HEALTH MONITORING OF TELESCOPIC HYDRAULIC CYLINDER

M.Tech. Thesis

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HEALTH MONITORING OF TELESCOPIC HYDRAULIC CYLINDER

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 **HEALTH MONITORING OF TELESCOPIC HYDRAULIC CYLINDER** 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 August 2020 to May 2022 under the supervision of **Dr. Pavan Kumar Kankar, Associate Professor, Department of Mechanical Engineering, Indian Institute of Technology** and **Dr. Ankur Miglani, Assistant Professor, Department of Mechanical Engineering, Indian Institute of Technology.**

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.

30/05/2022

(SHRISH TIWARI)

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

30/05/2022 | 30 MAY 2022

(Dr. PAVAN KUMAR KANKAR) | **(Dr. ANKUR MIGLANI)**

SHRISH TIWARI has successfully given his/her M.Tech. Oral Examination held on **24th MAY 2022.**

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(30/05/2022)

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(Dr. Sandeep Singh)
Date:

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I can't close my remarks without expressing my heartfelt gratitude and respect for my loving parents' blessings and attempts to keep my spirits up during my project. I'd want to express my profound gratitude to everyone who has helped me in any manner during this project, whether directly or indirectly.

With Regard,

Shrish Tiwari

Dedicated

to

My beloved

Grandfather

(Shri. Shiv Kumar Tiwari)

Abstract

A telescopic hydraulic cylinder is a specially designed hydraulic actuator that can provide a long-extended length with a very small, contracted length. A telescopic cylinder is made up of cylinder tubes and piston rods. It converts highly pressurized fluid energy into mechanical reciprocation. These telescopic cylinders work at very high pressure due to which designing them is a difficult task. To design a telescopic cylinder there is a lack of a standardized approach making the designing and production process a tedious job. This work proposes a simulation-based solution to the above issue. A MATLAB-Simscape model of a 2-stage double-acting telescopic hydraulic cylinder is developed. The development of the model is done using basic components necessarily required for a hydraulic actuator circuit to work.

Further, the model is optimized using the parameters of the experimental test rig. Using this test rig the simulation model is validated by comparing the characteristics plots obtained by both the processes. This validated model can be used further for diagnostics and designing purposes.

In continuation, the simulation model is simulated for two more input control signals. The characteristic plot of position and pressure are studied and interpreted. The information regarding the variation of pressure inside the telescopic hydraulic cylinder and the position of the cylinder under different loading control signals is useful to design an industry-ready hydraulic actuation setup.

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ACRONYMS

4/3 way - Four-Port Three Position

DC - Directional control

ISHM - Integrated System Health Management

LPM - Litre per Minute

PT1 - Pressure Transducer 1

PT2 - Pressure Transducer 2

Chapter 1: Introduction

1.1 Overview

Telescopic hydraulic cylinders are a special type of hydraulic actuator that converts hydraulic energy into mechanical energy by using the principle of pascal's law. These Telescopic hydraulic cylinders (THC) are widely used for various industrial purposes. They can be used to lift weight in dumper trucks or to position a missile launcher at a certain angle in order to precisely hit the target. From the excavator's arm to aircraft landing gear assembly THC can find its application in various places. Due to this loading conditions on THC are continuously varying which must be taken into consideration during the design of the system itself. In spite of all this, faults will arise during the operation, and to diagnose the faults simulation-based solution is proposed.

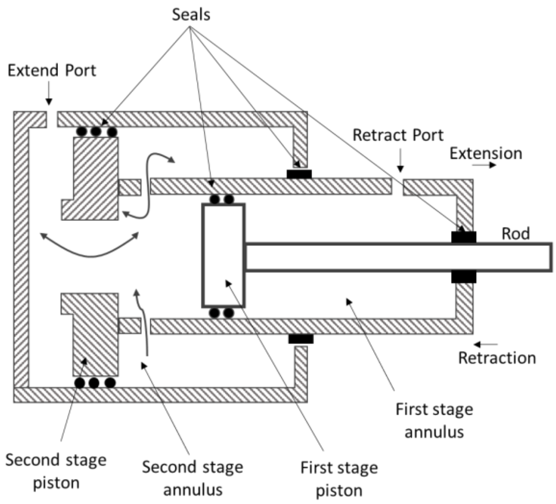

Fig 1.1: Schematic diagram of 2-stage double-acting telescopic hydraulic cylinder.

1.2 Telescopic Hydraulic Cylinder

A hydraulic cylinder (single-stage) is normally made up of a barrel and a piston rod on the other hand a telescopic hydraulic cylinder has several intermediate stages. The telescopic hydraulic cylinder consists of several-cylinder tubes and rams depending on the number of stages, ram, and piston rod. One of the most significant advantages of the telescopic cylinder over other types of cylinders is that it produces an extremely long output/stroke from a little constrained length. A telescoping hydraulic cylinder is employed when mounting space is restricted and the application requires a lengthy stroke.

Fig. 1.1 shows a schematic diagram of a cut section of a double-acting 2-stage telescopic hydraulic cylinder with internal construction and fluid passage. There are 2 cylinders one into another and a piston rod inside the 2nd stage cylinder. Both the 2nd cylinder and piston rod are having hydraulic seals made of nitrile rubber in order to avoid any leakage during the operation. High-pressure hydraulic fluid enters the 1st stage cylinder through port A and applies a force onto the 2nd stage cylinder which works as a piston for the stage 1 cylinder. Due to this force 2nd stage cylinder starts to extract from the stage 1 cylinder just like a conventional piston-cylinder arrangement. After extraction of the 2nd stage cylinder hydraulic fluid applies a force onto the piston rod which is present inside the 2nd stage cylinder. The area of the piston rod is less than the area of the 2nd stage cylinder so in order to move the piston rod value of pressure must be increased in the same ratio as the decrease of the area of the 2nd cylinder and piston rod. As the operating cylinder is double-acting after the extension of the piston rod the pressurized hydraulic fluid again enters the cylinder through Port B and exerts a force on the opposite side of the piston rod which results in retraction of the piston rod and after that this pressurized fluid applies force on the opposite side of 2nd stage cylinder which again resulting in retraction of 2nd stage cylinder and hereby the whole system gets back to the initial condition.

1.3 Types of Hydraulic Cylinder

On the basis of retracting force, hydraulic cylinders are classified as follows:

i. **Single-Acting Cylinders:** The single-acting cylinder is the most cost-effective and straightforward design. The hydraulic fluid is forced from only one end of the cylinder during both the extension and refutation processes in this type of hydraulic cylinder. When the fuel supply is taken off, the piston rod is returned to its original position by the springs, load, other cylinders, or the power of a flywheel. It requires less maintenance due to its simplified design.

ii. **Double-Acting Cylinders:** Working fluid is given alternatively on both ends of the piston head in double-acting hydraulic cylinders. Because they can work at practically any angle, double-acting hydraulic cylinders are more frequent. Because of the inherent mechanical issues of springs, single-acting cylinders cannot be used in big stroke applications.

iii. **Telescopic/Multi-Stage Cylinders:** Single-acting or double-acting multi-stage/telescopic cylinders are available. When deflated, they allow for a long stroke while taking up much less area. Because of the varied segmentation, the possibility for piston flexing on piston design is increased.

On the basis of basic design techniques of hydraulic cylinder construction, it can be roughly grouped as follows:

i. **Welded Body Cylinder:** The cylinder barrel and end caps are soldered together in this sort of cylinder. Welded cylinders have a thinner body, allowing them to fit into tighter spaces in machinery. This type of cylinder does not fail at high pressures and long strokes due to tie rod stretch.

ii. **Tie-Rod Cylinder:** Because of its flexibility to be completely disassembled for servicing and maintenance, tie-rod type hydraulic cylinders are most commonly used in industrial applications. This form of cylinder, however, is not necessarily configurable. The two end caps are held to the cylinder barrel by high-strength threaded steel rods.

1.4 Significance of Study

Nowadays in industries power transmission using hydraulic systems is the need of the hour. Hydraulic systems are broadly utilized in industries because of the quick reaction times and size to power ratio [1], [2].

Hydraulics is comparatively a new means of power transmission and control. Manufacturing and production of modern hydraulic machinery are at their peak these days. These systems can easily convert fluid energy to mechanical energy and vice versa and they are comparatively economical. Different manufacturers use different approaches to design and develop hydraulic components. There is no standardized method available to particularly design a hydraulic cylinder.

To account for the above-mentioned issues, a simulation model of 2-stage double-acting THC is developed using the MATLAB Simscape platform, and validation of the simulation model is performed using experimental data obtained from a specially designed test rig that is capable of simulating various types of real-life loading scenarios. The current study's scope and novelty can be summarized as follows:

- To study the parameters and working of various hydraulic components.
- To develop a simulation model of a 2-stage double-acting telescopic hydraulic cylinder along with various hydraulic components using MATLAB Simscape platform.
- To perform experiments on the specially designed test rig under square wave loading conditions and studied the pressure characteristics obtained during the operation.
- To optimize the simulation model using the parameters of the test rig and simulate the model under the same working conditions as the test rig.
- To study the pressure characteristics obtained from the simulation model under square wave loading conditions.
- To compare the pressure characteristics obtained from both experimentation and simulation in order to validate the simulation model.
- After validation the model is simulated for two more input signals and their position and pressure characteristics are studied.

1.5 Thesis Organization

The following is a chapter-by-chapter breakdown of the current work:

Chapter 1 highlights and introduces the working and classification of the telescopic hydraulic cylinder along with its applications. The categorization is been discussed based on retracting forces and design techniques. Along with the organization of the thesis, the goal of the current research endeavor has been highlighted.

Chapter 2 covers a brief discussion of the published literature related to the interest in the current work. This chapter reviews the history of hydraulic cylinders and faults reported to date along with the health monitoring techniques. In the last section of the chapter, the reviews of some literature using the same approach as current work is discussed.

Chapter 3 introduces the Simulink-Simsacpe blocks used to develop the simulation model along with the parameters used to mimic the real-life test rig conditions. A complete explanation of the construction and working of the simulation model is done in the further sections.

Chapter 4 gives a brief introduction of the experimental facility used to validate our simulation model. Hydraulic components used in the test rig are explained along with the input parameters to acquire the pressure characteristics plot.

Chapter 5 gives a proper explanation of the plots obtained under various input control signals. It gives a detailed overview of the plots and their technical interpretations. Input control signals, position, and pressure plots obtained after validation of model is discussed in the last section.

Chapter 6 concludes and summarizes the current research work on the basis of the results discussed earlier. It also gives a brief idea about the application of the developed model for designing and diagnostics purposes. In the last section, some ideas for future work in the related field is discussed.

Chapter 2: Literature Survey

2.1 Introduction

In this chapter, a detailed overview of published past work by various authors is discussed. There is a lot of published literature on Strength analysis and mathematical modeling of single-acting THC. During literature surveying, it was found that is no standardized method to design and analyze a hydraulic cylinder. Various researchers have used different approaches to study the buckling behavior and to model the hydraulic cylinder.

2.2 Hydraulic Cylinder History

The hydraulic cylinder's history begins in 17th century France with a Frenchman named Blaise Pascal (1623-1662). Pascal was interested in a variety of subjects, including physics, mathematics, and philosophy. Pascal's law, which bears his name, is one of these scientific breakthroughs. According to pascals law pressure applied in an enclosed fluid will be transmitted without a change in magnitude to every point in the fluid and every part of the container [3]. Pascal's scientific research led to the realisation that a confined fluid under equal pressure from all sides generated energy, which might be employed as a source of power. This realisation changed the way people thought about hydraulics and the capabilities and possibilities that were now possible [4].

Pascal's discovery was not presented to the masses for another century, thanks to British carpenter and engineer Joseph Bramah (1748-1814), who lived in the 18th century and was able to bring Pascal's findings and its consequences to people all over the world. He was able to accomplish this by employing hydraulic engineering to develop and construct a new sort of lock known as the challenge lock, which was created using new tools and techniques gained through hydraulic science and engineering. This made Bramah renowned, but what made him much more famous was his ability to mass-produce the hydraulic cylinder, something Pascal had failed to do 100 years previously. The printing press brought the hydraulic era to millions of people every day through newspapers and other written text, and it did so for years [4].

William Armstrong (1810-1900) was the first to push the frontiers of hydraulics in the nineteenth century, and he succeeded. He figured out how to utilise water that would otherwise be wasted or inefficiently used to power cranes. With his creation of the hydraulic accumulator, which is still a crucial aspect of hydraulic engineering, he was able to revolutionize our understanding of hydraulics in the modern era [4].

Finally, in the modern period, an American named Harry Vickers made the most current improvement in hydraulic technology (1898-1977). The balanced vane and pump were among his hydraulic engineering breakthroughs and achievements. This pump would allow hydraulic energy to expand into new realms of possibility, which hydraulics continues to advance from and rely on currently [4].

It has become more obvious over the last century that water is not the optimum fluid to utilize for hydraulic machinery and equipment. Oil is, in fact, the most effective if it is non-corrosive. Furthermore, oil was a better alternative than water since it was denser, could retain more weight, was less prone to evaporation than water, and remained cooler under the high temperatures of hydraulic machinery. Hydraulic machinery is also a very safe alternative for individuals who operate with it, making it even more effective and efficient for the twenty-first century. As fluid hydraulic engineering has progressed, hydraulic cylinders have likewise made significant progress. Over the years, every part of cylinders has completely transformed, including the production materials, how they are utilized, how they are constructed, how they are set up and operated, and how they are located. Furthermore, with developments in drills, mining, production, high-rise building construction, and aviation, cylinders have gone a long way in recent decades. Hydraulic power can be up to ten times stronger than electric power.

2.3 Faults Encountered in Telescopic Hydraulic Cylinder

THC operates at very high pressure and under transient loading conditions in a severe environment which results in various complications in the system over a period of time. In order to avoid downtime in the industries, regular maintenance, and inspection of telescopic hydraulic cylinder is important. Faults of various nature that are reported by the industries are listed here.

2.3.1 Seal Leakage

Seals are the weakest spot of any cylinder, as they are made from soft materials, and failure is likely for a variety of causes. They can be damaged by chipped piston rods or cracked by pressure spikes, and they can be broken down by heat. Chemical corrosion and/or wear from dirt and abrasives are additional factors. A failing seal is easy to notice. The shaft is wet, with only an oil jacket, and dirt may collect around the gland when dust and debris settle on the damp surface. Fluid deposition may also be discovered beneath the cylinder or machine [5].

2.3.2 Buckling

When the shaft diameter and structural strength are insufficient to bear the weights exerted on them, bent rods develop. Parts such as pistons and seals become mismatched as a result, and the capacity to control hydraulic fluid pressure is lost. The inevitable conclusion is a hydraulic fluid leak [5].

2.3.3 Damaged Bearings and Piston Rods

Normal wear and tear can cause piston and rod damage, but harm from rod bearings and pistons is a significant reason of cylinder failure. Bending or sideloading caused by poor installation might add unnecessary force to the cylinder, resulting in system failure [6].

2.3.4 Extreme Temperatures

Excessive heat and harsh cold temperatures, regardless of the work situation, might permanently harm your cylinders. Extreme events in temperature can compromise the sealing of your cylinder. Extremely hot or cold fluids might cause lubrication problems. Fluids that are too light or too thick might cause hydraulic cylinder problems due to leaking [6].

2.3.5 Internal Corrosion

If water is permitted to enter the hydraulic cylinder, interior corrosion of the barrel can occur. This level of damage is tough to fix and frequently necessitates part replacement. The easiest approach to avoid this level of harm is to avoid water entry, which is frequently caused by faulty seals. On the other side, such damage could indicate that cylinder is not suitable for the environment in which it is being utilized.

2.3.6 Contaminated Fluid

Debris and contaminants can wreak on practically all components, so the fluid that powers your cylinder is at threat of becoming contaminated. It can clog ports, degrade cylinder rod surface finishes, accelerate seal wear, and get between pistons & running surfaces. Hydraulic fluid deteriorates with time, but poor machine maintenance and factors like broken wiper seals can make fluid contamination more likely**.**

2.4 Health Monitoring

The health monitoring of a system can be explained as a technique used to examine the observation data. Health monitoring of a system is a collection of operations used to keep a system running well. It can be as simple as observing current system states and when performing maintenance and repairs as needed. Several sensors are obliged to conduct real-time structural integrity evaluations onboard the ship

ISHM stands for integrated system health management [7]. Health monitoring [8] and diagnosis is a support technique that is used to increase the safety as well as expand the serviceability of any machine or component. Due to heightened demands and quality requirements, most industries operate at their upper limit. This can frequently result in system faults or breakdowns, which are typically represented by continuous or critical alterations in the system's parameters, or by both. changes in the system's inherent dynamics Faults cause slow degradation. a reduction in system performance if not repaired in a timely manner, of course, this could lead to failures and a loss of output. equipment, as well as being a threat to human safety. Increased security and to incorporate safety from the start, reliability measures have been applied. As a result of the failures, extra maintenance research has been conducted procedures. To reduce the time and effort required for problem diagnosis, a number of computerized tools have been created and used, the majority of which are knowledge-based skilled systems based on precise models of system components and data from sensors [9]–[11]. Health monitoring is a management approach that entails regular assessments of the actual working condition of a plant's machinery and manufacturing systems, as well as plant management activities aimed at improving plant performance [12].

2.5 Modelling and Simulation Approach

According to the literature survey, it was found that there is no standardized method for analysis of THC. Various researchers have used different techniques for structural analysis to study the behavior of THC under different loading conditions. Hoblit [13] studied the critical buckling load on a hydraulic cylinder for the very first time. Uzny et al. have done strength analysis on THC for different mounting conditions at the end. For elastically mounted ends of THC, the static stability criterion was used to define a boundary value problem involving the system's stability using Lame’s theory. Using non-dimensional parameters results considering the influence of rigid mounting and the strength of the cylinder is presented [14]. In another study, the consequences of using various types of mounting on the stability of the THC are done. The maximum load that can be moved without damaging the THC is calculated and presented [15]. One more piece of literature considers the effect of torsional rigidity of THC under Euler's Load. Vibrations caused due to this load are studied and presented using non-dimensional characteristic curves [16]. Gupta et al. determined the buckling load of single-acting THC using the successive approximation method. Using this method number of iterations are performed to improve the accuracy of the method. For validation of the method, finite element analysis is performed [17]. Zhao et al. worked on fault and leakage extraction using the wavelet packet method of analysis. Pressure signals from outlet and inlet and displacement of hydraulic cylinder are acquired and used in this method [18]. For dynamic model simulation, mathematical modeling and simulation using Simulink is the most commonly used method found in the existing literature [19], [20]. But there is limited research based on the Simscape Dynamic model of THC. Lei et al worked on dynamic simulation modeling based on Simscape for single-acting THC to plot characteristic curves of position and speed [21]. In addition to this, another author worked on the simulation and modeling of THC for Elevators. A Simscape model of 3-stage single-acting THC is modeled and characteristics curves are plotted [22].

In this work, we are modeling a 2-stage double-acting THC using MATLAB Simscape platform and validating our simulation model using experimental data obtained from a specially designed test rig that is capable of simulating various types of real-life loading scenarios. This validated can be used for the design and development of multi-stage double-acting THC. A similar type of work is being performed by Kushwaha et al. [23] on the Fixed-Displacement pump. They developed a Simscape model and validated it experimentally. Using Simscape library various blocks are connected with each other to model a complete hydraulic circuit having a 2-stage double-acting THC as an actuator. Using the parameters and pressure signals from the experimental facility the model is optimized and validated.

Chapter 3: Modeling of Hydraulic System using MATLAB-Simscape

3.1 Introduction

A closed-loop hydraulic system involving a 2-stage double-acting telescopic hydraulic cylinder is developed using Simscape, in the Simulink environment of MATLAB 2021b. The system comprises a closed-loop flow circuit with a 2-stage double-acting telescopic hydraulic cylinder alongside the hydraulic fluid and sensors to capture the insights of the model. This model is used to simulate the actual operating conditions of a telescopic hydraulic cylinder under different types of input scenarios. To model this, various blocks namely Fixed-Displacement Pump, 4-way Directional Valve, Telescopic Cylinder Subsystem, etc. are connected with each other using numerous types of linkages on the MATLAB Sim-scape platform and these blocks are a representation of the actual physical component.

3.2 Simscape Blocks Used in Modeling

Blocks in the MATLAB-Simscape platform represent the actual physical component and provide options to insert customized parameters in order to replicate any component of the actual specification. Following blocks are used to model a complete hydraulic system [24] with a subsystem of 2-stage double-acting THC.

3.2.1 Reservoir

The Reservoir block symbolizes a pressurized hydraulic reservoir that stores fluid at a specific pressure. Regardless of volume changes, the pressure remains constant. The block compensates for the loss of pressure in the backflow line due to fittings, filters, or other local resistance. The pressure loss coefficient is used to specify the loss. The block calculates the fluid volume in the tank and sends it to the outside world through physical signal port V.

Parameters:

Pressurization level: The reservoir's internal pressure. 0 is the default value.

Initial fluid volume: The total volume of fluid in the tank at the start. 0.02 m^3 is the default value.

Return line diameter: The return line's diameter is measured in millimeters. 0.02 m is the default setting.

Pressure loss coefficient in return line: Pressure loss in the return line is accounted for by the value of the pressure loss coefficient. This value must be greater than 0 to be valid. 1 is the default value.

The loss is determined using an equation similar to that used in the Fixed Orifice block for computational robustness:

$$q = \frac{1}{\sqrt{k}} A_p \sqrt{\frac{2}{\rho} p_{loss}}$$

15 is the Critical Reynolds number.

where:

q	***Flow rate***
k	***Pressure loss coefficient***
A_p	***Area of passage***
ρ	***Fluid density***
p_{loss}	***Pressure loss***

Ports:

The ports on the block are as follows:

P

Pump line is connected to hydraulic conserving port.

R

Return line is connected to hydraulic conserving port.

V

Volume of fluid output via physical signal port.

3.2.2 Fixed-Displacement Pump

The Fixed-Displacement Pump is a machine that converts mechanical rotating energy into hydraulic (isothermal liquid) energy. The displacement of the pump [25] is set to a constant value, which you define using the Displacement option.

The pump [26] inlets are represented by ports P and T. The pump drive shaft is represented by Port S. The pressure rises from port T to port P during normal operation if the rotational speed at port S is also positive. The forward pump is the name given to this manner of operation.

Operation Modes:

There are four different operation modes available. The pressure gain from port T to port P (p) and the angular velocity at port S (ω) determine the working mode. The Operation Modes diagram corresponds to the p-chart quadrants. The modes are numbered a-d:

- Mode a: forward pump - A significant pressure rise is produced by positive shaft angular velocity.
- Mode b: reversal motor - A significant pressure drop causes a negative shaft angular velocity.
- Mode c: reversal pump - A significant pressure drop is produced by a negative angular velocity of the shaft.

- Mode d: forward motor - Positive axis A positive pressure decrease causes angular velocity.

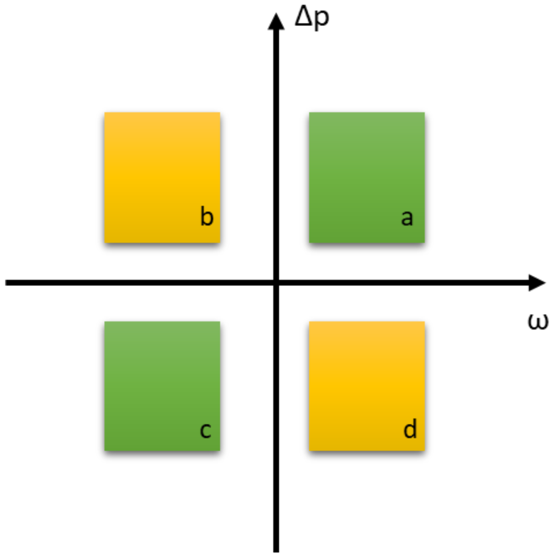

Fig. 3.1: 4-Operating Modes of Pump and Motor

In relation to the system reaction time, the pump response time is considered insignificant. The pump is handled as a quasi-stationary component since it is believed to achieve steady-state almost instantly.

Driving Torque and Flow Rate:

The volumetric flow rate generated by the pump is

$$q = q_{ideal} + q_{leak}$$

where:

- q denotes net volumetric flow rate.
- q_{ideal} denotes volumetric flow rate at an ideal condition.
- q_{leak} denotes the volumetric flow rate of internal leakage.

To power pump driving torque required is

$$\tau = \tau_{ideal} + \tau_{friction}$$

where:

- τ denotes nett driving torque.
- τ_{ideal} denotes driving torque at ideal condition.
- $\tau_{friction}$ denotes frictional torque.

Ideal Torque and Ideal Flow Rate:

The volumetric flow rate at ideal condition is

$$q_{ideal} = D\omega$$

and driving torque at ideal conditions is

$$\tau_{ideal} = D\Delta \mathrm{p}$$

where:

- D denotes displacement block parameter's value is supplied.
- ω is the rotary shaft's immediate angular velocity.
- Δp is immediate pressure rise from inlet to outlet.

Assumptions:

- The compressibility of fluid is minimal.
- The load on the shaft of the pump is an insignificant cause of spring forces, friction, and inertia.

3.2.3 4-way Directional Control Valve

A Directional Control (DC) valve with 4 ports and 3 positions, or flow channels, is represented by the 4-Way Directional Valve block. The ports connect to a double-acting actuator in a typical type (ports A and B), a pump (port P), and a tank (port T). Depending on the operating end of the actuator, hydraulic liquid can flow from the actuator to the tank via either A-T or B-T, and pump to the actuator via path P-A or P-B.

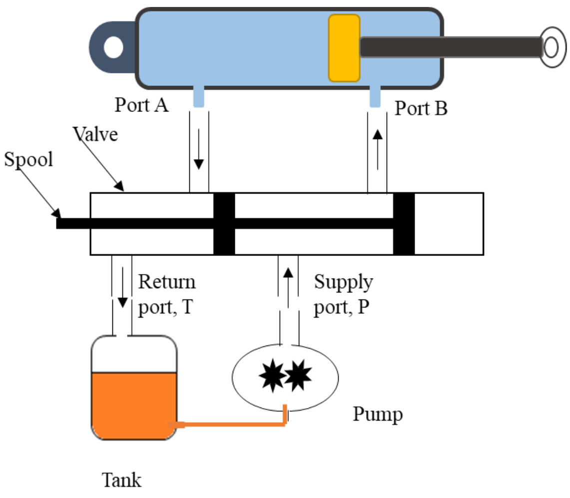

Fig. 3.2: Typical Valve Setup

One valve position corresponding to the B-T and P-A flow channels being maximum open and the A-T and P-B paths of flow being maximally closed in the default setup Fig. 3.2. Another valve position is the inverse of the previous one, with A-T and P-B maximally open B-T and P-A maximally closed Fig. 3.2. The 3rd valve position corresponding to the maximum closure of all flow pathways Fig. 3.2. A spool acts as the controlling member of the valve, determining whether the valve is at positions 1, 2, 3 or anywhere in during spool movement.

Valve Positions:

The spool displacement is controlled by physical signal port S. A zero displacement signal corresponds to valve position III in the default arrangement. The spool is shifted toward valve [27] position I when a positive displacement signal is received. The spool is shifted to valve position II by a negative displacement. The spool displacement has an indirect effect by adjusting the spool location in relation to each flow route, which is measured in length and referred to as the orifice aperture. The opening area of the respective flow route is determined by the orifice opening.

Assumptions:

- Fluid inertia is not taken into account.

- The effects of inertial, spring, and other forces on control members are neglected.
- Unless otherwise noted, all-valve orifices are presumed to have the same size.

3.2.4 Ideal Angular Velocity Source

The Ideal Angular Velocity Source illustrates an ideal angular velocity source that produces a velocity differential proportionate to the input basic signal at its terminals. The source is standard in the perception that it is believed to be significantly powerful to maintain a defined velocity independent of system torque.

Mechanical rotational conserving ports are connections R and C. The control signal that governs the source is applied through S port, which is a basic signal port. The signal at control S port is directly proportional to the relative velocity across the source. To create necessary velocity variation profile, you can utilize any of the Simulink® signal sources.

The positive direction of the block is from R port to C port. This denotes velocity is equal to $\omega R - \omega C$, where ωR and ωC are the absolute angular velocities at R port and C port, respectively, and torque at the source is upside if guided from R to C. If the source supplies energy to R port, the power produced by the source is at downside.

3.2.5 Pressure Relief Valve

Pressure Reduction of flow across valve that starts to vent in order to preserve a specific pressure drop between the inlet (A port) and output (B port) is represented by the valve block (B port). When pressure decrease from A to B and surpasses the valve pressure preset, the normally closed valve breaks open. The valve opening area is calculated using either a straightforward linear expression or a tabular function as a function of the pressure drop excess. The valve performs its function until its initial area stretches to maximum. A point near valve guideline range's limit, ahead of which the pressure fall is allowed to rise unabatedly once more.

Valve Flow Rate:

In the block, the causes of the pressure losses in the valve's passageways are neglected. Only the cumulative effect is addressed during simulation, regardless of their type (sudden area alterations, flow passage contortions). The discharge coefficient, a measure of the flow rate through the valve relative to the theoretical value that it would have in a perfect valve, captures this influence in the block. The flow rate through the valve can be calculated as follows:

$$q = C_D S \sqrt{\frac{2}{\rho}} \frac{\Delta p_{AB}}{[(\Delta p_{AB})^2 + p_{Crit}^2]^{\frac{1}{4}}}$$

where:

- q denotes volumetric flow rate of valve.
- C_D denotes discharge coefficient of block constraint.
- S denotes the open up area of the valve.
- Δp_{AB} denotes the difference in pressure between A port and B port.
- p_{Crit} denotes the pressure difference at which the stream conversion from laminar to turbulent stream.

The critical pressure is calculated based on the value of the Laminar conversion specification block constraint. If this constraint is set to the default value, $pressure\ ratio$:

$$p_{Crit} = (p_{Atm} + p_{Avg})(1 - \beta_{Crit})$$

where:

- p_{Atm} denotes atmosphere pressure.
- p_{Avg} denotes average pressures at A port and B port.
- β_{Crit} is Laminar flow pressure ratio

If, on the other hand, the Laminar conversion specification block constraint is set to $Reynolds\ Number$:

$$p_{Crit} = \frac{\rho}{2}\left(\frac{Re_{Crit}\nu}{C_D D_H}\right)^2$$

where:

- Re_{Crit} is Critical Reynolds number.
- ν is hydraulic network specified kinematic viscosity.
- D_H is hydraulic diameter at any given time:

$$D_H = \sqrt{\frac{4S}{\pi}}$$

3.2.6 Constant Volume Hydraulic Chamber

The Constant Volume Hydraulic Chamber represents a fixed-volume compartment as well as stiff or flexible walls, as found in valves, pipes, hoses, pumps, manifolds, and other hydraulic components. Use this unit in models that require fluid compressibility to be taken into account. The block parameters can be used to select the proper depiction of fluid compressibility.

The following equations are used to simulate fluid compressibility in its simplest form:

$$V_f = V_c + \frac{V_c}{E}p$$

$$q = \frac{V_c}{E} . \frac{dp}{dt}$$

where:

- q is rate of flow to the chamber.
- V_f is fluid volume in the chamber.
- V_c is geometrical volume in the chamber.
- E is bulk modulus of fluid
- p is gauge pressure inside the fluid chamber.

The previous equations must be improved if the pressure in the compartment is expected to decrease to negative amounts and move towards the cavitation threshold. The liquid in the compartment is represented as a mix of fluid and a tiny quantity of entrained, nondissolved gas in this block [28]. The bulk modulus of a mixture is calculated as:

$$E = \frac{E_{(l)}\left(1 + \alpha\left(\frac{p_a}{p_a + p}\right)^{\frac{1}{n}}\right)}{1 + \frac{\alpha\left((p_a)^{\frac{1}{n}}\right)}{n.(p_a + p)^{\frac{n+1}{n}}} E_l}$$

where:

- E_l is bulk modulus of pure liquid.
- p_a is atmospheric pressure.
- α is comparative gas content at atm pressure, $\alpha = V_G/V_L$
- V_G is volume if gas at atm pressure.
- V_L is liquid volume
- n is specific heat ration of gas.

Assumptions:

- There is no consideration for the inertia of pipe walls.
- A cylindrical shape is assumed for a chamber with compliant walls. Any shape can be achieved in a chamber with a hard wall.

3.2.7 Double-Acting Hydraulic Cylinder

The Double-Acting Hydraulic Cylinder is a reduced form of a double-acting hydraulic cylinder that was intended for usage where only the simple cylinder capabilities needs to be replicated in exchange for higher numerical efficiency. Fluid squeezability, friction, and leakages are thought to be insignificant for these reasons. To avoid any conceivable fluctuations at the ending of the stroke, the harsh stops are supposed to be completely inelastic. If such explanations are appropriate, the model is particularly well suited to real-time and hardware-in-the-loop simulation.

The equations that characterise the model are as follows:

$$F = A_A . p_A - A_B . p_B - F_c$$

$$q_A = A_A . \nu$$

$$q_B = A_B . \upsilon$$

$$\frac{dx}{dt} = \upsilon$$

$$\upsilon = \upsilon_R - \upsilon_C$$

$$F_C = \begin{cases} (x - x_E).K_P.\nu & if\ x > x_E, \nu > 0 \\ (x - x_R).K_p.\nu & if\ x < x_R, \nu < 0 \\ 0 & otherwise \end{cases}$$

$$x_E = S - x_O$$

$$x_R = -x_O$$

where,

- F is developed force by the cylinder.
- ν is velocity of cylinder rod.
- ν_R, ν_C are respective absolute velocities of rod and case of cylinder.

- A_A is A port side piston area.
- A_B is B port side piston area.
- p_A is A port side cylinder pressure.
- p_B is B port side cylinder pressure.
- q_A is rate of flow through A port.
- q_B is rate of flow through B port.
- x is position of piston.
- x_O is distance between piston and A cap in the start.
- F_C is force at hard stop.
- x_E is distance piston travels to full extend from its beginning position.
- x_R is distance piston travels to full retract from its beginning position.
- K_P is coefficient of penetration.
- S is stroke of piston.

The rigid model, which is simply a viscous muffler with the diffusion-dependent dampening constant, is used to describe the harsh stop in the Double-Acting Hydraulic Cylinder. The penetration coefficient is the name given to this coefficient. Because no oscillation is formed during an impact using an inelastic model, numerical robustness and efficiency are enhanced. However, an important aspect to consider when choosing an inelastic stop model is that striking bodies continue to move gradually towards each other if the link is loaded with squeezing force. This event is analogous to the impact of two bodies split by a large sheet of viscous fluid in real life. Before the bodies come into contact with the liquid, it takes some time to squeeze it.

Assumptions:

- Friction between moving parts isn't considered.
- Effects due to inertia is neglected.

- Fluid is assumed as incompressible.
- Leakage of fluid is neglected.
- Hard stops are considered as totally inelastic.

3.3 Construction of Simscape Model

To analyze the effect of hydraulic fluid input on the pressure signature of THC, a simple closed-loop flow network is constructed in Simscape as shown in Fig. **3.3** and Fig. **3.4**. It consists of a 2-stage double-acting THC which acts as a hydraulic actuator and converts hydraulic energy to linear motion using pascal's law. A reservoir containing hydraulic fluid is connected to a Fixed-Displacement pump which is responsible for creating pressure difference and therefore driving the fluid inside the hydraulic circuit. The Fixed-Displacement pump is driven by ideal angular velocity source at a constant speed. The pressurized hydraulic fluid enters the 4-way directional valve from the Fixed-Displacement pump responsible for supplying hydraulic fluid to the THC as per the operational requirements. A DC valve with 4 ports and 3 positions, or flow channels, is represented by the 4-Way Directional Valve block. The ports connect to a double-acting actuator in a typical type (ports A and B), a tank (T port), and a pump (P port). Depending up on the operating end of the actuator, liquid can run from the actuator to the tank via either A-T or B-T, and from pump to actuator via path P-B or P-A. A signal builder block is connected at port S of the 4-way Directional Valve which provides the signal to control the spool displacement. The signal builder is a block that generates desired SIMULINK signal and requires a SIMULINK-PS converter block so that the signal generated by the block is converted into a physical signal which can direct the spool displacement of the 4-way Directional Valve. As the spool gets the positive displacement signal pressurized hydraulic fluid enters 1st stage of telescopic block assembly resulting extension of the 1st stage (pascal's law). After the complete extension of 1st stage extension of 2nd stage of THC starts up to its maximum stroke length. For retraction, the signal builder generates a negative displacement signal which makes hydraulic fluid enter the THC from port B. This Pressurized Fluid through port B exerts a force on the piston rod's opposite side, causing it to retract back to initial conditions. The additional fluid goes back to the return line through Port T of the 4-way directional valve. The return line carries hydraulic fluid back to the reservoir. There is one pressure relief valve block connected between the output line of the Fixed-Displacement pump and the return line to the reservoir in order to maintain pressure between the preset limits of the model. An ideal force source block is connected to the R port of THC assembly in order to simulate loading on the THC assembly. Another signal builder block is connected to this Ideal force source in order to change the direction of applied load when the operation of THC assembly changes from extraction to retraction to simulate actual loading conditions. To extract motion signature of the THC assembly one position sensor block is also connected to port R. All these components blocks are connected with the help different types of linkages depending upon the need of the model. In order to connect Simscape blocks various types of linkages are used. To connect blocks of Simscape Fluids library hydraulic lines are used. Blocks from Simscape mechanical library are connected using mechanical linkages. Some Simulink signal blocks are connected using Simulink's standard line connections, e.g. Signal Builder block. Blocks that convert hydraulic energy into mechanical energy require both hydraulic and mechanical types of linkages e.g. Hydraulic Cylinder Block, Fixed-Displacement Pump. A 4-way Directional Valve is a block using one physical signal link along with hydraulic linkages.

3.3.1 Tables of Parameters

To customize the model blocks as per our requirements, parameters are inserted into the block options to replicate the actual physical component as we are using for experimental validation.

Table. 3.1: Parameters of Fixed-Displacement Pump

Fixed-Displacement Pump	Parameters
Nominal Displacement	5.13 cc/rev
Nominal Delivery @ 1500 rpm	7.7 lpm
Max Continuous Pressure	200 bar

Table. 3.2: Parameters of Telescopic Hydraulic Cylinder.

Telescopic Hydraulic Cylinder	Parameters
The bore Cross-section area of Stage 1 cylinder	5026.548 mm^2
The bore Cross-section area of Stage 2 cylinder	1963.495 mm^2
The cross-section area of the piston rod	706.85 mm^2
Mass of THC assembly	23.91 kg
Stroke length of stage 1	133 mm
Stroke length of stage 2	267 mm

Table. 3.3: Parameters of Hydraulic Fluid.

Hydraulic Fluid	Parameters
Kinematic Viscosity	68 cst
Relative Density	0.927 g/cc

3.4 Working of Simulation Model

To simulate the loading conditions as done by the PLC unit in the experimental setup (to be briefed in next section) a Signal Builder is used which generates Simulink signal which is converted into the physical signal by the use of the Simulink-PS converter block. This physical input signal governs the spool movement inside the 4-way Directional Valve. The Fixed-Displacement Pump creates a pressure difference which is responsible for the flow of pressurized hydraulic fluid into the circuit. This pressurized fluid reaches a 4-way Directional valve through Port P from where it is directed to THC assembly as per the input signal. The THC assembly converts the hydraulic energy of the fluid into mechanical translation. Pressure variation during the operation of THC is plotted w.r.t. to time in Simscape results and data inspector. These graphs are the major output of the simulation model. For double-acting THC pressure signature at both port A and port B must be taken into consideration. After one complete cycle, i.e. extraction and retraction of THC fluid return back to the reservoir via 4-way directional valve through return line. A constant load of 2100 N is continuously applied during both halves of the cycle using an ideal force source block. Position sensor block provides displacement vs time plot of the THC assembly. This displacement vs time plot is studied along with pressure plots to correlate them in order to understand operating conditions under provided loading conditions.

Table 3.4: Ports and their usage.

PORT	Usage
A	Telescopic Cylinder Extension Port
B	Telescopic Cylinder Retraction Port
C	Mechanical Linkage to Telescopic Cylinder (Frame)
R	Mechanical Linkage to Loading Cylinder
S	Signals Input to DC Valve
P	Suction Line to DC Valve
T	Return Line from DC Valve

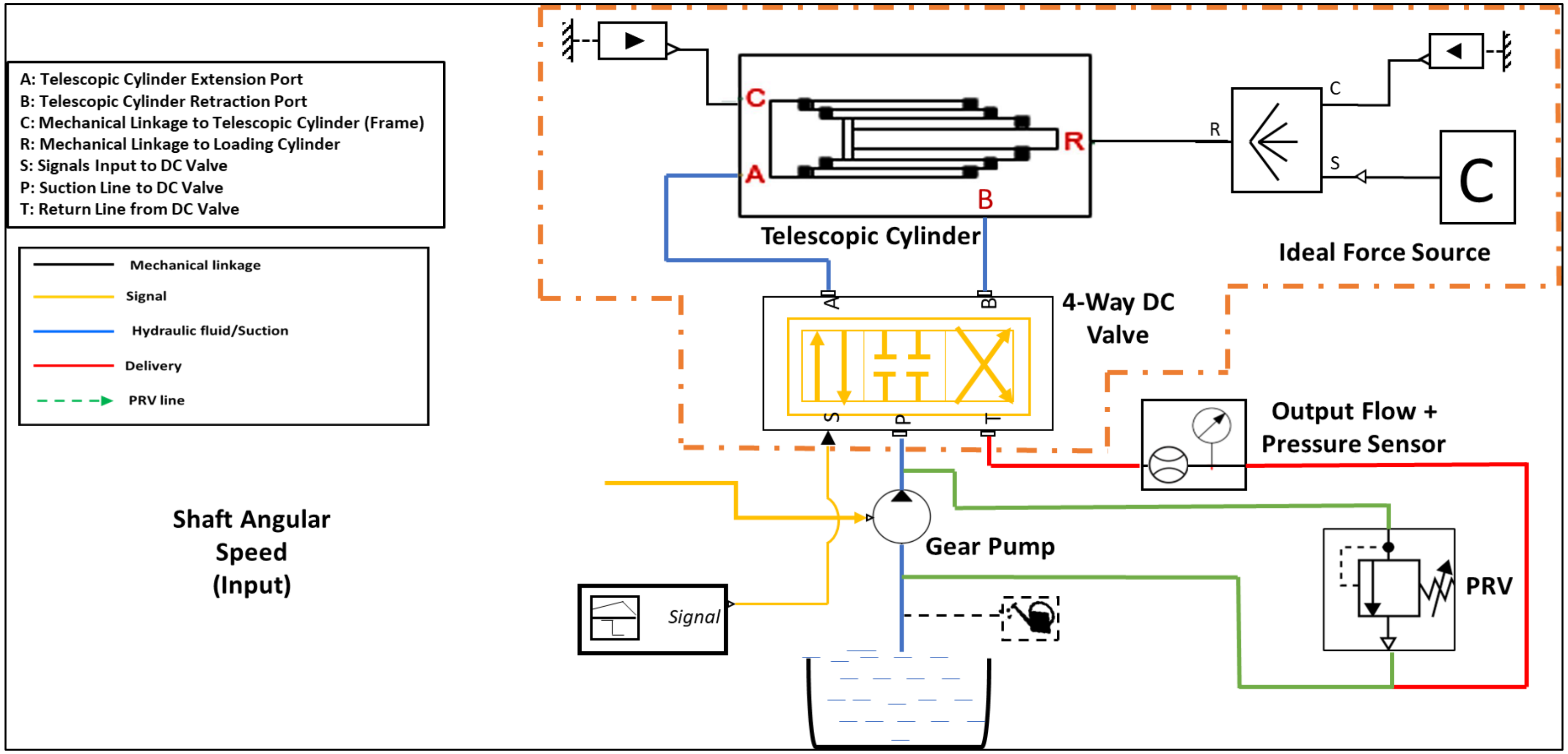

Fig. 3.3: Simscape Simulation Model

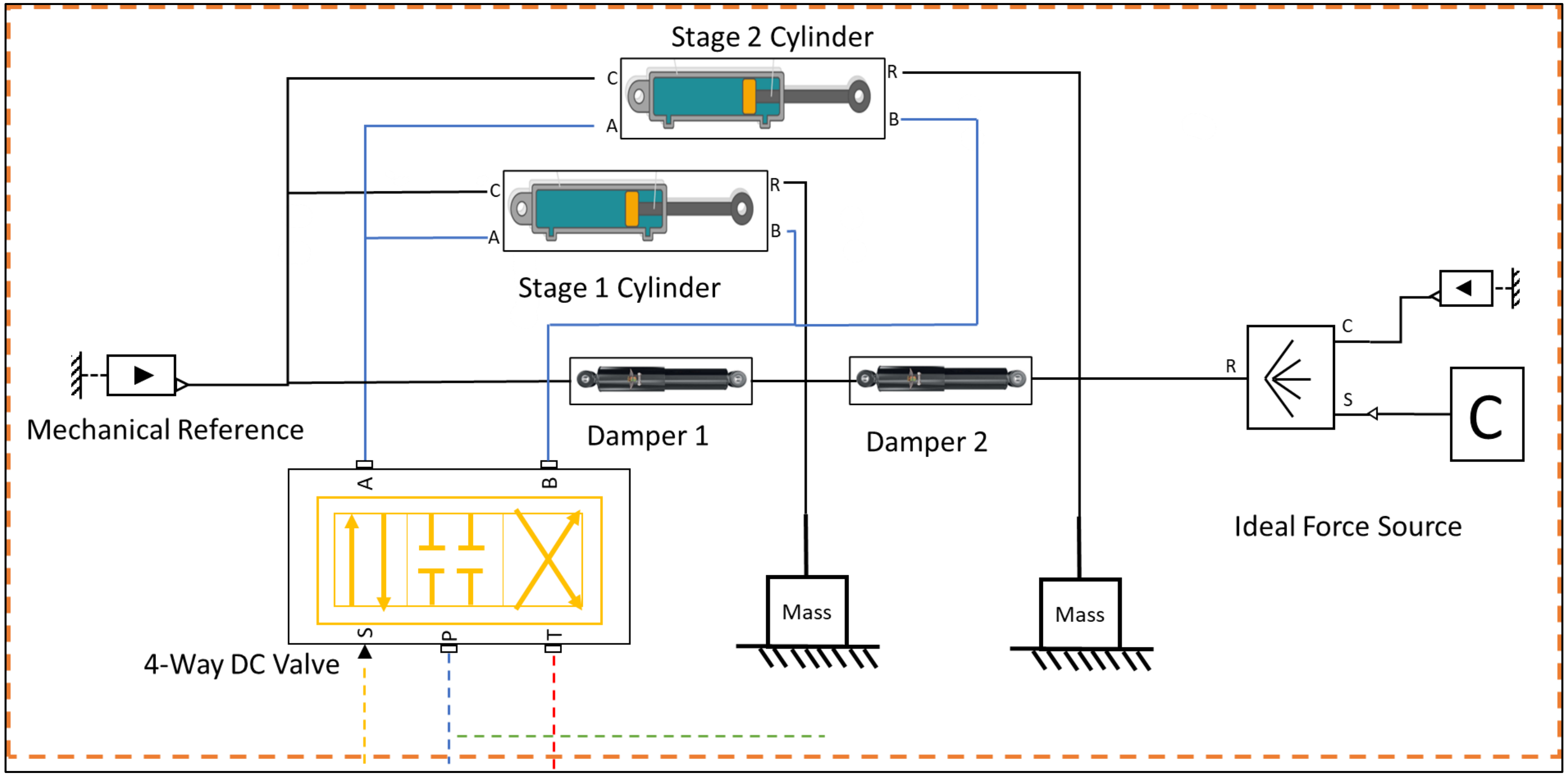

Fig. 3.4: Subsystem of THC Block

Chapter 4: Experimentation for Simulation Model Validation

4.1 Introduction

To validate the results obtained from the MATLAB Simscape simulation model, a specially designed experimental setup is used. This setup is capable of mimicking various real-life loading conditions on a 2-stage double-acting telescopic hydraulic cylinder. Using this setup one can obtain various types of insights during operation by use of various sensor data.

4.2 Experimental Facility

This experimental setup is having various types of sensors such as pressure sensors, LVDT sensors, load cells, temperature sensors, and flow meters. With the help of these sensors, we are monitoring parameters like pressure inside the telescopic cylinder, flow rate, and temperature of the hydraulic fluid, the load acting on the THC, and the position of the THC assembly. The data acquisition is done from these sensors with the help of the DAQ system. (HYDAC made). The rated maximum capacity of the experimental setup is 210 bar. It consists of a reservoir to store hydraulic fluid, a gear pump to generate high pressure, a 3-phase induction motor to power the pump, a proportional control valve to control the flow of the fluid as required, a pressure relief valve in order to maintain pressure within rated limits, a customized PLC coded HMI to control the whole setup. As we switch on the power supply HMI boots up then the first step is to turn on the motor a cooling fan also starts along with the motor. The HMI contains various types of predefined input loading signals to choose from. We can give the peak pressure and time period of the signal after choosing a particular input wave. In our case, we have chosen square wave input with a peak pressure of 200 bar at an interval of 1 sec. Along with these details we have to provide no. of cycles for which the setup has to operate. In the experimental setup, one complete cycle means either extraction or retraction. In other words, if we want one complete extraction and retraction of the THC we have to provide 2 as input for no. of cycles as in our case. Data of these two complete cycles are acquired in the DAQ system and used for further study.

For the validation purpose of our simulation model, we are mainly using pressure data of port A and port B w.r.t. Square wave loading signals. We are imposing square wave loading using an HMI system having customized PLC coding for the generation of required loading conditions. This square wave loading signal is having a peak of 200 bar at an interval of 1sec throughout the operation. This loading signal directs the proportional relief valve to apply the required loading on the THC by controlling the flow of the loading cylinder. We are acquiring data at the rate of 1000 samples per sec using the DAQ system. The data acquired is plotted against the time for study.

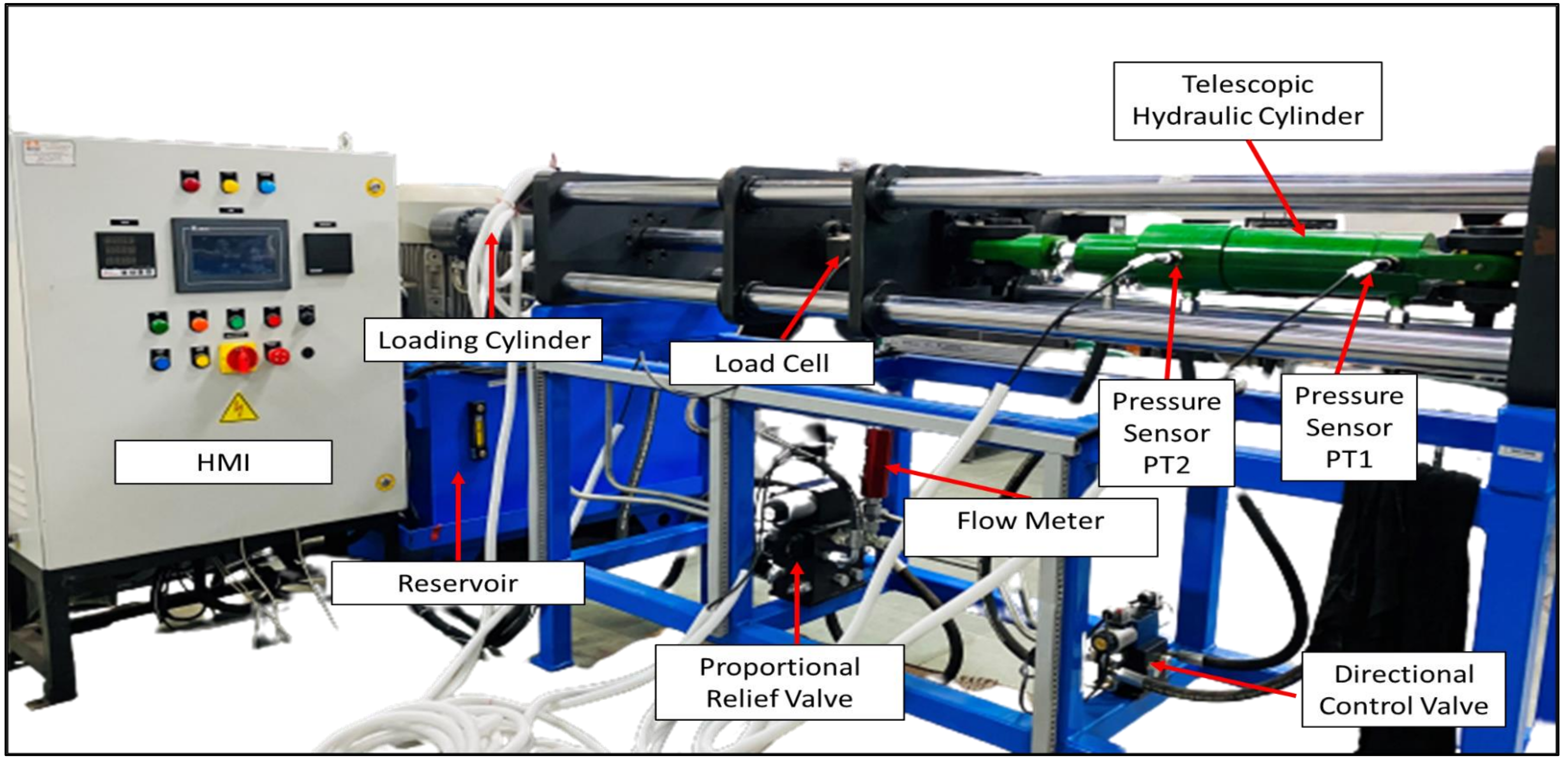

Fig. 4.1: 2-Stage Double-Acting THC Experimental Test Rig

Chapter 5: Results and Discussions

5.1 Introduction

This chapter explains the signals obtained both from simulation as well as from experimentation. A comparison between both the signal is also performed in order to validate the simulation results with the help of experimental results.

5.2 Simulation-based Results

Fig. 5.1 shows the square wave input signal to the 4-way Directional Valve. This input signal governs the spool movement inside the 4-way Directional valve to control the orifice opening which in turn controls the flow of fluid to the THC assembly. By doing that, we are trying to mimic the same loading conditions as done in the experiments. In the experimental setup, there is one loading cylinder that applies loading on the THC as directed by the PLC coded HMI. In the simulation model, we are following a reverse approach we are applying a constant load of 2100 N on the THC assembly and controlling the fluid flow which is entering the THC assembly using a square wave input signal to the 4-way directional valve.

Fig. 5.2 shows output pressure vs time plot of simulation w.r.t. provided input square wave loading signal. The graph shows the variation of pressure inside the THC assembly with an increase in time. The maximum pressure observed is 190 bar. The trend of the plot is clearly a square wave as shown in fig. The simulation time is 16 sec (approx), and we are observing 8 peaks at an equal interval from each other.

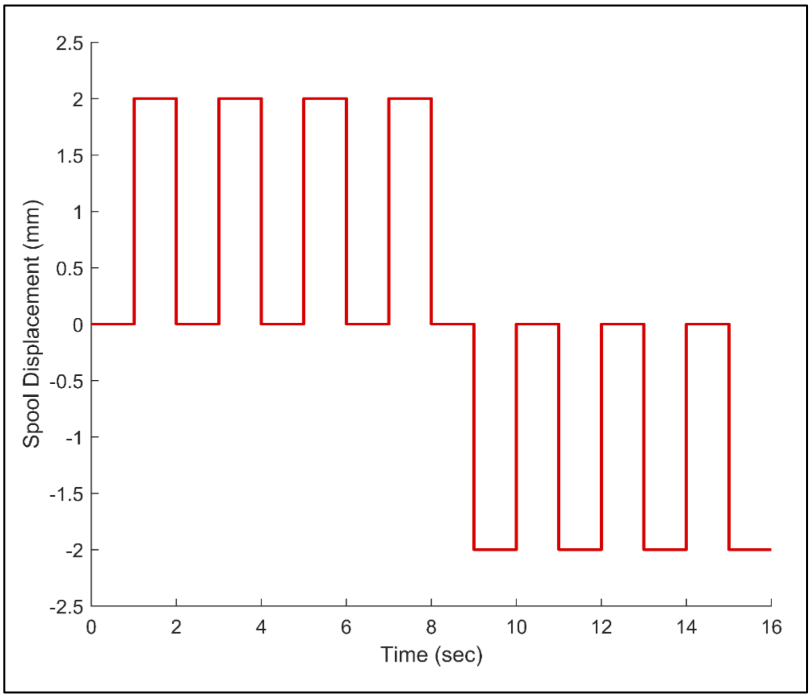

Fig. 5.1: Square Wave Input Signal for 4-way Directional Valve

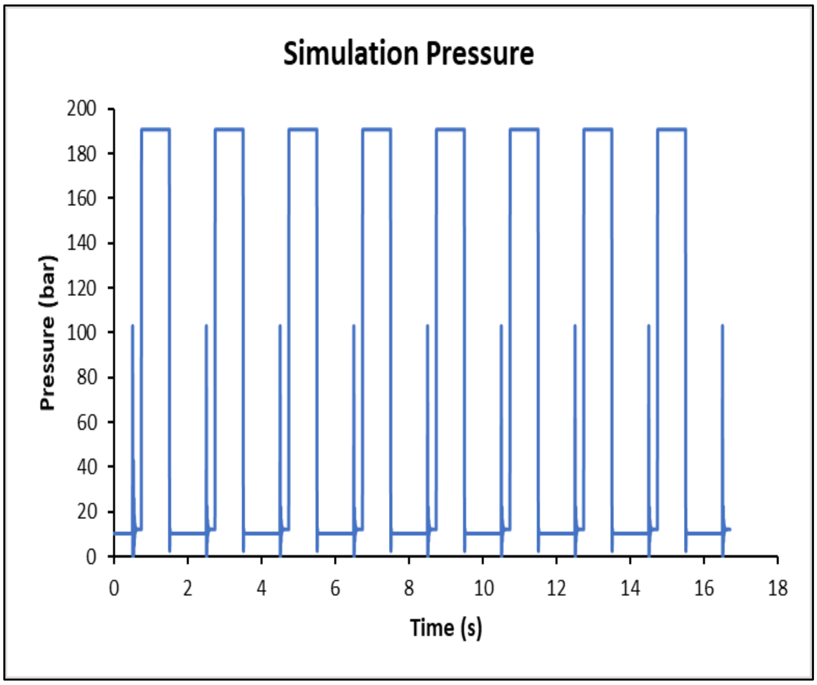

Fig. 5.2: Simulation Pressure Plot based on the Square Wave Control Signal

Fig. 5.3 shows a position-time graph of THC assembly of the simulation model. From the graphs it can be easily interpreted that there are 2 stages of opening. Its take approximately 3 seconds to completely open both the stage with the provided input signal. The pressurized fluid starts entering the cylinder as directed by the signal through port A during the first half of the input cycle. In next half of the cycle starts at 9^{th} sec when input signal goes to negative value port B opens and fluid starts entering through port B retraction starts.

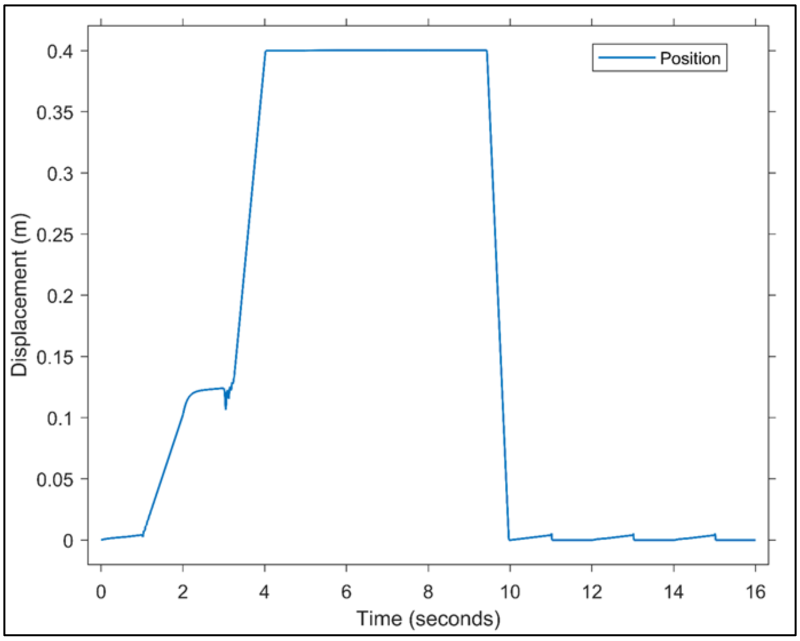

Fig. 5.3: Position-Time Graph of THC for Square Wave Input Control Signal.

We can interpret square wave loading in the THC as in Fig. 4.1 for the first 1 sec of the input signal there is almost zero flow as the spool of the 4-way directional valve is closed. For the next 1 sec i.e. 2^{nd} second of the input signal spool opens up in almost zero time and fluid started to flow to port A of the THC. This highly Pressurized fluid applies a force in the stage 1 cylinder which in turn extends out completely i.e. upto 133 mm Fig. 4.3. For the next second Fig 4.1 the flow stops from the 4-way Directional Valve resulting in stopping of the extension of cylinder. Again during next second Fig. 4.1 spool opens and flow starts. The 2^{nd} stage cylinder extension gets completed during this time period resulting in complete extension of THC assembly. After that input signals, up to 9 seconds won't affect the system as there is no other stage left for the extension. At the 9th-second signal direction changes and fluid starts entering through port B. this fluid applies a force on the opposite side of the piston resulting in retraction of the whole assembly.

5.3 Experiment-based Results

For experimental results, we are using square wave input loading signal parameters in the experimental setup using PLC coded HMI available. The maximum peak pressure for the input loading square wave is 200 bar with a time period of 1 sec. We are acquiring data from 2 pressure sensors at a sample rate of 1000 samples per sec. PT1 is a pressure sensor providing data for the extension of the THC and PT2 is another pressure sensor that provides data for retraction of the THC assembly. The data acquisition is done using a DAQ system (HYDAC made). This DAQ system records real-time data which we are using to study the insights of the THC. Using data from these two pressure sensors we are plotting a pressure vs time plot Fig. 4.4 of experimental signals.

Fig. 5.4 clearly shows a square wave trend with a time period of approximately 1 sec. The peak of the signal is hovering around 190 bar. The valleys of the signal are uneven Fig. 5.4 it can be interpreted as the next peak of the loading signal coming before the valley signal is completely executed by the system due to mechanical constraints. At around the 10th-second retraction of the THC starts due to which PT2 pressure becomes dominant. From 10th to 16th sec there are some small peaks of PT1 signal as well which can be due to flow reversal during retraction of the cylinder.

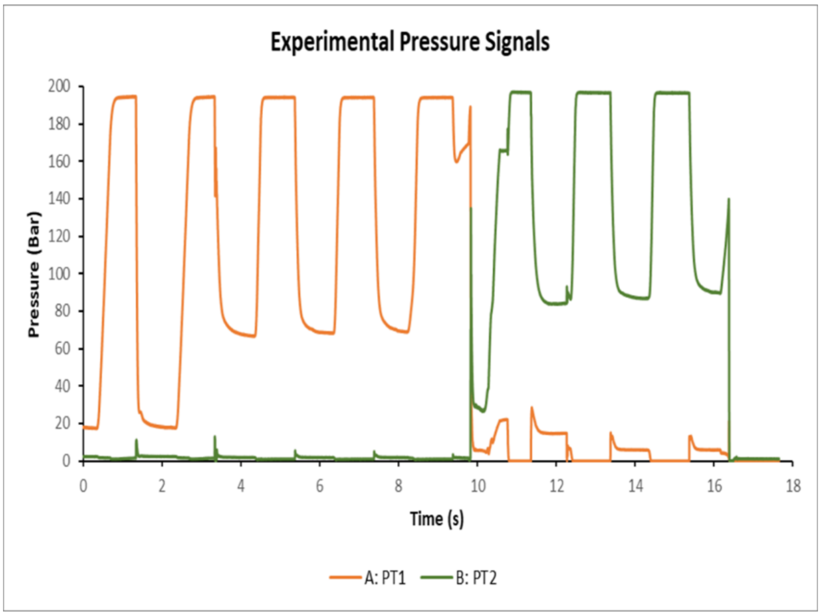

Fig.5.4: Experimental Pressure Signals for Square Wave Input Control Signal.

5.4 Model Validation

To validate the simulation model, we are optimizing our model using experimental setup parameters. With the help of these parameters, we trying to mimic the same working conditions as the experimental setup. These parameters include the piston area of both cylinders, the stroke length of both cylinders, the combined mass of the THC assembly, nominal displacement, speed and max pressure of the pump, kinematic viscosity, and relative density of hydraulic fluid used in the experimental setup.

To mimic the same conditions in the simulation model we are providing the same loading signal with the help of a signal builder block. This Input signal in Fig. 5.1 controls the spool movement inside the 4-way Directional Valve. A constant magnitude load of 2100 N is applied to the THC assembly throughout the operation. The real-time pressure insights of the simulation model are extracted from Simscape results and data inspector section Fig. 5.2 This simulation pressure vs time plot Fig.5.2 is studied in comparison with the experimental pressure vs time plot Fig. 5.4. We can clearly observe in Fig. 5.5 that the trends of both the graphs are similar and the number of peaks and valleys posses by the plot are also the same. There is some small variation between plots too which is due to material damping, dynamics of the system, and forces by the fluid flow that are not considered by the simulation model.

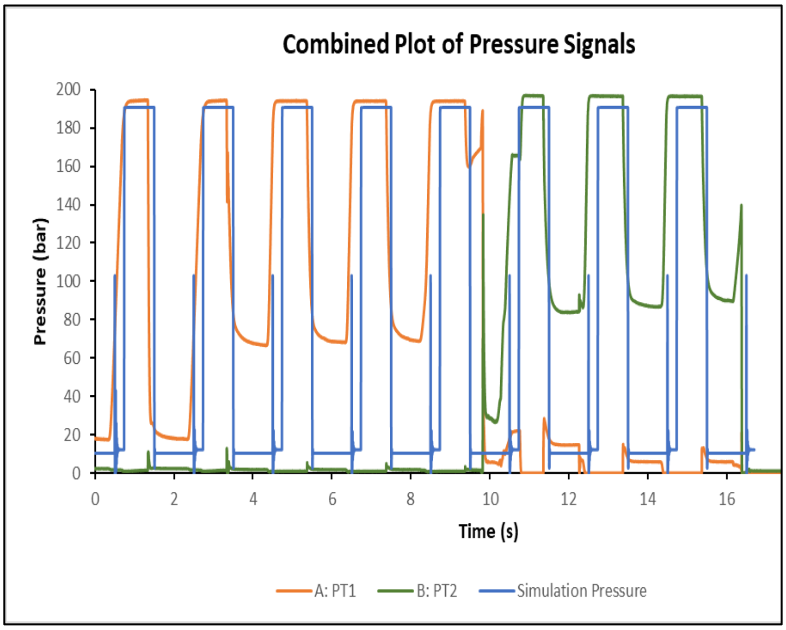

Fig. 5.5: Combined Pressure Plot of both Simulation and Experimentation

5.5 Simulation Results for Pulse Wave Input Signal

Fig. 5.6 shows the input signal which governs the spool inside the 4-way directional valve. This spool movement controls the flow of pressurized hydraulic fluid to the THC subsystem. As discussed earlier, in the simulation model we are providing input to the directional valve and applying a constant load of 2100 N on the THC. This setup of loading conditions is mimicking the 2100 N pulse waveform of loading on the THC in real-life situations. Fig. 5.6 shows the spool displacement vs time plot. For the first 4.4 seconds, the spool remains closed. On 4.4th second the signal instructs the 4-way directional valve to open the spool to 1.5 mm and maintain this position for the next 3.2 seconds. On 7.6th second the signal instructs the 4-way directional valve to close the spool. There is no movement till 8.4th second after that spool moves 1.5 mm to the opposite side so that the pressurized hydraulic fluid can enter through port B of the THC subsystem. The fluid that enters through port B is responsible for the retraction of the THC. Since the simulation modeling is based on double-acting THC, fluid that enters through port B exerts a force on the opposite side which makes THC retract back to its initial position.

Fig. 5.7 shows the pressure vs time plot of the pulse input signal Fig. 5.6. The pressure inside the THC starts rising as the valve opens at 4.4th second. As the pressure rises sufficiently to move the 1st stage of the THC assembly the pressure rise consolidates for around 1.5 seconds. In the meantime, 1st stage of the THC subsystem extends to the fullest. For the 2nd stage, the area of the piston reduces due to this the pressure must rise in order to push the 2nd stage out. Due to this, at the 6th second, there is a sharp rise in the pressure, and the 2nd stage cylinder extends out. Now for port B pressure signal which is at zero initially. At the 4th second, there is a slight peak in the signal because of the backpressure caused in the order to push hydraulic pressure back into the reservoir through port B so that the THC subsystem can extend. Pressure at port B rises at the 8th second to retract the THC assembly. During the rise of pressure at port B, there is a peak in port A pressure which is again due to the backpressure causing the hydraulic fluid to flow back to the reservoir through port A. When the highly pressurized hydraulic fluid from port B enters the THC subsystem it applies a force on the opposite side of the cylinder block causing it to retract back to its initial position. In Fig. 5.7 the port A pressure plot shows the closing of both stages one by one. At around the 9th second, the 2nd stage of the THC subsystem closes followed by the 1st stage. The pressure drop in the port A curve and pressure rise in the port B curve are visible during this time period in Fig. 5.7.

Fig. 5.8 shows the plot between the displacement of the THC with respect to time. Initially, the displacement is zero for up to 4 seconds. At around the 4th second when the hydraulic fluid starts entering the THC through port A 1st stage cylinder starts extending. In Fig. 5.8, there is a linear rise up to 133 mm which is the stroke length of the 1st stage cylinder. After that as pressure rises further extension of the 2nd stage cylinder begins. Higher pressure is required to push the 2nd stage therefore, the slope of the plot is more for the second stage. There is a slight curve at the peak around the 6th second caused due to damping. At the 8th second one small additional stage is seen in the plot which is due to back pressure caused by flow reversal for retraction of the THC assembly. From Fig. 5.7 the plot shows a peak in the port A curve at around the 8th second due to this an additional small stage is seen during retraction in Fig. 5.8. After that 2nd stage retracts first followed by the 1st stage.

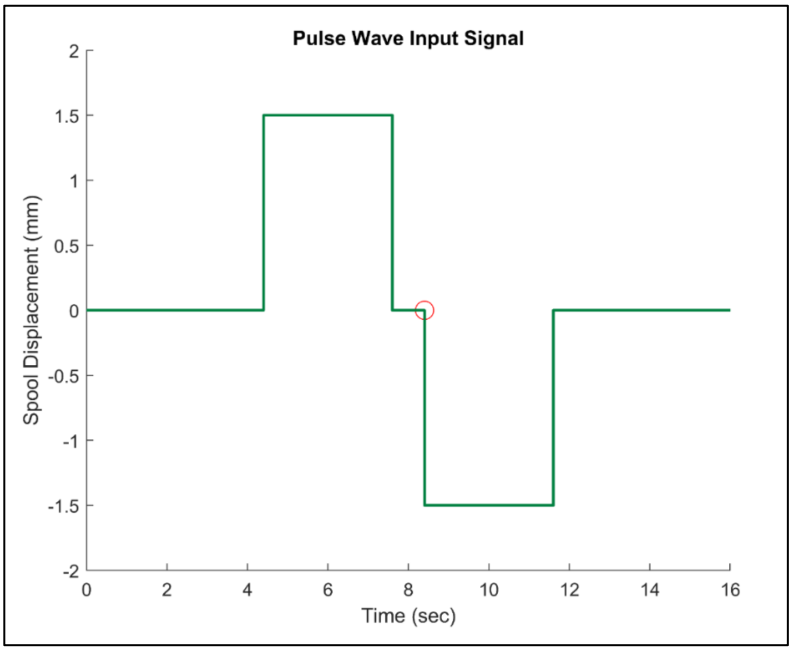

Fig. 5.6: Pulse Wave Input Control Signal for 4-way Directional Valve

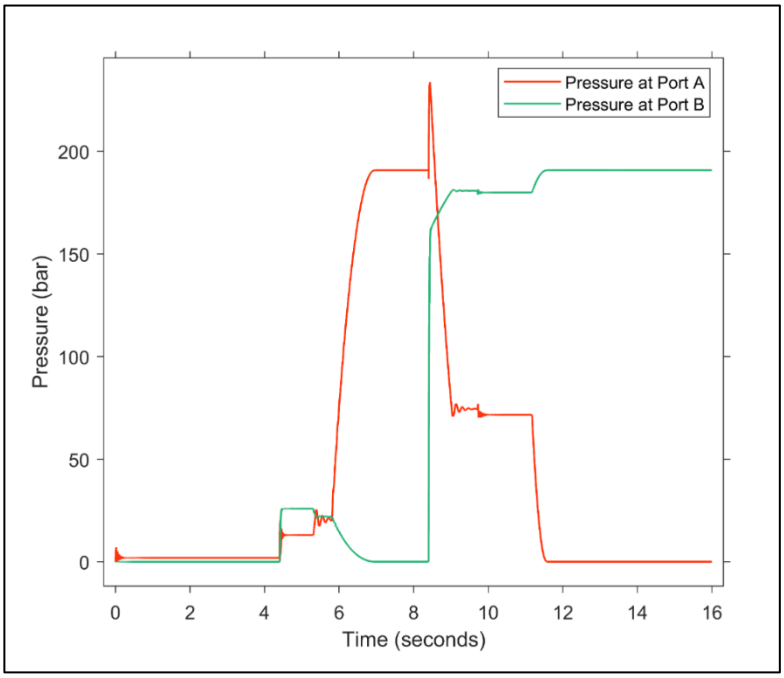

Fig. 5.7: Simulation Pressure Plot based on the Pulse Wave Control Signal

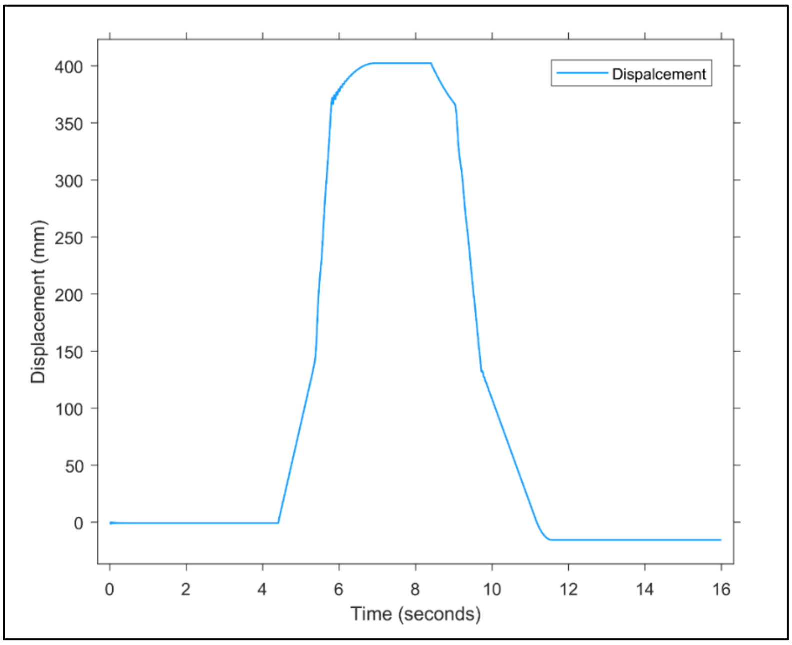

Fig. 5.8: Position-Time Graph of THC for given Input Control Signal.

Fig. 5.9 shows a control signal to operate the 4-way Directional Control Valve. The input signal that controls the spool inside the 4-way directional valve is shown in Figure 5.9. The flow of pressurized hydraulic fluid to the THC subsystem is controlled by this spool movement. As previously stated, we are sending input to the directional valve and putting a constant load of 2100 N to the THC in the simulation model. This control signal is taken from [21], [22], [29]. Lei et al and Abed Al-hady et al used this control signal in their respective work. Their study is focused on multistage single-acting THC. In the simulation model, the same control signal is used for double-acting THC and studying the output pressure and position signal obtained from the THC assembly. This is a special type of pulse signal which provides a slack of 0.15 seconds to open the spool from 0 to 3 mm. Initially, the valve is closed at 1.1 seconds signal instructs the 4-way directional valve to start the spool movement, and at 1.25 seconds the spool moves 3 mm from its initial position. This causes the highly pressurized hydraulic fluid to flow from port A to the THC subsystem. The flow is maintained till 10.65 seconds after that the spool starts closing and at around 10.95 seconds the spool is completely returned to its initial position. The spool stays at the initial position for a moment and at around 11.7 seconds it again starts to move but this time in the opposite direction. Due to this, the highly pressurized hydraulic fluid starts entering the THC assembly through port B. This causes the retraction of the THC subsystem.

Fig. 5.10 shows the pressure vs time plot of the custom input signal Fig. 5.9. The pressure inside the THC starts rising as the valve opens at 1.1 seconds. As the pressure rises sufficiently to move the 1st stage of the THC assembly the pressure rise consolidates for around 1 second. In the meantime, 1st stage of the THC subsystem extends to the fullest. For the 2nd stage, the area of the piston reduces due to this the pressure must rise in order to push the 2nd stage out. Due to this, at around the 3rd second, there is a sharp rise in the pressure, and the 2nd stage cylinder extends out. Now for port B pressure signal which is at zero initially. In the 1st second, there is a slight peak in the signal because of the backpressure caused in the order to push hydraulic pressure back into the reservoir through port B so that the THC subsystem can extend. Pressure at port B rises at around 11.5 seconds to retract the THC assembly. During the rise of pressure at port B, there is a peak in port A pressure which is again due to the backpressure causing the hydraulic fluid to flow back to the reservoir through port A. When the highly pressurized hydraulic fluid from port B enters the THC subsystem it applies a force on the opposite side of the cylinder block causing it to retract back to its initial position. In Fig. 5.10 the port A pressure plot shows the closing of both stages one by one. At around the 13th second, the 2nd stage of the THC subsystem closes followed by the 1st stage. The pressure drop in the port A curve and pressure rise in the port B curve is visible during this time period in Fig. 5.10.

Fig. 5.11 shows the plot between the displacement of the THC with respect to time. Initially, the displacement is zero for up to 1st second. At around the 1.1 seconds when the hydraulic fluid starts entering the THC through port A 1st stage cylinder starts extending. In Fig. 5.11, there is a linear rise up to 133 mm which is the stroke length of the 1st stage cylinder. After that as pressure rises further extension of the 2nd stage cylinder begins. Higher pressure is required to push the 2nd stage therefore, the slope of the plot is more for the second stage. There is a slight curve at the peak around the 3.5 second caused due to damping. At the 12th second one small additional stage is seen in the plot which is due to backpressure caused by flow reversal for retraction of the THC assembly. From Fig. 5.10 the plot shows a peak in the port A curve at around the 12th second due to this an additional small stage is seen during retraction in Fig. 5.10. After that 2nd stage retracts first followed by the 1st stage.

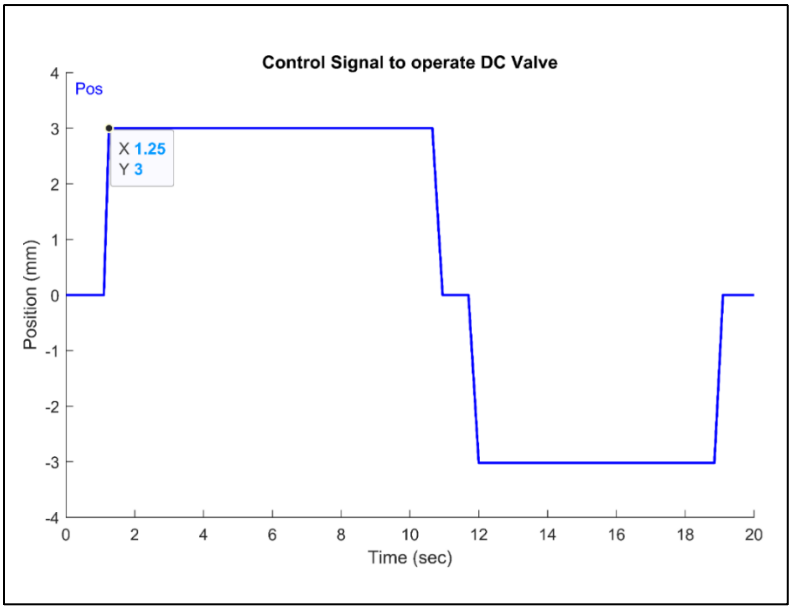

Fig. 5.9: Custom Pulse Wave Input Control Signal for 4-way Directional Valve

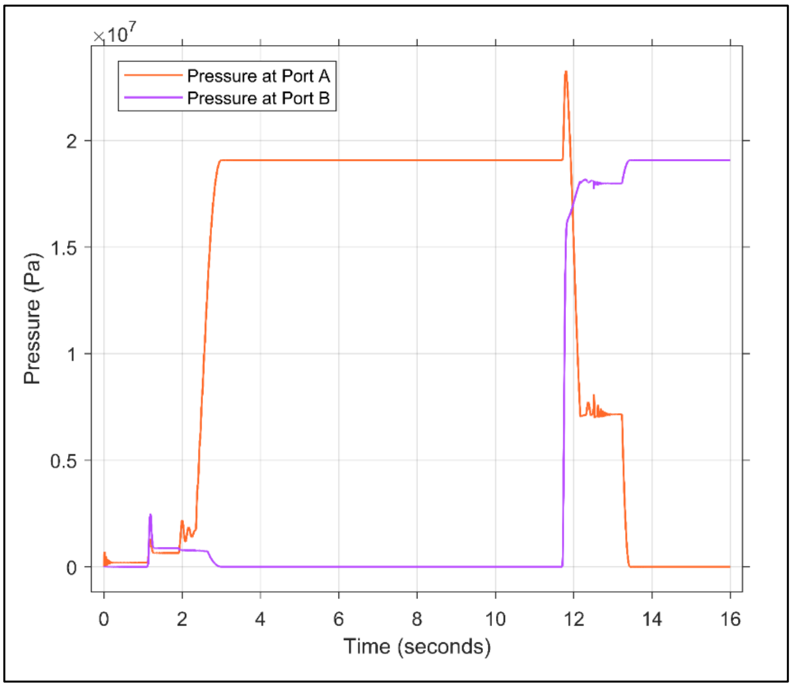

Fig. 5.10: Simulation Pressure Plot based on the Custom Pulse Wave Control Signal.

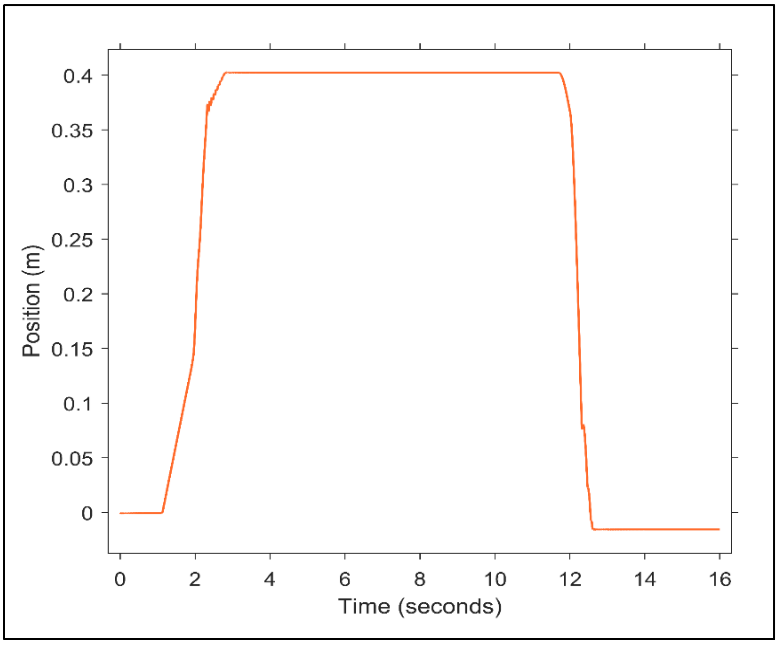

Fig. 5.11: Position-Time Graph of THC for given Input Control Signal.

Chapter 6: Conclusion and Future work

6.1 Conclusion

In this work, a detailed study on the working and modeling of hydraulic components is carried out using experimental and simulation help. A simulation model of a 2-stage double-acting THC is created using the MATLAB Simscape platform. This simulation model is validated experimentally using experimental setup parameters and mimicking the same input conditions. The output pressure plots of THC during one complete cycle of both simulation and experimentation are studied together. It was found that trends followed in both the plots are similar to each other. The number of peaks and valleys is the same in both plots. Fig. 5.5 represents both combined plots of both simulation and experimentation. The peak pressure observed in simulation is 190 bar while in experimentation it hovers around 189 bar to 191 bar which is quite acceptable. However, there are some small discrepancies between the simulation and experimental plots which is due to the limitation of the simulation to consider environmental factors and hidden losses.

Further, the simulation model is simulated under two more input control signals. One is the pulse wave loading control signal and the other is the loading control input signal used in two of the published literature. The characteristic plot of position and pressure are studied and their interpretation is explained in chapter 5.

6.2 Future Scope

This validated simulation model can be used to design multi-stage double-acting THC and study various fault severities without introducing faults in the actual system. The introduction of faults in the actual system is a tiresome and costly affair. For preliminary diagnostics of the system, one can refer to data generated by this simulation model after optimizing it using the parameters of the system under consideration. This modeling and simulation direction may be extended further toward the Digital Twin development of the hydraulic system capable of real-time monitoring of the system.

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