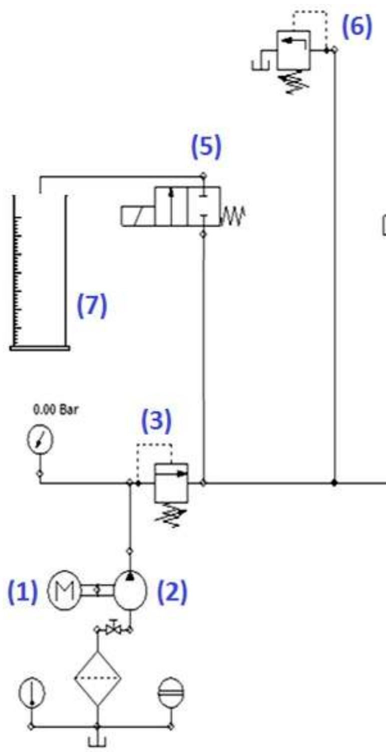

Figure 6. Hydraulic circuit of test set-up.

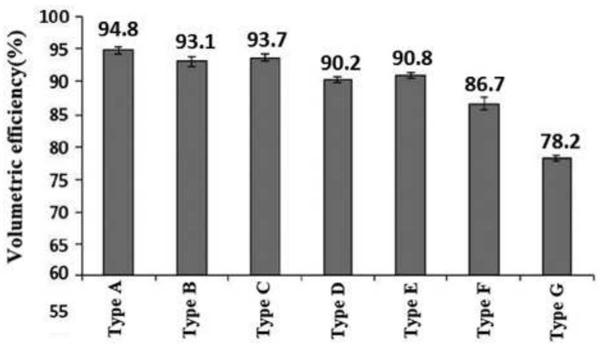

Figure 7. Volumetric efficiency of various types of pumps (grooved non-grooved piston configuration pumps) at 350 bar outlet pressure.

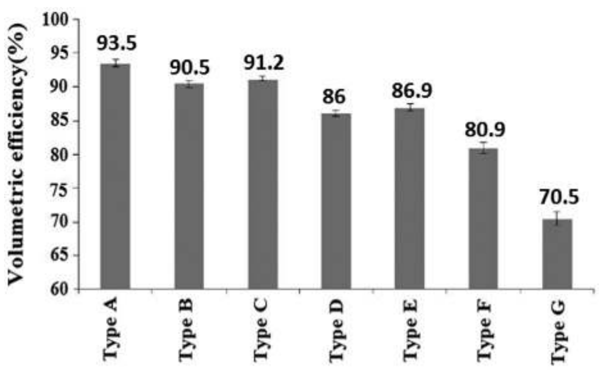

Figure 8. Volumetric efficiency of various types of pumps (grooved non-grooved piston configuration pumps) at 500 bar outlet pressure.

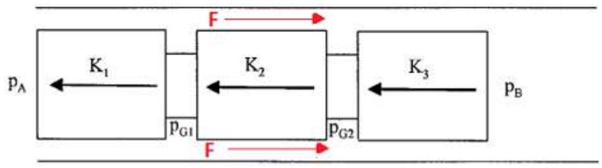

Figure 9. Schematic of three-grooved piston in the pump cylinder.

Note: According to the Figure, *K*: Clearance hydraulic conductance, P_A: Lower pressure side (tank side), P_B: Higher pressure side, P_{G1} and P_{G2}: the pressure in the groove, *F*: Looking force.

So when groove number increases, although the pressure drop decreases but the clearance hydraulic conductance which depends on geometry is higher because of strong locking force reduction and it causes an increase in the total leakage flow (Sweeney 1951, Manhajm and Sweeney 1955, Milani 2001).

Although another important consideration to be noticed is that the volumetric efficiency of the pumps with only one groove cut on piston surface, either at position 1 or 2, has not changed significantly compared to the non-grooved piston pumps at the outlet pressure of 350 bar. In addition, by increasing the number of grooves from 3 to 5, there is a considerable decrease in the volumetric efficiency of the pump at 500 bar outlet pressure.

3.4. The effects of groove position over the piston surface on the volumetric efficiency at the outlet pressure of 350 and 500 bar

In order to study the effect of groove positioning, as shown in Figure 3 seven different types of pistons were used to evaluate the piston performance. In the present study, at 350 bar outlet pressure, there is no significant difference between the pumps having grooves at positions 1 (93.1%) and 2 (93.7%) and non-grooved piston pumps (94.8%) in the volumetric efficiency. However, the volumetric efficiency of the pumps with groove at position 2 is slightly more than that of pumps with groove at position 1. Similar trend is observed for pumps with two grooves at positions 1 and 5 (90.8%) and two grooved piston pumps at positions 1 and 4 (90.2%). This is due to the fact that when grooves are placed near inner edge, the pressure will be higher and the piston–cylinder overall leakage tends to be increased. It is because of that when the piston is connected to the higher pressure side as the total force acting on the piston surface is much higher than the force when piston is connected to the tank side and then the overall leakage tends to increase (Kumar and Bergada 2013). The volumetric efficiency of the grooved piston pump at positions 1, 3 and 5 (86.7%) is lower than that of the pumps with grooves cut on the piston surface at positions 1 and 5 (90.8%). The main reason behind this reduction is the existence of higher number of grooves cut which belongs to piston configuration F.

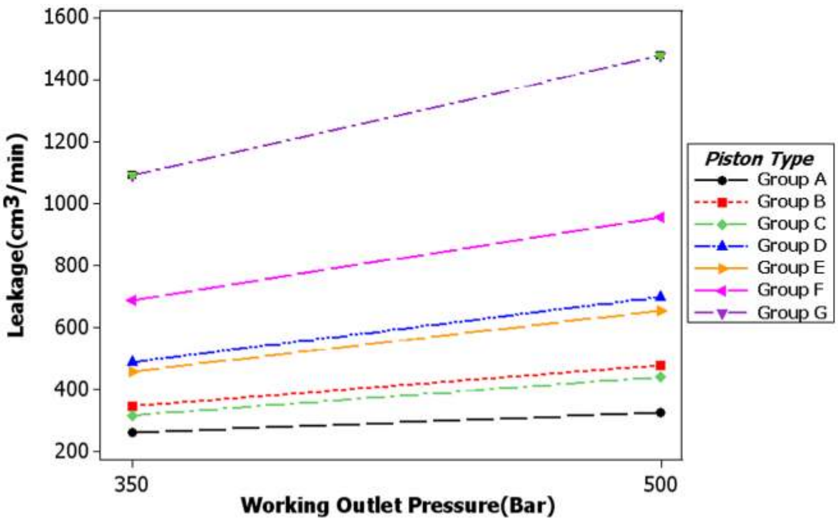

Figure 10. Pump leakage for different configurations of the pistons in two working outlet pressures.

According to the results, the volumetric efficiency of the pumps with one groove along the piston surface at positions 1 and 2 is 90.5 and 91.2%, respectively, at 500 bar outlet pressure. Also 86% volumetric efficiency was calculated for pump with two grooves at positions 1 and 4 and 86.9% volumetric efficiency belongs to a pump with two grooves at positions 1 and 5. Same as 350 bar pressure, the results concluded that when grooves are placed near inner edge the pressure will be higher and causes to increase the piston–cylinder leakage and reduction in volumetric efficiency of the pump. Similarly the volumetric efficiency of the grooved piston pump at positions 1, 3 and 5 (80.9%) is lower than that of the pumps with grooves cut on the piston surface at positions 1 and 5 (86.9%). On the other hand based on previous literature to avoid cavitation, it is important to consider the inclusion of grooves at the piston stroke length and near to the piston pressure side (Kumar and Bergada 2013). In addition, the grooves located at the central part of the piston, appear not to be useful regarding the pressure stabilisation, torque, force or even cavitation point of view (Kumar and Bergada 2013). The best performance among the configurations studied belongs to the pump with one groove at the inner edge at second groove position because of the minimum reduction in the volumetric efficiency between grooved piston pumps and also producing higher restoring torque at all pump configurations according to previous literature as the reciprocating movement of the pistons in radial pumps is similar to axial pumps (Kumar and Bergada 2013). Although it can be seen the overall leakage of this configuration is slightly higher than that of the non-groove configuration.

3.5. An analytical estimation of the leakage volume flow in relation to the number of grooves and the pressure level

The Figure 10 shows pump leakage for different configurations of pistons in two working outlet pressures (350 and 500 bar). Based on the figure, the increase in outlet pressure leads to an increase in leakage of the pump for all of the piston types. It can be found that the changing of outlet pressure from 350 to 500 bar is more effective for piston types F and G which have more grooves cut on the piston. According to the result, it is clear that by increasing the outlet pressure, piston–cylinder overall leakage for the pumps with higher number of grooves cut will be more affected. By comparison of the piston types B and C and piston types D and E, it can be seen that the groove position has not a significant effect on pump leakage during the increasing outlet pressure. Moreover, the figure shows that by increasing the number of grooves from 3 to 5 (piston types F to piston type G) the pump leakage was influenced more than that of when the number of grooves changes from 2 to 3 (piston types D and E to (piston type F) or 1 to 2 (piston types B and C to piston types D and E).

4. Conclusions

- The use of grooves cut on the piston surface produces a higher piston–cylinder overall leakage and then tend to reduce the volumetric efficiency of the pump.
- As the number of grooves being cut on piston surface increases, an increase in leakage will happen and the volumetric efficiency of the pump decreases.
- By increasing the outlet pressure from 350 to 500 bar, piston–cylinder overall leakage for the pumps with higher number of grooves cut will be more affected and the volumetric efficiency decreased significantly.
- It can be seen that the groove position has not a significant effect on pump leakage during the increasing outlet pressure
- When grooves are placed near inner edge the pressure will be higher and the piston–cylinder overall leakage tends to be increased.
- Among the configurations studied, the pump with one groove at the inner edge at second groove position is the one bringing the best performance although overall leakage is slightly higher that the non-groove

configuration because of the minimum reduction in the volumetric efficiency between grooved piston pumps and producing higher restoring torque at all pump configurations according to previous literature.

Disclosure statement

No potential conflict of interest was reported by the authors.

Notes on contributors

Alireza Shirneshan, the corresponding author of the paper, is a Researcher and University Faculty Member. He received his PhD from Science and Research Branch, Islamic Azad University in 2012. He is an affiliate assistant professor of Mechanical Engineering at Najafabad Branch, Islamic Azad University. His research interests are in the areas of Hydraulic, Energy, Biofuels and Internal Combustion Engines.

Mojtaba Kolahdoozan, the corresponding author of the paper, is a researcher and University Faculty Member. He received his PhD from Science and Research Branch, Islamic Azad University in 2012. He is an affiliate assistant professor of Mechanical Engineering at Najafabad Branch, Islamic Azad University. His research interests are in the areas of Hydraulic, Machining, MEMS.

Mohammad Javad Saheban Alahadi, the author of the paper is a Mechanical Engineer. He received his MSc from Najafabad Branch, Islamic Azad University in 2015. His research interests are in the areas of Hydraulic and Manufacturing.

References

Berger, E.J., Sadeghi, F., and Krousgrill, C.M., 1996. Finite element modeling of engagement of rough and grooved wet clutches. *Journal of Tribology*, 118, 137–146.

Blackburn, J., Reethof, G., and Shearer, J. 1960. *Fluid power control*, NewYork: MITPress – Johnwiley.

Borghi, M., 2001. Hydraulic locking-in spool-type valves: tapered clearances analysis. *Proceedings of the Institution of Mechanical Engineers*, 215 Part I, 157–168.

Borghi, M., Cantore, G., Milani, M., and Paoluzzi, P. 1998. Numerical analysis of the lateral forces acting on spools of hydraulic components. *FPST5, Fluid Power Systems and Technology, ASME*.

Fang, Y. and Shirakashi, M., 1995. Mixed lubrication characteristics between the piston and cylinder in hydraulic piston pump-motor. *Journal of Tribology*, 117, 80–85.

Ivantysynova, M. and Huang, C., 2002. Investigation of the gap flow in displacement machines considering elastohydrodynamic effect. *Proceedings of the JFPS International Symposium on Fluid Power*, 2002, 219–229.

Ivantysynova, M. and Lasaar, R., 2004. An investigation into micro- and macrogeometric design of piston/cylinder assembly of swash plate machines. *International Journal of Fluid Power*, 5, 23–36.

Kumar, S. and Bergada, J.M., 2013. The effect of piston grooves performance in an axial piston pumps via CFD analysis. *International Journal of Mechanical Sciences*, 66, 168–179.

Kumar, S., Bergada, J.M., and Watton, J., 2009. Axial piston pump grooved slipper analysis by CFD simulation of three-dimensional NVS equation in cylindrical coordinates. *Computers & Fluids*, 38, 648–663.

Manhajm, J. and Sweeney, D.C., 1955. An investigation of hydraulic lock. *Proceedings of the Institution of Mechanical Engineers*, 169, 865–879.

Merrit, H., 1967. *Hydraulic control systems*. New York: John Wiley.

Milani, M., 2001. Designing hydraulic locking balancing grooves. *Proceedings of the Institution of Mechanical Engineers, Part I: Journal of Systems and Control Engineering*, 215, 453–465.

Prata, A., Fernandes, J., and Fagotti, F., 2000. Dynamic Analysis of Piston Secondary Motion for Small Reciprocating Compressors. *Journal of Tribology*, 122, 752–760.

Sadashivappa, K., Singaperumal, M., and Narayanasamy, K., 2001. Piston eccentricity and friction force measurement in a hydraulic cylinder in dynamic conditions considering the form deviations on a piston. *Mechatronics*, 11, 251–266.

Sweeney, D., 1951. Preliminary investigations of hydroliclock. *Engineering*, 172, 513–516.