FAILURE PREDICTION IN AN AXIAL PISTON PUMP

M.Tech. Thesis

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A THESIS

Submitted in partial fulfillment of the requirements for the award of the degree
of
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CANDIDATE'S DECLARATION

I hereby certify that the work which is being presented in the thesis entitled **FAILURE PREDICTION IN AN AXIAL PISTON PUMP** in the partial fulfillment of the requirements for the award of the degree of **MASTER OF TECHNOLOGY** and submitted in the **DEPARTMENT OF MECHANICAL ENGINEERING, Indian Institute of Technology Indore**, is an authentic record of my own work carried out during the time period from July 2023 to June 2025 under the supervision **of Prof. Pavan Kumar Kankar, Professor, Department of Mechanical Engineering**

The matter presented in this thesis has not been submitted by me for the award of any other degree of this or any other institute.

Prashant Singh
2/06/25

Signature of the student with date
(SINGH PRASHANT ARVIND KUMAR)

This is to certify that the above statement made by the candidate is correct to the best of my/our knowledge.

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Signature of the Supervisor of
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SINGH PRASHANT ARVIND KUMAR has successfully given his M.Tech. Oral Examination held on **23rd May 2025**.

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With Regards,

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ABSTRACT

This research is about the analysis of internal leakage occurs in the axial piston pumps particularly the impact of varying annular gap sizes between the piston and cylinder on pump performance. A mathematical model is developed using the Conservation of momentum (Navier–Stokes) and Conservation of mass (continuity Equation) to identify leakage flow characteristics, calculating for both pressure-driven (Poiseuille) and shear-driven (Couette) flow components. Controlled clearances of 50 µm, 80 µm, 100 µm and 180 µm were introduced to simulate wear, and the theoretical predictions were validated through experimental measurements. The study reveals that pressure-driven flow is the primary contributor to leakage, while shear-driven flow becomes significant at higher operating speeds. Sensitivity analysis of the swashplate angle further clarifies its role in displacement control. The model demonstrates high accuracy in predicting discharge and leakage behavior by early detection of wear and enabling effective predictive maintenance. This work contributes to enhancing the efficiency, reliability and operational life of hydraulic systems employing axial piston pumps.

TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

NOMENCLATURE

Symbols

H_p	**Total stroke length of the piston in one complete rotation of the cylinder block (m)**
S_p	**Piston displacement (m)**
R	**Length from the center of the cylinder bore to the cylinder block center (m)**
β	**Swashplate angle (radian)**
φ	**Phase angle (radian)**
ω	**Angular speed (rpm)**
u_p	**Piston speed (m/s)**
t	**Time (s)**
θ	**Angular position**
u(r)	**Fow or leakage velocity (m/s)**
P	**Cylinder Pressure (bar)**
r	**Radial distance (m)**
R_c	**Radius of Cylinder bore (m)**
R_p	**Radius of piston (m)**

d	**Diameter of Cylinder (m)**
m	**No. of piston**
h	**Annular gap (m)**
μ	**Dynamic viscosity (Pa-s)**
K	**Pressure gradient (Pa/m)**
γ	**Phase delay (radian)**

Chapter 1

Introduction

1.1. Overview

Hydraulic pumps are mainly designed to transform an input prime mover energy into hydraulic energy, and this energy is in the form of pressure and flow. Axial piston pumps are key elements in hydraulic systems, delivering high performance and accuracy control. They use two main principles: The swash plate and the bent axis design. In both cases pistons move back and forth within cylinder bores as the rotor spins, creating fluid movement.

In the employment of a swash plate, the pistons which are driven by the swash plate are held by the lead angle of the angled swash plate relative to the plate plane. Variation of the swash plate angle sets the displacement volume and therefore the flow rate. On the other hand, bent-axis type is based on a swivel angle between the drive shaft and the piston assembly which determines the displacement volume.

Axial piston machines can work as both pumps and motors by making them versatile for various applications. They can be used in both open circuit and closed-loop hydraulic circuits by using system parameters which significantly affect the performance. Open-loop systems draw fluid from a reservoir, whereas closed-loop circuits continuously recirculate fluid, offering higher efficiency in applications such as mobile hydraulic and Industrial machinery.

Because of high-pressure applications and with complex geometry and reliability, axial piston pumps are widely used in construction machines, jets, and automotive applications. Its ability to provide precise control over hydraulic power makes them a preferred choice in modern hydraulic systems.

1.2. Background

Axial piston pumps are widely used in most of the industrial applications because of high efficiency, durability, and capabilities to operate at high pressures ~315 bar. They are requisite for heavy duty machines, including excavators, wheel loaders and industrial presses. These axial piston pumps have precise control and continuous operation making them ideal for demanding environments. Their compact design, variable displacement capability enhances their efficiency while ensuring reliable performance. Typical applications include hydraulic drive systems, injection molding machines and power generation. Their ability to work under extreme conditions which makes them a preferred choice for industries that require high-performance hydraulic solutions.

Axial piston pumps consist of several key components that enable their efficient operation:

- **Cylinder Block:** The piston barrel called as cylinder block houses multiple pistons and features central cavities that allow hydraulic fluid movement. As the pistons reciprocate, they generate the pumping flow, enabling efficient fluid displacement in the hydraulic system.

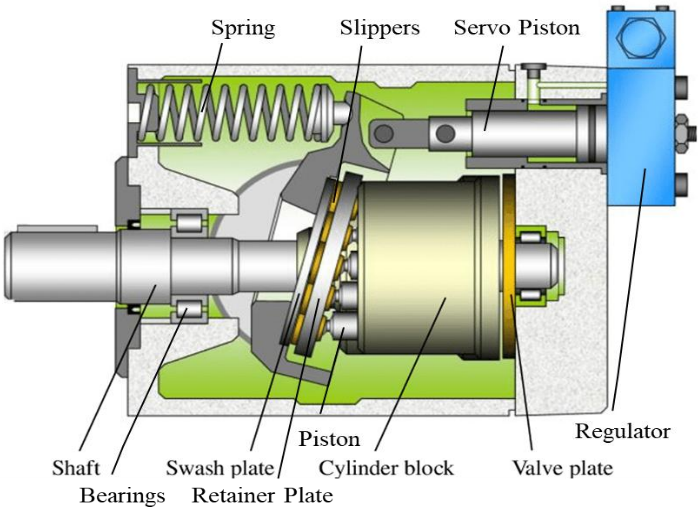

Fig. 1.1. Sectional view of an axial-piston pump with key components labelled.[4]

- **Shaft:** It is part of a rotatory/driven element, rod-like structure having a spline that is connected to a cylinder block to rotate together.
    - It is driven by an external body such as a motor.
    - It is mounted or rested on bearings inside the housing.

- **Swashplate:** It is a flat disc shaped component attached to a shaft with the help of bearings. In an axial piston pump that uses a swash plate mechanism and this plate sets the tilt angle of the piston housings that influencing the piston stroke length. By adjusting swash plate angle the output of the pump and its performance can be accurately regulated and controlled.

- It is a plate on which a piston shoe with a retaining plate moves.
- So due to the continuous movement of the piston shoe on the plate friction develops, therefore small hole is provided on the shoe which lubricates the plate and reduces wear.

- **Pistons:** The pistons within the pump revolve around the central shaft. Because one end of each piston is attached on a plane angled by the swash plate and thus the pistons change their distance from the valve plate during rotation. This changing distance results in a constant shift in the volume available for hydraulic fluid inside the piston cylinder chamber. As a result pistons repeatedly go through the intake and discharge cycle which enabling a continuous pumping operation.

- **Bearing:** A bearing is used to rest the shaft in conjunction with the cylinder block, swash plate, retaining plate, and pistons. It causes the drive shaft to move or rotate freely without friction.
- **Valve plate:** valve plate is attached at the end of the cylinder block opposite of the pistons. It has precisely machined kidney shaped slots which regulate hydraulic fluid flow, directing it to designated intake and discharge ports. As the piston barrel rotates the valve plate ensures a seamless transition between high and low-pressure zones, optimizing pump efficiency. Its design plays a

crucial role in maintaining smooth, controlled hydraulic operation in high-performance systems.

- **Retaining plate:** Hydraulic piston pump's retainer plate made to secure and retain the pistons inside the rotating cylindrical barrel of the Axial piston pump assembly. It has equal holes respective to piston which allows them to slide inside the cylinder block when pump operates. This plate helps the pistons to remain in their respective slots while having the necessary motion inside the cylinder block of the pump. It also prevents the pistons from drive out or shifting during pump operation ensuring proper pump functionality and hydraulic fluid flow.

- **Spring:** In axial piston pump the spring in the comparator plays a crucial role in pressure control. It provides a resistance force against hydraulic pressure helping maintain balance in the control mechanism. As pressure increases, it compresses the spring, adjusting the swashplate angle to reduce pump displacement. This ensures stable operation and prevents over-pressurization. Typically, a precision-calibrated compression spring is used for accurate and reliable pressure compensation.

- **Servo piston:** In an axial piston pump the servo piston works with the comparator to control the swashplate angle. It responds to changes in hydraulic pressure while moving the swash plate to adjust pump displacement.

The spring in the comparator provides a balancing force while the servo piston moves based on the pressure difference.

- **Slipper:** The slipper is a circular pad attached to the end of each piston in an axial piston pump. It has a central recess or pocket surrounded by land and an orifice connects the pocket to the piston bore. The slipper is often slightly non-flat with a parabolic or chamfered profile. It may feature an inner and outer land with fluid grooves between them. Slipper can consist of a spherical joint are retained using swash plates.

- **Regulator:** In an axial piston pump regulator is often a pressure compensator that controls output flow of the pump and pressure by adjusting the swash plate angle. This swash plate influences the discharge of fluid delivered by each piston and thus regulating flow and pressure. The regulator can be mechanically or electronically controlled by allowing for precise adjustment of the pump performance.

  Axial piston pumps can use different types of regulators including:

  - Mechanical Pressure Compensators: These depends on the springs and levers to adjust the swash plate angle in response to pressure changes.

- Electronic Control Systems: Sensors, microcontrollers, and actuators are used here to control the swash plate angle enabling precise and adaptive control of flow and pressure.

1.3. Working Principle

An axial piston pump operates by rotating a cylinder block containing pistons aligned parallel to the drive shaft. As the cylinder block turns pistons move in and out of their bores due to the angled swash plate drawing in and expelling hydraulic fluid with displacement adjusted by changing the swash plate angle.

Axial piston pump is one type Variable displacement Hydraulic pumps which we are using is operated by adjusting the amount of hydraulic fluid they deliver based on system demand. Unlike fixed displacement pumps which produce a constant flow regardless of the load with variable displacement pumps can change their output by altering the internal configuration of their components. A key element in this operation is the swash plate which is a tilted plate connected to pistons. By changing the angle of the swash plate stroke length of the pistons varies which in turn adjusts the volume of fluid displaced per cycle. Another crucial component is the pressure compensator also called the regulator which senses pressure changes in the hydraulic system and automatically adjusts the displacement to maintain the desired pressure and flow.

This ability to regulate output makes these pumps highly energy-efficient, as they supply only what is necessary at any given moment.

Additional elements like check valves and control valves help manage fluid direction and prevent backflow, ensuring smooth and reliable operation. Overall the working principle focus on dynamic flow control, enhancing performance, efficiency and system longevity.

1.4. Applications

Hydraulic pumps are the essential components in various industries such as agriculture, constructions and manufacturing and converts mechanical energy into hydraulic energy by pressurizing and transporting fluid enabling actuators to perform mechanical work. Their function ensures efficient force transmission and motion control in systems requiring precise power delivery and fluid regulation. There are various applications as mentioned below.

1) **Construction Equipment:** Hydraulic pumps power excavators, loaders and cranes by delivering high force for lifting and digging. Their precision and reliability are essential for material handling, earthmoving and structural work in dynamic and rugged construction environments.

2) **Agricultural Machinery:** Used in tractors, sprayers, and log splitters, hydraulic pumps drive implements and

attachments. They enable efficient soil preparation, harvesting, and wood processing reducing manual labour and enhancing agricultural productivity across various field operations.

3) **Automotive Industry:** Hydraulic power presses, brakes and steering systems in vehicles. They ensure precise control, safety and comfort through shock absorbers and lifting systems, while supporting manufacturing through hydraulic forming and metal shaping processes.

4) **Manufacturing & Automation:** Hydraulic pumps operate presses, injection molding and robotic systems. They provide consistent force and precise motion control, enabling high-efficiency and high-volume production in modern automated industrial facilities and heavy-duty processing lines.

5) **Aerospace and Defence:** Hydraulic systems actuate flaps, landing gear and control surfaces in aircraft. They ensure responsive, reliable operation in extreme conditions, supporting both flight dynamics and military ground or airborne equipment with compact and powerful actuation.

6) **Marine Industry:** Used in steering gears, winches and hatch covers, hydraulic pumps provide smooth, powerful control on ships. They enhance manoeuvrability, cargo handling and marine operations even under corrosive and high-load maritime conditions.

7) **Mining Operations:** Hydraulic pumps drive crushers, drills and shovels, delivering constant power for extraction and material movement. They perform reliably in harsh, remote environments requiring rugged, continuous-duty systems for demanding mining operations.

8) **Energy Sector:** Pumps control wind turbine blades, hydroelectric gates and drilling rigs. Hydraulic actuation ensures safe, efficient operation in renewable and fossil energy systems, regulating pressure, motion and safety mechanisms with precision.

9) **Material Handling Equipment:** Forklifts and lift tables use hydraulic pumps for lifting and transporting heavy loads. These systems provide controlled movement and efficient load handling in logistics, warehouses and industrial settings.

10) **Hydraulic Presses:** Used in stamping, forming and molding, hydraulic pumps enable uniform pressure distribution. They support precision manufacturing in automotive, aerospace and metalworking industries requiring high force and repeatability.

11) **Earthmoving Equipment:** Backhoes and trenchers rely on hydraulics for digging and levelling. Pumps enable strong, responsive actuation needed for soil displacement, site preparation and road construction work.

1.4. Various defects in the Axial piston pump

Axial piston pump defects including wear, cavitation, seal leakage and overheating. These issue reduces efficiency and lifespan and are often caused due to contamination, poor lubrication or high system pressure. Regular maintenance prevents failures. So, various defects happen in various parts due to the failure of components mentioned in Table 1.1:

Table 1.1. Various Defects in the Axial Piston Pumps

Sr. No.	Components	Effects	Causes	Remedies
1.	Piston-Cylinder	Internal leakage, loss of pressure, reduced flow	Abrasion, Wear or scoring, Excessive operating temperature, Improper material tolerance	Improve lubrication, replace parts, and conduct regular inspections
2.	Swash plate	Inconsistent or reduced displacement	Surface wear, improper angle adjustment	Replace worn swash plates, Check and recalibrate, Lubricate pivot

3.	Slipper	Loss of hydraulic sealing, reduced volumetric efficiency, increased wear on swash plate	Excessive wear, poor lubrication, contamination, or misalignment	Replace worn slippers, ensure proper lubrication, Inspect for piston misalignment
4.	Valve plate	Improper timing of fluid flow, cavitation	Erosion, wear from high-pressure drops, dirty fluid	lap valve plate, maintain cleanliness, Inspect for proper sealing
5.	Bearings	Noise, vibration, potential shaft misalignment	Overload, contamination, lubrication failure	Replace, Use correct grade of lubricant, prevent contamination, Ensure proper alignment
6.	Seals	External leakage	Wear, aging, improper installation	Replace, monitor seal condition

There are other types of failures apart from these, which are given below:

- **Piston-shoe wear or detachment**
    a) **Cause:** High load, poor lubrication, or contamination.
    b) **Effect:** Loss of contact with the swash plate, resulting in loss of displacement and erratic flow.

- **Shaft Seal Leakage**
    a) **Cause:** Worn seals or excessive shaft movement.
    b) **Effect:** Fluid leaks, contamination ingress, pressure loss.

- **Cylinder Block cracking**
    a) **Cause:** Hydraulic shock or manufacturing defect.
    b) **Effect:** Catastrophic failure and complete pump breakdown.

- **Overheating**
    a) **Cause:** Internal leakage, low fluid levels, or cooling system failure.
    b) **Effect:** Reduced efficiency, accelerated wear, and component distortion.

- **Cavitation**
    a) **Cause:** Low inlet pressure or blocked suction line.
    b) **Effect:** Pitting of internal surfaces, noise, and loss of efficiency.

1.5. Problem Statement

Axial Piston Pumps (APPs) are built with the extremely fine tolerances between the piston and the cylinder to ensure the proper lubrication and minimal internal leakage. This precision allows them to maintain a tight seal even at high operating speeds typically ~1500 rpm. These pumps operate under high pressure conditions which makes them efficient but also more prone to wear. Thus close fitting components can be easily damaged by contaminants in the hydraulic fluid such as dirt or metal particles. Such wear increases the clearance between moving parts leading to internal leakage and reduced performance. To ensure long term reliability maintaining a clean fluid and utilizing proper filtration are essential.

1.6. Research Objectives

The primary objective of this research is to investigate internal leakage characteristics in an axial piston pump as influenced by the annular gap between the piston and cylinder mating surfaces, aiming to enhance failure prediction and condition monitoring methodologies. Controlled annular clearances, specifically 50 µm, 80 µm, 100 µm, and 180 µm will be introduced through precision machining to simulate varying degrees of wear or dimensional deviation. A mathematical model will be developed based on the Navier Stokes and continuity equations to predict leakage flow through these micro gaps. The model will account for both pressure driven and shear driven flow components enabling a comprehensive understanding of fluid transport in the annular domain.

Theoretical leakage predictions will be validated through experimental measurements obtained under controlled operational conditions. Furthermore, the study will quantify the proportion of shear driven (Couette) and pressure driven (Poiseuille) flow across different gap sizes and operating conditions, providing insight into the transition of dominant flow mechanisms with wear progression. Comparative analysis between experimental and model based results will support model refinement and enhance reliability. Ultimately this research aims to establish leakage behaviour as a quantitative indicator for early detection of wear enabling predictive maintenance strategies in axial piston pump systems.

1.7. Thesis Structure

Chapter 2. Literature Survey: Researchers have explored leakage faults in axial piston pumps. Gupta *et al.* modeled performance degradation from leakage in single and multiple cylinders. Pathak *et al.* studied leakage due to increasing piston-cylinder eccentricity. Bergada *et al.* experimentally analysed dynamic pressure and flow ripples, validating leakage models. These studies highlight the impact on pump efficiency and performance.

Chapter 3. Mathematical Modelling: Mathematical modeling analyzes axial piston pump leakage using assumptions and sinusoidal piston motion. Velocity and individual piston leakage are derived via the Navier-Stokes equation, while total discharge sums contributions. Piston stroke is calculated geometrically using the swashplate angle, aiding accurate leakage and flow predictions.

- **Chapter 4. Model Validation & Results:** This section presents the mathematical validation and results, including sensitivity analysis of the swash plate angle, evaluation of model parameters, and comparison of model predictions with experimental discharge data across various leakage gaps. It also compares velocity profiles for pressure- and shear-driven flow under different leakage conditions.

- **Chapter 5. Conclusion and Research Scope:** The study confirms the model's accuracy in predicting discharge and flow velocity across leakage gaps, highlighting the swash plate angle's impact and the significance of pressure and shear-driven flow components. Future research can investigate the effects of Reynolds number variation due to leakage severity and geometric modifications in pistons (tapered, spherical, eccentric) and extend the modeling approach to other axial piston pump components using the same governing equations.

Chapter 2

Literature Survey

2.1. Overview

This study focuses on internal leakage in the annular gap of piston-cylinder assemblies in variable-displacement axial piston pumps. While previous research has explored leakage effects caused by wear therefore the underlying flow dynamics within the gap remain insufficiently understood. A thorough review of existing literature reveals critical gaps and highlighting the need for further investigation in this area. In particular the interaction between pressure driven and shear driven flow components during piston motion has not been adequately studied. This research aims to analyse these flow behaviours via assessing their variation with increasing gap size and enhance leakage prediction and condition monitoring capabilities.

2.2. Background

Failures in the piston and cylinder interface of axial piston pumps often result in fluid leakage subsequently diminishing both efficiency and overall system performance. To improve the longevity and dependability of these pumps researchers have conducted extensive investigations into the root causes of such failures. Studies cited in references [1-12] explore the mechanical behavior of pump components under fault conditions, with a specific emphasis on leakage intensity and its implications. Therefore an approach to studying leakage was presented by Burton et al. [13], who introduced “artificial leakage” using a servo valve with high-frequency responsiveness. This valve diverted a regulated portion of the pump’s outlet flow directly to the tank, simulating leakage similar to that from a degraded piston. By modulating the flow in a controlled waveform, the system effectively replicated piston leakage from the high-pressure chamber to the low-pressure drain side. Control algorithms mimicked wear at various levels in a single piston, providing a testbed for leakage analysis. Bergada et al. [14] contributed by initially formulating equations to quantify leakage across all key interfaces in piston pumps. These analytical models were then validated against numerical simulations. Following validation, the equations were linked to describe the dynamic relationships between pressure and flow within the pump. The comprehensive model estimates both instantaneous chamber pressures and leakages across all pump clearances over time. Kumar S. [15] introduced a finite volume-based Reynolds equation model focused on the piston–cylinder clearance. The model incorporated the relative motion between the piston and barrel. Several groove geometries were examined to determine optimal designs that reduce leakage, minimize cavitation risk, and maximize restoring torque. This model aids in refining piston design to boost efficiency. D. Bensaad et al. [16] presented an innovative fault diagnosis framework using an Extended Kalman Filter (EKF). Their model based technique identifies pistons suffering from internal leakage by estimating chamber pressures using a nonlinear dynamic model. The EKF tracks real time pressure changes and generates residuals to detect anomalies and isolating faulty pistons. Tested on a HYDAC experimental platform, the model proved highly accurate, even with multiple concurrent faults, offering a cost-effective approach to predictive maintenance. Rishabh et al. [17] conducted a study utilizing simulation and mathematical modeling to assess how increased leakage severity affects one or more cylinders in a pump. Leakage was modeled as laminar flow through an annular clearance between the piston and cylinder. Their findings indicated that as the number of faulty pistons increased, the system's critical pressure threshold decreased until stabilizing after four defective cylinders. Dynamic analysis of outlet pressure signals revealed that leakage altered the waveform's characteristics, particularly during the discharge phase. Fast Fourier Transform (FFT) was used alongside average pressure and flow readings for leakage diagnosis. Additionally, Rishabh's [18] work proposed a model-based methodology to reorganize the arrangement of faulty pistons to reduce pressure fluctuations. A MATLAB Simulink model simulated pump behavior under varying leakage levels (20 μm, 60 μm, 100 μm) and piston configurations, such as inline, random, and uniform arrangements. Experimental data supported the model's accuracy. Fault configuration was assessed using a configuration ratio, a normalized spatial metric reflecting piston placement. Results showed that uniformly distributing faulty pistons minimized peak pressure swings and associated vibrations. Guanglin et al. [19] made important advances by developing a thermal-hydraulic simulation model that predicts temperature behavior and evaluates pump performance under various thermal conditions. Their study concentrated on the heat generated at lubricated interfaces using energy transformation principles and modeled heat transfer via the control volume technique. Implemented in AMESim due to its suitability for thermal-fluid applications, the model demonstrated that temperature changes had minimal influence on mechanical losses but significantly impacted volumetric efficiency. They concluded that modifying piston design considering material selection, surface area, and thermal properties could effectively reduce thermal buildup and improve efficiency. Despite substantial progress in modeling and simulation, a gap remains in the current literature. Specifically, a detailed analysis of the annular clearance region using a hybrid Couette–Poiseuille flow model is lacking. Such an investigation would clarify the velocity distribution of leakage and identify the contribution of shear driven (Couette flow) versus pressure driven (Poiseuille flow) components. Understanding these two flow is essential to improve the energy loss estimates which refine leakage models and optimize the overall efficiency of axial piston pumps. In summary a range of analytical and experimental and simulation based studies have enriched our understanding of axial piston pump failures caused by piston and cylinder leakage. These include detailed modeling of fluid dynamics within the pump, fault diagnosis techniques, and thermal performance assessments. However, the integration of combined Couette and Poiseuille flow analysis within the leakage pathway remains an unexplored frontier. Filling this research gap could lead to more precise leakage modeling, enabling improved system performance, reduced energy dissipation, and enhanced pump reliability.

2.3. Research Objectives

Axial piston pump are essential in various industrial applications for its performance depending on the integrity of key components such as the piston and cylinder interface. Regular maintenance is essential to prevent costly breakdowns and ensure long term functionality. In the case under review wear at the piston and cylinder interface was identified as the primary cause of malfunction leading to fluid leakage thus reduced efficiency and increased strain on the pump. This wear compromised the pressure generation of the pump and flow that resulting in operational inefficiencies and overheating. This highlights the importance of routine inspections to detect and address wear before it escalates and optimizing performance and extending the pump's lifespan. To ensure optimal performance under these conditions this study proposes a methodology to quantify the overall discharge of a hydraulic pump with a compromised piston and cylinder interface by utilizing a mathematical model and relevant boundary conditions. By identifying the failure modes in the gap between piston-cylinder and correlating them with the severity of leakage therefore the study aims to minimize wear induced factors that contribute to internal leakage and vibrations. This model facilitates the determination of the dominant flow characteristics responsible for pump failure and enabling the enhancement of pump reliability and energy efficiency. Ultimately this approach contributes to the improvement of the overall performance and longevity of hydraulic pumps.

The following aims have been set for the proposed study:

- Develop a mathematical model that integrates Poiseuille flow and Couette flow using the Navier Stokes equations having appropriate boundary conditions in the cylindrical coordinates. The model will include velocity, leakage, and discharge equations. This unified equation after combining leakage and discharge phenomena in a single piston aims to replicate functional failures and operating conditions of axial piston pumps.

- Utilize the velocity equation derived from the Navier–Stokes and continuity equations in cylindrical coordinate systems to determine the velocity profile. This profile will help assess the contributions of Poiseuille and Couette flows under varying severity of leakage. The analysis will provide a brief discussion on leakage dominance.

- Investigate how the Reynolds number of the fluid under annular gaps varies with increasing leakage severity and operating speeds. This analysis will determine whether the flow behaviour remains laminar or transitions to turbulent flow under different conditions.

- Analyze failures and operational conditions to develop strategies for refining pump designs, aiming to enhance energy efficiency and operational versatility.

Chapter 3

Mathematical Modelling

3.1. Introduction

In this section of the mathematical modelling of the axial piston pump we will discuss the kinematic relationship between the rotating cylinder block and the axial motion of the pistons governed by the swashplate angle. The schematic illustrates how the pistons convert rotary motion into reciprocating motion thus resulting in fluid displacement. This analysis forms the basis for understanding flow generation and dynamic behaviour within the pump. Also, we present the governing equations for piston velocity, piston-cylinder leakage, and volumetric discharge in an axial piston pump. The developed model captures the essential functional characteristics, operational constraints, and dynamic conditions of the pump, thereby providing an accurate representation of its real-world performance. By applying the velocity equation, leakage in an individual piston-cylinder pair can be quantitatively estimated, facilitating validation against experimental data that reflects the actual failure behaviour of the pump. MATLAB is employed to implement the mathematical model and assess its accuracy by analyzing total discharge under various leakage scenarios. This methodology enables precise identification of the pump's condition during failure and supports the refinement of condition monitoring strategies, ultimately contributing to enhanced energy efficiency and improved adaptability of axial piston pumps.

3.2. Methodology for the finding of piston stroke

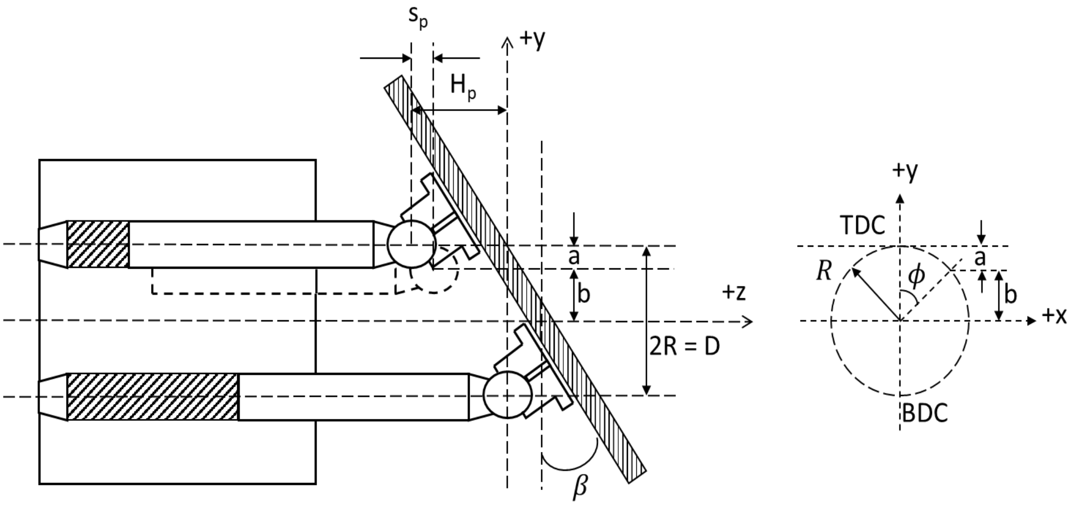

Fig. 3.1. Schematic diagram of the Axial-piston pump illustrates the relationship between piston-cylinder motion and the location of the piston in a circular array from TDC to BDC.

In the axial piston pump nine pistons are symmetrically arranged in a circular array within the rotating cylinder block. These pistons are spaced evenly with a phase angle of 40° between each and they are aligned along the central shaft commonly referred to as the driveline axis. The movement of each piston is governed by the inclination of the swashplate which is a key component in controlling the pump operation. The pistons are connected to the swashplate through slippers and as the swashplate rotates it induces a reciprocating motion in each piston. This motion is responsible for the alternating phases of fluid intake called suction and fluid expel called discharge. During operation when a piston reaches the top dead center (TDC) it initiates the suction phase and drawing fluid into the cylinder. As the piston moves toward the bottom dead center

(BDC) the discharge phase begins and pushing the fluid out of the cylinder. For mathematical modeling, it is assumed that the first piston begins its motion at TDC. This assumption simplifies the analysis by providing a fixed reference for describing the angular positions and motion of the remaining pistons throughout the rotation cycle. To maintain analytical simplicity the swashplate angle is considered constant under specific operating conditions via including a fixed annular gap and a given system pressure. While this assumption facilitates easier modeling thus it does not fully capture the complexities of real world pump operation such as transitions between laminar and turbulent flow regimes or fluctuations in internal pressure. Furthermore the flow within the pump is assumed to be laminar for this study. Although idealized this assumption allows for a clearer understanding of the pistons coordinated motion and the fundamental mechanics of fluid displacement in the pump. Overall the model provides essential insights into the swashplate-driven synchronization of suction and discharge across all pistons.

The coordinate system and geometric parameters illustrated in Fig. 3.1. are utilized as the basis for evaluating the piston stroke and velocity as follows:

$$S_P = R \cdot tan\,\beta \cdot (1 - cos\,\phi) \tag{3.1}$$

$$u_P = \dot{S}_P = \frac{dS_P}{dt} = \frac{ds_P}{d\phi} \cdot \frac{d\phi}{dt} = \frac{ds_P}{d\phi} \cdot \omega \tag{3.2}$$

$$u_P = R \cdot \omega \cdot \tan\beta \cdot \sin\phi \tag{3.3}$$

where, $\phi = \omega t$

3.3. Assumptions

The formulation of the Axial piston pump's mathematical model is based on the following assumptions and simplifications are as follows:

1. The Hydraulic working fluid properties having viscosity, density, and bulk modulus are assumed to be independent of temperature and remain constant throughout the operation of pump.

2. Flow is laminar and the fluid's inertia is neglected.

3. The gap between the piston and the cylinder is assumed to be annular. Leakage is assumed to occur only through the annular gap between the piston and cylinder surface of the axial piston pump.

4. All other faults such as those resulting from the slipper, valve-plate, and swash-plate leakage are being neglected.

5. No eccentricity is assumed between the piston and the cylinder annulus of APP's.

6. The pump operates at a constant rpm and for a given annular gap and system pressure the swash plate angle is assumed to be constant.

3.4. Mathematical Model for Leakage

In this section analytical models for both leakage and discharge are systematically developed by idealizing the annular gap between the piston and cylinder as a concentric annulus. This assumption implies zero eccentricity in the radial direction by simplifying the geometry and allowing for a tractable solution using classical fluid dynamics principles. The resulting models quantify the volumetric flow through the clearance which comprises both the intended discharge and the undesired leakage component. A unified expression is formulated to compute the net discharge from the axial piston pump by integrating the effects of both components. The derivation of the leakage flow is based on the application of fundamental fluid mechanics principles. Specifically the mass conservation law (continuity equation) and momentum conservation law (Navier-Stokes equations) are applied in cylindrical coordinates which appropriate for the annular geometry of the piston and cylinder assembly. The fluid is assumed to be incompressible and Newtonian and the flow is considered laminar, steady, and axisymmetric. Additional assumptions include negligible inertial forces due to the small scale of the clearance and the dominance of viscous effects. Under these assumptions the Navier–Stokes equations are significantly simplified thus allowing the derivation of an analytical expression for the velocity profile and volumetric flow rate through the annular gap. The flow is typically driven by a combination of pressure gradient (axial direction) and piston motion (resulting in Couette flow due to wall shear). The superposition of these two mechanisms governs the net leakage rate which varies as a function of gap height, fluid viscosity, pressure difference, and piston velocity.

The velocity equation for leakage and leakage flow can be derived under the following assumptions in the Navier-Stokes Equation and the Continuity Equation are as follows:

a) Neglecting gravity

b) $\left(\frac{\partial}{\partial \theta} = 0\right)$ axial symmetry

c) Radial and angular velocity components are zero, (i.e., $u_r = 0$, $u_\theta = 0$) for fully developed laminar flow through the annulus.

$$\therefore \frac{\partial u}{\partial x} = 0 \text{ , u = u(r) only} \tag{3.4}$$

$$\therefore \frac{\partial P}{\partial r} = 0 \text{ , P = P(x) only} \tag{3.5}$$

Applying all these assumptions to the Navier-Stokes equation in cylindrical coordinates, it simplifies to:

$$\frac{dP}{dx} = \frac{\mu}{r}\frac{d}{dr}\left(r\frac{du}{dr}\right) \tag{3.6}$$

Integrating equation (3.6) twice w.r.t r, the axial velocity equation is derived as follows:

$$u(r) = \frac{Kr^2}{4\mu} + K_1 \ln r + K_2 \tag{3.7}$$

where, K = $\frac{\partial P}{\partial x}$ = (constant) < 0.

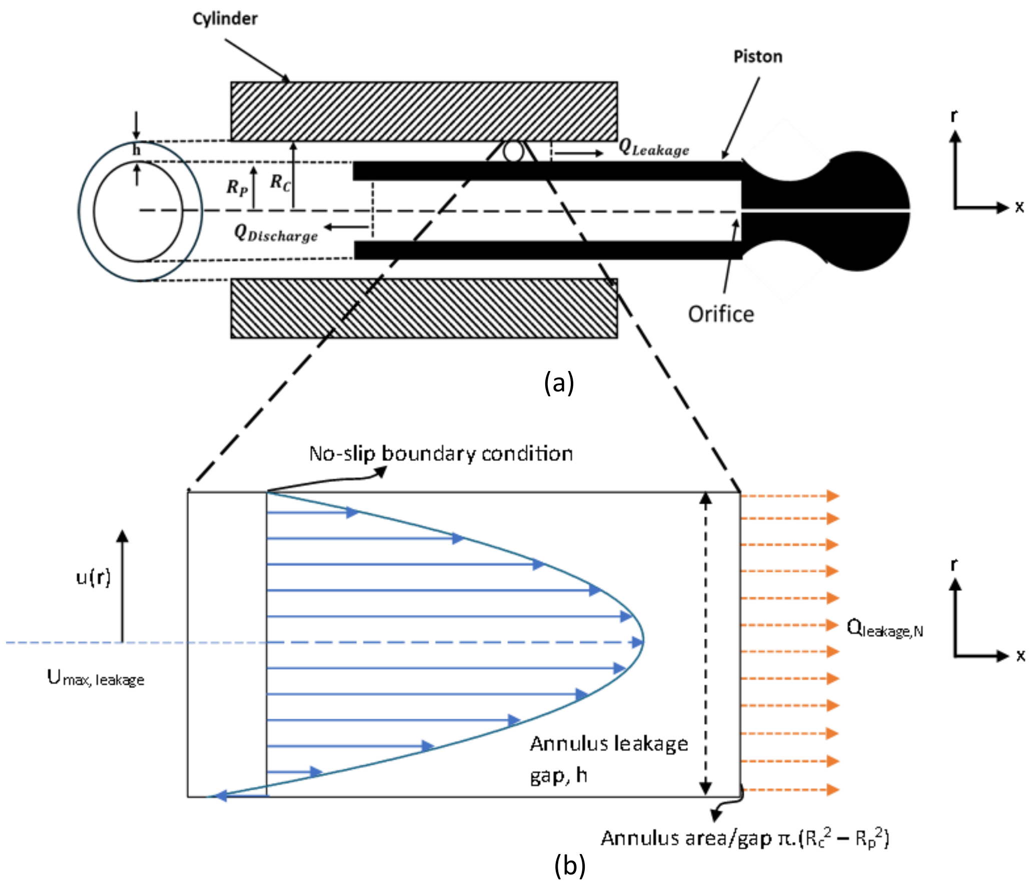

Fig.3.2. (a) Sectional view of piston-cylinder arrangement illustrates the fluid leakage ($Q_{Leakage}$) through the annular gap width h between the piston and cylinder, along with discharge flow ($Q_{Discharge}$). (b) It shows the leakage flow in-depth and the fluid's velocity profile in the annular gap between the piston and the cylinder.

Sectional view of piston-cylinder arrangement provides a comprehensive visualization of the fluid flow phenomena within the piston-cylinder interface of an axial piston pump. Sectional view of the assembly which illustrating the occurrence of fluid leakage through the narrow annular gap between the piston and the cylinder wall. This gap denoted by h which is responsible for permitting a portion of the working fluid to bypass the intended flow path, resulting in a leakage flow rate $Q_{Leakage}$. Simultaneously, the figure shows the discharge flow $Q_{Discharge}$, which represents the useful flow delivered by the piston. Enlarged view of the leakage flow region emphasizing the velocity distribution of the fluid within the annular clearance. In this configuration the outer boundary corresponding to the inner surface of the stationary cylinder is located at radius ($r = R_c$) and has zero velocity ($u = 0$) while the inner boundary defined by the surface of the moving piston at radius ($r = R_p$) which moves with a constant velocity ($u = -u_p$) in the axial direction. The fluid flow in this region is governed by the superposition of pressure-driven and shear-driven mechanisms, resulting in a velocity profile that combines characteristics of Poiseuille and Couette flow.

Using the boundary conditions mentioned above at the surface of the piston and cylinder for the piston-cylinder arrangement, the constants can be determined as given below:

$$K_1 = \frac{u_P}{\ln\left(\frac{Rc}{R_P}\right)} \tag{3.8}$$

$$K_2 = \frac{-u_P.\ln R_C}{\ln\left(\frac{Rc}{R_P}\right)} \tag{3.9}$$

The final velocity equation for the leakage flow obtained is as follows:

$$u(r) = -\frac{K}{4\mu}\left[(h + R_P)^2 - r^2 - \frac{(h+2hR_P)}{ln(1+\alpha)} \cdot ln\left(\frac{h+R_p}{r}\right)\right] + \frac{u_P}{ln(1+\alpha)} \cdot ln\left(\frac{h+R_p}{r}\right) \tag{3.10}$$

where, $\alpha = \frac{h}{R_P}$

The velocity distribution u(r) within the annular gap of a cylindrical piston-cylinder system can be described by an analytical expression that accounts for two flow mechanisms. The first is pressure-driven flow, commonly referred to as Poiseuille flow, which arises due to an axial pressure gradient (K). This component results in a parabolic velocity profile characteristic of viscous flow in confined cylindrical geometries. The second mechanism is shear-driven flow, or Couette flow, which is induced by the relative axial motion of the piston. As the piston translates with a constant velocity u_p, it exerts tangential shear forces on the adjacent fluid layers, generating a linear velocity distribution superimposed on the pressure-driven component. The combined effect of these two mechanisms produces a combined velocity profile u(r) that captures the full dynamics of the fluid within the annular clearance.

Using Eq. (3.10), the leakage flow equation can be determined as follows:

$$Q_{Leakage} = \frac{-K \cdot \pi}{8\mu} \cdot hR_P^3(2+\alpha)\left[1 + (1+\alpha)^2 - \frac{(\alpha)(2+\alpha)}{\ln(1+\alpha)}\right] + u_P \cdot \pi \left[R_P^2 - \frac{hRp(2+\alpha)}{2\ln(1+\alpha)}\right] \quad (3.11)$$

The derived equation for leakage flow $Q_{Leakage}$ in a piston-cylinder system integrates two fundamental flow components, pressure-driven and shear-driven mechanisms, in a cylindrical coordinate framework. The first component represents classical Poiseuille flow, driven by an axial pressure gradient (K) , and incorporates corrections specific to cylindrical geometry through logarithmic terms. This term accounts for the influence of both the piston radius R_p and the radial clearance h, ensuring that geometric curvature effects are properly modeled. The second component corresponds to Couette flow, generated by the axial motion of the piston at a velocity u_p. This contribution addresses the fluid shear induced by the moving boundary and incorporates modifications for the curvature of the annular domain. By combining these two mechanisms the model provides a more comprehensive representation of leakage flow compared to traditional formulations. Earlier studies often assumed a planar gap and considered only pressure driven flow neglecting both curvature effects and piston induced shear. Such simplifications may lead to significant deviations in predicted leakage and particularly in systems with tight tolerances and dynamically moving pistons. The present formulation then by applying appropriate boundary conditions in cylindrical coordinates and capturing both pressure and shear influences which offers enhanced predictive accuracy for leakage behavior in practical hydraulic applications where small clearances and piston motion are critical factors.

3.5. Mathematical Model for Pump's Discharge

The reciprocating motion of the pistons facilitates two fundamental stages within each complete pump cycle the suction stroke and the discharge stroke. During the upward movement of a piston toward its top dead centre (TDC) a reduction in chamber pressure induces fluid intake that marking the suction phase. In contrast as the piston descends toward its bottom dead centre (BDC) then the enclosed fluid is pressurized and expelled which defines the discharge phase. For dynamic modelling the initial phase reference is established by designating the first piston at TDC at time zero. This serves as the temporal baseline for defining the phase offsets of the remaining eight pistons which enabling a systematic and consistent representation of their cyclic motion. For this analysis the internal flow within the piston and cylinder interface is assumed to remain within the laminar regime by ensuring the applicability of classical viscous flow models and avoiding complications associated with turbulent flow behaviour. Simplified model provides a foundational understanding of the coordinated motion of the piston and operation of the pump that emphasizing the role of the swashplate in synchronizing suction and discharge phases across the array of pistons.

The coordinate system and geometric parameters, as depicted in Fig. 3.1, are considered in the derivation of the net discharge flow for the axial-piston pump. Using equation (3.3) Discharge flow can be determined as follows:

$$Q_P = u_P \text{ . Area} = \frac{\pi\, d^2}{4} \cdot R \cdot \omega \cdot tan\, \beta \cdot sin(\omega t) \qquad (3.12)$$

The discharge flow from each piston is delayed by the phase angle with the flow rate of the i^{th} piston given by Eq. (3.13):

$$Q_P = \frac{\pi\,\mathrm{d}^2}{4} \cdot R \cdot \omega \cdot tan\,\beta \cdot sin(\omega t - (i-1)\gamma) \quad (3.13)$$

The net discharge flow equation, which accounts for the combined output from all nine pistons is given by Eq. (3.14)

$$Q_d = \frac{\pi\,\mathrm{d}^2}{4} \cdot R \cdot \omega \cdot tan\,\beta \cdot \sum_{\mathrm{i}=0}^{n-1} \; sin(\omega t - i\gamma) \quad (3.14)$$

where, $\gamma = \frac{2\pi}{m}$ (radians)

Equation (3.14) determines the net discharge flow rate (Q_d) of an axial piston pump by aggregating the individual flow contributions from multiple pistons arranged in a circular configuration. Each piston undergoes reciprocating motion governed by a sinusoidal function, reflecting the inclined swash plate mechanism that drives its axial displacement. The pistons are angularly phased related to one another which typically spaced uniformly around the cylinder block that resulting in a temporally astonish flow contribution. This phase offset clarifies that the cumulative flow output remains continuous and exhibits minimal pulsation. The overall discharge flow rate is influenced by key parameters such as stroke length, radius of piston, number of piston, rotational speed and swash plate angle as well as the dynamic interaction between the piston and cylinder geometry under varying operating conditions.

3.6. Summary

This chapter develops a mathematical model of an axial piston pump focusing on velocity, leakage and discharge flow characteristics. It derives the piston motion using sinusoidal functions based on the pump geometry and rotational configuration. Leakage through the piston and cylinder annular gap is modeled using fluid dynamics principles to quantify volumetric losses. The net discharge flow rate is calculated by summing the phase shifted contributions of individual pistons that ensuring continuous output. The model incorporates key parameters such as piston dimensions, swash plate angle and rotational speed. Schematic diagrams and geometric representations support the theoretical framework and enhance understanding of the pump's internal dynamics.

Chapter 4

Model Validation & Results

4.1. Introduction

This chapter focuses on validating experimental results with a mathematical model to study leakage and flow behavior in an axial piston pump. The analysis considers key operational conditions such as a system pressure of 100 bar and an annular gap of 100 µm to investigate fluid velocity profiles across the piston stroke from Bottom Dead Center (BDC) to the Top Dead Center (TDC). A detailed sensitivity analysis of the swashplate angle is carried out to understand its influence on the dynamic behavior of the pump and to refine the model accuracy under different working conditions. The results reveal how changes in swashplate angle directly affect flow characteristics and discharge rates. Several plots of volumetric discharge under varying pressures and annular gaps illustrate the fluid flow patterns and leakage characteristics along with model accuracy. The study also compares leakage across different gap sizes such as 50 µm, 80 µm, 100 µm, and 180 µm and thus demonstrating that pressure driven flow contributes more significantly to overall leakage than shear driven flow. The findings confirm the capability of model to represent real system behavior and provide a solid foundation for further investigations into leakage prediction with design optimization and condition monitoring of axial piston pumps used in high-pressure hydraulic applications.

4.2. Sensitivity Analysis for the Swashplate Angle

The sensitivity analysis on the swashplate angle enriches the understanding of the behavior of model under varying operating conditions. This approach also provides valuable insights into the actual functional characteristics of the swashplate mechanism.

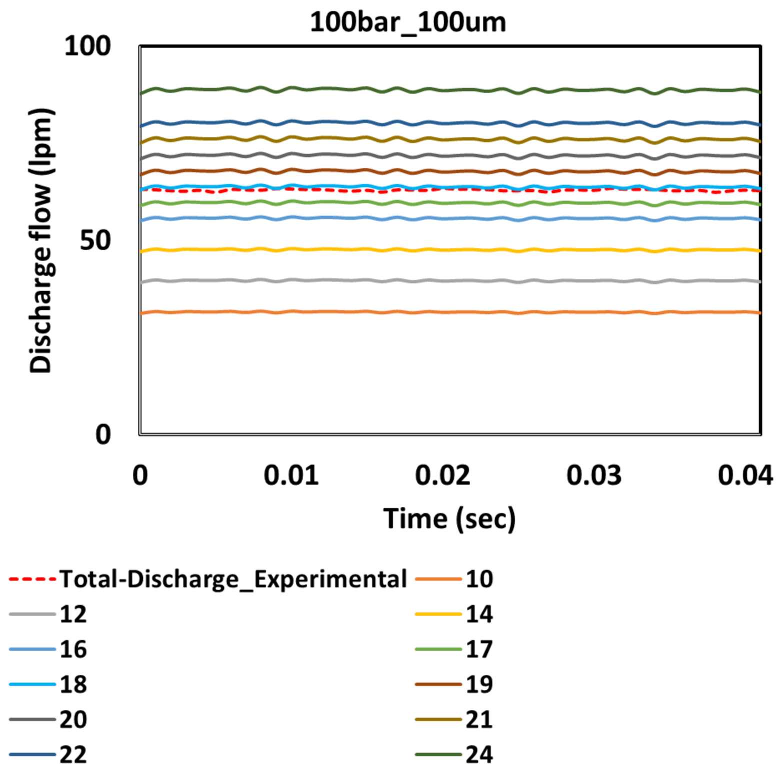

Fig. 4.1(a) Total discharge of Axial piston pump at various swashplate angles under 100 bar pressure with 100 µm annular gap.

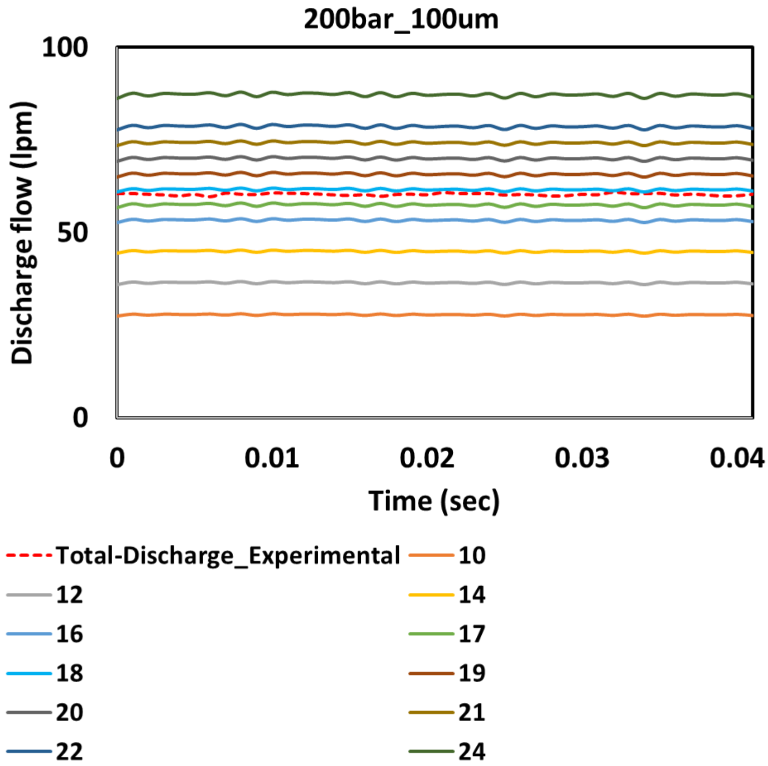

Fig. 4.1(b) Total discharge of Axial piston pump at various swashplate angles under 200 bar pressure with 100 µm annular gap.

The sensitivity analysis which is presented in the Fig.4.1(a) and Fig.4.1(b) investigates the effect of varying swashplate angles on the total discharge of an axial piston pump with a fixed annular gap of 100 µm under two different operating pressures: 100 bar and 200 bar. Therefore the results show that discharge increases consistently with the swashplate angle which reflecting the direct relationship between swashplate angle and piston stroke length.

At higher pressure (~200 bar) the discharge is notably greater for the same angles indicating enhanced volumetric efficiency under increased load conditions. The modeled discharge data were obtained for an annular gap across various swashplate angles and operating pressures. Comparison with experimental discharge measurements revealed a close match with a specific modeled swashplate angle for both case having 100 bar and 200 bar. This correlation indicates the actual swashplate angle at which the pump operates under real conditions and effectively delineates the operational range of the swashplate angle. Therefore the strong agreement between experimental and modeled data validates the accuracy and predictive capability of the model. Minor oscillations in the curves suggest the model also captures flow fluctuations likely due to internal leakage or dynamic effects. Thus overall the analysis provides valuable insights into the operational behavior of the swashplate mechanism and supports the model's effectiveness in predicting real-world pump performance.

4.3 Model Validation

4.3.1 Model Validation Parameters

To evaluate the effectiveness of the proposed model in detecting internal leakage faults a thorough validation was carried out by comparing the model of the axial piston pump for total discharge with experimental measurements obtained from a controlled test facility. The experimental setup includes an axial piston pump featuring nine piston and cylinder assemblies that replicating realistic operational conditions. The model was developed to simulate leakage behavior under identical boundary conditions and input parameters as those applied during the experimental runs. This makes a consistent and accurate basis for comparison. Total discharge from the pump was quantified at the outlet under steady-state conditions and the output of the model was assessed against these empirical results. Such a direct comparison allows for precise validation of the model predictive capability regarding internal leakage particularly in the piston and cylinder interface. The close alignment between the simulated and measured data indicates the robustness and suitability of the model for fault diagnosis and condition monitoring of axial piston pumps.

The parameters, detailed in Table 4.1., form the foundational input for the axial piston pump model, enabling it to accurately replicate real-world operating behavior.

Table 4.1. Parameters are utilized as model inputs for validation purposes.

Parameter	Properties	Symbol	Magnitude
Fluid Properties	Fluid type	-	VEEDOL 68
	Fluid temperature	T	28° C
	Density of fluid	ρ (fluid)	926.7 kg/m^3
	Kinematic viscosity	ν	68cst
Pump specifications	No. of pistons	i	9
	Cylinder pitch radius	r_p	42.64 mm
	Cylinder bore diameter	R_c	17.95 mm
	Piston radius	R_p	8.97 mm
	Piston length	L_p	37.92 mm
	Phase delay	γ	40°
	Annular gaps	h	50 µm, 80µm, 100 µm,180 µm
	Length from the center of the cylinder bore to the cylinder block center	R	30.305 mm
Operating conditions	Rotational speed	ω	1470 rpm
	Operating Pressure	P	100, 200 bar

4.3.2 Comparison of the model and experimental results

In this section a comparative analysis is presented between the total discharge predicted by the mathematical model and the discharge measured experimentally under identical operating parameters and boundary conditions. This comparison serves to evaluate the model predictive accuracy. The study involves a series of experiments in which the piston surface was machined to create annular gaps of 50 µm, 80 µm, 100 µm and 180 µm. Initially the clearance between the piston and cylinder mating surfaces was approximately 5 µm. Incremental increases in clearance were introduced deliberately to investigate leakage characteristics with the objective of developing insights for predictive maintenance strategies.

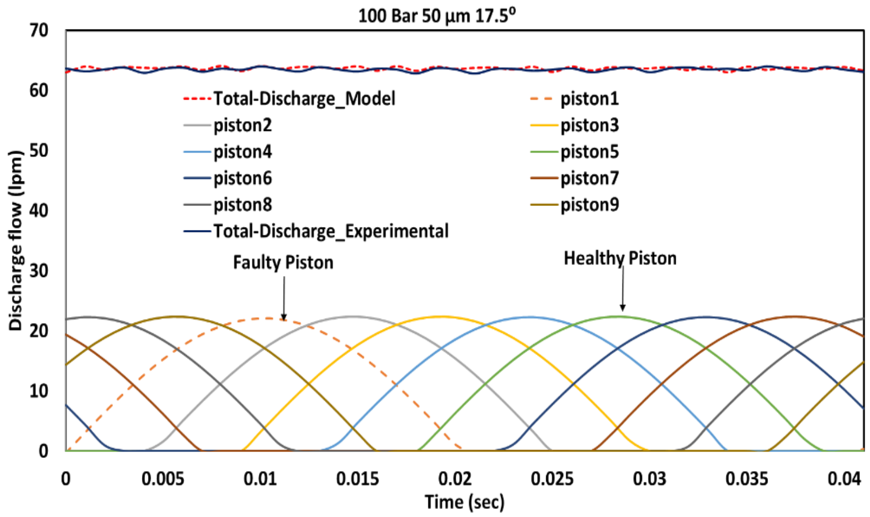

Fig. 4.2(a) Comparison of Total discharge for 50 µm annular gap at 100 bar operating pressure.

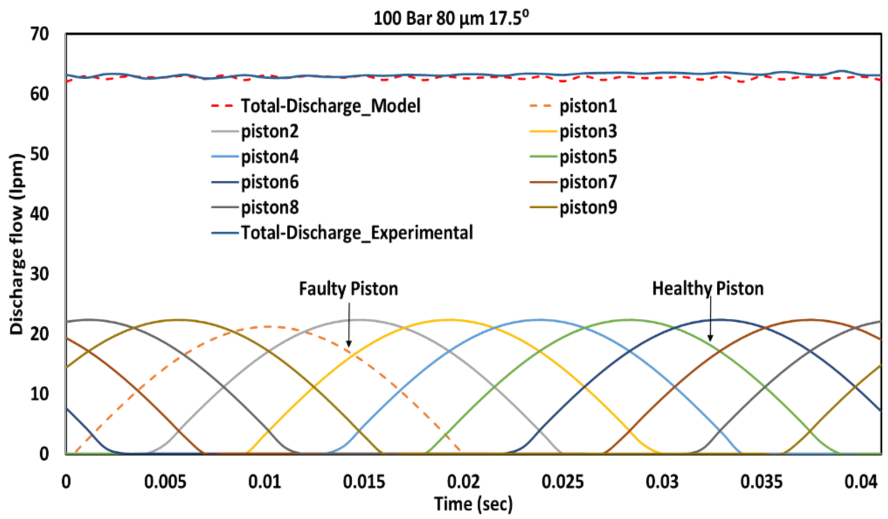

Fig. 4.2(b) Comparison of Total discharge for 80 µm annular gap at 100 bar operating pressure.

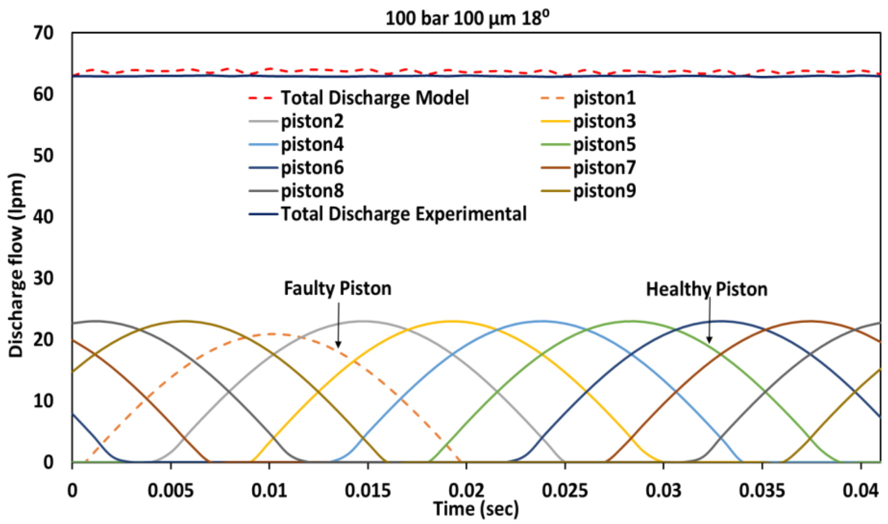

Fig. 4.2(c) Comparison of Total discharge for 100 µm annular gap at 100 bar operating pressure.

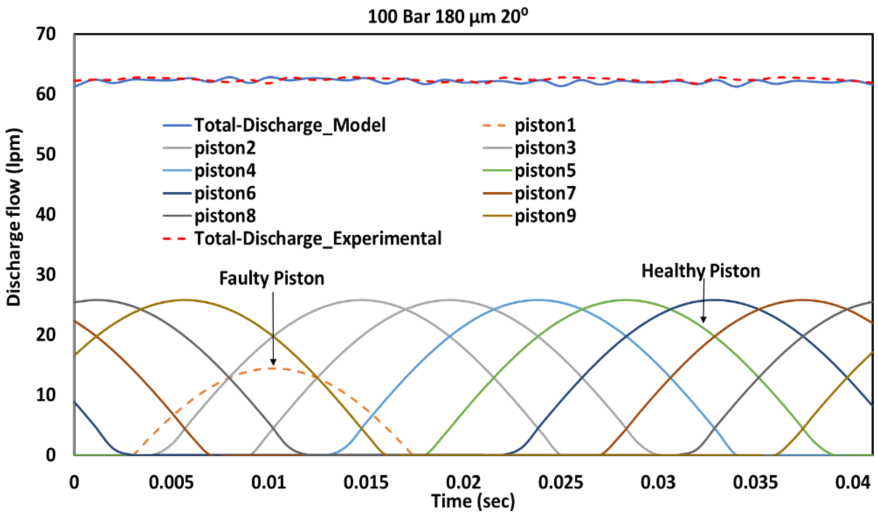

Fig. 4.2(d) Comparison of Total discharge for 180 µm annular gap at 100 bar operating pressure.

The set of graphs illustrates the impact of increasing annular gap size on the performance of an axial piston pump operating at a constant pressure of 100 bar. Ideally, axial piston pumps achieve maximum volumetric efficiency at very low clearances around 5 µm, where leakage shown in orange dotted line is minimal, and the sealing between the piston and bore is most effective. In Fig. 4.2(a), with a clearance of 50 µm and a swashplate angle of 17.5°, the pump operates above the optimal clearance, here a slight decrease in total discharge. In Fig. 4.2(b), although the swashplate angle remains unchanged, increasing the annular gap to 80 µm results in a discernible reduction in total volumetric output. This decline is attributed to elevated internal leakage through the expanded clearance between the piston and cylinder bore. Fig. 4.2(c) further illustrates the impact of leakage with a gap of 100 µm and a slightly increased swashplate angle of 18°.

The compensation in swashplate angle is insufficient to offset the intensified fluid losses, as volumetric efficiency continues to degrade. In Fig. 4.2(d), the annular gap reaches 180 µm, and the swashplate angle is increased to 20° in an attempt to maintain discharge performance. However, the leakage losses at this clearance are substantial, leading to the lowest overall discharge among all cases. These results clearly demonstrate that under constant pressure conditions (100 bar), increasing annular gap size significantly compromises pump performance. The expanded leakage path associated with larger clearances leads to reduced volumetric efficiency. Also, the comparison between experimental discharge and model-predicted discharge shows excellent agreement, validating the accuracy of the simulation model. The discharge values closely match across various annular gap sizes, confirming that the model reliably captures internal leakage trends observed in real axial piston pump operation under constant pressure conditions.

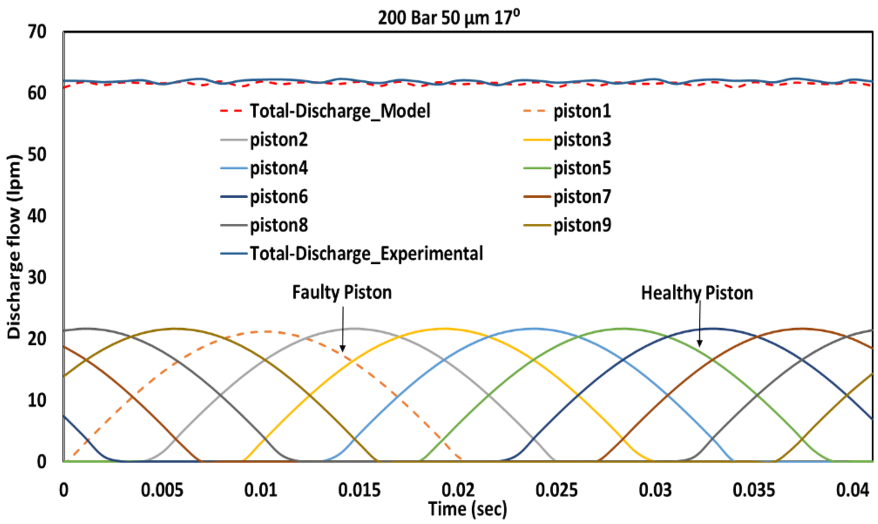

Fig. 4.2(e) Comparison of Total discharge for 50 µm annular gap at 200 bar operating pressure.

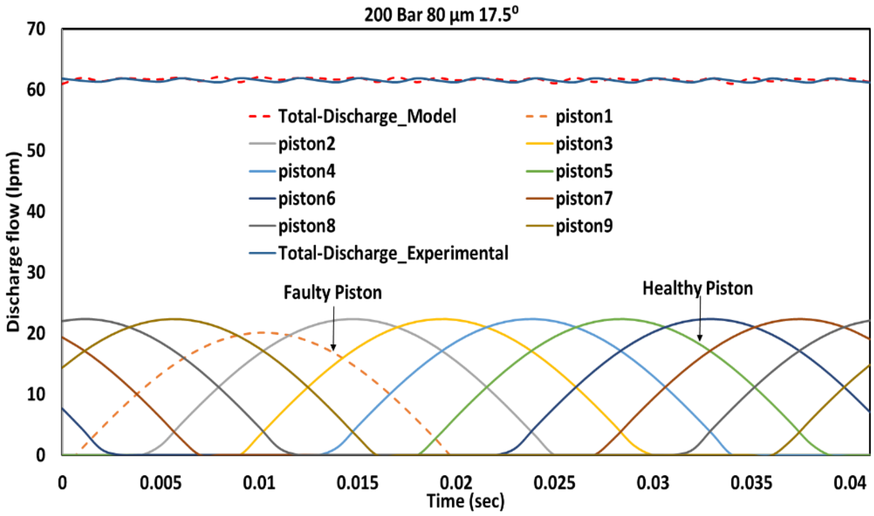

Fig. 4.2(f) Comparison of Total discharge for 80 µm annular gap at 200 bar operating pressure.

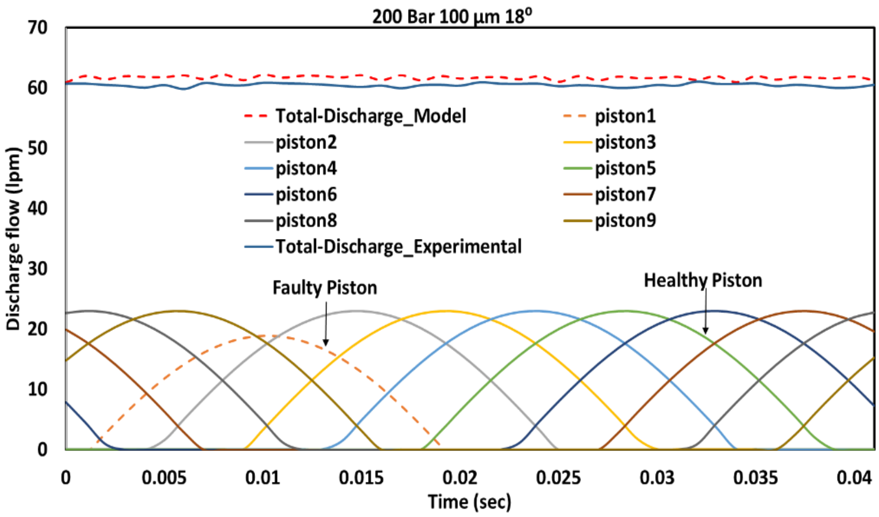

Fig. 4.2(g) Comparison of Total discharge for 100 µm annular gap at 200 bar operating pressure.

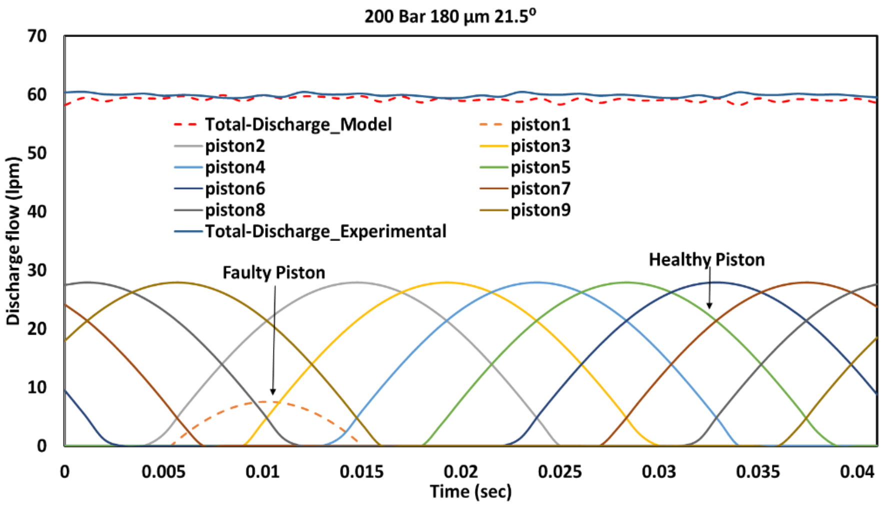

Fig. 4.2(h) Comparison of Total discharge for 180 µm annular gap at 200 bar operating pressure.

Similarly at an operating pressure of 200 bar the model predicted discharge shows a close correlation with experimental discharge data thereby validating the reliability and accuracy of the simulation model. As illustrated from Fig. 4.2(e) through Fig. 4.2(h) the annular gap increases progressively which results in slight variations in the swashplate angle due to its self regulating characteristics. In Fig. 4.2(e) with an annular gap of approximately 50 µm the internal leakage remains minimal as indicated by the orange dotted line and hence the total discharge is not significantly affected. However in Fig. 4.2(f) the annular gap increases to 80 µm which introduces a marked rise in internal leakage resulting in a noticeable reduction in the pump's overall discharge. The trend continues in Fig.4.2(g) and Fig. 4.2(h) where the gap further increases to 100 µm and 180 µm respectively. This expansion of clearance significantly amplifies internal leakage, which adversely impacts the volumetric efficiency and discharge performance of the axial piston pump. Additionally, when comparing the pump performance at 100 bar and 200 bar, it is evident that higher operating pressure exacerbates internal leakage. At elevated pressures, the fluid force is greater, causing more pronounced leakage through the enlarged gaps, thereby leading to a more substantial reduction in discharge and efficiency.

4.4. Velocity distribution at various positions of the piston within the cylinder bore.

This section analyzes the leakage behavior (velocity profile) within the mating interface of the piston and cylinder assembly. The velocity profile u(r), Eq. 3.10, is computed under consistent operating conditions and boundary constraints, incorporating a phase angle shift of 45° from Bottom Dead Center (BDC) to Top Dead Center (TDC), corresponding to the discharge phase of the cycle.

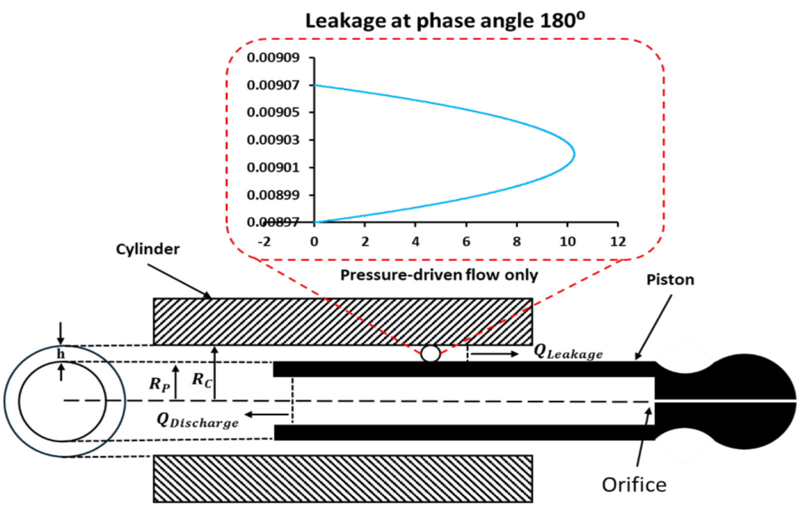

Fig. 4.3(a) Leakage profile at phase angle 180° (BDC)

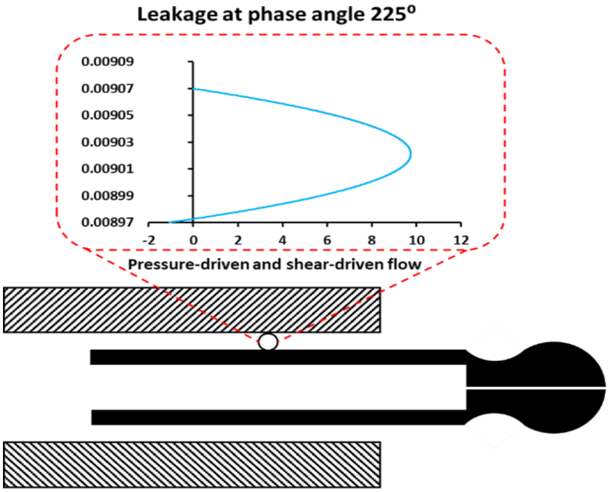

Fig. 4.3(b) Leakage profile at phase angle 225°

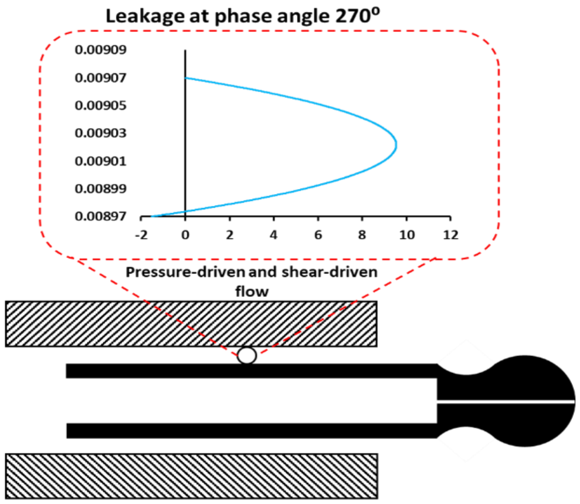

Fig. 4.3(c) Leakage profile at phase angle 270°

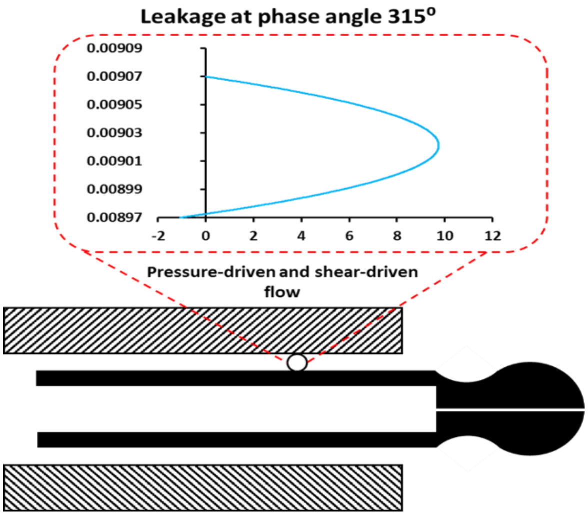

Fig. 4.3(d) Leakage profile at phase angle 315°

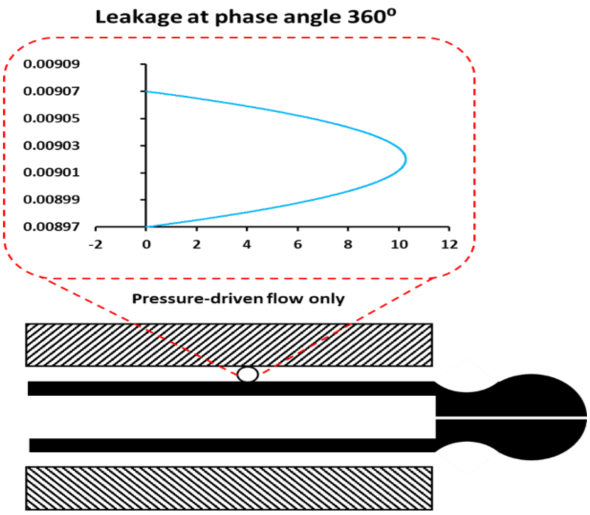

Fig. 4.3(e) Leakage profile at phase angle 360° (TDC)

As illustrated in Figures 4.3(a) to 4.3(e) the velocity profile shown by blue line within the annular gap between the piston and cylinder undergoes significant changes as the piston moves from Bottom Dead Center (BDC) to Top Dead Center (TDC) corresponding to a phase angle transition from 180° to 360°. These changes are primarily influenced by the combined effects of pressure driven and shear driven flow mechanisms. The pressure driven component arises due to the axial pressure gradient across the gap while the shear driven component is induced by the relative motion between the piston surface and the fluid generating viscous shear forces. Under a fixed annular gap size and constant operating pressure and it is observed that the shear term evolves with piston motion. From 180° to 270° as shown in Figures 4.3(a) to 4.3(c) the shear contribution increases progressively and reaches its maximum magnitude at 270° corresponding to mid stroke. Beyond this point as the piston continues toward TDC (270° to 360°) the shear effect begins to decline and eventually approaching zero at 360° as depicted in Figure 4.3(e). This transition reflects the dynamic interaction between fluid viscosity and piston kinematics, which significantly influences the overall velocity distribution and leakage behavior in the piston-cylinder interface during the discharge phase.

4.5. Combine results for Leakage and proportion of Pressure and Shear driven flow.

This section analyses the combined influence of leakage and induced flow proportion under specific leakage conditions at a constant operating speed focusing on variations in annular gap sizes and system pressure. The investigation considers annular gaps of 100 µm and 180 µm operating at pressure levels of 100 bar and 200 bar. The objective is to evaluate how these geometric and pressure variations affect the overall leakage behavior and the resulting flow characteristics. The analysis provides insights into fluid structure interactions and performance implications for systems operating under high pressure conditions with varying clearances.

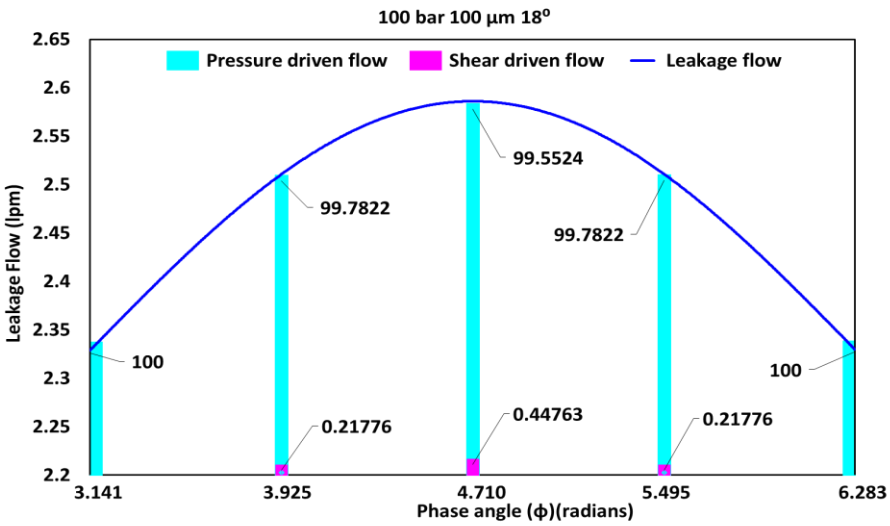

Fig. 4.4(a) Comparison of pressure driven and shear driven velocity for 100µm annular gap at 100 bar operating pressure.

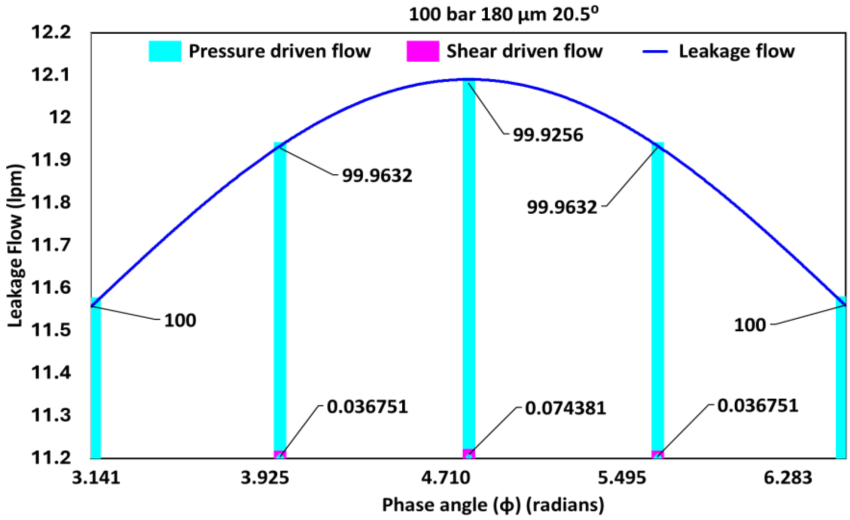

Fig. 4.4(b) Comparison of pressure driven and shear driven velocity for 180μm annular gap at 100 bar operating pressure.

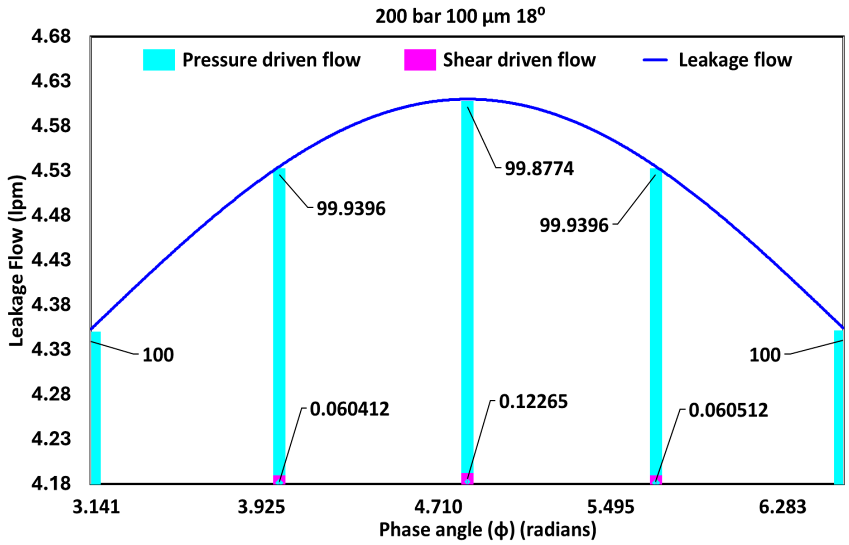

Fig. 4.4(c) Comparison of pressure driven and shear driven velocity for 100μm annular gap at 200 bar operating pressure.

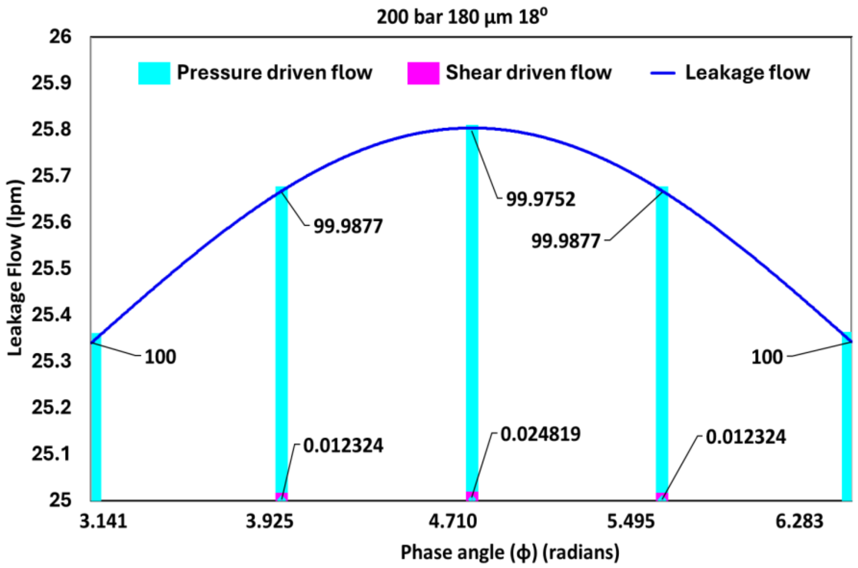

Fig. 4.4(d) Comparison of pressure driven and shear driven velocity for 180μm annular gap at 200 bar operating pressure.

As depicted in Fig. 4.4(a) through Fig. 4.4(d) the leakage behavior within the axial piston pump is governed by the combined effects of shear driven and pressure driven flow components. An increase in the annular gap while maintaining constant operating pressures of 100 bar and 200 bar leads to a substantial reduction in the shear driven component due to the diminished viscous shear forces resulting from the increased clearance. The pressure driven component, however remains relatively dominant. A detailed examination of the flow variation over the phase angle range from 3.14 to 6.28 radians reveals that the pressure driven flow decreases until approximately 4.71 radians. This reduction corresponds to the influence of shear driven flow induced by the relative motion between the piston and the cylinder bore mating surface. Beyond the 4.71 radians phase angle the pressure driven component begins to increase again and reaches its peak constituting nearly 100% of the total flow at 6.28 radians. This trend indicates that the shear effects temporarily influence the flow during the intermediate phase angles and pressure driven flow remains the predominant mechanism throughout the cycle. Thus in the context of internal leakage pressure induced flow plays a significantly greater role than the shear driven flow especially at higher pressures and larger annular gaps.

4.6. Summary

This chapter presents a comprehensive analysis on the performance of axial piston pump under induced internal leakage conditions. The developed model is validated against experimental data that demonstrating high accuracy and reliability for predictive maintenance applications. A sensitivity analysis of the swashplate angle was conducted to determine the operational range and its effect on pump behavior. Comparative evaluation between simulation and experimental results highlights precision of the model in capturing leakage flow characteristics as the internal gap increases. Furthermore the study examines the distribution and transition of pressure-driven and shear-driven flows with detailing their profiles during piston motion from BDC to TDC.

Chapter 5

Conclusion and Research Scope

5.1. Introduction

A mathematical model has been developed to analyse the leakage and discharge behavior in the axial piston pump under varying operational conditions. This model incorporates consistent constraints and operating parameters to evaluate internal leakage across different annular gap dimensions and a range of operating pressures. By simulating both healthy and faulty conditions the model effectively distinguishes performance deviations associated with faults diagnosis. This approach captures a comprehensive assessment of leakage characteristics and helps to identify the dominant factors which influencing pump efficiency and fluid discharge. The analysis provides valuable insights into the relationship between structural variations such as changes in the annular gap and their impact on volumetric efficiency. Furthermore the model supports predictive maintenance by quantifying the extent of internal leakage thereby facilitating early detection of potential failures. Overall the mathematical framework serves as a diagnostic and predictive tool for enhancing the reliability and performance of axial piston pumps in hydraulic systems.

5.2. Leakage at high operating speed

This section presents an analysis of internal leakage behavior in an axial piston pump operating at elevated rotational speeds specifically around at 6000 rpm. As the pump speed increases a notable rise in leakage flow is observed which attributed primarily to enhanced shear forces within the fluid film. The increase in relative motion between components enlarges the shear driven contribution to the total leakage which becomes significantly more pronounced at high speeds. Quantitative analysis indicates that the shear driven term can account for approximately 10% of the total leakage flow under these conditions marking a substantial deviation from low-speed operation where pressure driven flow dominates. This shift highlights the critical influence of dynamic viscosity and shear rate on leakage mechanisms at high operating speeds. The findings tell the necessity of accounting for speed dependent effects in leakage modeling particularly in the high performance hydraulic systems where precise control and efficiency are essential.

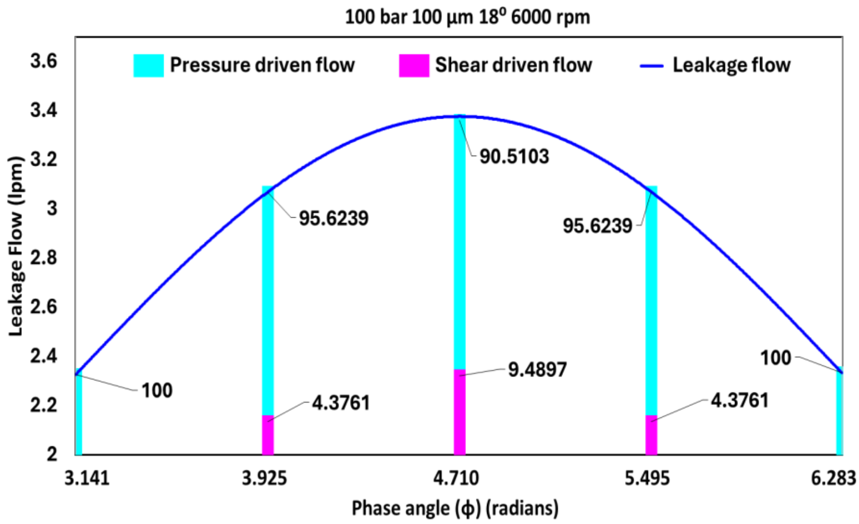

Fig. 5.1 Comparison of pressure-driven and shear-driven velocity for a 100µm annular gap at 100 bar operating pressure and 6000 RPM.

When the rotational speed of an axial piston pump increases to ~12000 to 15000 RPM shear driven flow resulting from the relative motion between the piston and cylinder bore becomes a major contributor to internal leakage. This adversely affects the volumetric efficiency of the pump. However operating at such high speeds presents significant challenges. The elevated speed causes excessive heat generation due to increased friction and fluid shear leading to a rise in fluid temperature. Additionally standard induction motors may reach their thermal or torque limits that resulting in automatic shutdowns to prevent damage. These factors collectively restrict the practical operating range of the pump.

5.3. Conclusion

The close correlation between the developed mathematical model and experimental results validates the accuracy and reliability of the proposed approach. The study confirms that pressure driven flow is the dominant source of internal leakage in axial piston pumps while shear driven flow plays a secondary role under normal operating conditions. However at extremely high rotational speeds (~12000–15000 rpm) shear effects become more pronounced although practical constraints such as fluid overheating and induction motor limitations restrict continuous operation in this range. By incorporating the shear induced flow term the model not only captures leakage behavior accurately but also enriches the ability to predict potential failure with supporting performance assessment and design optimization.

5.4. Future Work or Research Scope

Several recommendations for future research on piston-cylinder leakage in axial piston pumps are suggested below:

- As highlighted in previous studies clearance between the piston and cylinder is a major contributor to internal leakage in axial piston pumps. However several other factors also influence leakage behavior. One important factor is piston eccentricity which can disrupt the uniformity of the fluid film and lead to increased leakage. Additionally the orientation and geometry of the piston play a crucial role for instance a tapered pistons or those with nonstandard surface profiles such as spherical pistons can alter the flow characteristics within the cylinder bore thereby affecting leakage rates. The combined effect of these design features especially when eccentricity is present alongside unconventional piston geometries which can significantly exacerbate internal leakage in axial piston pumps.
- Future work can also focus on fluid flow characteristics by calculating the Reynolds number for the annular gap between the piston and cylinder. By evaluating the Reynolds number at varying gap sizes and under different operating pressures, it is possible to investigate the influence of these parameters on flow behavior and validate assumptions regarding laminar or turbulent flow made in previous studies.

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