B. TECH. PROJECT REPORT

on

"Analyzing the impact of increasing leakage fault in Single and Multiple Cylinders and Pressure Fluctuation Reduction via Piston Rearrangement in an Axial Piston Pump"

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“Analyzing the impact of increasing leakage fault in Single and Multiple Cylinders and Pressure Fluctuation Reduction via Piston Rearrangement in an Axial Piston Pump”

A PROJECT REPORT

Submitted in partial fulfillment of the requirements for the award of the degree of

BACHELOR OF TECHNOLOGY IN MECHANICAL ENGINEERING

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CANDIDATE'S DECLARATION

I hereby declare that the project entitled " Impact of Increasing Leakage Fault in single and Multiple Cylinders and Pressure Fluctuation Reduction via Piston Rearrangement in an Axial Piston Pump" submitted in partial fulfillment for the award of the degree of Bachelor of Technology in mechanical Engineering completed under the supervision of **Dr. Pavan Kumar Kankar, Professor, Mechanical Engineering, IIT Indore and Dr. B.K Chourasia, Professor, Mechanical engineering, JEC, Jabalpur** is an authentic work. Further, I declare that I have not submitted this work for the award of any degree elsewhere.

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CERTIFICATE BY BTP Guide(s)

It is certified that the above statement made by the students is correct to the best of my/our knowledge.

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PREFACE

This report on **“Analyzing the impact of increasing leakage fault in single and multiple cylinders and pressure fluctuation Reduction via Piston Rearrangement in an Axial Piston Pump**” was prepared under the supervision of **Dr. Pavan Kumar Kankar, Professor, Mechanical Engineering, IIT Indore and Dr. B.K Chourasia, Professor, Mechanical engineering, JEC, Jabalpur**. This project is all about the analysis of various types of failure occurring in an Axial Piston Pump despite its wide applications in Industrial, Automobile and Manufacturing fields. It consists of the complete analysis of the cause and prevention of failure in various components of the piston pump. It also comprises a simulation-based study on the performance of the piston pump by analyzing the impact of increasing severity of leakage in single and multiple piston cylinders and pressure fluctuation via piston rearrangement.

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ACKNOWLEDGEMENTS

To acknowledge one's debt is not an easy task, as it may not be possible to do it merely by words. But it would be an injustice if I did not acknowledge the debt of my supervisor, **Dr. Pavan Kumar Kankar**, without whose help this project could not have been done at all. It is a matter of great honor for me to express my gratitude to my supervisor, **Dr. Pavan Kumar Kankar, Professor, Department of Mechanical Engineering, IIT Indore and Dr. B.K Chourasia, Professor, Mechanical Engineering, JEC Jabalpur**, for their creative guidance, constant encouragement, parental affection, and untiring help throughout the course of this work. Their guidance and tremendous support helped me in all the time of research and writing of this thesis. I would like to express my deepest gratitude to **Dr. Ankur Miglani, Associate Professor, Mechanical Engineering, IIT Indore and Mr. Neeraj Sonkar**, **Research Scholar, IIT Indore**, for their immense help, guidance, feedback sessions, patience throughout the work, and moral support. I am sincerely thankful for the collective efforts and support that have played a pivotal role in the successful execution of this project.

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ABSTRACT

The transition towards digitalization and intelligent networking in modern industries emphasizes the necessity for advanced diagnostic technologies to ensure the safe functioning of machinery and mechanical components, particularly hydraulic piston pumps. These pumps play an important role in hydraulic transmission systems as they can be used diversely in power generation and transmission. They work upon the principle of Pascal's Law or law of equal pressure. Oil is preferred over water, as oil is noncorrosive, denser and better performing working fluid than water.

This thesis is all about the complex working of various components of the axial piston pumps (APPs), especially the swash plate type. It consists of the detailed discussion on the classification of the hydraulic pumps, various types of failure occurring in various components of the pump along with the preventive measures taken against them.

Key challenges and failures occurring in an APP are discussed in detail. Some of the issues include loose slippers, slipper wear, shaft failure, valve plate failure, cylinder block wear and many others. Also, the factors causing these issues are addressed in this thesis such as lack of lubrication, overspeed, overheating, particle and chemical contamination.

In summary, this abstract highlight the complex role of Axial Piston Pumps in modern day industries, emphasizing the need to have a comprehensive study of the failure occurring in various components of the piston pump.

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CHAPTER 1

INTRODUCTION

1.1 Hydraulic Pumps:

Hydraulic pumps are used in various fields such as space navigation, national defense, airplanes, construction machines, ships, wind turbines, and various high-tech fields [2,5]. Their compact structure, high-pressure rating, and convenient flow regulation make them indispensable [2,5].

Hydraulics means the use of fluid to generate power to run any machinery [15,16]. It is based on Pascal's Law. Pascal's law or law of equal pressure states that pressure exerted or applied to any part of the enclosed fluid is equally transmitted to the other part of the same fluid without any change in magnitude. It does not depend on the shape of the container [15,16].

In hydraulic systems, oil is preferred over water as oil offers some indispensable properties which make it the best choice.

Some of the advantages of using oil over water are as follows:

i. Oil has non-corrosive properties, unlike water which causes corrosion over the mechanical equipment when used [15,16].
ii. The density of oil is more than that of water and thus capable of withstanding high loads [15,16].
iii. Oil is also capable of exhibiting elevated evaporating temperatures [15,16].

It works on the principle that vacuum is created at the inlet port due to which the hydraulic fluid is sucked inside the pump and then is forced out of the delivery line [15,16].

1.2 Classification of Hydraulic Pumps:

Hydraulic pumps are classified into two types:

i. Hydrodynamic pumps

ii. Hydrostatic pumps

Hydrodynamic Pumps:

Hydrodynamic Pumps use a rotating element (motor) within the casing. It imparts radial velocity to the fluid and then the pressurized fluid is transmitted to the discharge nozzle to generate and transmit power [15,16].

Hydrostatic Pumps:

Hydrostatic pumps are positive displacement pumps, in which the flow rate is directly proportional to the speed and number of cycles of the pump [15,16]. Hydrostatic pumps either consist of rotating vanes or pistons which take fluid inside the void and force it to the delivery or discharge line [15,16].

They are further classified into Rotary and Reciprocating Pumps.

Rotary Pumps:

It works on the principle that the fluid is got inside the void of the pump in the suction stroke and through rotating vanes it is forced outside the discharge port [15,16].

In rotary pumps, the vanes are attached to the rotor which rotates and hence makes vanes rotate which are fixed coaxially with it [15,16]. The fluid is drawn inside the void and then expelled out by the centrifugal action of the rotating vanes [15,16].

Reciprocating Pumps:

Reciprocating pumps are the positive displacement pumps. It consists of reciprocating pistons or plungers [15,16]. The fluid is drawn into the void and then is expelled out by the reciprocating action of the pistons [15,16]. One of the reciprocating pumps is an Axial Piston Pump.

Hydraulic piston pumps enable high pressure flow fluid to accomplish power transmission [15,16]. These pumps exhibit structural variations, they are further classified into radial and axial piston pumps [15,16]. Additionally, they can be classified into swash plate and bent axis type based on their structure and characteristics [15,16].

In this discussion, the focus will be on swash plate type hydraulic piston pump to clarify its internal structure. We will also be discussing various types of failure occurring is different components of the pumps along with their prevention measures.

1.3 Axial Piston Pumps:

APP is a sophisticated system with interconnected components including shaft, cylinder block, piston, slipper, swashplate, bearing and valve plate [2,5]. Their collaborative efforts in reciprocating motion and fluid dynamics enable the pump to operate effectively.

The axial piston pump consists of several major components within its rotary system. The pump consists of the rotating drive shaft which is responsible for the operation of the pump. The shaft is further connected to the motor which makes it rotate along its axis. The cylinder block is another important element that surrounds the shaft and rotates along with the rotation of the shaft.

A cylinder block consists of a series of pistons aligned in a circular array at some angle. These pistons are only responsible for the reciprocation of the pump. Each piston is attached to a slipper via a ball and socket joint. Also, the slipper retainer helps in avoiding the direct contact of the slipper with the pistons.

The slippers are also attached to the inclined swash plate. As the cylinder block rotates, the pistons in it also rotate but due to the inclination of the swash plate at some angle, the rotation of the cylinder block is changed into the reciprocation of the pistons. There are many frictional pairs in the piston pump including slipper- pistons, slipper-swash plate anda piston –cylinder block.

The faults like loose slippers, slipper wear, center spring failure and valve plate failure caused due to lack of lubrication, overheating, over speeding, wear, abrasive contaminants, use of inappropriate fluid, aerated fluid and many others can adversely impact the performance of piston pumps [2,5].

The discussion mainly highlights the challenges arising from these faults in various components of an axial piston pump like slipper, cylinder block, piston, swash plate etc. emphasizing the importance of addressing these issues to optimize pump performance and efficiency also to minimize the energy losses caused by these faults within the pump [2,5]

CHAPTER 2

FAILURE ANALYSIS OF SLIPPER

2.1 Slipper of an Axial Piston Pump:

Slipper ensures the smooth operation of APP. It is attached to the piston via a ball and socket joint (refer fig 2.1) [5,6]. This integral connection extends to the swash plate, forming a synchronized mechanism that facilitates the reciprocating movement of the piston [5,6].

Slipper serves a dual purpose by reducing wear between the swash plate and the piston [5,6]. Its presence not only facilitates the desired speed but also acts as a protective barrier, thereby increasing the longevity and efficiency of the pump [5,6].

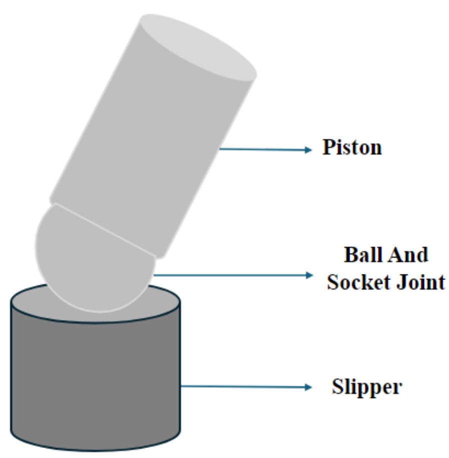


Fig. 2.1 Piston- Slipper Assembly

Recognizing the importance of the slipper in this complex system, it ensures optimal performance and longevity of the axial piston pump [5,6]. But constant high load applications cause the slippers to wear and lead to their failure. Various types of slipper failure along with their causes are illustrated in Table 1.

Table 1. Various types of Slipper Failure [5,6]:

Failure Type	Failure Cause
I. Adhesive wear	a). Seah plate and the slipper friction makes the oil film thinner which causes wear. b). Lack of lubrication and inappropriate fluid. c). Over speeding of the hydraulic pump beyond a fixed limit. d). High case pressure of the hydraulic fluid. e). Cavitation within the hydraulic fluid
II. Particle Contamination	a). Particles contaminated within the hydraulic fluid. b). Lack of maintenance of the hydraulic fluid

III. Chemical Contamination	a). Presence of chemical pollutants within hydraulic fluid. b). Lack of maintenance of the hydraulic fluid.
IV. Overheating	a). Over speeding of the hydraulic pump. b). Lack of lubrication within the piston slipper assembly.
V. Separation	a). Adhesion causes separation of piston from cylindrical block. b). Over speeding c). Contamination within the hydraulic fluid d). Lack of lubrication e). High case pressure within the pump.
VI. Discoloration	a). Overheating of the pump b). Lack of Lubrication within the pump's assembly.
VII. Slipper Overturning and Collision	a). Intricate multi-degree motion of slipper with its coupling. b). Complex working conditions.

2.2 Adhesive Wear:

Adhesive wear or slipper wear refers to the wear and tear resulting from the relative friction between the slipper and the Swash plate and smearing and galling (refer Danfoss Failure analysis manual for the figure of adhesive wear on the slipper surface [6]) of the slipper face during the pump's operation [5,6].

The hydraulic oil is absorbed and discharged by the piston cavity. The sudden start and stop of the piston pump can impact slippers, resulting in thinning of the oil film between swash plate and slipper and hence leading to friction.

This friction, exacerbated by the existing clearances, contributes to slipper wear. The slipper face exhibits adhesive wear, indicative of insufficient lubrication or the use of improper fluid. In essence, this underscores the importance of ensuring adequate lubrication and appropriate fluid properties to mitigate wear and maintain the efficient operation of the hydraulic piston pump.

2.2.1 Prevention of Adhesive Wear on Slipper:

To prevent and mitigate adhesive wear in slippers of hydraulic piston pumps, consider the following measures:

a). Microstructures to the slippers:

The abrasion of slippers with swash plate and the pistons can be reduced by introducing three types of slipper microstructures, which also helps in increasing the lubrication performance of the slipper [10,11].

The three types of slipper microstructures are as follows [10,11];

1. Micro Chamfering
2. Micro filleting
3. Micro stepping

Simulation results from the studies have found that the slipper without any microstructures lean forward or tilt and touch the swash plate hence overturn [10,11]. Leading to the partial abrasion and collision and ultimately to the failure. The first two are better than the third one.

Results from the simulation reflect that the slippers without microstructures lean towards the yoke and hence overturns, leading to the pump failure. While the one with microstructure, prevents this contact by forming an oil film [2,7].

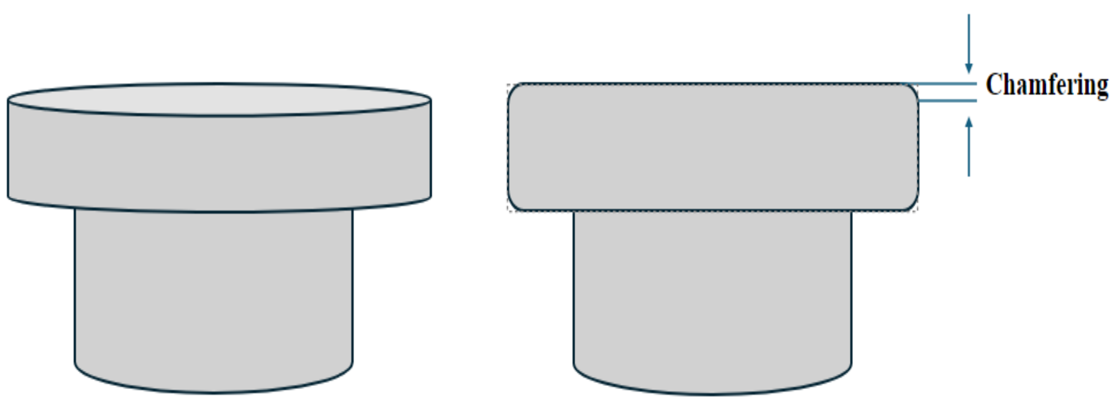


Fig 2.2 Slipper without Microstructure and with micro chamfering

The slipper rotates at a high rotational speed leading to heavy tilting of the slipper due to the generation of centrifugal force [12].

Since, the rotational speed cannot be reduced as it will lead to poor working and operation of the piston pump. Therefore, studies have found that slipper tilts can only be prevented at high speeds, by increasing the slipper retainer's down force. [12].

Some studies have also found that slippers with Diamond like Coating (DLC) on the surface are prone to less wear as compared to that of the standard ones [12].

b). New Slipper Retainer Mechanism to prevent the slipper wear:

This new slipper-retainer mechanism is different from the old mechanism as it consists of the integrated unit which does not allow the surface-to-surface contact of the slippers with the swash plate and hence protecting it from the partial abrasion and wear [12].

In traditional mechanism, the retainer rubs against the bearing plate of the swash plate [12]. The new retainer consists of three plates, the driving, adjusting and sliding plate connected via bolts [12]. To see the complete mechanism along with the figure, refer to the paper of Chao, Q., Zhang, J., Xu, B., Wang, Q., Lyu, F. and Li, K., 2022 on this new approach of integrated slipper retainer mechanism [12].

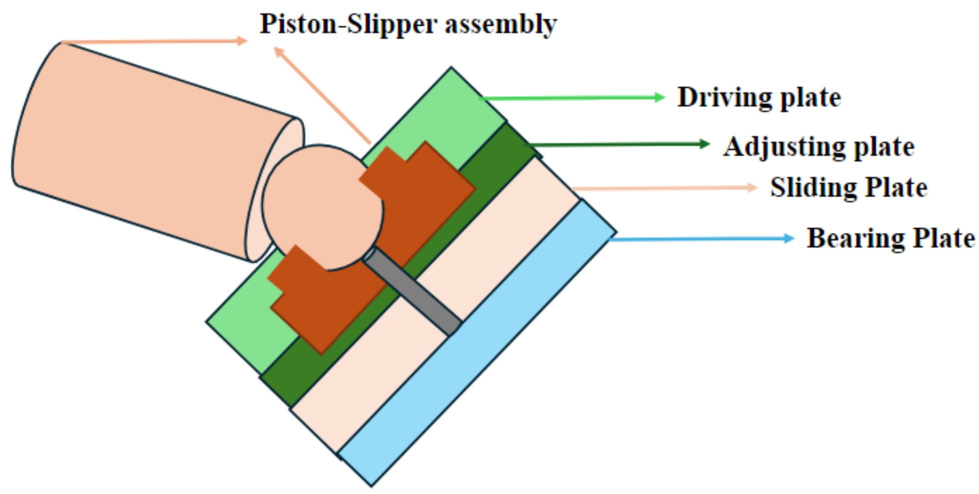


Fig 2.3 New Slipper-Retainer Mechanism to reduce slipper wear

The oil is supplied to the sliding and bearing plate through the holes proved as shown in fig 2.3 and it is further transferred to the pump casing to lubricate the components of the pump. The hydrodynamic pressure is generated due to the rotation of the sliding plate with the bearing plate [12].

The combined effect of hydrostatic and hydrodynamic forces balances the forces acting on the sliding plate [12]. Thus, the direct contact between the slipper and the swash plate can be avoided by this new slipper retainer mechanism. It also helps with proper lubrication and thus protecting the pump from failure and slipper wear [12].

Lubrication: Ensure a consistent and sufficient supply of high-quality lubricating oil to the slipper and swashplate interface to reduce friction and minimize wear [5,6].

Right fluid: A correct fluid must be selected to prevent damage to the pump which has low viscosity and high efficiency.

Regular Maintenance: A regular maintenance schedule must be implemented to prevent damage to the pump and increase its efficiency.

2.3 Particle Contamination:

Particle contamination in the slipper is a serious issue, primarily stemming from the utilization of an unsuitable hydraulic fluid and insufficient lubrication practices (refer Danfoss failure manual [6] for the figure of particle contamination).

The improper choice of hydraulic fluid can lead to accelerated wear and tear and scratches, causing friction-induced abrasions on the slipper's surface [6]. Furthermore, the absence of adequate lubrication exacerbates the problem, as the slipper is deprived of the necessary protective film that would otherwise mitigate the detrimental effects of friction and ensure smoother operation [6].

These scratches not only compromise the structural integrity of the slipper but also impede the seamless functioning of the piston pump, potentially resulting in diminished performance and efficiency [6].

Addressing this issue necessitates a comprehensive evaluation of the hydraulic fluid compatibility, coupled with a robust lubrication regimen, to safeguard the slipper from premature wear and maintain the optimal operational condition of the pump [6].

The extent of damage can be avoided by a simple method that involves running a fingernail or a lead pencil over the slipper [6].

2.3.1 Methods to avoid particle contamination in slipper:

1. Regular maintenance of the fluid
2. Regular replacement of the fluid

2.4 Chemical Contamination:

Chemical contamination occurs in the pump due to chemical pollutants in the working fluid [6]. When the discoloration of the pump happens, it means the chemical pollutants have entered the pump via a hydraulic fluid (refer Danfoss manual for the figure of chemical contamination [6)].

2.5 Overheating:

In cases where the slipper exhibits discoloration due to overheating, dark bands may be evident pointing towards a critical concern [6].

This discoloration is symptomatic of an oil cooling system failure, where the mechanism designed to properly cool the oil has faltered, leading to an excessive rise in temperature within the system [6]. Consequently, the elevated temperatures adversely affect the slipper, causing observable dark bands on its surface [6].

Over speeding is identified as another significant factor contributing to overheating of the slipper [6]. Excessive rotational speed can lead to increased friction and heat generation within the pump's components [6].

2.6 Slipper overturning and Collision:

Slipper overturns and collides due to its complex dual degree rotation with other components [9].

This overturning leads to wear and decrease in the performance of the slipper.

Slipper overturning leads to the reduction of oil lubrication, viscosity and bearing capacity which leads to abrasion failure. And ultimately resulted in poor efficiency of the pump.

To reduce the wear failure of slippers, many works have been done which are as follows:

1. A slipper rotation testing device was designed, which upon measurements reflected the slipper's rotation at macro and micro levels. It was found that at the macro level, the slippers follow a circumferential motion while at the micro level, spinning motion [9].
2. A fully coupled model to understand the spin behavior of slippers was examined [9].
3. One of the studies pointed out that the key cause of slipper failure is its collision [9].
4. Micro chamfering would be helpful to reduce the slipper wear [9].

2.7 Failure Analysis of Loose slipper

The problem of loose slippers, in the context of a hydraulic piston pump, arises from excessive clearance between the slipper and the piston ball head. This problem becomes serious when the slipper gradually loosens to a certain extent, potentially causing a drop in the efficiency of the pump.

There are two primary reasons for slippers becoming loose, which are as follows:

a) During long-term operation of a hydraulic piston pump, dust particles may accumulate, causing the gap to increase.
b) The manufacturing process is the second reason. The slippers and pistons must be manufactured very precisely to prevent loose slipper defects.

CHAPTER 3

FAILURE ANALYSIS OF SWASH PLATE

3.1 Types of Failure in Swash Plate:

Common types of failure occurring in Swash Plate are:

1). Contamination

2). Adhesive wear

3.1.1 Contamination:

Due to contaminants in the fluid, scratches appear on the swash plate. These Scratches lead to the poor functioning of the hydraulic piston pumps. They can be detected using lead pencil or fingernail [6].

3.1.2 Adhesive Wear:

Adhesive wear occurs on the swash plate mainly due to lack of lubrication. The hydraulic oil plays a major role in the smooth operation of an axial piston pump [1].

3.2 Prevention methods against contamination:

1). Flush System:

To prevent contamination in the hydraulic fluid leading to scratches on the swash plate, flushing is required of the whole hydraulic system. This includes cleaning all the intricate parts of the system and replacing it with a fresh hydraulic fluid [6].

2). Replace the Fluid:

After cleaning the hydraulic system, drain out all the contaminated fluid and replace it with a fresh hydraulic fluid. Due to this, all the impurities and contaminants will be drained out with the previous fluid and a new non- contaminated fluid will function [6].

3.3 Prevention Methods against Adhesive Wear:

1). Improved Lubrication:

To reduce the adhesive wear on the swash plate, proper lubrication must be there.

2). Use of Proper Fluid:

Adhesive wear on the swash plate can be reduced if proper fluid is used. Proper fluid should be non- contaminated and should lead to proper lubrication [1].

CHAPTER 4

FAILURE ANALYSIS OF SLIPPER RETAINER

4.1 Types of Failure in Slipper Retainer:

Failure in Slipper Retainer occurs due to following reasons:

1. Over speeding
2. Breaking
3. Scoring

4.1.1 Over Speeding:

Over speeding of the pump leads to discoloration of the slipper retainer. Discoloration of the slipper retainer may also occur by running the pump when it is low in fluid. By regulating the speed of the pump, discoloration can be avoided [1].

4.1.2 Breaking:

Slipper retainer breaking is caused by over speeding. It is also a clear indicative of the feasibility of slipper separation [1].

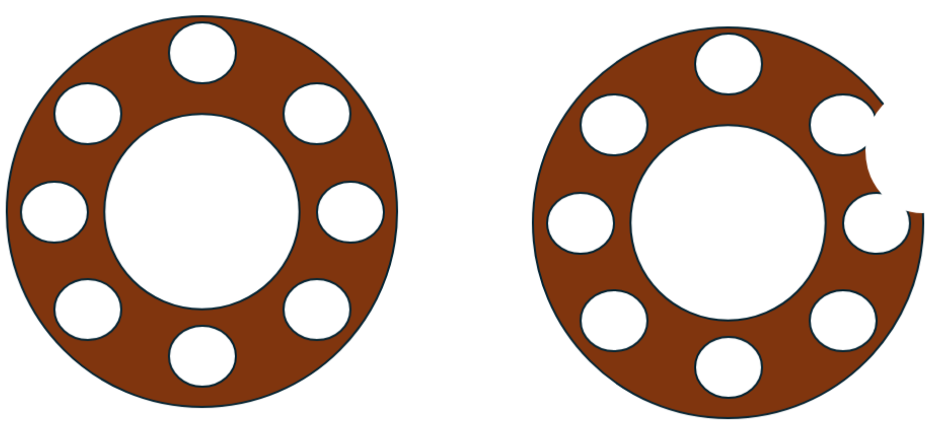


Fig 4.1 Healthy and Broken Slipper retainer

4.1.3 Scoring:

Scoring refers to the grooved wear pattern caused by piston on the slipper retainer. Scoring occurs due to particle contamination, that is, presence of contaminated particles in hydraulic fluid. Due to scoring, slipper separation occurs which ultimately leads to the pump failure [1].

4.2 Preventive Measures against failure of Slipper Retainer:

Two slippers getting stuck during the operation of the pump causes failure of the slipper retainer [17]. Slippers stuck due to uneven loading and high-pressure conditions. As the slippers get stuck, the pistons also get stuck [17]. Pistons with grooves are used by many manufacturers to avoid being stuck [17].

It was found by an analysis of the pistons and the cylinder block frictional pairs that pistons and the cylinder block friction materials do not match [17]. The friction material of the piston was 15CrMo and that of the cylinder block was QT500 [17]. To solve the problem, 38CrMoAl was chosen as the friction material of the piston [17]. Hence, the problem of the slipper retainer break was solved.

CHAPTER 5

FAILURE ANALYSIS OF CYLINDER BLOCK

5.1 Types of Failure in Cylinder Block:

Failure in cylinder block mainly occurs due to following reason:

1. Contamination
2. Pitting
3. Lack of Lubrication
4. Chemical Reaction

5.1.1 Contamination:

Due to contaminated particles in the hydraulic fluid, the port face of the cylinder block gets scratches on it, leading to poor functioning of the pump. Scratches or grooves on the port face of a cylinder block can be detected by a lead pencil or fingernail.

If detected, replace the part. To reduce the failure of a cylinder block caused by contamination, clean the system and drain out all the impure hydraulic oil by replacing it with the fresh one.

5.1.2 Pitting:

Pitting means formation of small holes or scratches on cylinder block's surface. It is caused due to over speeding of the pump, low charge pressure of the hydraulic fluid used and cavitation or aerated fluids [1].

5.1.3 Lack of Lubrication:

One of the main causes of failure of a cylinder block assembly is lack of lubrication. Since lack of lubrication between the piston- cylinder block assembly leads to the increase in friction between them and hence leading to failure of the pump. Therefore, to improve the efficiency of the cylinder block, proper lubrication must be done [1].

5.1.4 Chemical reaction:

Due to the presence of chemical particles and contaminants in the hydraulic fluid, chemical reactions occur between the particles and the yellow metal leading to system failure. To reduce the chemical reaction, proper flushing of the system along with the replacement of hydraulic fluid is required [1].

The cylinder block undergoes friction with the pistons and especially with the valve plate [13]. This friction can be reduced by coating high Lead Bronze around the cylinder [13]. Bronze alloys and high Lead help the cylinder block under high load conditions [13]. This anti-friction coating is preferred as it offers good lubrication properties [13]. The coating is done around the cylinder block by a casting process [13].

Inside the furnace, the cylinder block made up of steel is correctly bonded with the bronze tablets [13]. Afterwards, the cylinder block is gone through drilling, boring for the piston rods assembly [13]. And finally undergoes the heat treatment process for the surface hardening [13].

CHAPTER 6

FAILURE ANALYSIS OF VALVE PLATE

6.1 Types of Failure in valve plate:

Failure of the Valve Plate occurs due to following reasons:

(a) Contamination
(b) Cavitation
(c) Lack of Lubrication

6.1.1 Contamination:

The presence of contaminated or debris particles in hydraulic fluid leads to the occurrence of scratches on the surface of the valve plate. Surface grooves appear on the valve plate face. Also, the impurities present in the hydraulic fluid may be stuck to the surface leading to embedded contamination [1].

These scratches can be detected using lead pencils or fingernails [1]. To reduce the contamination of the hydraulic fluid, flush the system, drain out all the hydraulic fluid from it and replace the system with a fresh hydraulic oil [1].

6.1.2 Cavitation:

Cavitation occurs on the valve plate due to over speeding of the pump, aerated hydraulic fluid and improper pressure conditions [1]. In such a failure, replace the valve plate with a fresh valve plate [1].

6.1.3 Lack of Lubrication:

Adhesive wear occurs on the valve plate due to lack of lubrication. In this case, fluid replacement is preferred to improve the functioning of an axial piston pump [1].

CHAPTER 7

FAILURE ANALYSIS OF SHAFTS

7.1 Types of Failure in Shafts:

Failure in shafts occur due to following reasons:

1. Torsional Load
2. Axial Load
3. Radial Load
4. Interference
5. Misalignment
6. Corrosion

7.1.1 Torsional Load:

Failure of the shaft occurs due to torsional loading of the shaft. Torsional loading refers to the appliance of the load along the sideways [1].

It mainly occurs due to overloading of the pressure of the pump [1]. To reduce the torsional loading on the shaft, prevent excessive loading of the shaft, use proper fluids with perfect lubrication [1].

7.1.2 Axial Load:

When the bearing assembly receives an improper axial loading, it pushes the shaft towards the rear end of the pump and, hence damaging it. Improper fit of the pump with the prime moving element often leads to damage of the shaft from axial loading.

Proper cautions should be abided while fitting the pump with the prime mover to stop the shaft's damage. Ensure flushing of the system along with replacement of the fluid before operation [1].

7.1.3 Radial Load;

During installation of the pump, misalignment of the pump's assembly with the mating components often leads to damage of the shaft due to radial loading. Radial loading refers to the appliance of the load on the shaft radially. In such a damage, replace the shaft by ensuring proper alignment of all the pump's assembly [1].

7.1.4 Interference:

The contamination due to shaft and mating components causes the interference marks on the surface of the shaft. Interference can cause failure of the bearing along with the shaft [1].

7.1.5 Misalignment:

Whenever there is misalignment of the shaft with the mating components, it causes failure of the shaft. In such a case, reinstallation of the pump with proper alignment of the shaft is preferred [1].

7.1.6 Corrosion:

Inadequate lubrication of the spline teeth of a shaft cause building up of corrosion layer on the spline teeth and thus leading to failure of the shafts [1].

CHAPTER 8

FAILURE ANALYSIS OF BALL GUIDES AND GEARS

8.1 Failure analysis of Ball Guides:

Due to the presence of contaminated particles in the hydraulic fluid, scoring/wear appears on the inner and outer surface of the ball guides [1].

To overcome this failure, proper lubrication techniques must be followed along with regular flushing of the system and replacement of the hydraulic oil [1].

8.2 Failure analysis of Gears:

Failure of the gears occur due to following reasons:

1. Overloading
2. Fatigue and wear

8.2.1 Overloading:

Due to excessive pressure on the gears, overloading occurs and hence causing the failure of the gears due to fatigue and wear. Fatigue failure of the gears may also occur due to misalignment of the shaft with the mating components [1].

8.2.2 Fatigue and Wear:

Due to misalignment of the shaft with the mating components, gears are damaged due to fatigue and wear [1].

CHAPTER 9

FAILURE ANALYSIS OF PISTON

9.1 Types of Failure in Piston:

Failure in piston of an axial piston pump occurs due to following reasons:

1. Scoring
2. Over Speeding

9.1.1 Scoring:

Scoring occurs in piston mainly due to lack of lubrication. As the system overheats, viscosity of the hydraulic oil decreases leading to the reduction of oil film thickness and thus contributing to the loss of lubrication [1].

This lack of lubrication causes the piston rings to wear, and scratches appear on the outer surface of the piston body leading to scoring [1].

9.1.2 Over Speeding:

Due to over speeding of the pump, reciprocating motion of the piston increases. As a result of which the piston rings and the body fractures and lead to failure of the pump [1].

9.2 Failure in some of the other parts of an Axial Piston Pump:

1. **Twisted Sync Shaft**
2. **Broken Rollers**

9.2.1 Twisted Sync Shaft:

Twisting of the Synchronizing shaft occurs when a sudden rotational overload appears on it through the motor shaft. This Causes the detachment of the block and valve segment from the end cap [1]. Wear Marks appear on the piston stems [1].

9.2.2 Broken Rollers:

Due to over speeding of the pump, rollers from the synchronizing shaft break away [1].

CHAPTER 10

ABOUT THE SIMSCAPE MODEL

10.1 Literature Review:

Axial piston pumps (APPs), also known as swash plate pumps, are extensively utilized across various industries like aerospace, shipping, automobile, and construction machinery due to their high-pressure ratings, favorable force-to-weight ratio, and ability to efficiently transmit power using pressurized fluids.

These pumps are carefully designed to keep the parts well-lubricated and tightly sealed, so they work well even when spinning fast. However, dirt and particles can cause wear and tear because the parts move very closely together. This wear can make gaps bigger, leading to leaks.
Researchers have studied this wear to see how it affects how well the pumps work and how long they last.

Bergada paper gives the profound understanding of the flow losses occurring in an axial piston pump. Equations are made to test the leakage flow rate and then are compared with that of the experimental results [21].

Phanindra Garimella, Bin Yao, paper presents a model-based approach to detect the failure occurring in hydraulic pumps.

X. Zhang, J. Cho, S. S. Nair, and N. D. Manring, paper is focused on a new model which gives a mathematical expression for the damping mechanism in the pump.

Tang H, Yang W, Wang Z presents a leakage detection model of piston pumps under variable load conditions [20].

10.2 The Simscape model:

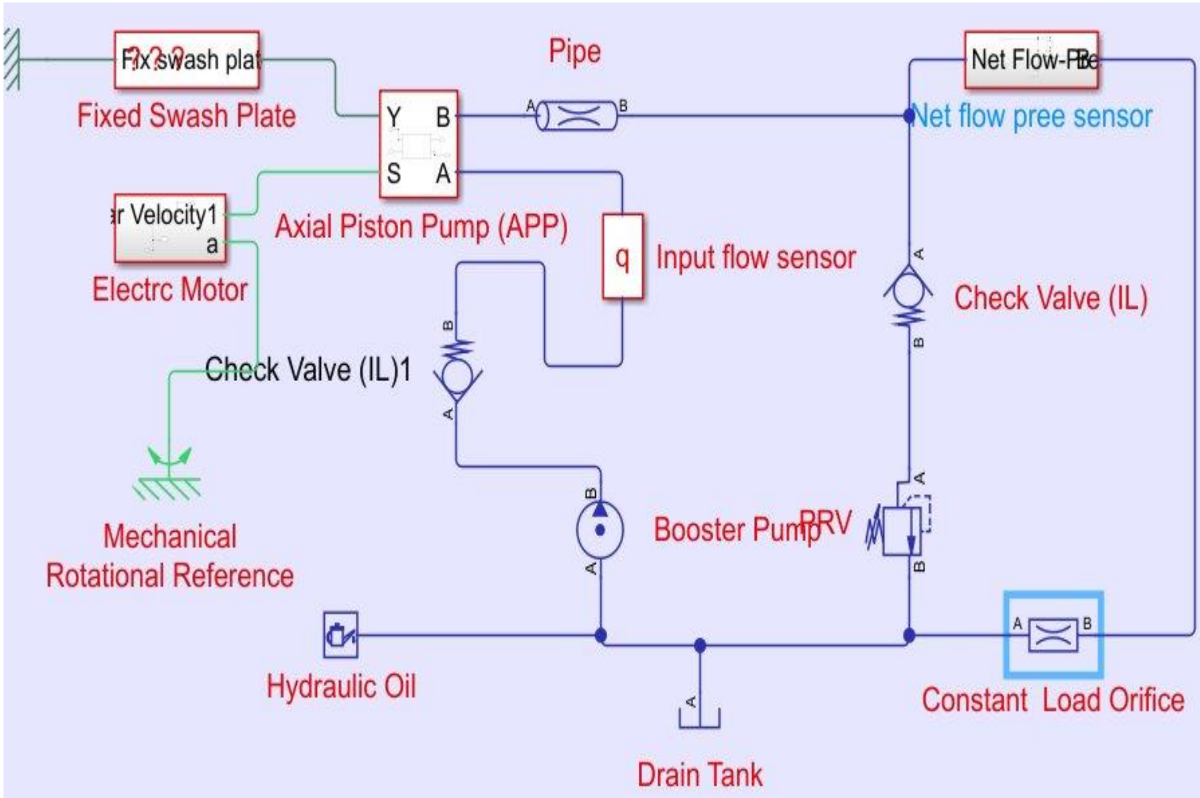


Fig 10.1 Simscape Model

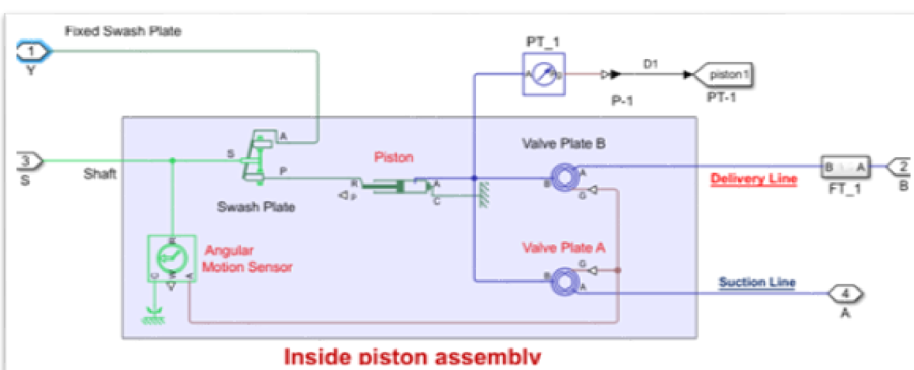


Fig 10.2 Inside Piston Assembly

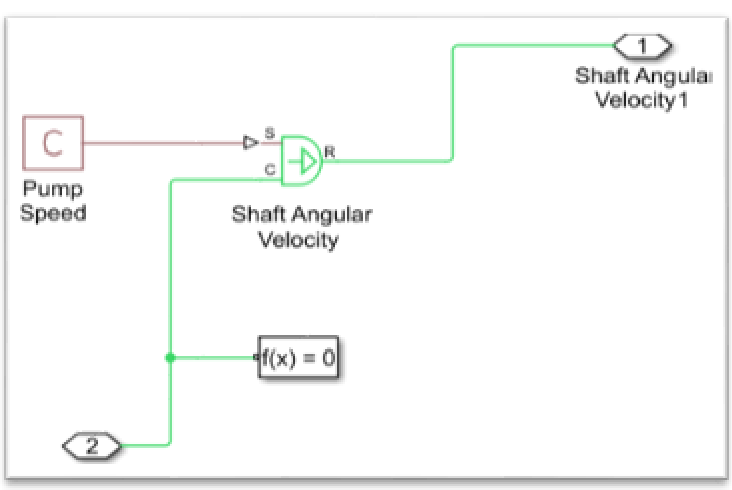


Fig 10.3 Electric Motor Sub-System

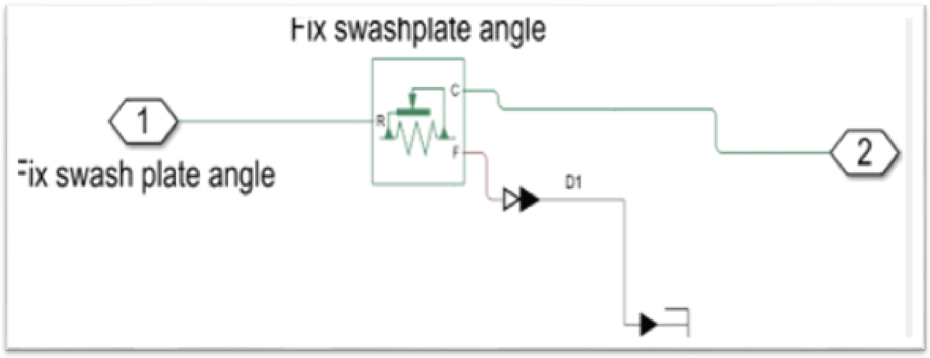


Fig 10.4. Fixed Swash Plate Angle Sub-System

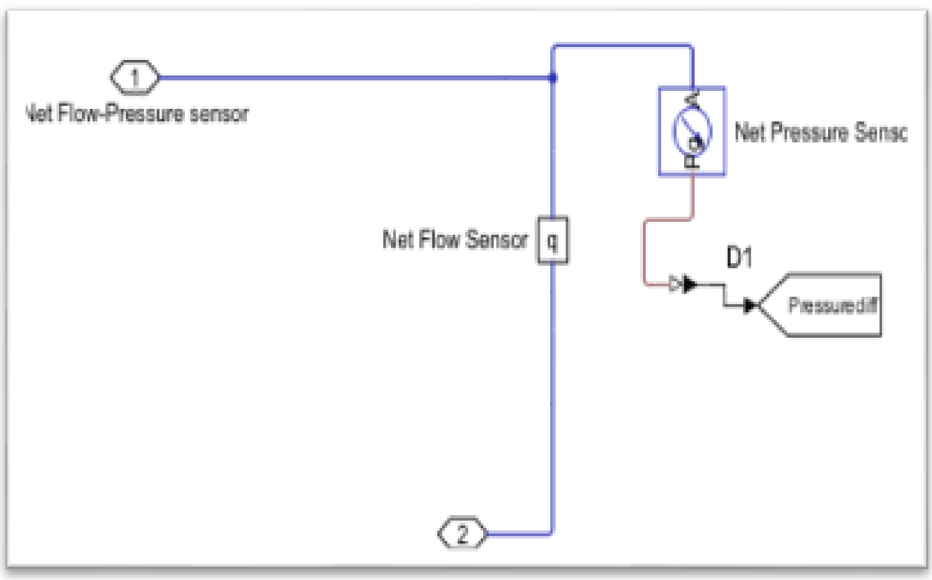


Fig 10.5 Net Flow Pressure Sensor Sub- System

A closed loop flow network was developed for the axial piston pump using SIMSCAPE environment of Matlab Simulink. It consists of a variable displacement pump. This model helps in accounting the clearance between the piston and the cylinder block. The flow is assumed to be laminar, also it is assumed that the annulus is uniform.

The model helps in simulating varying degrees of leakage severity ranging from 1 µm to 100 µm. The simulation results depict the graph between the leakage flow rate versus time and outlet pressure of the pump versus time. Various blocks used in the model represent the physical component of the pump like the piston-cylinder assembly etc.

10.3 Analytical Model for piston-cylinder Leakage when the flow is pressure driven:

The mathematical model for annular leakage flow in Axial Piston Pumps (APPs) incorporates several assumptions and simplifications to streamline the analysis process. These include:

The properties of the hydraulic fluid, such as density, viscosity, and bulk modulus are assumed to be temperature-independent, ensuring they remain constant during pump operation.

1. Flow is assumed to be laminar, and fluid inertia effects are neglected. Leakage is only considered through the annulus.

2. The leakage flow path is modeled as a uniform annulus with zero eccentricity. The pump operates at a constant rotational speed and a constant swash plate angle. The hydraulic circuit operates under a constant load condition.

3. The mathematical model for annular leakage flow is integrated into the system network model to simulate leakage from single cylinders or combinations thereof.

4. The leakage flow path and the severity of faults in single and multiple cylinders are depicted schematically. The expression for the leakage flow rate is derived from the continuity and the Navier–Stokes equations for incompressible fluid flow.

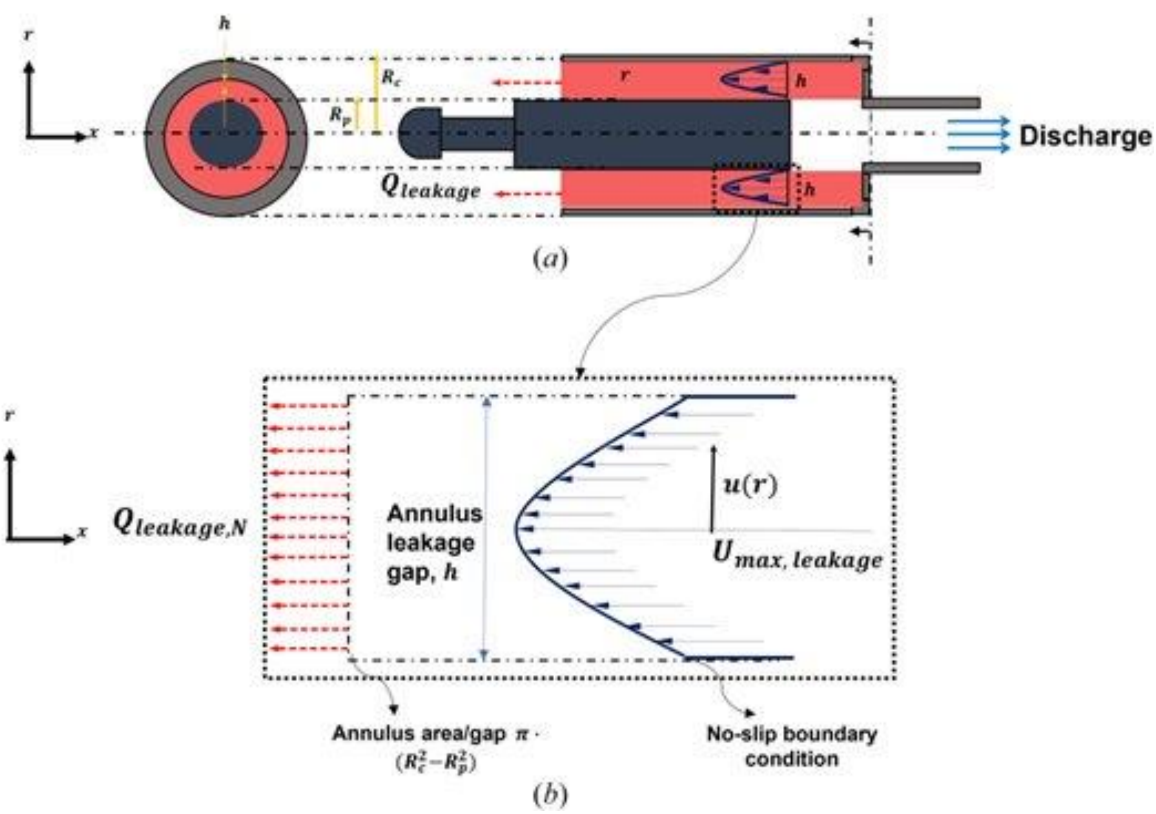


Fig. 10.6 Annular Leakage through piston-cylinder assembly

The Leakage flow rate is found out to be as follows:

$$Q_{\text{leakage}, N} = \frac{-K}{8} h R_p^3 \left(2 + \frac{h}{R_p}\right)\left[1 + \left(1 + \frac{h}{R_p}\right)^2 - \frac{h R_p \left(2 + \frac{h}{R_p}\right)}{\ln\left(1 + \frac{h}{R_p}\right)}\right]$$

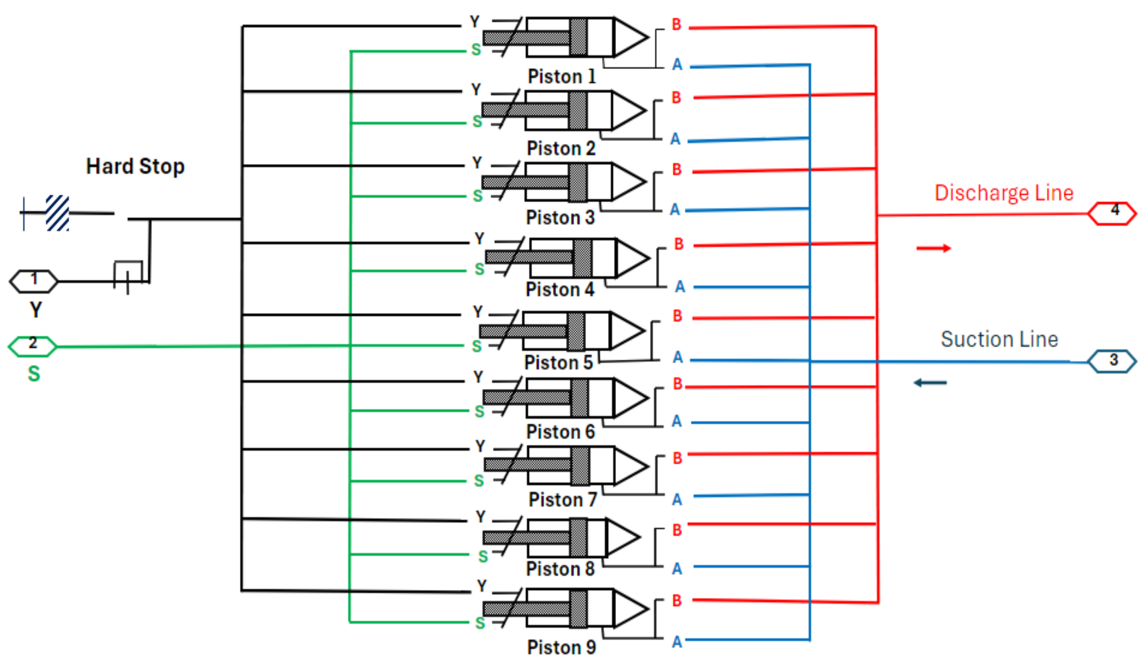


Fig. 10.7 Piston Cylinder Assembly

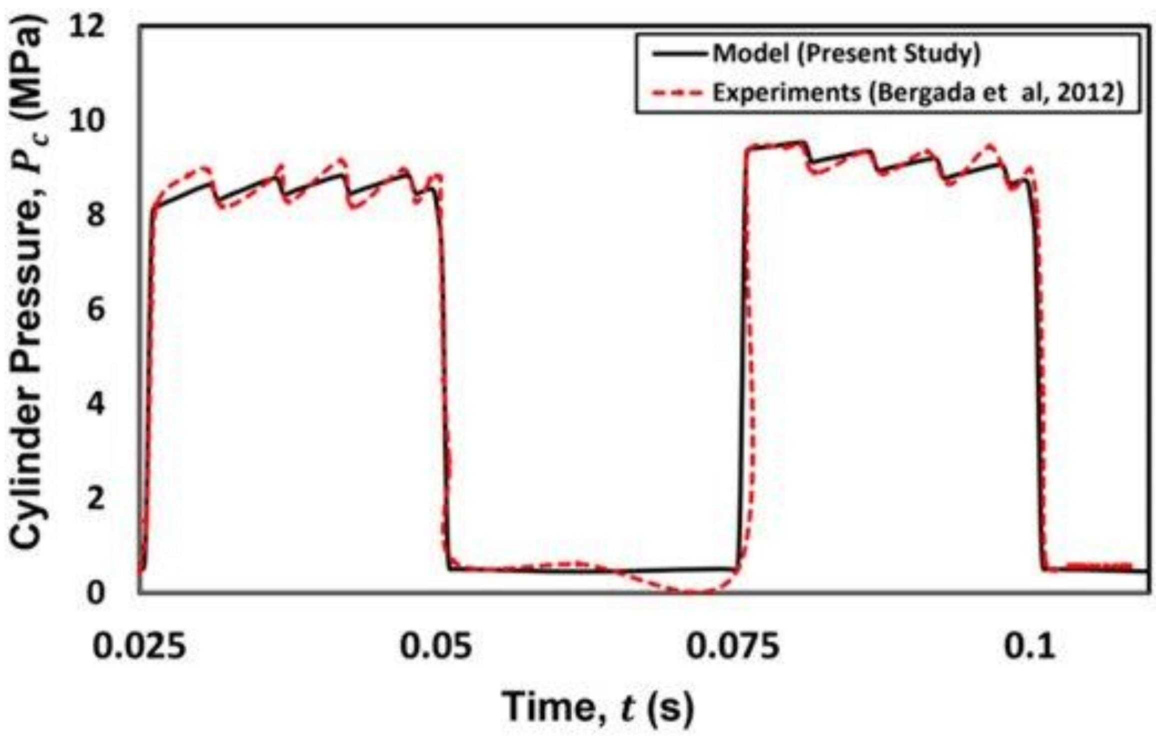
10.4 Validation of the Model:

The model's validity is evaluated by comparing its predicted pressure waveforms with both experimental and numerical results conducted by Bergada et al. Their experimental setup involved a Vickers PVB5 pump with nine pistons operating at 1000 rpm and a fixed swash plate angle of 20 degrees, utilizing hydraulic mineral oil ISO 32 under a constant load condition of 10 MPa.

Transient pressure measurements were taken in each cylinder using absolute pressure transducers, and pressure and flow rate measurements were captured at the pump discharge line. Dynamic pressure signals were measured for a single cylinder and at the pump outlet, simulating a leakage fault scenario with varying levels of wear.

To compare the model predictions with experimental results, a representative case of a 5 µm annular clearance was selected. Parameters from the experimental setup were used as inputs to the model.

The comparison between model predictions and experimental results is illustrated in the figure below, depicting time-varying pressure signatures in a single cylinder and the net pressure at the pump outlet pressure versus time graph for a fault severity of 5 µm in a single cylinder.



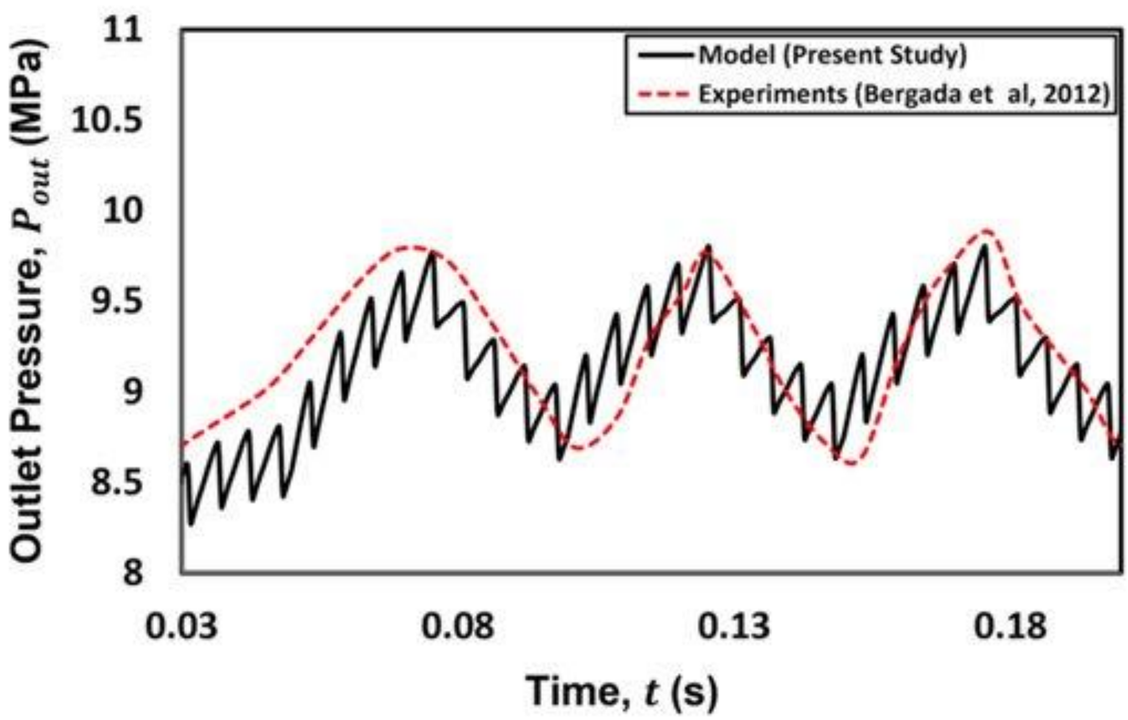


Fig. 10.8 Comparison of the Present study and that of the experiments

CHAPTER 11

PERFORMANCE PREDICTION BY INCREASING THE LEAKAGE FAULT

The plots in figures below illustrate the impact of increasing severity of leakage fault, displayed separately for single and multiple pistons. In the top, dynamic flow rate signals are depicted, while the bottom showcases dynamic pressure signals. Each plot includes signals for individual cylinders as well as the net output signal, with light lines representing a healthy condition and bold lines representing a faulty condition.

For example, in Figure 11.1, the bottom waveforms represent the rectified sinewaves of the flow rate from each of the nine pistons individually, with a phase delay of 0.0026 s (corresponding to a 40-degree phase delay). The top waveform (solid green curve) represents the net flow at the pump outlet, reflecting the cumulative effect of all nine pistons. Over a pumping cycle duration of 0.024 s, nine ripples are observed in the signal. The bold curve represents the flow rate waveform of a single faulty cylinder (cylinder #1 with a fault severity of 60 µm), showing a drop-in flow rate and a corresponding reduction in net flow rate.

Figure 11.2 demonstrates that with the same level of fault severity (60 µm) in three cylinders, the net flow rate reduces significantly. Similar trends are observed in the bottom two plots for pressure at the pump outlet, as shown in Figures 11.3 and11.4.

These plots provide a basic understanding of the effects of leakage fault severity on the time-varying behavior of the pump. Changes in dynamic behavior, such as variations in amplitude and frequency of outlet flow rate and pressure waveforms, are characterized to simulate different degrees of oil leakage fault, varying annular clearance (h) from 1 µm to 100 µm in increments of 10 µm. A healthy pump condition with h = 1 µm serves as the baseline case for comparison against other scenarios.

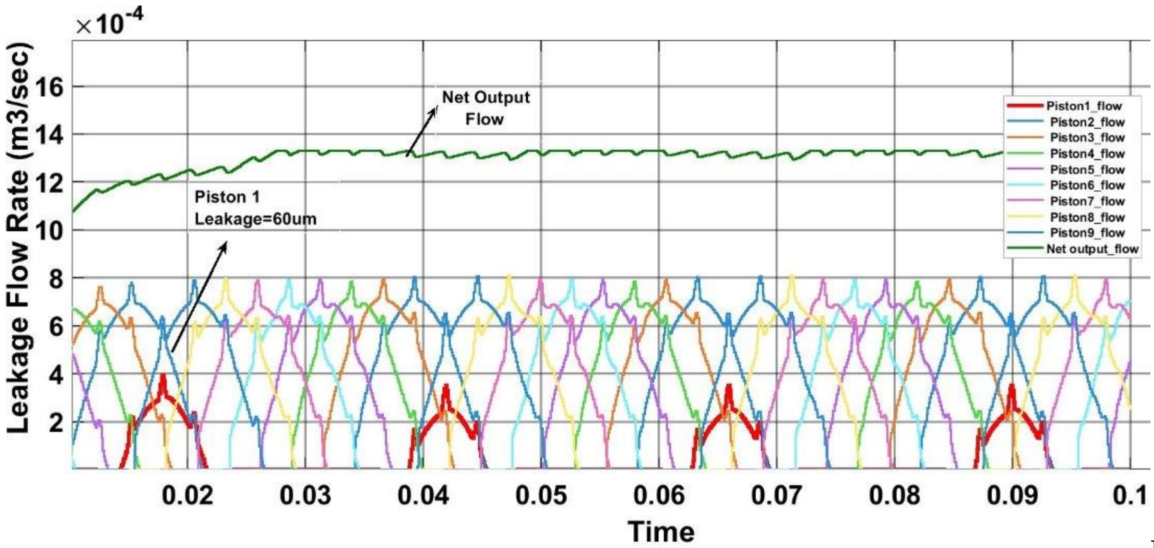


Fig. 11.1 Leakage Flow Rate for a Single faulty cylinder

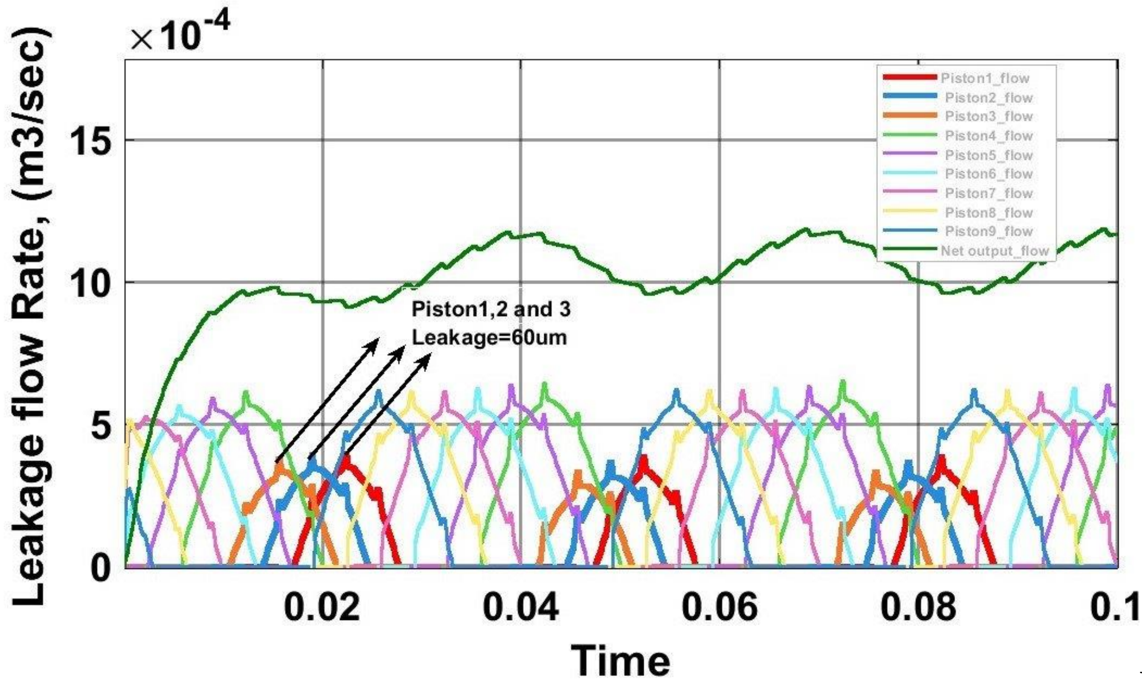


Fig. 11.2 Leakage Flow Rate for Multiple faulty cylinders

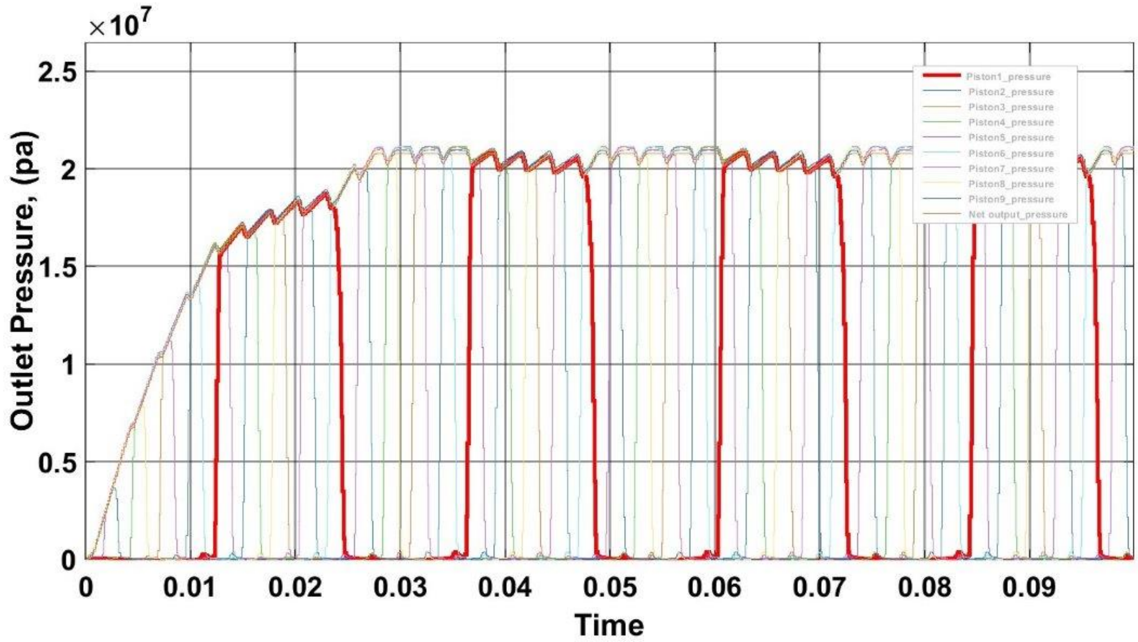


Fig. 11.3 Outlet pressure for a single faulty cylinder

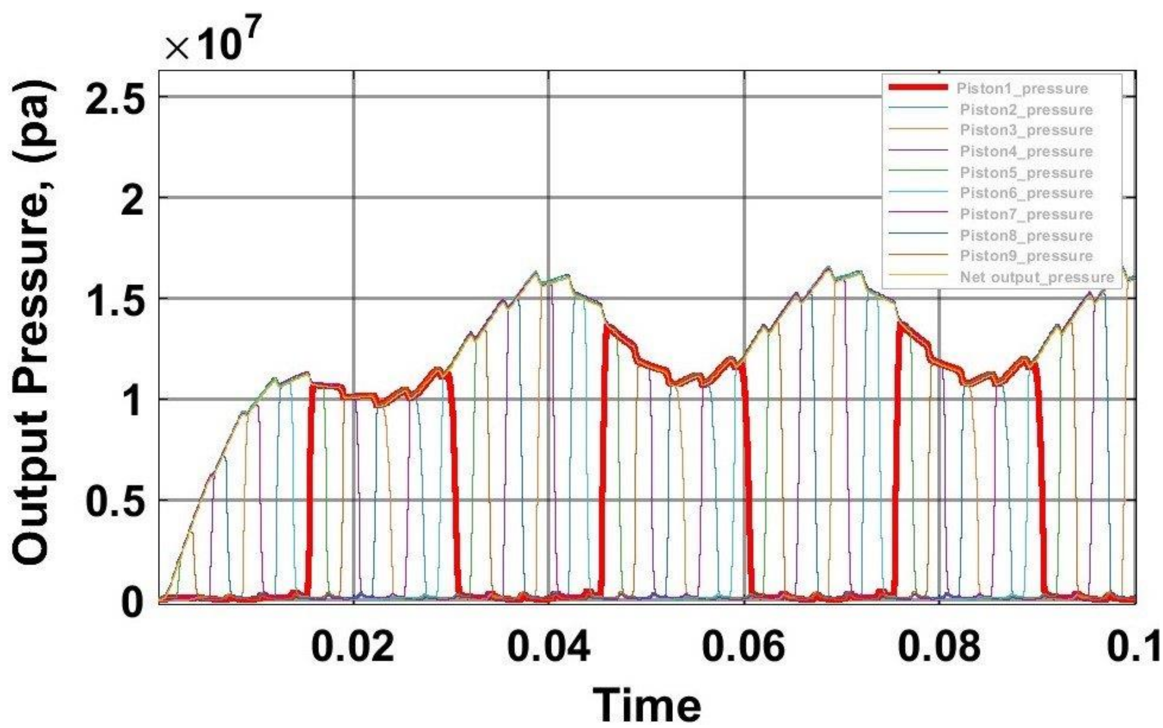


Fig. 11.4 Outlet Pressure for multiple faulty cylinder

CHAPTER 12

INCREASING THE SEVERITY OF LEAKAGE

The figures below illustrate the changes in dynamic pressure waveforms at the pump outlet as the severity of oil leakage fault in a single cylinder increase. Comparing healthy and faulty conditions, it's noticeable that the fault causes a reduction in both outlet flow and pressure, with the pressure signal being significantly affected.

Typically, pressure sensors aren't individually installed for cylinders due to the complex design implications, making outlet pressure signals the preferred monitoring choice. Analyzing a single pumping cycle (from 0.082 s to 0.1 s), several observations can be made: initially, as the fault severity increases, no pressure reduction is observed up to a certain threshold (approximately 40 µm), beyond which a nonlinear decrease occurs.

This reduction becomes more pronounced as the fault severity increases further. Notably, at extreme fault severities (around 80 µm and beyond), the pressure drops drastically and struggles to recover fully during a pumping cycle. This drop occurs nonlinearly in three stages: no reduction (up to 40 µm), moderate reduction (40 µm to 80 µm), and severe reduction (beyond 80 µm).

The alterations in pressure waveform time signature during fault stages II and III are particularly interesting, indicating changes not only in pressure magnitude but also in waveform dynamics. Specifically, the pressure-build-up time increases with fault severity, leading to alterations in the pressure waveform's peak-to-peak amplitude and time-mean value.

As fault severity increases, the pressure waveform undergoes changes, including increased peak-to-peak amplitude, decreased time-mean value, and prolonged pressure-build-up time.

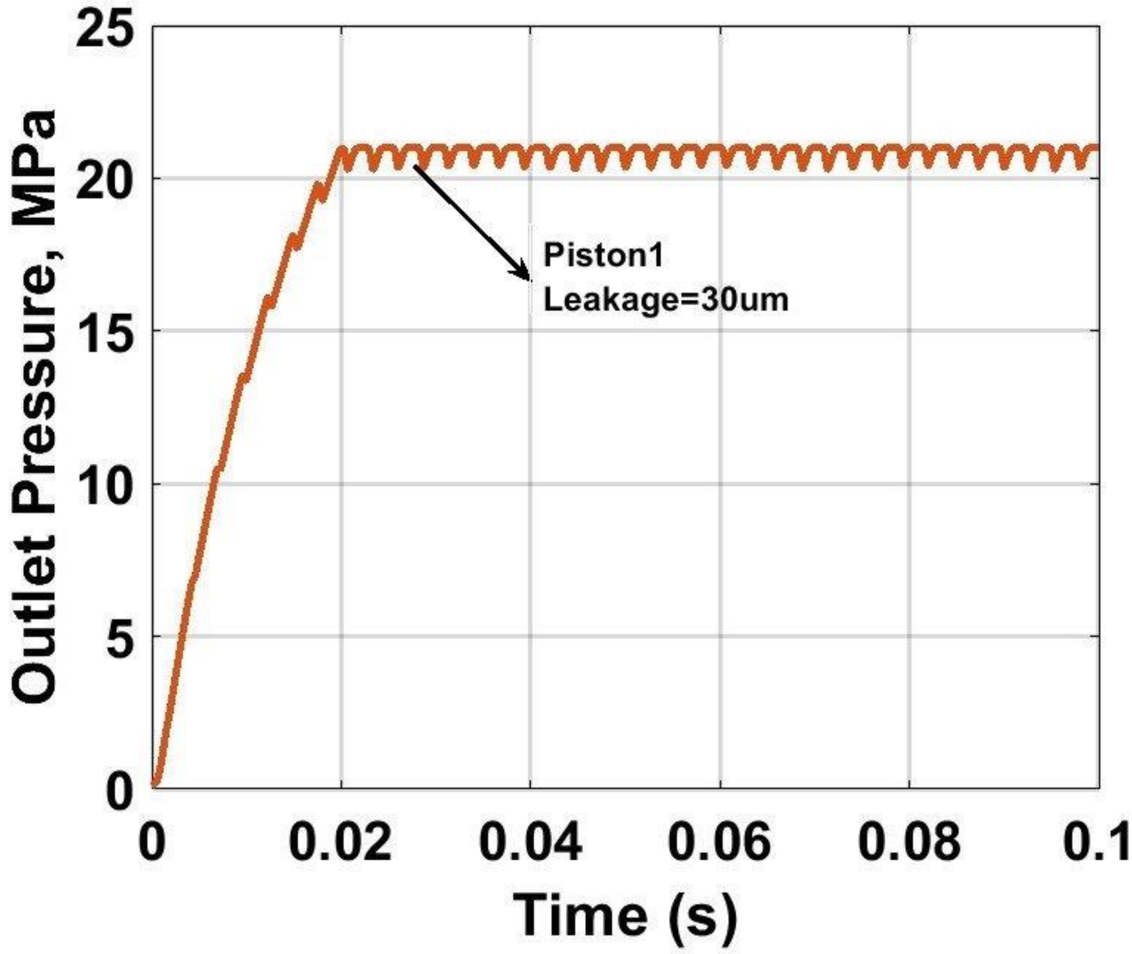


Fig. 12.1 Net Outlet Pressure with leakage of 30 µm at Piston 1

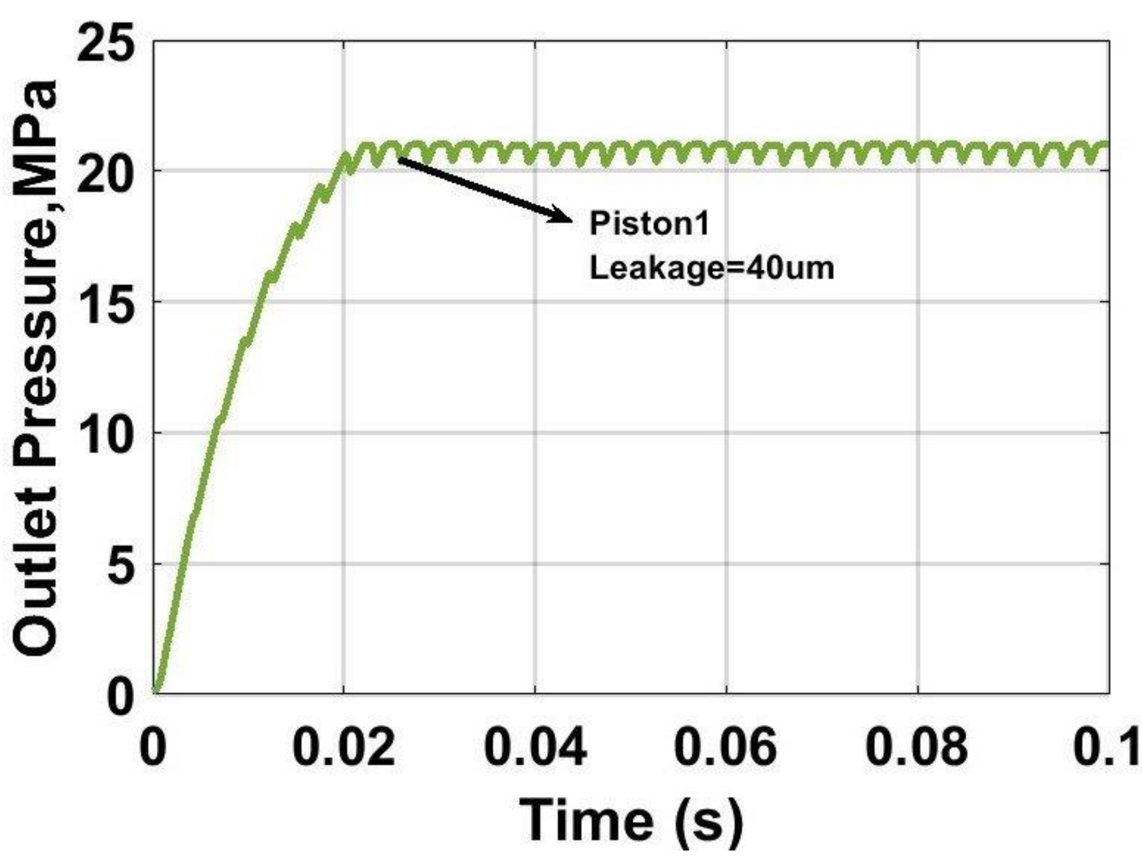


Fig. 12.2 Net Outlet Pressure with leakage of 40 µm at Piston 1

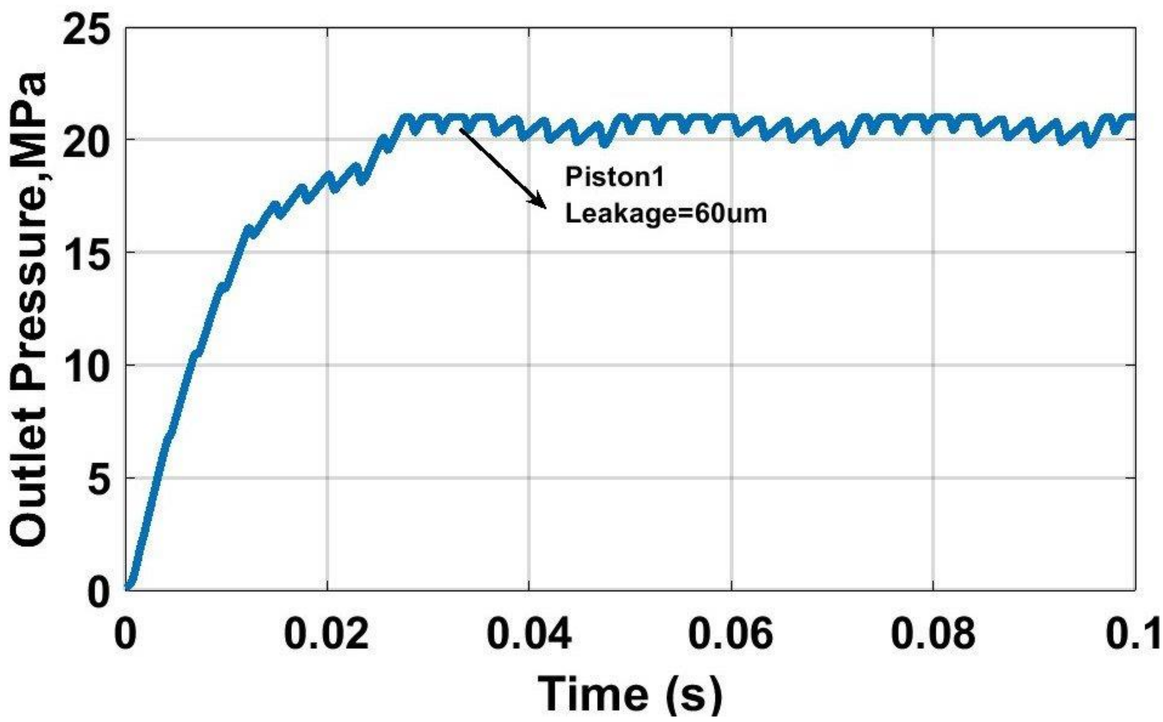


Fig. 12.3 Net Outlet Pressure with leakage of 60 μm at Piston 1

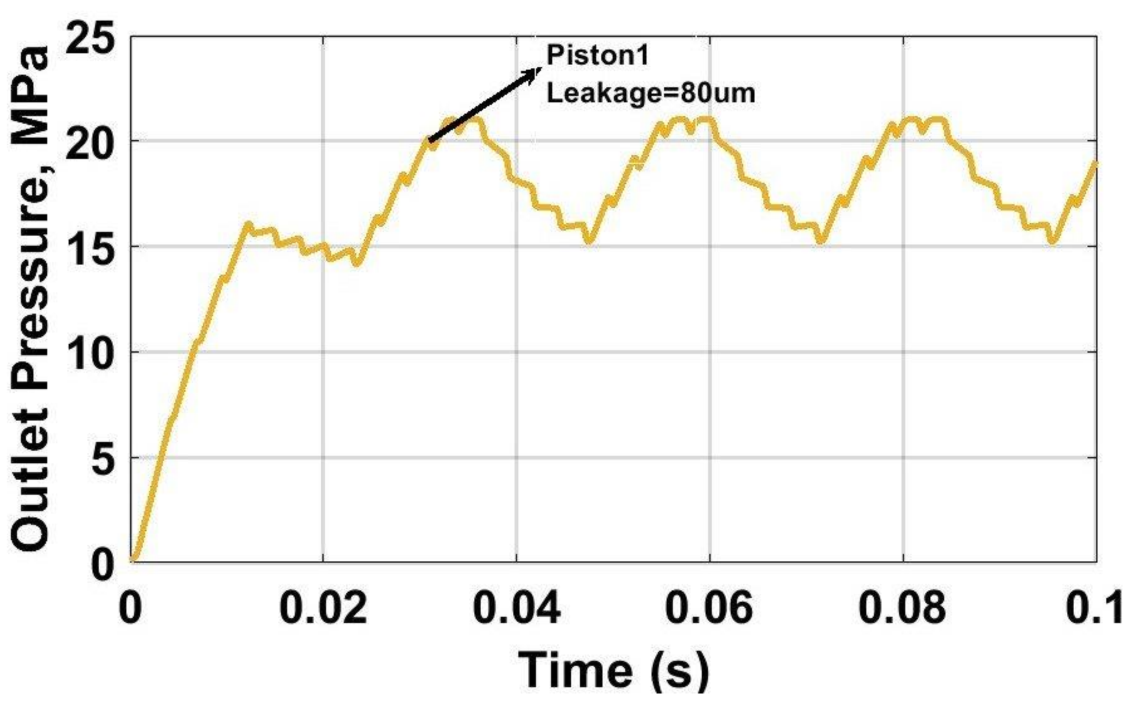


Fig. 12.4 Net Outlet Pressure with leakage of 80 μm at Piston 1

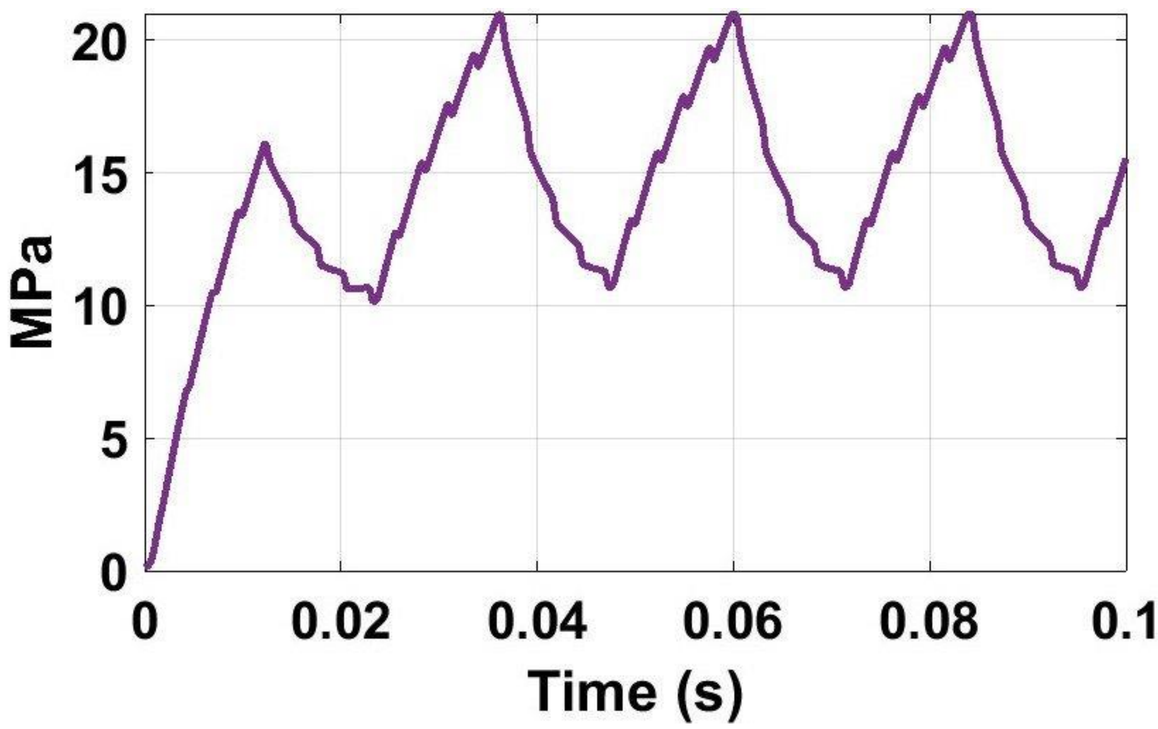


Fig. 12.5 Net Outlet Pressure with leakage of 100 µm at Piston 1

Observations:

- **Stage I (h<40um):**

The pressure signal is identical to the healthy signal

- **Stage II (40um<h<80um):**

The pressure signal drops markedly with increasing faulty severity

- **Stage III (h>80um):**

The pressure signal drops to such an extent that it barely recovers.

CHAPTER 13

PERFORMANCE PREDICTION BY THE REARRANGEMNT OF PISTONS

In evaluating the performance of an axial piston pump, three distinct arrangements of faulty pistons were examined: inline, random, and uniform. The inline configuration entailed faulting pistons 1, 2, and 3; the random arrangement involved pistons 1, 3, and 8 being faulty, while the uniform setup designated pistons 1, 4, and 7 as faulty.

Through comprehensive simulations of leakage flow rate and outlet pressure for each arrangement, it was discerned that the uniform piston arrangement exhibited the least pressure fluctuations in comparison to both the random and inline configurations.

This finding underscores the significance of piston placement in mitigating pressure irregularities within the pump system. The uniform distribution of faulty pistons evidently fosters a more stable hydraulic environment, thereby enhancing the overall operational efficiency and reliability of the pump.

By contrast, the random and inline configurations manifest higher pressure fluctuations, indicative of less optimal performance and potentially increased susceptibility to operational challenges.

Consequently, the insights gleaned from these comparative analyses underscore the importance of strategic piston arrangement in optimizing pump functionality and minimizing adverse effects stemming from faulty components.

Arrangement of Pistons	Type of arrangement	Pressure Fluctuations
Inline	**Worst**	**High**
Random	**Intermediate**	**Medium**
Uniform	**Best**	**Low**

INLINE ARRANGMENT:

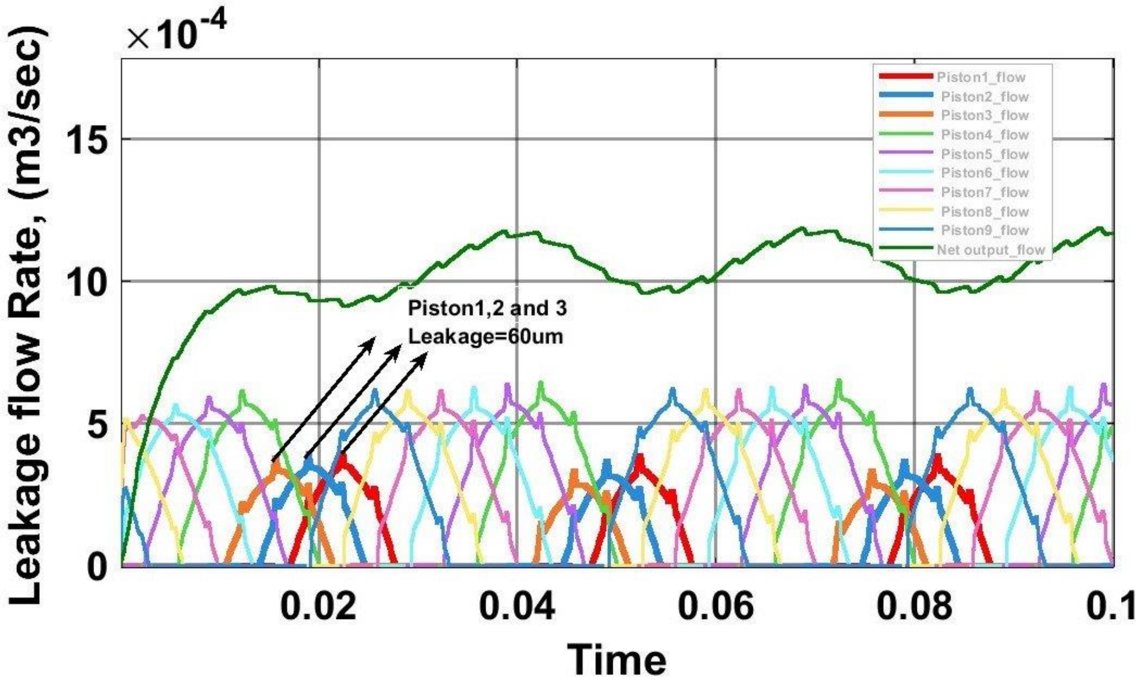


Fig. 13.1 Leakage flow rate with Inline Arrangement

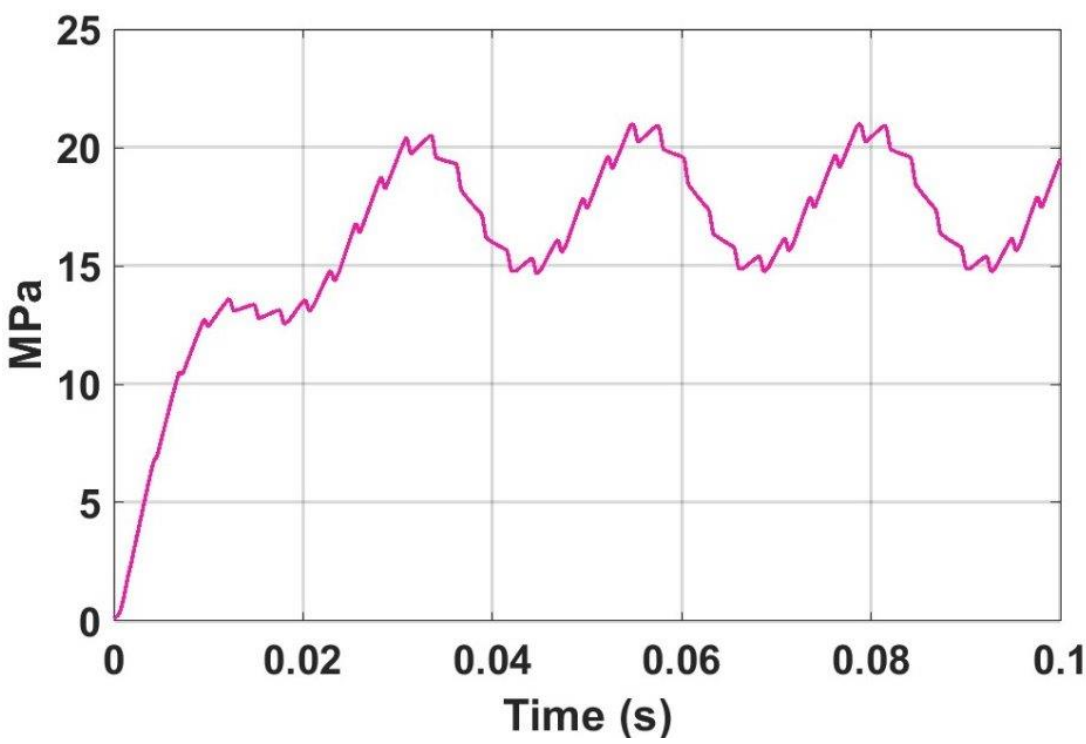


Fig. 13.2 Outlet Pressure of the Pump with Inline Arrangement

RANDOM ARRANGEMENT :

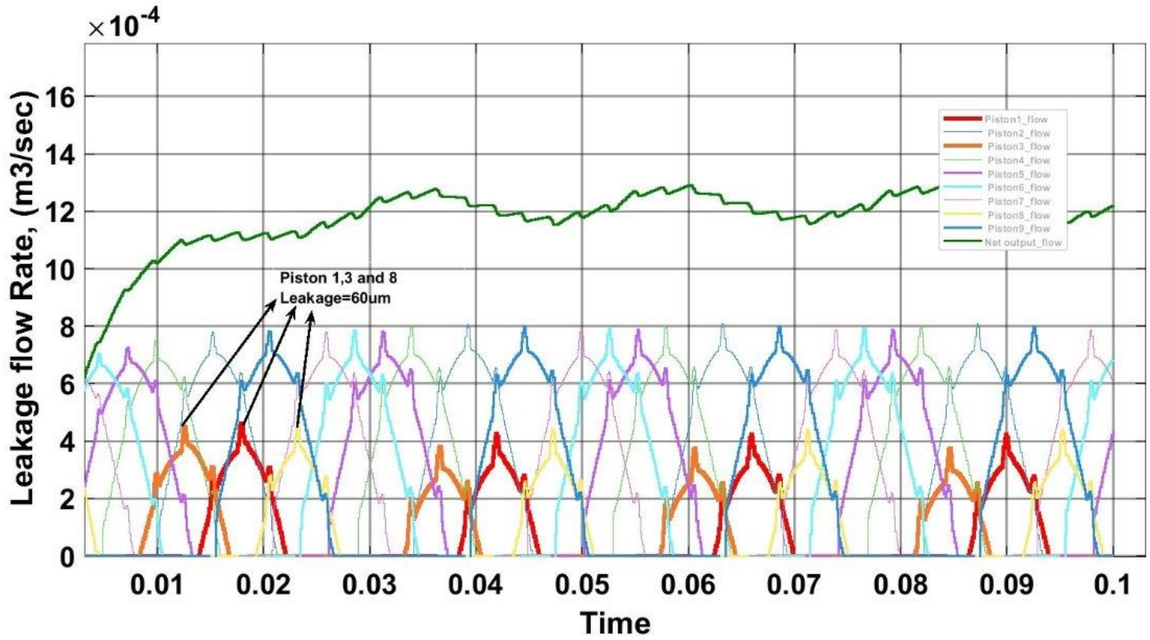


Fig. 13. 3 Leakage flow rate with Random Arrangement

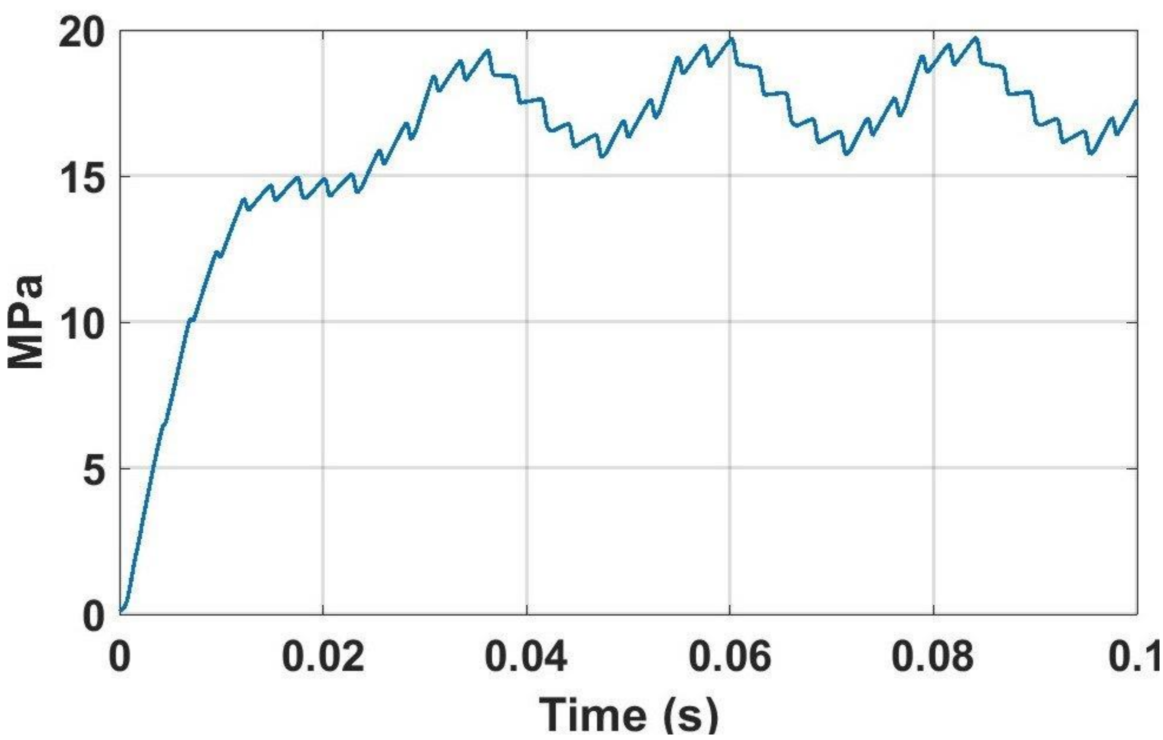


Fig. 13.4 Outlet Pressure of the Pump with Random Arrangement

UNIFORM ARRANGEMENT:

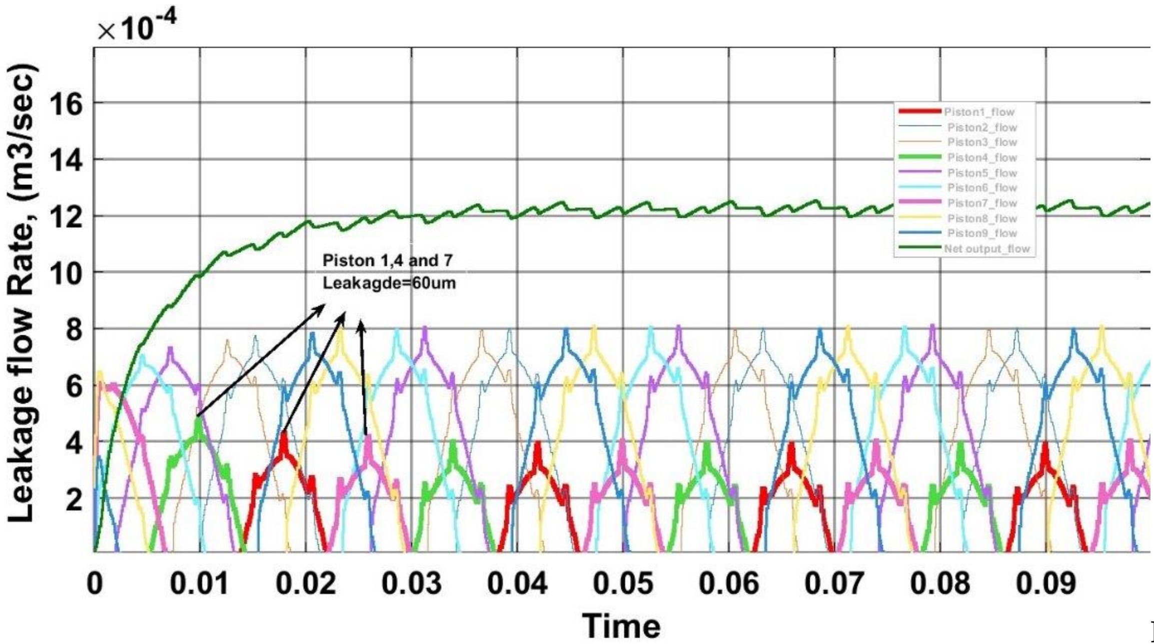


Fig. 13.5 Leakage Flow Rate with Uniform Arrangement

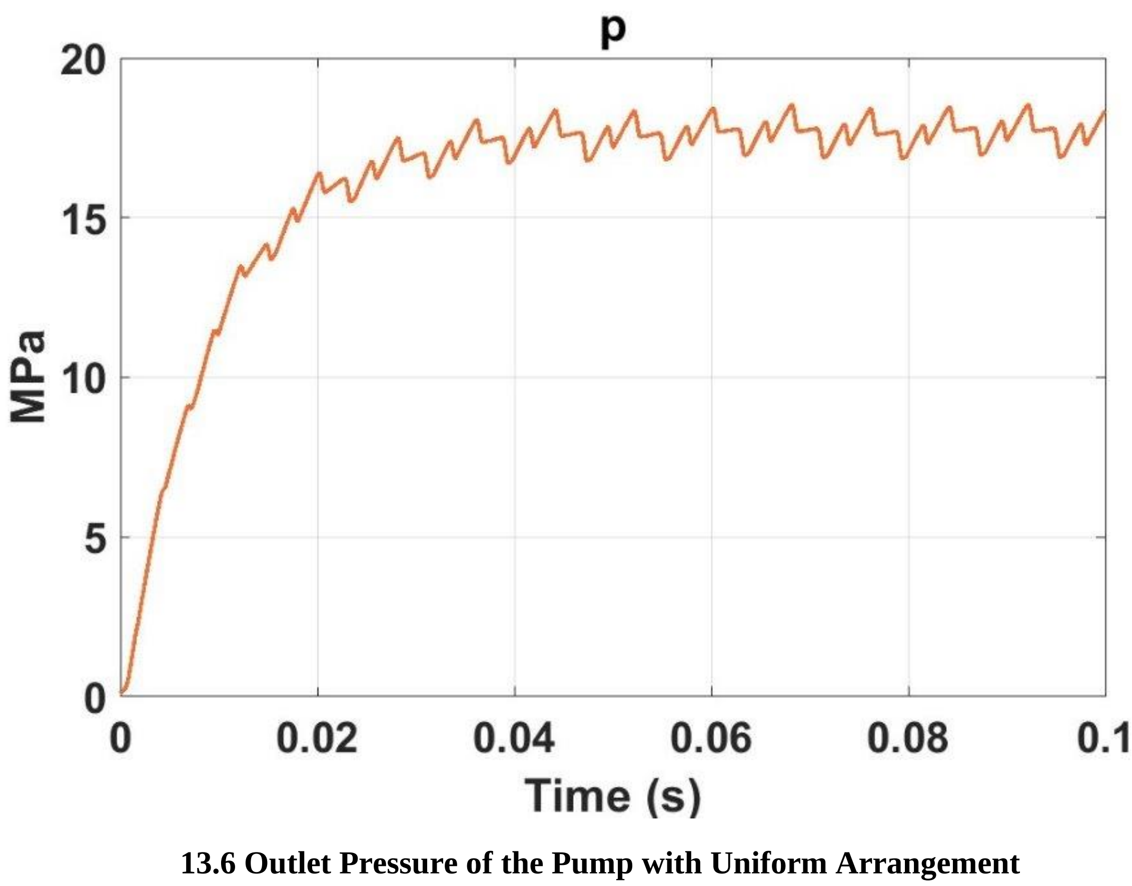


Fig. 13.6 Outlet Pressure of the Pump with Uniform Arrangement

CHAPTER 14

ANALYTICAL MODEL OF THE PISTON CYLINDER ARRANGEMENT WHEN THE FLOW IS BOTH PRESSURE AND SHEAR DRIVEN

The analytical simulations previously provided illustrated ideal conditions where the piston and cylinder remained stationary, and leakage between them resulted solely from pressure-driven flow. However, real-world scenarios differ significantly. In practical settings, the cylinder is typically fixed while the piston reciprocates between TDC and BDC. In this dynamic environment, flow comprises both pressure-driven and shear-driven components due to piston movement. Additionally, the cylinder rotates along with the drive shaft's rotation.

The analytical model presented reflects this ideal condition where the piston moves opposite to the flow direction, allowing for the derivation of formulas governing leakage flow rates. However, in practice, the situation is more complex.

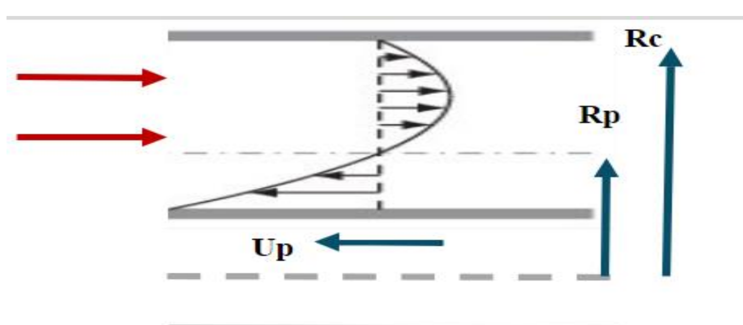


Fig. 14.1 Annular leakage for both pressure and shear driven flow

By using the continuity equation we get,

du/dx = 0 or u = u(r)

dp/dr = 0 i.e., P = P(x)

Therefore, the expression for velocity in terms of r is:

$U(r) = Kr^2/4\mu + K1 \ln(r) + k2$

Applying the Boundary conditions,

At r = Rc, u = 0

At r = Rp, u = -Up

We get values of K1 and K2 and by putting the values of K1 and K2, we get:

$$u(r) = -\frac{K}{4\mu}\left[Rc^2 - r^2 + \frac{(Rc^2 - Rp^2)}{\ln\left(\frac{Rp}{Rc}\right)}\ln\left(\frac{Rc}{r}\right) + \frac{Up\ln\left(\frac{Rc}{r}\right)}{\ln\left(\frac{Rp}{Rc}\right)}\right]$$

The Leakage flow rate for real conditions have come out to be:

$$Q = -\frac{K}{2\mu}\left[\frac{((h+Rp)^4 - Rp^4 - 2hRp^2)}{4} - \left(\frac{(2Rp^2+h^2+2hRp)}{4} - \frac{\ln(1+\frac{h}{Rp})Rp^2}{4}\right)\left(\frac{h^2+2hRp+Up}{\ln(1+\frac{h}{Rp})}\right)\right]$$

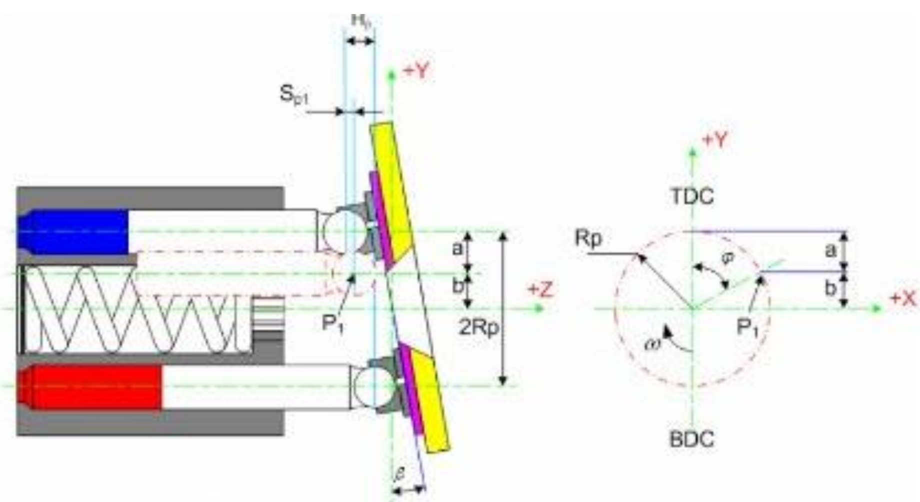


Fig 14.2 Displacement of the piston

Φ = Piston's angular position

β = Swash Plate Angle

Sp = Displacement of the piston = a tanβ

B = Rp cos Φ

a = Rp - b = Rp[1 - cosΦ]

Sp = Rp tanβ [1 - cosΦ]

Up = dSp/dt

The expression for the velocity of the piston is as follows:

$$Up = Rp \tan(\beta) \sin(\phi) \omega,$$

And the final leakage flow rate with the expression of Up comes out to be:

$$Q = -\frac{K}{2\mu}\left[\frac{((h+Rp)^4 - Rp^4 - 2hRp^2)}{4} - \left(\frac{(2Rp^2+h^2+2hRp)}{4} - \frac{\ln(1+\frac{h}{Rp})Rp^2}{4}\right)\left(\frac{h^2+2hRp+Rp\tan(\beta)\sin(\phi)\omega}{\ln(1+\frac{h}{Rp})}\right)\right]$$

FUTURE SCOPE

In terms of future scope, the next step involves conducting simulations based on the analytical model under real-world conditions, where the piston reciprocates and the cylinder rotates, to validate its applicability and accuracy. This will provide valuable insights into how well the model captures the complexities introduced by dynamic piston and cylinder movements, and how it compares to actual system behavior.

By gathering comprehensive experimental data, including leakage flow rates, pressure distributions, and fluid dynamics, researchers can refine the model and improve its predictive capabilities.

Furthermore, exploring the effects of varying parameters, such as changes in speed and external radius, will enhance our understanding of the system's response under different operating conditions.

This comprehensive analysis will not only deepen our insight into the fundamental principles governing piston-cylinder dynamics but also provide practical guidance for optimizing system performance and reliability in real-world applications.

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