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# Modeling And Simulation Swash Plate Pump Response Characteristics in Load Sensing And Pressure Compensated Hydraulic System

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***ABSTRACT:*** *Fluid Power is widely employed in applications required high loads such as tractors, cranes, and airplanes. In load sensing hydraulic systems, loads are controlled by adjusting a pump-valve arrangement. In this paper, the swash plate pump hydraulic characteristics will be determined, the pump and its fluid gains will be derived to obtain the pump overall transfer function. Firstly, the swash plate pump mechanism is analyzed and its dynamic model is constructed; the pump pressure and flow rate are plotted and the possible improvement is introduced. The load sensing unit parameters such as orifice width, orifice area, maximum passage area, and piston area at X and Y will be examined to identify their influence on the pump characteristics; and the optimum parameters will be introduced. All results are developed and simulated numerically.*

***Keywords:*** *Swashplate pump, load sensing control, pump characteristics, pump flow rate, pump delivery pressure.*

## I. INTRODUCTION

In the past twenty years, a remarkable attention has been directed toward the variable displacement swash plate pumps. These studies covered several aspects of the pump such as the pump kinematics and pump characteristics control to achieve the optimum performance. One interesting study has been presented by Tanglin [1]. Tanglin proposal based on controlling the pump characteristics by controlling the prime mover speed. Akers and Lin [2]have modeled the axial piston pump and controller the pump pressure as they have used a single stage electro-hydraulic valve to control the pump flow rate. They used a step function and tested the non-stationary response by implementing the optimum control theory for the control open loop. There was a remarkable improvement in the pump performance. One more study was conducted by Kaliftas and Costopolus [3]. They modeled and tested the dynamic and static pump characteristics. The pump was equipped with a pressure regulator. The results have been collected numerically and matched with the pump operations' curves. In [4], Johnson and Manring have modeled the axial swash plate pump and investigated the effect of a defined loads on the pump characteristics. In the last few years, the swash plate pump with the conical arrangement was introduced. In the conical arrangement, there is an angle between axial coordinate and the pistons' axis. This arrangement has a remarkable advantage over the cylindrical arrangement as the detachment force has been reduced. The detachment force tends to remove the piston from its slipper pad. The conical pump has been studied extensively, some good examples can be obtained in [5,6,7,8]. Khalil el al. [6] has modeled and simulated the swash plate and investigated the effect of the pump kinematics on the pump flow rate and pressure ripples. Also, Khalil in another study has investigated the influence of the controller specifications on the pump characteristics [6,7, 9]. Further study has been conducted by Chikhalsouk. The study has concluded a simplified geometrical pump kinematic expressions and accordingly, the pump characteristics have been investigated. Another important technique in swash plate pump characteristics control is load sensing control. The load sensing swash plate pump consists of the pump and control valves, which automatically adjust the pump characteristics i.e. the pump flow rate and pressure to meet the load requirements, and this maintains the hydraulic flow quality, the pump efficiency and avoids the throttling overflow losses. Thus, the load sensing swash plate pump has relatively fewer energy losses and higher efficiency in comparison to other types of hydraulic pumps. In order to improve the pump operating stability and its performance, it is very vital to optimize the pump construction. The complex structure for the swash plate sensing pump makes the mathematical modeling is pretty challenging. Hence, the conventional development approaches are hard to comply with the modern pumps' designrequirements.

Load sensing systems have become very popular in the past two decades, especially in automobile and heavy duty machinery [10, 11, 12]. One important advantage of these systems is the high potential of energy saving; where is the pump meets the power requirements to the various loads with the least control losses. However, a kind of interaction amongst loads and an instability is reported [13].In load sensing pump, the pressure is sent back to the pump controller to adjust the swash plate swiveling angle and pump flow rate accordingly.

The remarkable advancements in computer simulation grant researchers with an effective and convenient platform to design the pump[14, 15]. The simulation tests enable the researchers to discover the potential design problems and enhance the required parameters to achieve the desired performance and then building the prototype. Some of the computer simulations that can significantly simplify the development process and shorten its cycle, and substantially reducing the development costs which lead to the ideal design scheme [16, 17]. In the past few years, the computer simulation has been employed in improving the design of the hydraulic pump and has shown a remarkable success [18, 19]. For example in [20], Cho et al. have used AMESim software to model and simulate the conical swash plate pump and studied and analyzed the piston pressure fluctuation. In [21],Baek et al. have extended their studies to include the piston behaviors under disk eccentricity ratio. The axial piston pump characteristics and the effect of the internal leak on the pump pressure and flow rate were investigated by Bergada [22]. Casoli et al. have modeled and simulated an excavator's arm powered by swash plate pump by using gray box technique, and have obtained fast excavation cycle[23]. The pressure control by pressure compensator for swash plate pump was studied by Mandal in [24]. In [25], Xu et al. have studied the optimum structure for anti-overturning slipper of swash plate pump to enhance the service life and the swash plate pump reliability. Roccatello et al. have modeled and simulated multi subsystems of a swash plate pump based on software technology and tested the swash plate pump dynamic response characteristics [26]. The Zhu et al.'s study has added a significant analysis of the swash plate load sensing pump [27], however, more work needs to be achieved.Other studies have concentrated on the structure and optimum parameters of the load sensing swash plate pump [28, 29].

In this paper, the swash plate pump is selected for the research object. The load sensing and swash plate pump structure are explained. The mathematical model is developed. The test rig specification and structure are detailed. The pump characteristics are obtained. The load sensing unit parameters influence on the pump characteristics are studied. Finally, the optimum load sensing parameters are recommended.

**Load Sensing Unit with Swash Plate Pump**

The load sensing arrangement consists of a variable displacement pump i.e. swash plate pump, an actuatingyoke, a pump control piston, and a critically lapped adjusting valve. The orifice is controlled by two pressures, which are upstream(pump pressure) and downstream (load pressure).When the load increases, there will be an increase in pump feedback sensing load pressure. Hence, there will be a change in the pressure acting on the spool. The spool is experiencinga unbalance force and shifting it to the right and connecting to the tank. On the other hand, a decrease in the fluid pressure will increase the swash plate swivelingangle, which increases the pump flow rate. This increase in the pump flow experiences a higher resistance in the controlling orifice, which increases the pressure. The pump feedback line pressure is sensed, which affects the compensator piston right handed side. The pressure keeps building up till the balance occurs through the compensator. Accordingly, the orifice pressure drops and the flow is adjusted reaching to its original level (control flow). In other words, in load sense control, the pump flow increases or decreases to main the differential pressureacross the orifice. Thus, the flow will be constant for the same orifice opening regardless the load condition or the prime mover revolution. Accordingly, the pump will use only the enough power to maintain suitable flow. A limiting pressure control is a controlled method to set the maximum system pressure. The pressure limiting valve is installed to hinder pump pressure to go beyond the preset pressure. The load sensing valve preloaded spring is calculated with the following equation as:

$$f_v \geq \Delta p.A_v \tag{1}$$

where

$\Delta p$ desired differential pressure across the control valve in Pa

$A_v$ control valve cross-sectional area in m$^2$

3. Pressure and Flow Control Arrangement for the Swash Plate Pump
The control unit consists of a 3-way hydraulic directional valve and a double-acting servo cylinder that controls the pump yoke. The load sensing is enabled by keeping a predetermined pressure differential through the flow control valve. These pressures act upon the sides of the directional valve and move it correspondingly to the pressure difference.

**The control unit elements are:**

**A.** Flow Control Variable Area Orifice: The variable orifice consists of a cylindrical sharp-corner spool and a rectangular opening in a sleeve. The orifice flow rate is proportional to the pressure differential through the orifice faces and its opening. The connections A and B are maintaining hydraulic channels with the orifice inlet/outlet. The positive direction is from A to B, where the positive signal at spool "S" opens/closes the orifice according to the orifice orientation parameters.

**B.** Load Sensing Fixed Orifice: Is s squared corner fixed area orifice. The flow rate is proportional to the pressure differential through t the orifice. Ports A and B are maintaining hydraulic channels linked with the orifice inlet/outlet. The positive direction is from port A to port B, which means that when the flow from A to B is counted positive, and the differential pressure is computed as $P = P_A - P_B$.

**C.** Pressure Relief Valve: it opens as the pressure exceeds the valve preset pressure, where value control member is pushed against its seat and initiating a gap between the outlet and inlet. This relief sends some of the fluid back to the tank and decreases the pressure at the inlet. Yet, the pressure starts to be built as the flow rate is insufficient. At this condition, the area increases till the control element reaches its peak. Connections A and B are maintaining channels and the working direction from A to B.

**D.** Double Acting Calve Actuator: It uses as a pilot actuator for pressure/directional/flow control valves, where all flow consumption and forces can be neglected except spring force. The actuator has two single acting actuators advancing against each other. Each actuator contains a centering spring with its washer and a piston. When a port is under a control pressure, one centering spring is compressed across its washer only, while will exert no force. When a pressure control is released on the both ports, the spool centers between them. This design enables each actuator to have its own design parameters. When a control pressure applies, the piston will move against its spring. The different control unit elements are listed in Figure 2.

**4. Swash Plate PumpStructure**

As shown in Figure 1, the swash plate pump consists of a certain number of pistons moving inside their cylinders and combined in a common block named the cylinders block. The cylinders/pistons are enclosed in an annular array within the cylinders' block at an equal interval around the axial coordinate. The cylinders block is retained securely aginst the port plate by the aid of the cylinder block spring compressed force. A ball and socket joint links the piston base to its slipper. The slippers are maintained in touch with the swash plate and the swash plates swiveling angle is controlled by a servomechanism depending on the delivery pressure/ flow rate demands. In actual applications, the loadon the hydraulic actuator may alter from instance to instance, which will require the swash plate pump to produce a several operating pressure/ flow rate accordingly.

## II. Mathematical Modeling

The analytic method is the most popular method to model and analyzes the hydraulic pumps in general and swash plate pumps in particular [ 30, 31], state space method [32], and bond graph approach [ 33]. In this work, the dynamic model for the load sensing pump will be built based on analytic approach.

1) Load Sensing Valve Dynamic Equation:
The load sensing valve dynamic expression can be written as:

$$M_v.\ddot{x}_v + C_v.\dot{x}_v + K_v.x_v = (P_s - P_l)A_v - F_0 \tag{2}$$

Let's name total inputs as $f_e$, hence, the load sensing valve transform function can be obtained by Laplace transform

$$G_1(s) = \frac{X_v(s)}{F_e(s)} = \frac{\frac{1}{K_v}}{\frac{s^2}{\omega^2_{nv}} + 2.\varsigma_v \frac{s}{\omega_{nv}} + 1} \tag{3}$$

2) Swash plate dynamic equation:
The dynamic equation of swash plate[20] can be expressed as:

$$k_q.x_v = A_1.\dot{x}_p + \frac{J.V}{A_1.l_0^{\,2}.\beta}.\dddot{x}_p + (K_p + C_0)\frac{J}{A_1.l_0^{\,2}}.\ddot{x}_p \tag{4}$$

Hence, the swash plate transferfunction can be obtained by Laplace transform

$$G_2(s) = \frac{X_p(s)}{X_v(s)} = \frac{\frac{K_q}{A_1}}{s(\frac{s^2}{\omega^2_{np}} + 2.\varsigma_p\frac{s}{\omega_{np}} + 1)} \tag{5}$$

3) Pump output flow rate characteristics. The pump flow gain equation of the pump [ 34] can be written as:

$$Q_{pump} = -K_Q.n.x_p \tag{6}$$

The pump output flow rate gain can be obtained by Laplace transform as

$$G_3(s) = \frac{-Q_{pump}(s)}{X_p(s)} = K_Q.n \tag{7}$$

4) Pump pressure characteristics. The pump flow rate variation causes the pressure variation. Hence, the pressure differential equation can be written as follows:

$$-Q_p + Q_l - c_l.P_s = \frac{V_t}{\beta}\dot{P}_s \tag{8}$$

Consequently, the pump pressure may be expressed as

$$G_4(s) = \frac{P_s(s)}{-Q_p(s) + Q_l(s)} = \frac{\frac{1}{c_l}}{1 + \frac{s}{\omega_t}} \tag{9}$$

Open loop transfer function of the load sensing swash plate pump is the product of the four gains i.e.

$$G(s) = \frac{(\frac{1}{K_v})}{(\frac{s^2}{\omega^2_{nv}} + 2.\varsigma_v\frac{s}{\omega_{nv}} + 1)}.\frac{(\frac{K_q}{A_1})}{s(\frac{s^2}{\omega^2_{np}} + 2.\varsigma_p\frac{s}{\omega_{np}} + 1)}.\frac{(K_Q.n)(\frac{1}{c_l})}{(1 + \frac{s}{\omega_t})} \tag{10}$$

Equation (10) represents the load sensing swash plate pump transfer function. Two oscillatoryelements, one inertial element, and one amplifying element are the cascade gains for the open loop of the equivalent transfer function for the pump. The pump response in the time domain can be represented by the fundamental frequency of the elements, at the same time, the vibration characteristics of the system can be figured out by the elements damping coefficients.

Hence and in order to improve the system performance, there is a need to lessen the influence of the first order inertial element and secure that the second order vibratory element plays the leading role in the system. Increasing the corner frequency of the first order to increase the pressure/ flow coefficient of the load system can be one a suitable approach. Also, decreasing the load sensing valve spring stiffness or decreasing the piston area of the variable hydraulic cylinder are other powerful approaches.

**5. Test Rig Construction**

The objective of the test rig is to investigate the interaction between the swash plate pump and the control unit, and the same time achieving the load sensing and the pressure control goals. To secure the vital accuracy, the pump model should be derived in the way that considers this interaction among the pistons, the swash plate, and the port plates, which makes it very critical to constructing the comprehensive pump model.

**5. 1Test Rig Outline**

The schematic representation of the system test rig is illustrated in Figure 3. The pump model is symbolized with the name Swash Plate Pump. The prime mover, which is the source of pump mechanical power, is simulated with the Ideal Angular Velocity source. The pump outputs/ delivery are a hydraulic pipeline and two variable orifices. The flow rate is adjusted by the flow control orifices; in this work, the opening is remained fixed during the simulation. The load sensing pump needs to reserve a constant differential pressure across the orifice regardless the swash plate loading. The loading pump is represented in the simulation as the load orifice block. In order to investigate the control unit for several load conditions, the load orifice area is changed during the simulation. The changing profile is utilized by the Load Signal Builder block. The control unit is symbolized in the test rig by a three-way valve pressure controlled, a pressure relief valve, flow control orifice, and an orifice. The control unit senses signals on the pump pressure and the load pressure, measured post the valve of flow control. According to these pressures, the yoke displacement is generated, which limits the angular swash plate swiveling angle i.e. the piston stroke. In this way, the differential pressuresmaintained across the flow control valve and eliminate the pump to generate an excessive pressure set by the manufacturers. The fundamental test rigs parameters are listed in Table 1.

**5.2: Test Rig Model**

The models of the swash plate pump and the control unit are built.

**5. 2. 1. Swash Plate pump Model**

The pump under investigation is a swash plate pump. The pump block diagram is illustrated in Figure (4). In Figure (4), S is symbolizing the pump driving shaft (port 3), Y for the yoke linked to the inclined swash plate (port 1), and B is the pump discharge pressure (port 2). All pistons are symbolized by a subsystem named piston. These pistons' models are identicals and linked to pump external pump's ports. Every piston is linked to the low port (suction port i.e. A), which is simulated by with the Ideal Hydraulic Pressure Source block. The booster pressure set point is 5 bar. The yoke is linked to the pistons' actuator ports, hence operating on the inclined swash plate. The yoke displacement is controlled by the hard stop function.

**5.2.2: Pressure and Flow Control Subdivision Model**

The control subsystem is exhibited on Figure (2), such that is constructed by a three-way directional valve, double acting hydraulic valve actuator, pressure relief valve, and non-variable orifice blocks. The port B and A are linked to the up and downstream of the control orifice in Figure 2. The flow control orifice differential pressure is selected to be 20bar. In Figure (2), the lower position of the 3-way directional valve needs to be preliminary open, to push the swash plate pump to boost its displacement at the beginning of the operation. To conduct the load sensing objective, the pressure rise at the LSP opening have to open the 3-directional valve lower position. These are the major consideration in valve port to system connection. The other parameters such as orifice area, valve length, and spring stiffness are adjusted during the simulation to guarantee the effectiveness and numerical stability. The pump needs to be operated within the power limitations, and hence, the pressure limiting function is applied in joint with the non-variable orifice and pressure relief valve blocks. The relief valve set value is selected to be 250bars. This value is suitable to decrease the pressure of the double acting valve actuator, which is linked to the yoke, as the pressure increases at the non-variable orifice. This action decreases the pump displacement.

**5.3: Test Rig Data**

The load sensing system is investigated by simulating the derived models with appropriate parameters, which are summarized in Table (2).

**5.3.1: Simulation Data and Results**

The simulation cycle contains several elements defined by several load conditions that affect the variable area block whose opening is adjusted by spool position in Load Signal Builder block, as illustratedin Figures(5)Figure (6)shows the load sensing evolution with respect to time. From the figure, the simulation cycle starts with zero opening for 0.25 sec, and then, the opening increases linearly and becomes 2.8 mm at 0.31 sec. This opening remains constant for 0.20 sec, and another linear increase takes place and reaching the maximum valve 5.2 mm at 0.52 sec. it remains constant for 0.25 sec and then decreases to 1, and another decrease (-0.8 mm). Finally, the opening settles at 2.5 mm. At the beginning of the cycle, the pump shaft rotates at 260 rad/sec and the initial position of the yoke is selected to be 5 mm. The pump flow rate and pressure are illustrated in Figures (7-1) and (7-2). As can be noticed from the Pressure Profile, the servo cylinder begins to increase the pump displacement, the pump pressure increase to reach 200 bars at 0.24 sec, where the differential pressure across the flow control valve gets close to the pressure set value; the load sensing valve opens at this instant.

Afterward, the pressure settles at 75 bars and this lasts till 0.5 sec, hence, the pump pressure drops to 50 bars and this last for 0.25 sec (till 0.75 sec). Another two pressure increase cycles take place at 0.85 sec and 1.1 sec, respectively. Through these three cycles, the pump holds almost the same flow rate in spite of alternations of the opening of the load valve. The limiting function dominates the pressure rise as it reaches 270 bars and the pump reacts to the load sensing regime allowing the pressure to decrease. The simulation results exhibit that the pump flow rate and pressure characteristics and the load sensing variable pump are compatible. Hence, the swash plate pump characteristics can be dynamically simulated and analyzed by using the model.

## 6. Load Sensing System Components Simulation and Optimization

In this section, the pump flow and pressure characteristics govern the pump fundamental parameters are discussed; then, a series of simulation runs are conducted to investigate the impact of the several loads sensing unit parameters on the pump external characteristics i.e. the pump flow rate and the pressure. As mentioned earlier, the load sensing unit consists of the 3-way directional valve, the double acting servo cylinder which stroke/ stroke the control piston, the variable control orifice, the load sensing fixed orifice, and the pressure relief valve. The objective of this simulation is to identify the most sensitive parameter for improving the pump characteristics. The simulation matrix is formulated and the parameters are selected. The parameters are Flow Control Orifice with Variable Area Slot /Orifice Width, Load Sensing Fixed Orifice/ Orifice Area, Pressure Relief Valve/ Maximum Passage Area, and Hydraulic Double Acting Valve Actuator/ Piston Area at X and Y.

The parameters are grouped in Table (1), and the simulation results are plotted accordingly in Figure (8). Figure (8-1) illustrates the influence of flow control orifice width on the pump pressure. The orifice width is increased consecutively starting from the original value and increased to 1.2, 1.3, 1.4, 1.5, and 2 cm. Increasing the orifice width improves the pump pressure and the pressure fluctuation can be noticed at a high value of the width, which is 2 cm. Figure (8-2) shows the influence of flow control orifice width on the pump flow rate. It is notable that pump flow rate increase as the orifice width increase and the maximum width generates an excessive flow fluctuation which is not recommended for hydraulic systems. Figures (8-3 and 8-4) demonstrate the pump characteristics at different values for Load Sensing Fixed Orifice areas. The selected areas are 4, 6, 8, and 10 $mm^2$, respectively. The increase in the area has an impact on the pump flow rate at 1.1 sec and increases of the flow rate, however, there is a minimal fluctuation in the flow rate. Moreover, increasing the area has a positive impact on the pressure and the pressure overshooting dropped by 50 %. The effect of the pressure relief maximum area on the pump flow rate and pressure are represented in Figures (8-5 and 8-6). The simulation results show that there is no impact on the relief valve area on the pump characteristics. Figures (8-7 and 8-8) establish the relationships between the pump characteristics and the actuator area. Increasing the area would decrease the pump flow rate and pressure by 5% with every 5 $mm^2$ area increase. For optimum pump characteristics, it can be summarized the following observations: Increasing the variable orifice width up to 1.5 cm, increasing the fixed orifice area up to 10 $mm^2$, and the small increase in actuator area (up to 85 $mm^2$ ). On the other hand, no impact on the relief valve area on the pump characteristics as used for maintaining the pump pressure at the preset value.

## 7. Conclusions

The swash plate pump hydraulic characteristics were determined, the pump several gains and overall transfer functions were derived and determined. The load sensing unit parameters such as orifice width, orifice area, maximum passage area, and piston area at X and Y were first identified to simulate the pump characteristics. Then a set of simulation runs were conducted to obtain the optimum characteristics. It was found that increasing the orifice width reasonably would improve the pump characteristics. Also, the increase in the fixed orifice area has a positive impact. Moreover, the small increase in the actuator area could improve the pump flow rate and the pressure. However, there is no influence for the relief valve on the pump performance.

### Figures Captions:

LIST OF FIGURES

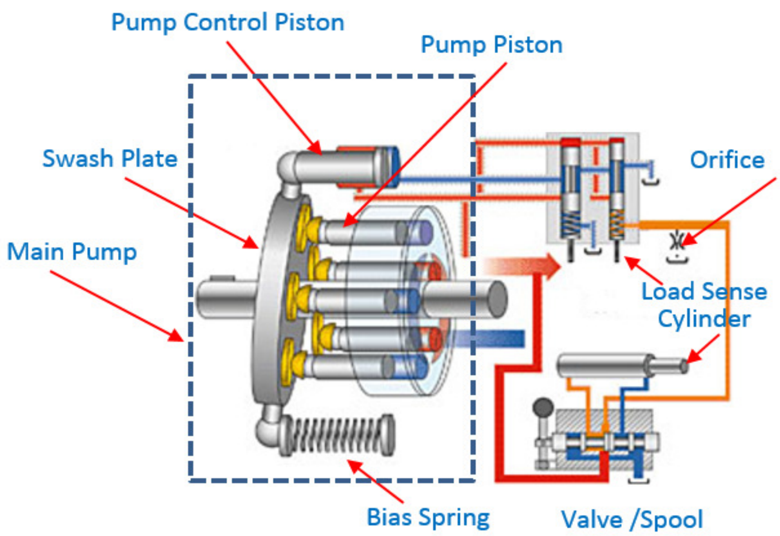


Figure 1: Load sensing unit with swash plate

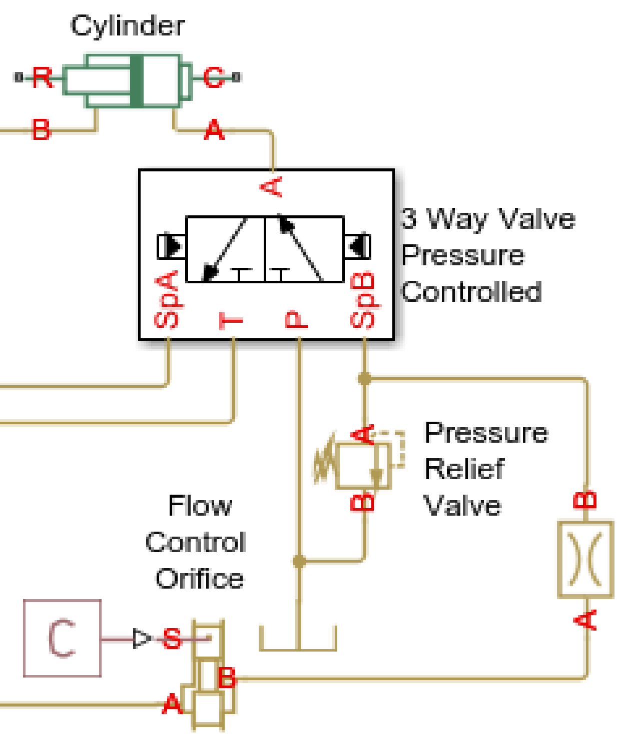


**Figure 2:** Schematic Representation of the Swash Plate Response Characteristics Control Unit

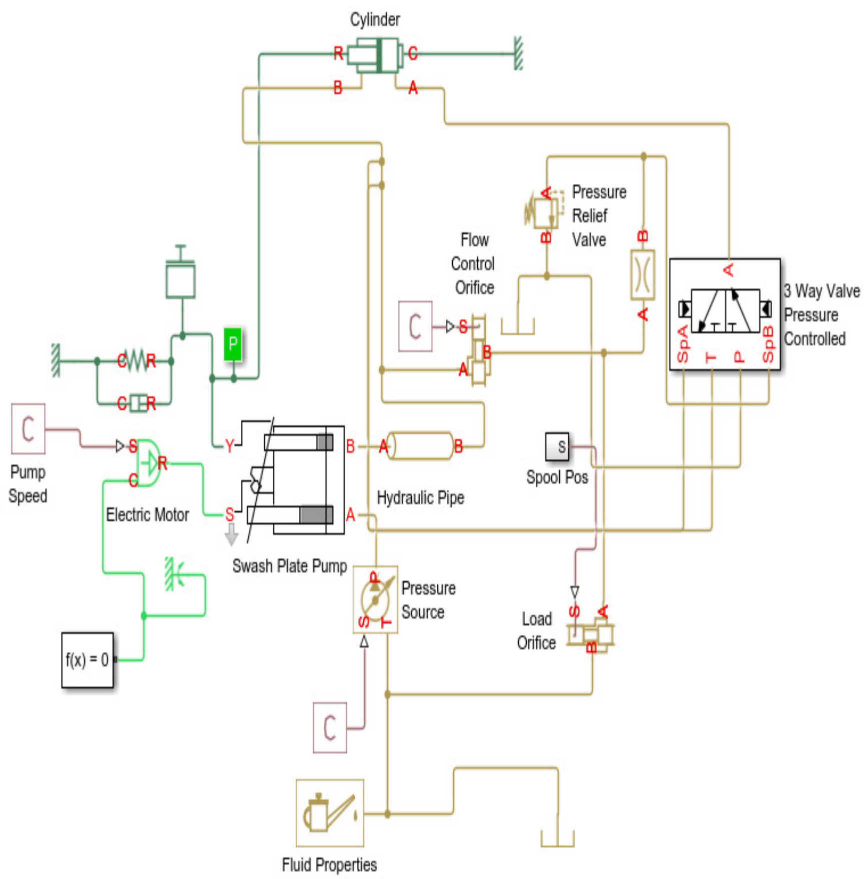


**Figure 3:** Schematic Representation of the Test Rig

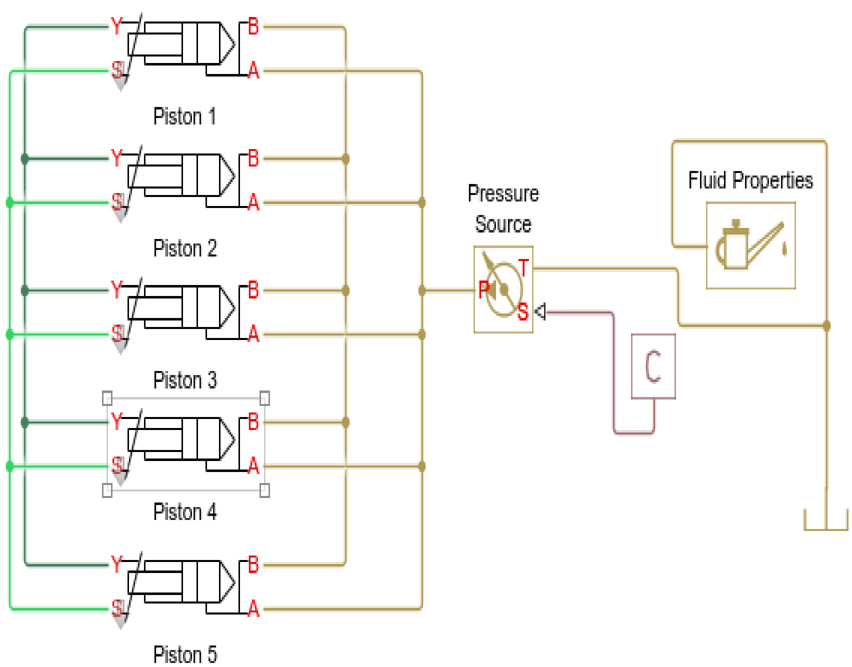


**Figure 4:** Schematic Representation of the Swash Plate

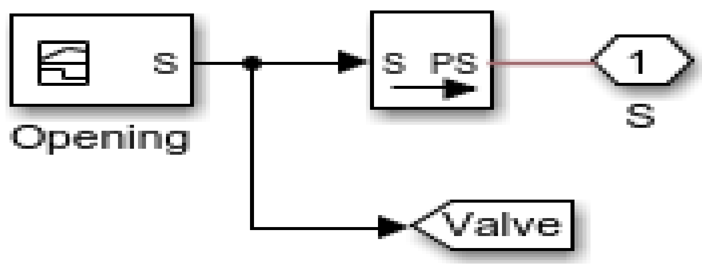


**Figure 5**: Schematic Representation of theSpool Block

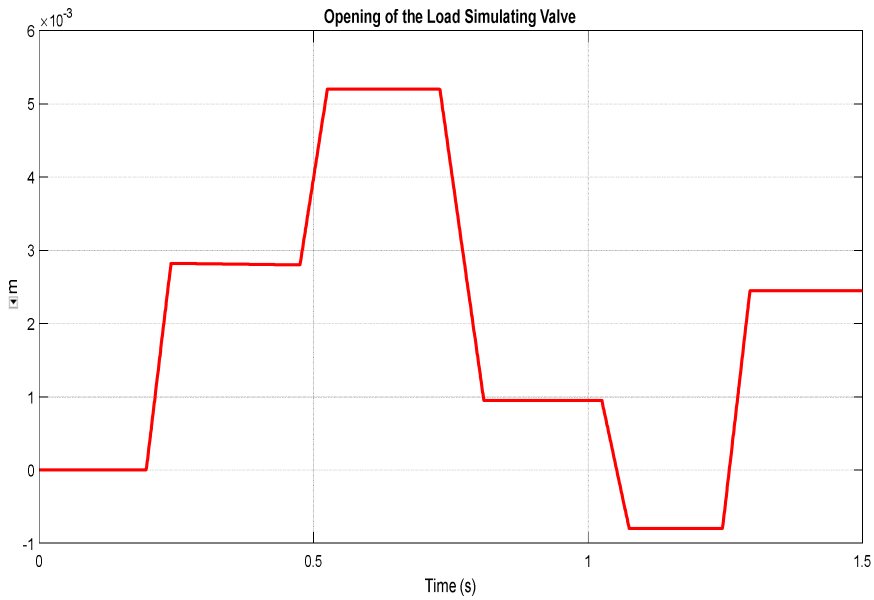


Figure 6: Load Sensing Valve Opening with Time

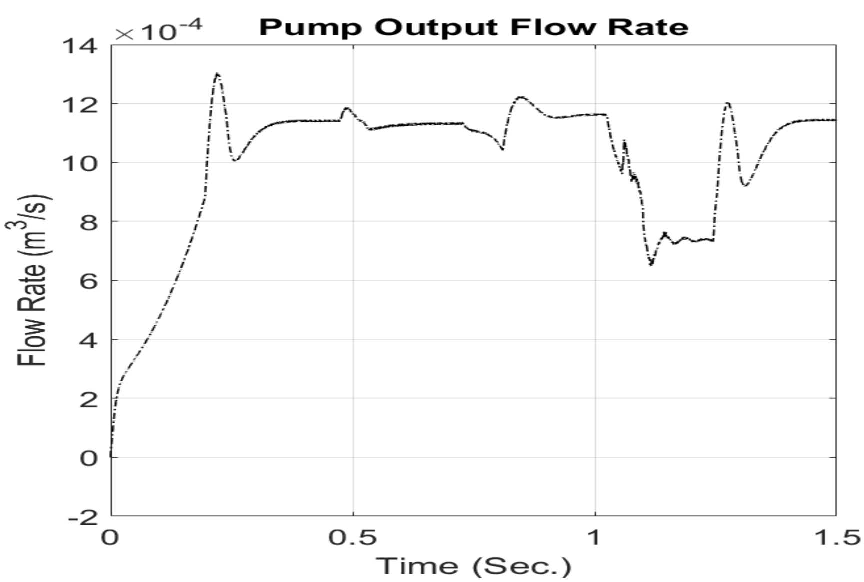


**Figure 7-1:** Pump Flow Rate Profile

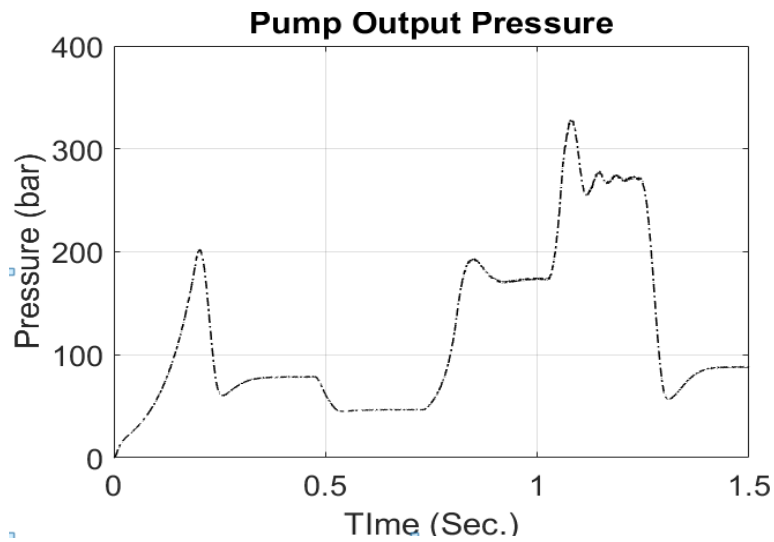


**Figure 7-2:** Pump Pressure Profile

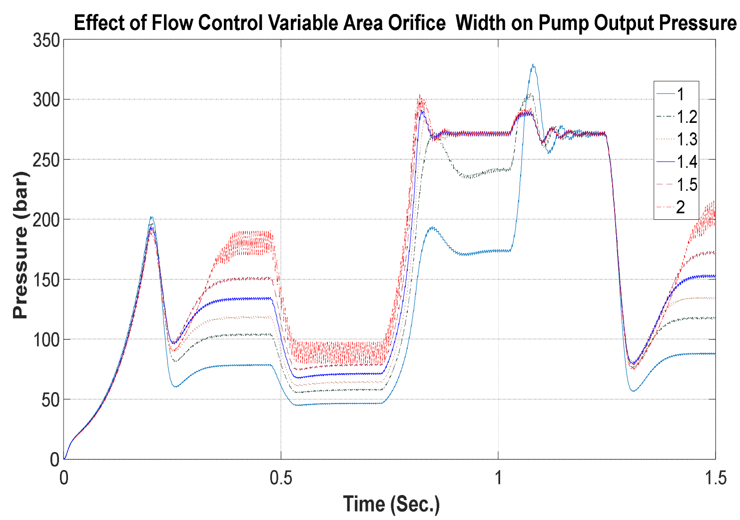


Figure 8-1: Pump Pressure Profile with Several Control Orifice Widths

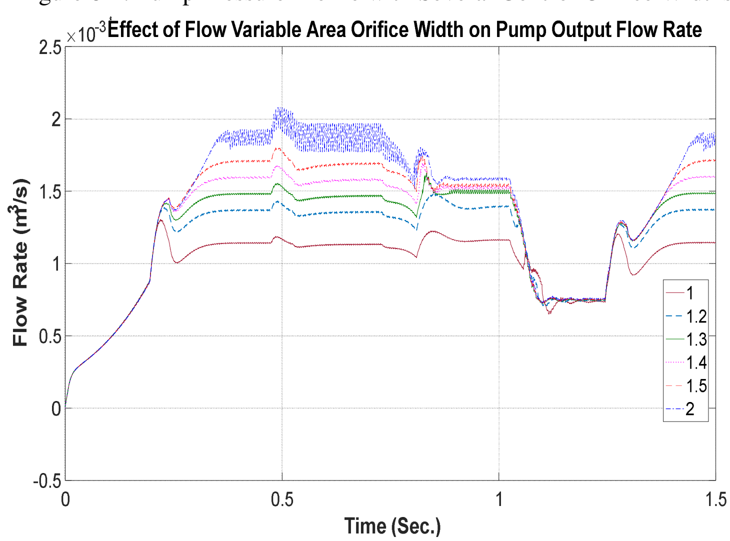


**Figure 8-2:** Pump Flow Rate Profile with Several Control Orifice Widths

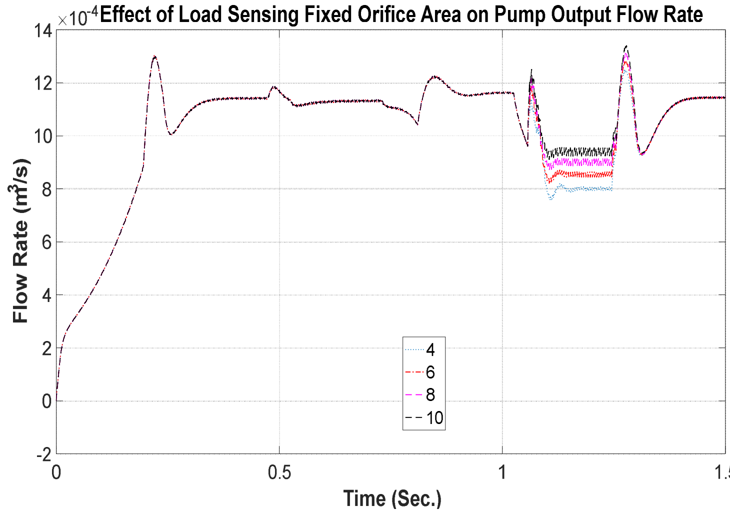


**Figure 8-3:**Pump Output Flow Rate with Several Load Sensing Fixed Orifice/ Orifice Area Values

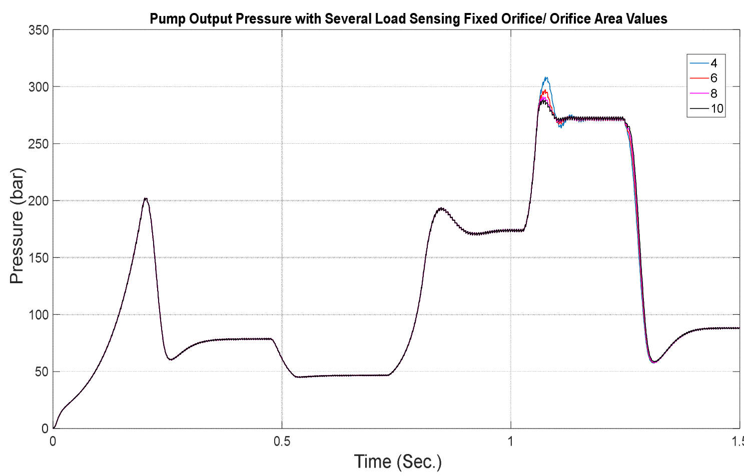


Figure 8-4: Pump Output Pressure with Several Load Sensing Fixed Orifice/ Orifice Area Values

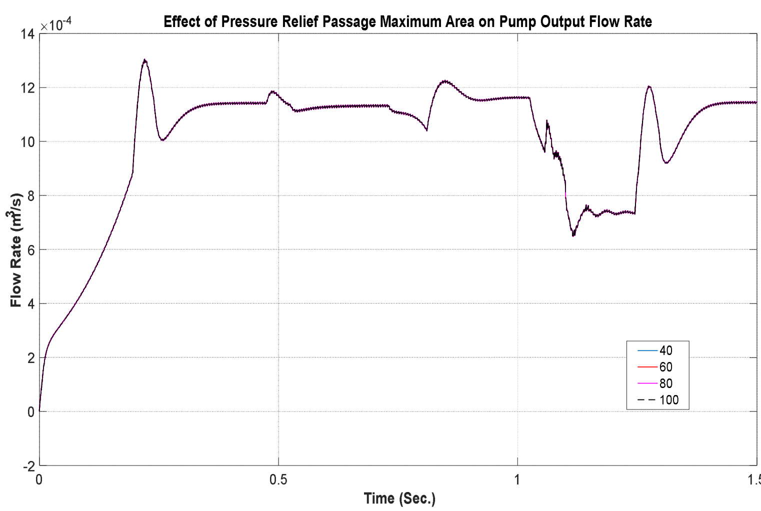


**Figure 8-5:**Pump Output Flow Rate with Different Pressure Relief Valve/ Maximum Passage Area Values

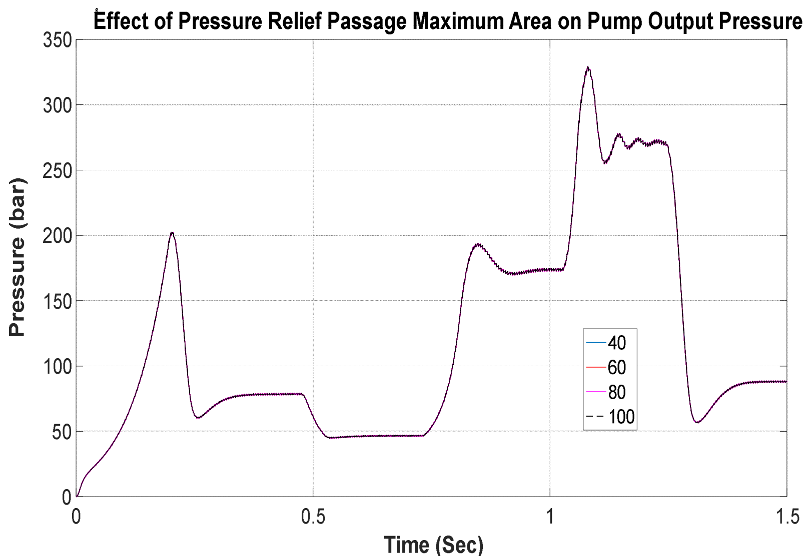


**Figure 8-6:**Pump Output Pressure with Different Pressure Relief Valve/ Maximum Passage Area Values

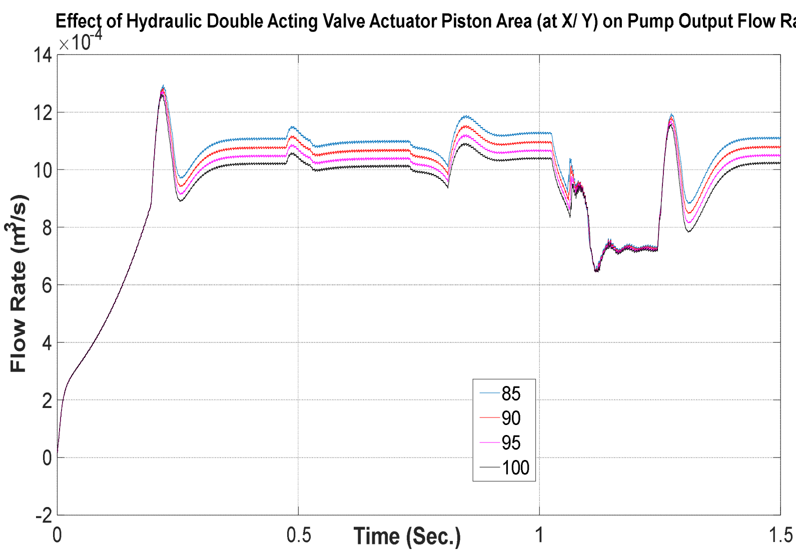


Figure 8-7: Pump Output Flow Rate with Several Double Acting Valve Actuator/ Piston Area Values

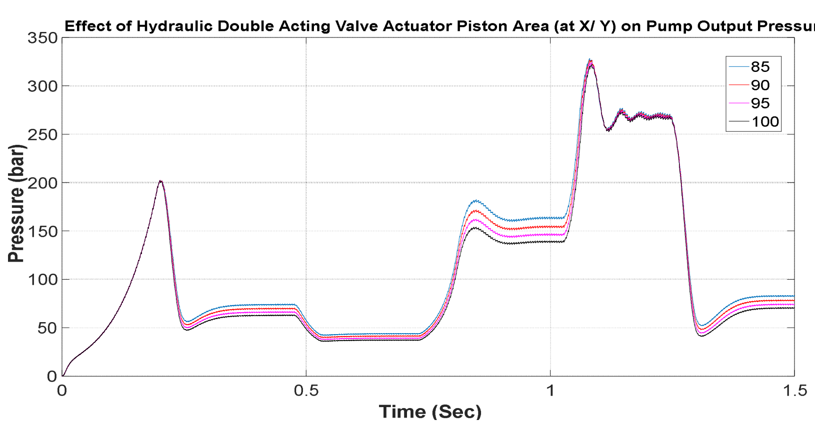


**Figure 8-8:**Pump Output Pressure with Several Double Acting Valve Actuator/ Piston Area Values

Appendix

| | Component | Parameter | Unit | #1 | #2 | #3 | #4 | #5 | #6 |
|---|---|---|---|---|---|---|---|---|---|
| Flow Control Orifice with Variable Area Slot /Orifice Width | Flow Control Orifice with Variable Area Slot | Orifice Width | cm | **1** | **1.2** | **1.3** | **1.4** | **1.5** | **2** |
| | Load Sensing Fixed Orifice | Orifice Area | $mm^2$ | 2 | | | | | |
| | Pressure Relief Valve | Maximum Passage Area | $mm^2$ | 20 | | | | | |
| | Hydraulic Double Acting Valve Actuator | Piston Area at X/ Y | $mm^2$ | 80 | | | | | |
| Load Sensing Fixed Orifice/ Orifice Area | Flow Control Orifice with Variable Area Slot | Orifice Width | cm | 1 | | | | | |
| | Load Sensing Fixed Orifice | Orifice Area | $mm^2$ | **2** | **4** | **6** | **8** | **10** | |
| | Pressure Relief Valve | Maximum Passage Area | $mm^2$ | 20 | | | | | |
| | Hydraulic Double Acting Valve Actuator | Piston Area at X/ Y | $mm^2$ | 80 | | | | | |
| Pressure Relief Valve/ Maximum Passage Area | Flow Control Orifice with Variable Area Slot | Orifice Width | cm | 1 | | | | | |
| | Load Sensing Fixed Orifice | Orifice Area | $mm^2$ | 2 | | | | | |
| | Pressure Relief Valve | Maximum Passage Area | $mm^2$ | **40** | | **60** | **80** | **100** | |
| | Hydraulic Double Acting Valve Actuator | Piston Area at X/ Y | $mm^2$ | 80 | | | | | |
| Hydraulic Double Acting Valve Actuator/ Piston Area at X/ Y | Flow Control Orifice with Variable Area Slot | Orifice Width | cm | 1 | | | | | |
| | Load Sensing Fixed Orifice | Orifice Area | $mm^2$ | 2 | | | | | |
| | Pressure Relief Valve | Maximum Passage Area | $mm^2$ | 20 | | | | | |
| | Hydraulic Double Acting Valve Actuator | Piston Area at X/ Y | $mm^2$ | **80** | **85** | **90** | **95** | **100** | |

**Table 1**: Load Sensing Components Simulation Matrix

Nomenclature:

| | |
|---|---|
| Flow Control Orifice with Variable Area Width | 1 cm |
| Load Sensing Fixed Orifice Area | 2 $mm^2$ |
| Pressure Relief ValveArea | 20 $mm^2$ |
| Hydraulic Double Acting Valve Actuator Piston Area at X and Y | 80 $mm^2$ |
| $f_v$ : Load sensing valve preload force | N |
| $\ddot{x}_v$ , $\dot{x}_v$ , $x_v$ ( $\ddot{x}_p$ , $\dot{x}_p$ , $x_p$ ): Acceleration/Velocity/ displacement of the spool ($n^{th}$piston) | |
| $M_v$ : Proportional valve spool mass | 0.1 kg |
| $C_v$ : Proportional valve viscous friction coefficient | 90 N.m/s |
| $K_v$ : Proportional valve spring stiffness | 20 kN/m |
| $P_l$ : Leak Pressure | |
| $A_v$ : Spool cross-sectional area | $m^2$ |
| $\varsigma_v$ , $\varsigma_p$ : Damping coefficient (spool, swash plate) | |
| $\omega_{nv}$ , $\omega_{np}$ , $\omega_t$ : Natural frequency (valve, swash plate, and pressure ) | Rad/sec |
| $Q_p$ : Pump volume flow rate | $m^3$/sec |
| $A_1$ : Control piston cross-sectional area | $m^2$ |
| $J$ : Average Swash plate inertia | $1.06X10^{-3}$ N.m. $sec^2$/rad |
| $V$ : Volume of pump at the high-pressure side | $m^3$ |
| $\beta$ : Effective bulk modulus | $1X10^9$ Pa |
| $K_p$ : swash plate equivalent stiffness | N/m |
| $Q_l$ : Volumetric leak | $m^3$/sec |

| | |
|---|---|
| $c_l$ : Leak coefficient | 4.3 X$10^{-13}$ $m^3$/Pa.sec |
| $P_s$ , $\dot{P}_s$ : suction pressure, instantaneous suction pressure | Pa, Pa/s |
| $V_t$ : Volume of the pump | $m^3$ |
| Pump maximum displacement | 7.9 $cm^3$/rad |
| Pitch radius | 5 cm |
| Piton diameter | 2.8 cm |
| n: Pistons' number | 9 |
| Maximum piston stroke | 6 cm |
| Maximum swash plate angle | $37^0$ |
| Arm length between the swash plate pivoting point and the actuator | 5.6 cm |
| Swash plate actuator stroke | 5 cm |
| Orifice diameter at the BDC | 7 mm |
| Prime mover speed | 260 rad/sec |
| Hydraulic fluid type | Skydrol LD-4 |
| Pump maximum pressure | 260 bar |
| Pump maximum volume flow rate | 1.1 l/s |
| Piston dead volume | 1 $cm^3$ |
| Leakage area | 1X$10^{-12}$ $m^2$ |
| Pressure control unit piston area A/B | 9/4.2 $m^2$ |
| Orifice with variable area slot orifice width | 0.003 |
| Flow discharge coefficient | 0.7 |
| Orifice with variable area slot orifice initial direction | 0.002 |
| Critical Reynolds number | 12 |
| Orifice with variable area slot orifice leakage area | 1X$10^{-9}$ $m^2$ |

**Table 2:** Important Parameters

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