ELECTRO HYDRAULIC SERVO VALVE

A final year project report

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ABSTRACT

Fuel flow control is one of the most important aspects in aerospace applications and its accurate calibration is of the essence. Due to very high fuel inlet pressures, a normal proportional cannot be used and the servo valve is used to ensure safe and accurate functioning.

To propose a working solution we are required to redesign an EHSV to make it light weight and more compact. All the flow and design parameter constraints are given by the industry (NESCOM). The working of the design can be explained by starting off with the torque motor. The torque motor in our project would receive an input signal from ranging from 0 to 150mA. Upon receiving a signal within this range the torque motor would rotate the armature to a certain angle which would be calibrated upon the input range and output flow range of 100 to 1100 L/Hr. Upon rotation of armature the armature comes closer to the nozzle. In the neutral position, the pressure difference ΔP is equal to zero. However, when the armature by a small angle and comes closer to the nozzle a pressure increase happens on one side of the spool. This increase in pressure creates a pressure difference across the ends of the spool and causes the spool to move. Movement of the spool creates the opening from the inlet towards the outlet port and the resulting flow can be calculated by a flow meter.

We have simulated the working at 60 bar pressure and the pressure contours are monitored. The working prototype, however, has been designed for 15 bar inlet pressure as a safety precaution. The resulting flow at the input current signals has an error of only 5%.

PREFACE

This report presents a detailed objective summary of the process and parameters used for designing and fabricating our project i.e. an Electro Hydraulic Servo Valve.

The report is divided into four chapters and further headings corresponding to the departments divided for the purpose of executing the project, these four chapters are Introduction, Literature review, Methodology and Results.

In the first chapter, the need of the project along with a brief history and the detailed deliverables are explained. In the second chapter, theory behind the servo valves is explained in detail. Everything about the work already done by the researchers about servo valves is compiled and then the reasons for selecting the final specifications have been elaborated while mentioning the references used in the report.

Third chapter starts with the working principle of the valve which is then used to calculate the important parameters using different softwares resulting in the design of our project. Formulas and governing equations have been stated while the detailed calculations are in appendices.

Finally, the results are explained along with conclusions and recommendations in the last chapter. The combination of figures, graphs and detailed explanations will surely make it interesting to read the report.

Although, the report gives an exhaustive account of the project progress, any omissions or mistakes noted are deeply regretted.

AKNOWLEDGMENTS

We are deeply thankful to our Supervisor, Dr. Hussain Imran for offering us this project and helping us throughout the course of the project in accomplishing our goals. His guidance, support and motivation enabled us in achieving the objectives of the project.

We are also thankful to Dr Emad Ud Din, who helped us developing a practical strategy for the completion of this project when we were facing problems due to limitation of resources. Apart from them, we are also thankful to our committee members for their valuable feedback. We would specially like to thank Syed Engineering Works for their machining services which helped us a lot in fabrication of the project.

Finally, we would like to thank SMME and all those factors that have contributed towards our ability to undertake and complete this project.

ORIGINALITY REPORT

We hereby declare that no portion of the work of this project or report is a work of plagiarism and the workings and findings have been originally produced. The project has been done in sole partnership with NESCOM and has not been a support project of any similar work serving towards a similar degree's requirement from any institute. Any reference used in the project has been clearly cited and we take responsibility if found otherwise.

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ABBREVIATIONS

EHSV Electro Hydraulic Servo Valve.

GUI Graphical User Interface.

MATLAB MATrix LABoratory.

NESCOM National Engineering and Scientific Commission.

NASA National Aeronautics and Space Administration.

NOMENCLATURE

x_f	Flapper Nozzle Gap
d_n	Nozzle Diameter
ΔP	Differential Pressure
A_s	Area of Spool
X_v	Spool Position
P_1	Inlet Pressure
P_2	Outlet Pressure
Q_1	Inlet Flowrate
Q_2	Outlet Flowrate
θ	Jet Flow Angle
A_v	Flow Area
F_{ax}	Flow Force (static and dynamic)
l	Distance between ports
C_d	Discharge Coefficient
g	Acceleration due to Gravity
A	Area of orifice
r	Radius of orifice
P_s	Working Pressure
P_a	Ambient Pressure
V	Flow Velocity
γ	Specific Weight
HL_{major}	Major Losses

HL_{minor} Major Losses

f Friction Fcator

D Diameter of pipe

K Loss Coefficient

R Reynolds Number

μ Dynamic Viscosity

P_i Pressure inside Water Tank

P_o Pressure outside Water Tank

PE_o Potential Energy outside Hole

PE_i Potential Energy inside Hole

CHAPTER 1

INTRODUCTION

The servo valve is the key component enabling the creation of closed loop electrohydraulic motion control systems (or ‘servomechanisms’, the traditional term now largely fallen out of use). ‘Servo valve’ has come to mean a valve whose main spool is positioned in proportion to the electrical input to the valve, where the spool movement is achieved through internal hydraulic actuation. The spool movement changes the size of metering orifices, thus enabling the valve to control flow; however, this flow is dependent on the pressure difference across the orifice unless some form of pressure compensation is used.

Brief History of Servo Valves:

Many of the basic design ideas in single or two-stage servo valve design had been conceived by the mid-1950’s: 60 years ago. The two-stage mechanical feedback servo valve became established through the 1960’s for aerospace and then high performance industrial applications. The single stage valve, with proportional solenoid or linear force motor direct spool valve, became established in the 1970’s and 80’s as a lower cost solution for industrial applications, increasingly with electrical spool position feedback and integrated electronics.

The torque motor driven two-stage valve has been remarkably successful and long lived. Nevertheless, manual assembly and adjustment of torque motors has always proved necessary, which is one motivation for investigating alternative technology, principally harnessing active materials. Also, in a few applications, the potential for faster dynamics that piezoelectric or some other active materials promise is attractive, but this is very much the minority of cases. Despite 60 years of research into alternatives, the torque motor has survived, although the gradual improvements in piezoelectric actuator technology, including drive electronics and hysteresis compensation methods, may eventually provide a viable competitor. [3]

Our Project:

Our project was the construction of an electro hydraulic servo valve (EHSV) using a servo motor instead of the typically used torque motor to be used as a flow control valve.

Aims & Objectives:

- To redesign a single stage flapper nozzle servo valve under specified dimensional and flow constraints.
- To produce a design with maximum result repeatability and reproducibility
- To model the design and perform and perform flow simulations.
- To fabricate the model.

All the flow and design parameter constraints are given by the industry (NESCOM), which are as follows:

- ▶ Single Stage Flapper Nozzle
- ▶ Under-Lapped valve condition
- ▶ Input current= 0 ~ 150 mA
- ▶ Fuel inlet pressure= 6 MPa
- ▶ Fuel flow= 100 ~1100 L/Hr
- ▶ Inlet fuel filter= 5 microns
- ▶ Fuel grade = Jet A1/JP-8
- ▶ Outlet Pressure = 23 bar
- ▶ Coil resistance=78Ω ± 3Ω
- ▶ Insulation resistance ≥ 55MΩ
- ▶ Flow hysteresis ≤ 52 L/hr
- ▶ Response time ≤ 100 ms
- ▶ Length ≤ 134 mm

- Width ≤ 114
- Height ≤ 76
- Weight ≤ 1.5 kg
- Qualified for military and aerospace standards
- Class of cleanliness of operating fluid: Grade # 6 of NASA 1638

Table 1: Current-Flowrate from Industry

Current Isv (mA)	Flow (L/hr)	
	Min	Max
0	50-60	110-115
5	120-124	135-140
10	125-130	180-190
15	140-145	220-230
20	160-175	260-270
30	200-205	320-330
50	333-340	467-480
70	467-480	613-623
90	560-565	760-770
100	640-645	833-843
110	700-705	885-895
130	800-805	1000-1010
150	880-885	1100-1110

CHAPTER 2

LITERATURE REVIEW

An electrohydraulic servo valve is a valve device with either a flapper nozzle or a jet pipe (used to position the servo). These type of servo valves are called servo valves because they are controlled through an electrical signal that govern the servo position. The main usage of servo valves is where accurate position control is required, such as flight control. Position control is primarily achieved through a closed loop control system, comprising of feedback sensor, command sensor, analog or digital controller, and the servo valve. Servo valves are also used to control hydraulic actuators or motors. When a servo valve is used to control an actuator, the servo valve – actuator combination is then termed as a servo actuator. The primary advantage of servo valves is that a low power electric signal is used to accurately position an actuator or a motor. The disadvantage, however, includes the complexity and the cost associated with due to detail parts manufactured and very tight tolerances. Therefore, servo valves should be used only when accurate positioning and fast response is required.

A schematic diagram of a servo actuator is shown in the figure below. The actuator included shows how the servo valve – actuator components work together in a system. In a servo mechanism the primary components are a torque motor, flapper nozzle or jet pipe, and one or more spools. The stage is comprised of a flapper nozzle/jet pipe and the spool valve. The stage provide the hydraulic force amplification that is the flapper goes from lower current signal to spool differential pressure and the spool valve amplifies the differential pressure on the actuator. The servo valve shown in Figure is a 2 stage servo valve with an actuator. Most of the servo valves in use are 2 stage, but 3 stage designs also exist. A 3 stage servo valve normally has an additional in between the 1st spool and the actuator. The 1st spool valve acts as a pilot spool and provides a spool differential pressure to the 2nd spool valve. [6]

The servo valve shown in the figure below uses a flapper nozzle. A servo valve has a hydraulic pressure inlet and an electrical input for the torque motor. The input current controls the flapper position. The flapper position controls the pressure in Chambers A & B of the servo. So, a current

(+ or -) will position the flapper, leading to a delta pressure on the servo, which cause the servo to move in one direction or the other. Movement of the servo ports hydraulic pressure to one side of the actuator or the other, while porting the opposite side of the actuator to return. Operation of a servo valve is described in more detail below.

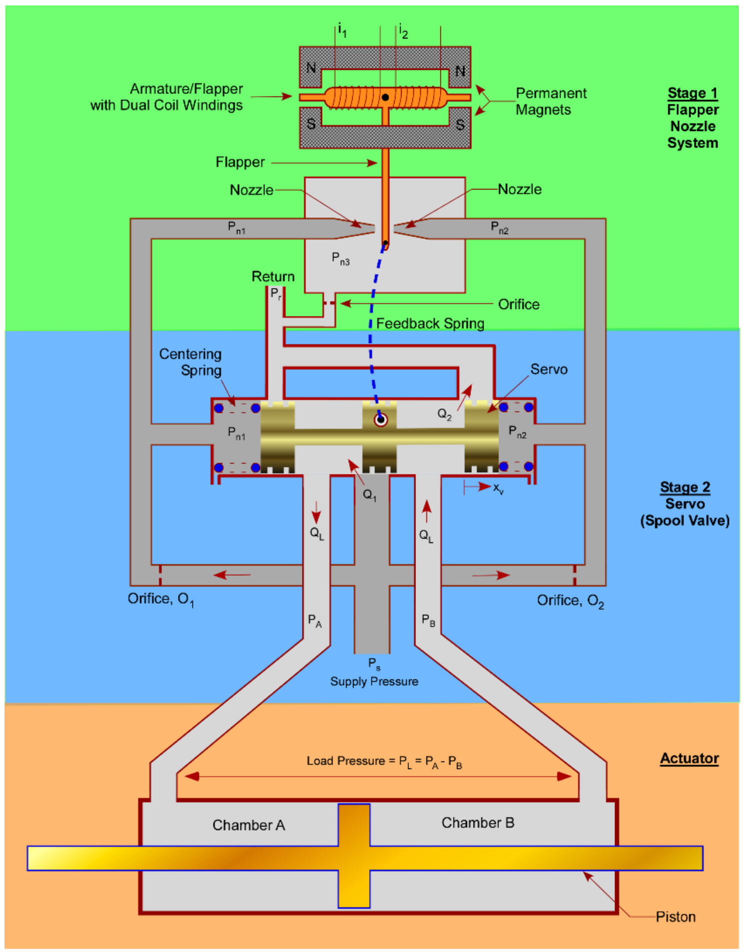

Figure 1: 2 stage servo valve with actuator

Flapper Nozzle System

Flapper position is controlled by the electromagnetic torque motor (see top portion of Figure 1). The torque developed by the torque motor is proportional to the applied current. Currents are generally small, in the milliamp range. A torque motor consists of two permanent magnets with a coil winding attached to a magnetically permeable armature. The armature is part of the flapper piece. When a current is applied to the coils, magnetic flux acting on the ends of the armature is developed. The direction of the magnetic flux (force) depends on the sign (direction) of the current. The magnetic flux will cause the armature tips to be attracted to the ends of the permanent magnets (current direction determines which magnetic pole is attracting and which one is repelling). This magnetic force creates an applied torque on the flapper assembly, which is proportional to applied current. In the absence of any other forces, the magnetic force would cause the armature to contact the permanent magnet and effectively lock in this position. However, other forces are acting on the nozzle, such that flapper position is determined through a torque balance consisting of magnetic flux (force), hydraulic flow forces through each nozzle, friction on the flapper hinge point, and any spring (wire) connecting the flapper to the spool (which is almost always installed used in servo valves to improve performance and stability).

As the applied current is increased, the armature and flapper will rotate. As the flapper moves closer to one nozzle, the flow area through this nozzle is decreased while the flow area through the other nozzle increases. The flapper generally rotates over very small angles (~ 0.01 rad) and the gap (G in the figure) is around 0.002 – 0.003 inches. If the gap, G, between the magnet and the flapper end gets too large, the torque motor may latch and become inoperative due to limited available torque from the torque motor.

The flapper nozzle consists of the flapper, two inlet orifices (O1 and O2), two outlet nozzles (n1 and n2), nozzle backpressure nozzle (n3) and usually a feedback spring. As described above, the torque motor positions the flapper, which in turns controls the flow through the nozzles. The inlet orfices, O1 and O2, are important as they create a pressure volume whose pressure is controlled by the flapper. [7]

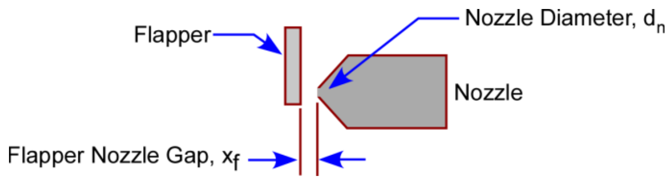

Figure 2: Flapper Nozzle Distance

Referring to figure 2, for the flapper nozzle to control flow in a linear manner, the relationship

$$x_f \cdot \pi d_n << \frac{\pi}{4} d_n^2$$

must be maintained. This relationship implies that the circumferential area created by the flapper distance to the nozzle must be smaller than the nozzle diameter, such that the circumferential area controls flow and not the nozzle diameter. In this way, the flow area varies linearly with flapper position. Also, the torque motor materials, windings and overall design features lead to accurate control of torque such that small movements of the flapper are possible. This leads to accurate control of the pilot spool, which in turns provides accurate control of the actuator.

The goal of the flapper and nozzles is to control the pressure acting on both sides of the pilot spool. When the flapper is in the neutral position, the nozzle flow areas are equal and the pressures P_{n1} and P_{n2} are equal. When the flow areas and inlet nozzle pressures are equal, the flow forces through each nozzle keep the flapper centered in the neutral position. For a zero lapped pilot spool valve, there would be no flow into or out of the actuator chambers. As the flapper moves towards one of the nozzles, the outlet flow area is reduced for this nozzle. Outlet flow area increases for the other nozzle. For example, looking at Figure 1 let the flapper move towards the n1 nozzle. This will reduce the outlet flow area and the pressure P_{n1} will increase. At the same time, the outlet flow area at the n2 nozzle will increase and the pressure P_{n2} will decrease. A delta pressure $\Delta p = P_{n1} - P_{n2}$ will occur across the pilot spool piston and the pilot spool will displace to the right. High pressure fluid will then flow to the P_A actuator chamber while the P_B actuator chamber is ported to return. Depending on the size of the flapper and nozzles, the Δp across the pilot spool is limited in magnitude (200-300 lb range for medium size aerospace applications).

Most servo valves incorporate a feedback spring (wire) between the pilot spool and the flapper. This wire is shown as a dotted blue line in Figure 1. Examining Figure 1, if the flapper moves to the left, the Δp on the pilot spool moves the spool to the right. The feedback wire will then pull the flapper back towards the neutral position. Hence the feedback wire provides a stabilizing force to the flapper and helps improve stability and response of the flapper system. This same affect can be done electronically by putting a feedback sensor (usually a linear variable differential transducer) on the pilot spool. The output of the sensor is fed back electronically to reduce the current command and allow the flapper to move back to the neutral position. [1]

Servo

The servo part of the valve is exactly the same as any servo or spool valve. The function of the servo is the same for either a flapper nozzle or a jet pipe servo actuator. The relationship between flow and Δp through the servo valve is governed by the orifice flow equation. Servo position is determined by a force balance on the spool, which includes the Δp created from the flapper nozzle or jet pipe, friction forces, spring forces and flow forces acting on the spool. For a complete description of a servo, see Servo, Hydraulic – Description. [8]

When the spool is in the neutral position, the servo valve is in the null position. In some applications, a compression spring is installed on each side of the servo to help keep the servo centered. In other applications (spoiler panel servo valves, for example), a spring is installed in one side only which will push the servo in one direction. For flight spoilers the spring would bias the actuator to the retract position. So, in the absence of electrical commands, the spring pushes the servo towards the retract command position allowing hydraulic fluid to flow to the retract chamber. The applied current required to overcome the spring force and return the servo to the null (no flow) position is referred to as the null bias. The null bias current will drift in service due to changes in supply pressure, operating temperatures, wear and other factors. Good servo valve design practice is to keep long term null bias shifts to within ±3% of rated current.

Flow characteristics and the effects of load flow and load pressure drop are determined by the servo or spool portion of the valve. The equations that describe the relationship between these parameters are given in Servo valve, Hydraulic – Equations. To understand the theoretical performance of a servo valve, these mathematical relationships must be understood.

The mechanism shown in the figure below is often called a servo mechanism in which there is a spool inserted in a cylinder having drilled ports. This cylinder is also called outer sleeve or bushing. The movement of the spool causes the flow area to change which in turn changes the flow rate through the valve. The movement of the spool can be done by many methods, some of which are:

Solenoid controlled: In which there are only two positions, fully open or fully closed.

Proportional solenoid controlled: In which the spool position is directly proportional to the applied current

Servo Valve arrangement: In which hydraulic differential pressure is used to control the movement of the spool.

Mechanical lever controlled method is also sometimes used.

When analyzing the flow through servo valve ports, orifice flow equation is used. As we can see in the figure above, the outlet flow port area is controlled by the movement of the spool, and the position of the spool (x_V) determines the size of the outlet orifice area (A_V).

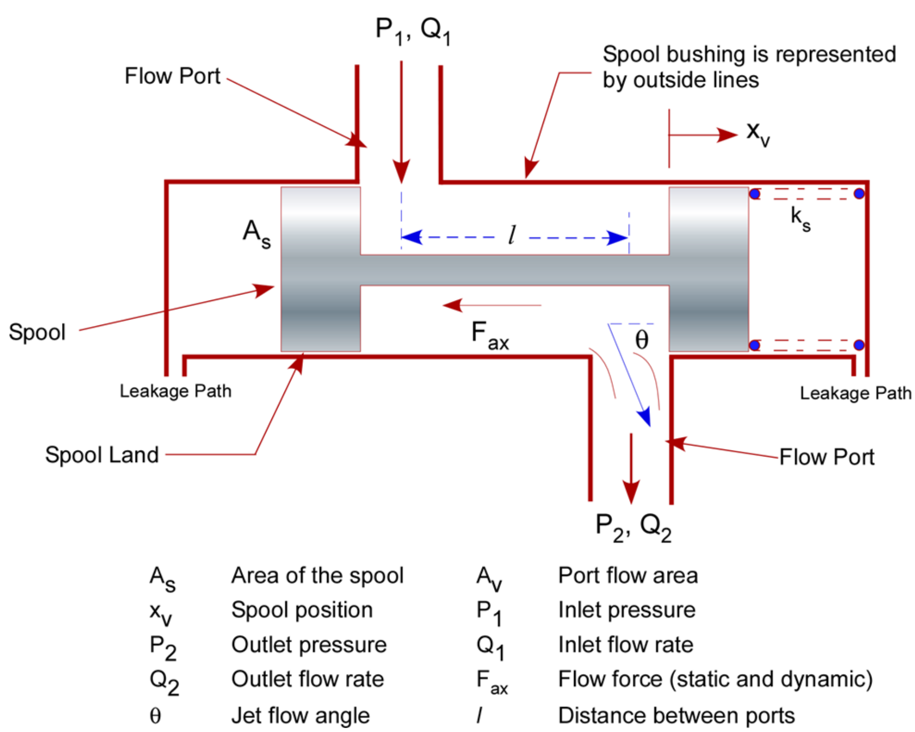

Figure 3: Servo Mechanism

When the fluid flows through the outlet orifice port, there is a significant pressure drop due to which turbulent flow is mostly assumed. Also, the outlet port area is kept smaller than the inlet port area because the flow controlling port is the outlet port in a servo valve.

Servo mechanisms can be zero lapped, over lapped or under lapped according to the position of the spool relative to the outer sleeve.

Zero Lapped: In the zero lapped servos, the width of the larger diameter spool portion (called land of spool) is equal to the width of the flow port of the outer sleeve. In this arrangement, there is only one position of the spool for which there can be zero flow i.e when the land of spool is exactly on the flow port.

Over lapped: In the over lapped servos, width of the land of spool is greater than the width of flow port of the outer sleeve. In this arrangement, there can be multiple positions of the spool for which the flow is zero. This results in a dead band in the graph of flow area vs pressure. In the over lapped servos, the spool must move by a minimum amount equal to the size of the overlap before the flow occurs.

Under lapped: In the under lapped servos, the width of the land of spool is smaller than the width of flow port of outer sleeve. In this arrangement, there is flow at all positions of the spool, because the spool land cannot block the flow port completely.

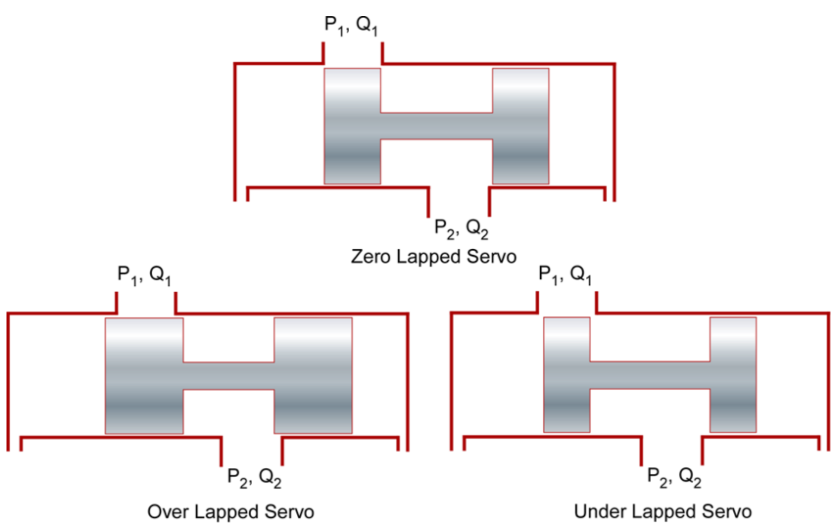

Figure 4: Servo Conditions

The benefits of a under lapped valve is faster response time with a cost of high leakage flows. Since we need a faster response time, so we will be using under lapped system for our case. There will be leakage flow as a disadvantage but it will not affect the system significantly if we look at the flow conditions for the output.

Servo valve Flow Characteristics

Using the turbulent orifice equation, the flow expression for flow through servo flow ports is

$$Q_2 = Q\left(x_v, \Delta p\right) = \alpha_d A\left(x_v\right)\sqrt{2/\rho}\sqrt{p_1 - p_2}$$

Where $A(x_v)$ the area of the valve orifice (servo port). The flow area depends on port geometry, which varies with manufacturer, valve type, and spool position. Inspection of the equation (1) indicates that the flow rate varies proportionally with area if the Δp is held constant, and that the flow rate varies with the square root of Δp if the flow area is held constant. Figure 5 shows notional charts of the flow behavior for a servo which are similar to orifice flow graphs.

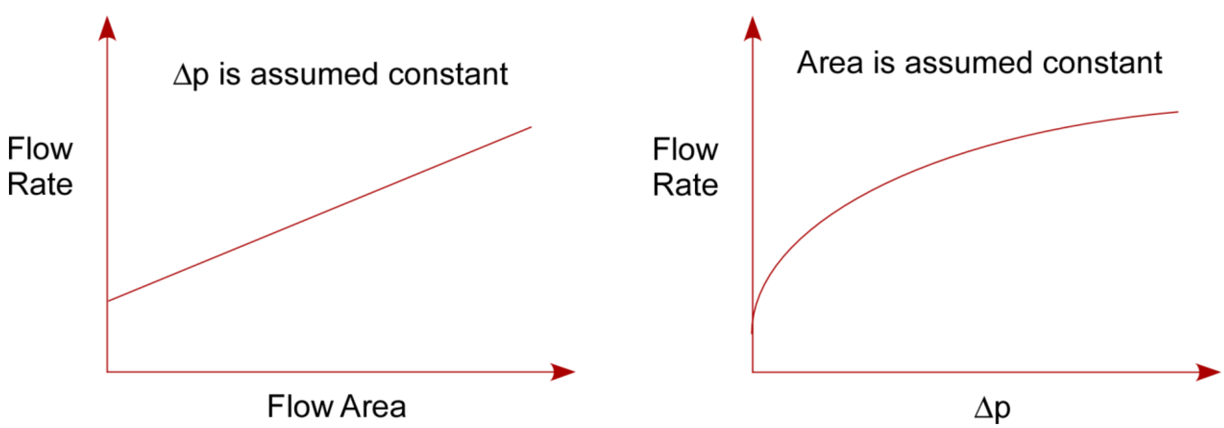

Figure 5: Servo Flow Behavior

The effects of a lapping can be seen in Figure 6. Figure 6 assumes a 4 way servo and illustrates ideal flow curves. In the figure, control flow (flow through the valve port) is plotted against valve position (where Δp is assumed constant) for under lapped, zero lapped and overlapped valve. For a zero lapped valve, the curve goes through the origin. For an overlapped valve, the flow is zero until the valve spool has moved sufficiently to allow flow. For the under lapped valve, there is flow through both directions of the servo yielding a zero flow to the load at the null position. However, as the under lapped valve moves off of null, flow to the load will change quickly (higher gain) and change less rapidly once the spool has moved to the point where flow through one side goes to zero

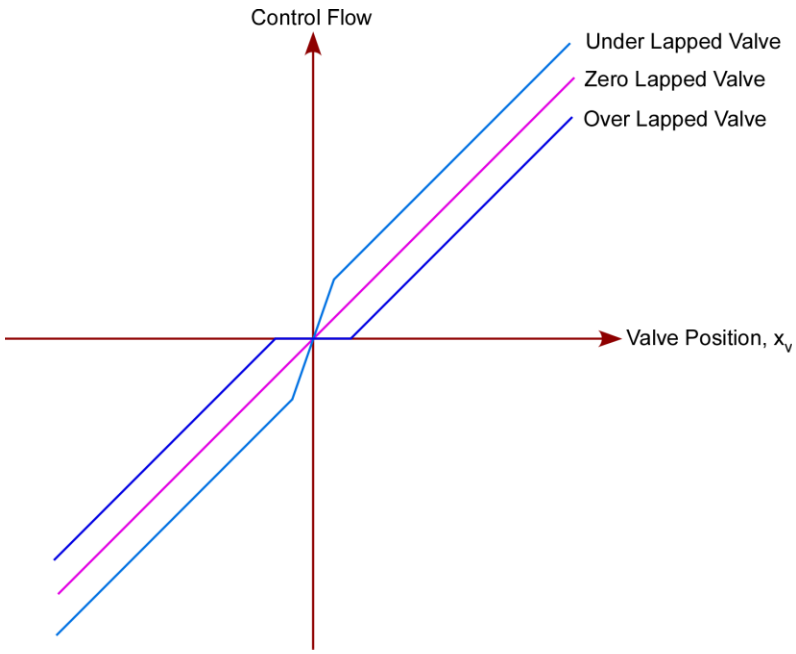

Figure 6: Servo Flow Curves

For a two position (open/close) solenoid servo, the flow area is either A_{min}(which is usually zero) and A_{max}. Changing between A_{min} and A_{max} takes less than 100 milliseconds in most applications (see Figure 7 below).

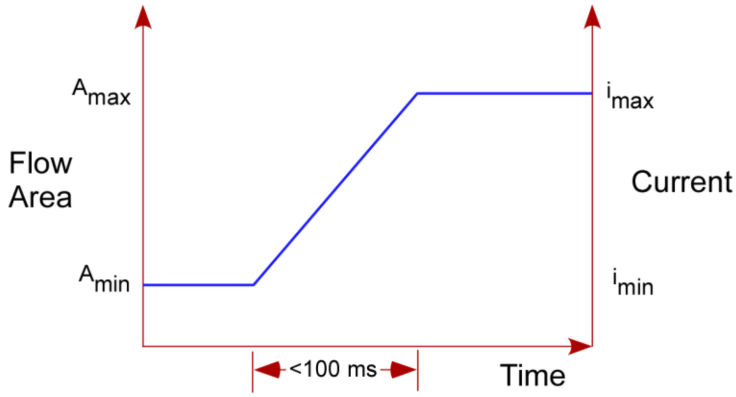

Figure 7: Change in Flow Area Graph

Lateral Spool Forces

If an unequal pressure distribution occurs in the small clearance between the spool outer diameter and the inner diameter of the bushing, a lateral force will be applied to the spool. This force can sometimes result in a spool experiencing a "hydraulic lock" condition. Here the lateral force is so great that the spool cannot be moved as long as hydraulic pressure is applied. Minor machining tolerances on the spool outer diameter can result in slightly varying leakage flow across the spool periphery setting up the unequal pressure distribution on the spool periphery. This type of lateral force is commonly minimized by machining grooves around the spool outer diameter since the grooves allow the pressure to equalize around the periphery of the spool. The grooves are perpendicular to the bore the spool rides in (see Figure 7).

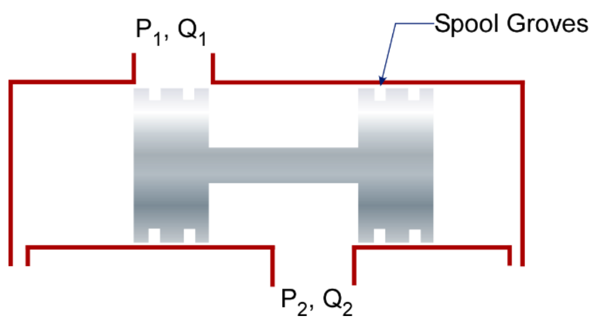

Figure 8: Spool Grooves

Servo/Torque Motor

Torque motor is used as an actuator in our valve. A torque motor is a specialized form of electric motor which can operate indefinitely while stalled, that is, with the rotor blocked from turning, without incurring damage. In this mode of operation, the motor will apply a steady torque to the load. [2]

A servomotor is a closed-loop servomechanism that uses position feedback to control its motion and final position. The input to its control is a signal (either analogue or digital) representing the position commanded for the output shaft.

The motor is paired with some type of encoder to provide position and speed feedback. In the simplest case, only the position is measured. The measured position of the output is compared to the command position, the external input to the controller. If the output position differs from that required, an error signal is generated which then causes the motor to rotate in either direction, as needed to bring the output shaft to the appropriate position. As the positions approach, the error signal reduces to zero and the motor stops.

The very simplest servomotors use position-only sensing via a potentiometer and bang-bang control of their motor; the motor always rotates at full speed (or is stopped). This type of servomotor is not widely used in industrial motion control, but it forms the basis of the simple and cheap servos used for radio-controlled models. [9]

Servo motors VS Stepper Motors

A servomotor consumes power as it rotates to the commanded position but then the servomotor rests. Stepper motors continue to consume power to lock in and hold the commanded position.

Servomotors are generally used as a high-performance alternative to the stepper motor. Stepper motors have some inherent ability to control position, as they have built-in output steps. This often allows them to be used as an open-loop position control, without any feedback encoder, as their drive signal specifies the number of steps of movement to rotate, but for this the controller needs to 'know' the position of the stepper motor on power up. Therefore, on first power up, the controller will have to activate the stepper motor and turn it to a known position, e.g. until it activates an end limit switch. This can be observed when switching on an inkjet printer; the controller will move the ink jet carrier to the extreme left and right to establish the end positions. A servomotor will immediately turn to whatever angle the controller instructs it to, regardless of the initial position at power up.

The lack of feedback of a stepper motor limits its performance, as the stepper motor can only drive a load that is well within its capacity, otherwise missed steps under load may lead to positioning errors and the system may have to be restarted or recalibrated. The encoder and controller of a servomotor are an additional cost, but they optimize the performance of the overall system (for all of speed, power and accuracy) relative to the capacity of the basic motor. With larger systems, where a powerful motor represents an increasing proportion of the system cost, servomotors have the advantage.

There has been increasing popularity in closed loop stepper motors in recent years. They act like servomotors but have some differences in their software control to get smooth motion. The top 3 manufacturers of closed loop stepper motor systems employ magnetic encoders as their feedback device of choice due to low cost and resistance to vibration. The main benefit of a closed loop stepper motor is the cost to performance ratio. There is also no need to tune the PID controller on a closed loop stepper system.

Many applications, such as laser cutting machines, may be offered in two ranges, the low-priced range using stepper motors and the high-performance range using servomotors.

CHAPTER 3

METHODOLOGY

Design

Our design is a single stage flapper nozzle mechanism, whose schematic diagram is given below:

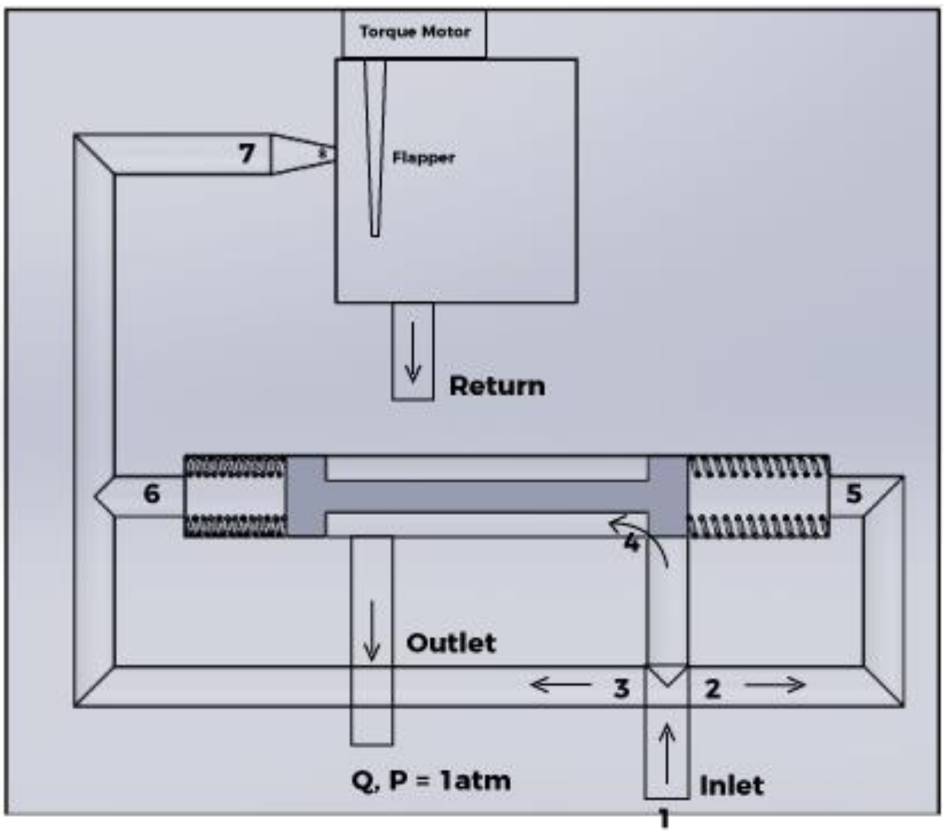

Figure 9: Single Stage Flapper Nozzle Schematic Diagram

Working

The general working of the design can be explained by starting off with the torque motor. The torque motor in our project would receive an input signal from ranging from 0 to 150mA. Upon receiving a signal within this range, the torque motor would rotate the armature to a certain angle which would be calibrated upon the input range and output flow range of 100 to 1100 L/Hr. Upon rotation of armature the armature comes closer to one of the nozzles. In the neutral position, as can be seen in the design pictures attached with the report, the pressure difference P1 – P2 is equal to zero. However, when the armature by a small angle and comes closer to one of the nozzles a pressure increase happens one side of the spool. This increase in pressure creates a pressure difference across the ends of the spool and causes the spool to move. Movement of the spool links the supply pressure to the outlet port.

Linking the working of the design to the schematic above point 1 is the entry point for the fluid. Upon entering the fluid divides into three flow paths. Path 4 is linked to the outlet but in the neutral position is blocked by the spool. Path 2 generates fluid pressure at the right end of the spool and path 3 generates the fluid pressure at the left end of the spool and fluid from this path also flows to the nozzle within the tank which, within the schematic diagram, can be seen as point 7. From within the tank, the fluid flows out from the Return.

CAD Model

Initial Model:

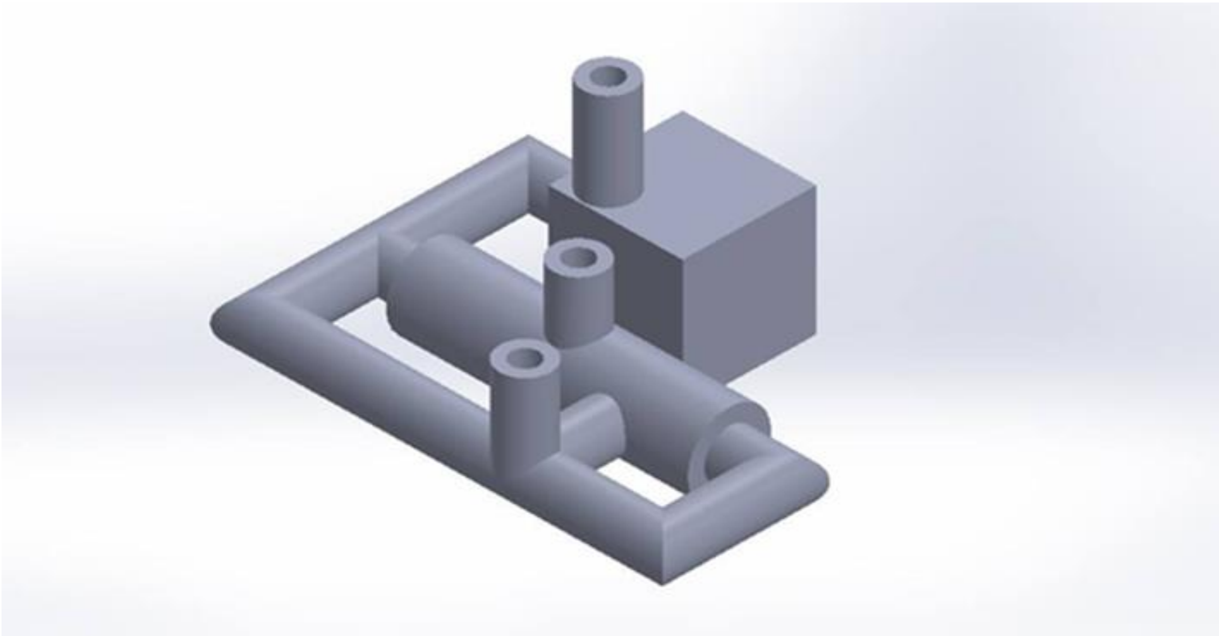

Figure 10: Isometric View of Initial Model

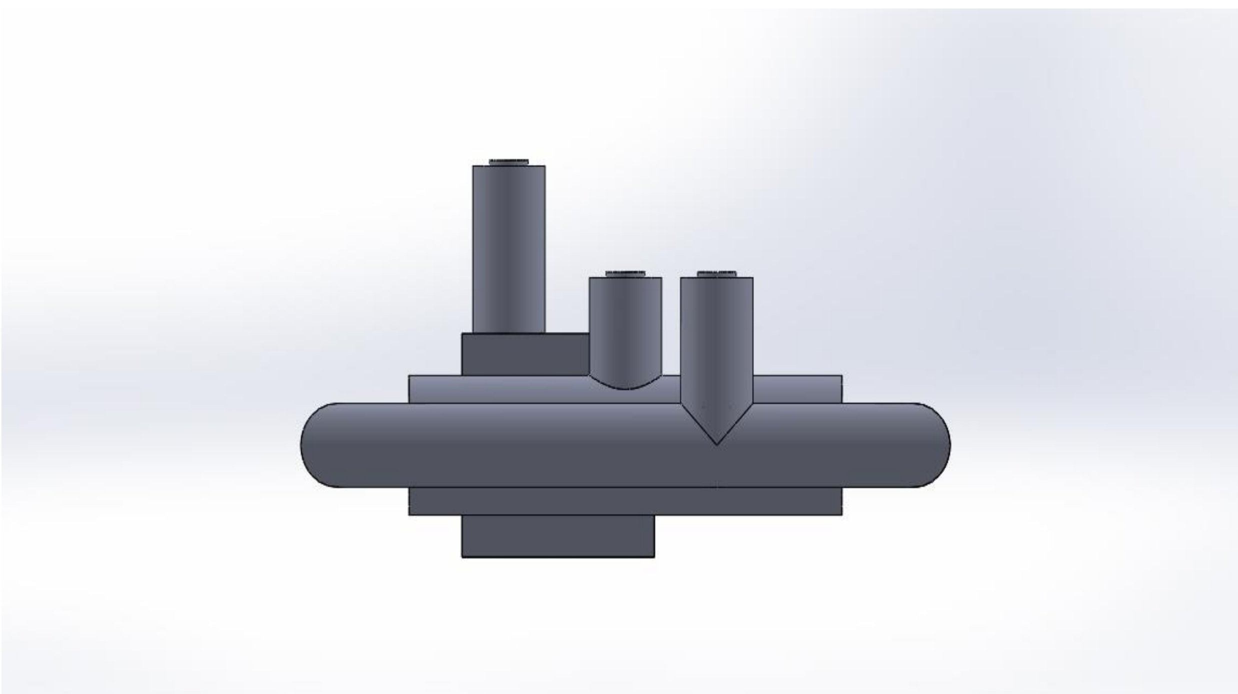

Figure 11: Front View of Initial Model

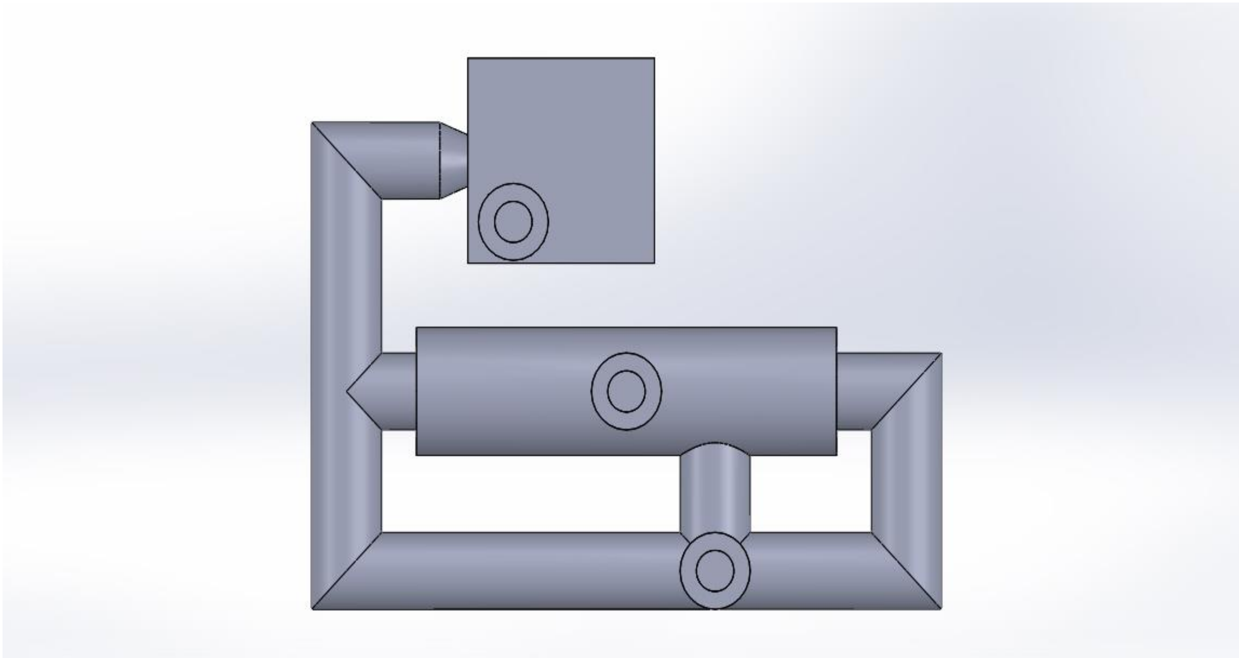

Figure 12: Top View of Initial Model

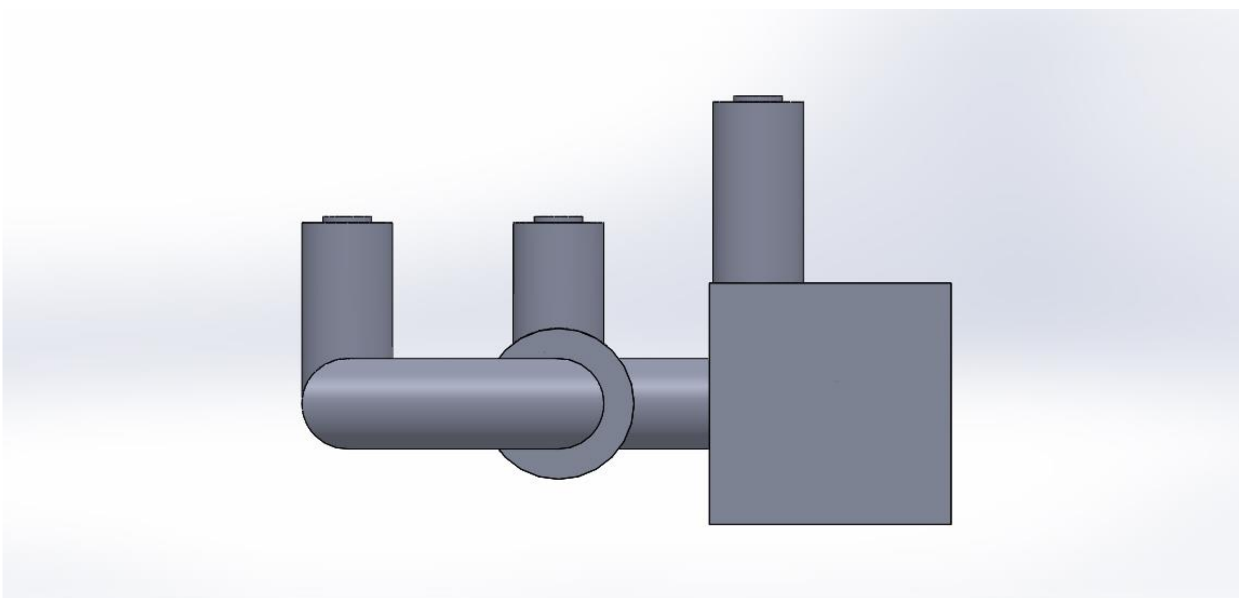

Figure 13: Side View of Initial Model

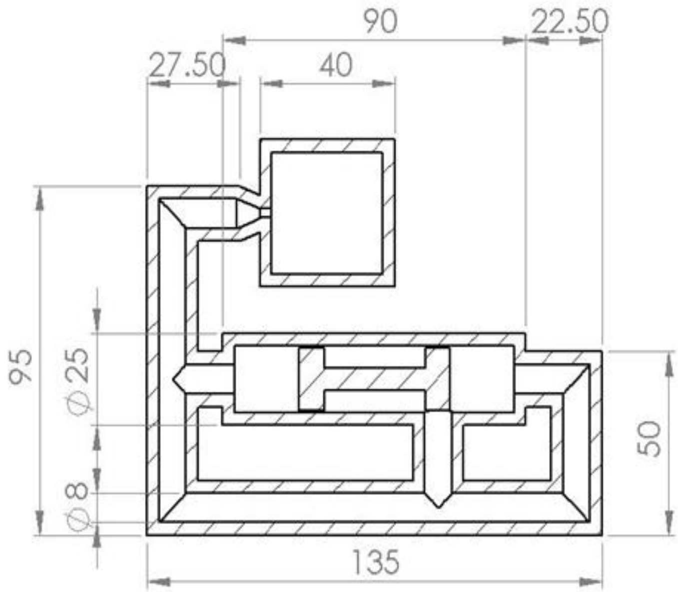

Figure 14: Cross Sectional View with Basic Dimensions

Revised Model:

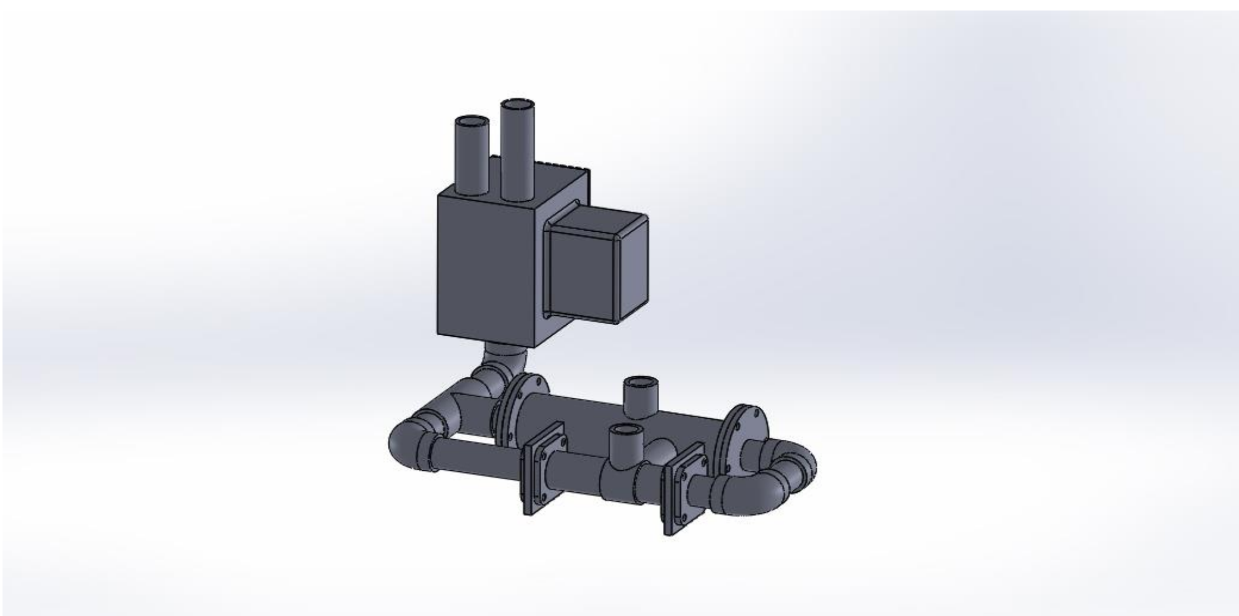

Figure 15: Isometric View of Revised Model

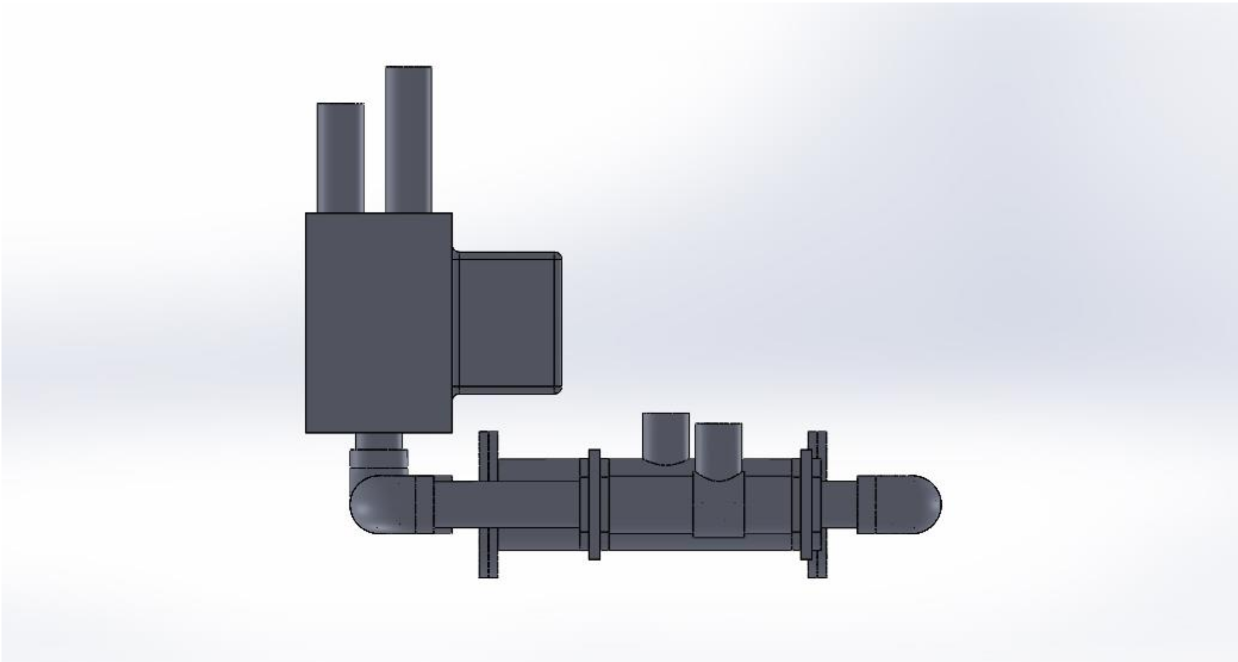

Figure 16: Front View of Revised Model

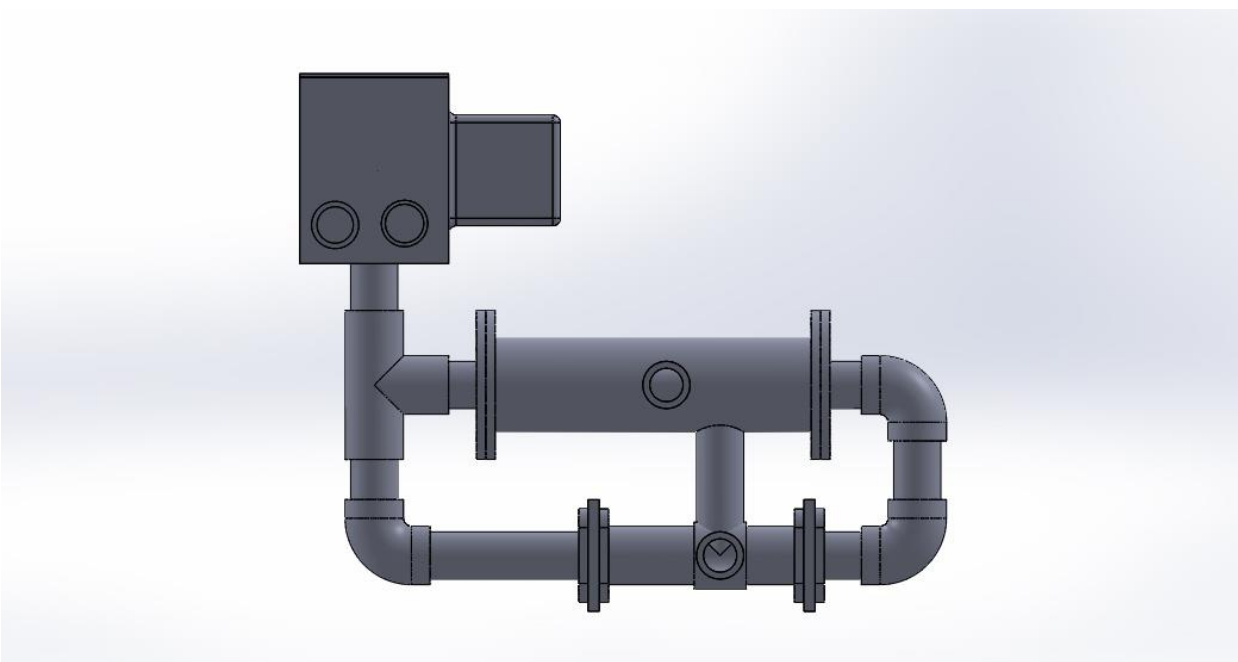

Figure 17: Top View of Revised Model

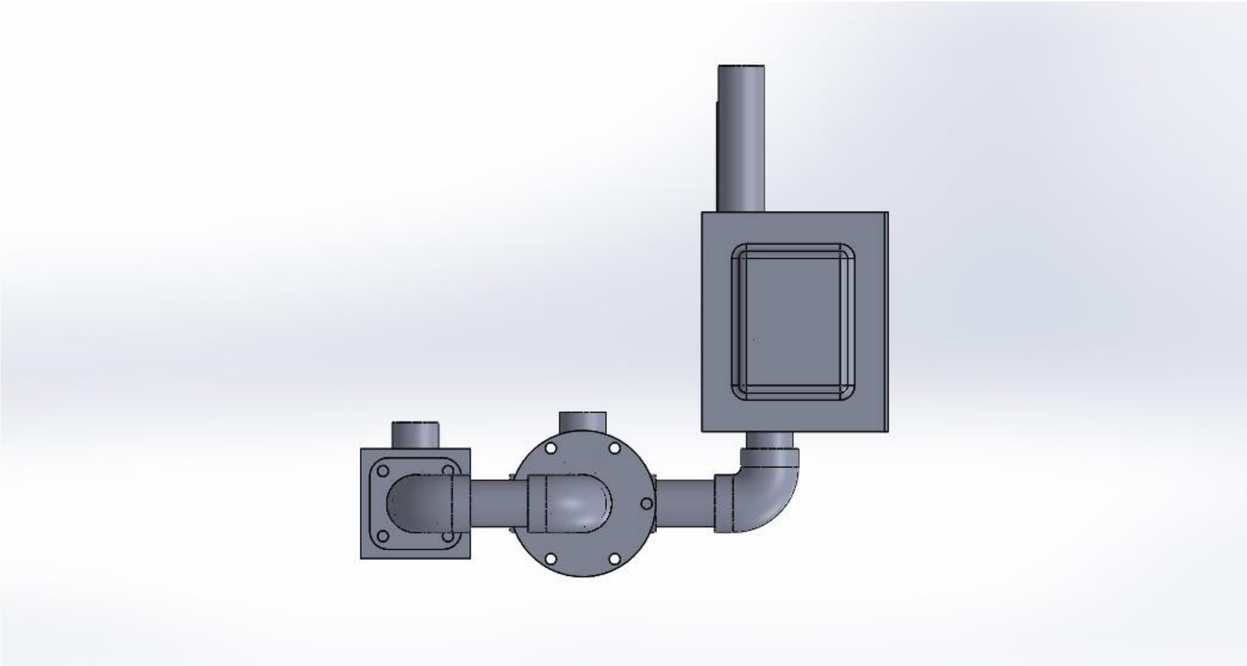

Figure 18: Side View of Revised Model

Theoretical Analysis:

For theoretical analysis we used MATLAB to write the governing equations and solve them for a any input variables. This gave us flexibility with regard to changing the design parameters and solving the governing equations for those parameters. For any future change with respect to design, the theoretical results can easily be gathered just by making the respective changes in the MATLAB code and solving for results.

Governing Equations:

The governing equations involved include the orifice flow equation and calculating the major and minor losses across the length of the pipe and the bends involve in the fluid flow. Below are the equations used for theoretical analysis:

$$Q = Cd\ A\sqrt{\frac{2g}{r}} \times \sqrt{Ps - Pa}$$

$$Q = AV$$

$$HL_{major} = \frac{\Delta P}{\gamma} = f\frac{l}{D}\frac{V^2}{2g}$$

$$HL_{minor} = K\frac{V^2}{2g}$$

$$R = \frac{\rho VD}{\mu}$$

MATLAB Results

Referring to the Figure 9 (schematic diagram), following results were obtained in MATLAB.

- Reynolds number is 7.980729e+03
- Nozzle velocity is 4 m/s
- Pressure at position 7 is 1.117741e+03 kPa
- Pressure at position 6 is 1.370893 e+03 kPa
- Pressure at position 5 is 1.253575 e+03 kPa
- Force at position 5 is 1.420978 e+00 kN
- Force at position 6 is 1.553962 e+00 kN

Fabrication

Fabrication of servo valve requires high quality and accurate machining. The machining operations used in the fabrication of our valve were mainly lathe and milling. The spool, sleeve and the flanges were made from aluminum. The rest of the valve was made from mild steel and stainless steel.

Due to the limitations of resources, we made some modifications in our design.

Talking about the main cylinder and spool, in our original design the spool and cylinder were to be machined within such tolerances as to move without friction within the cylinder and to not cause any leakage as well. However due to unavailability of machining at such part tolerance values we modified the spool to have two O-rings at each ends of the spool to prevent leakage. This increased friction for movement of the spool but operation at very high pressure differences reduces the impact of friction in spool movement.

We also modified our design to screw a small aluminum cylinder at the right end of the spool for the centering spring to clip on to it to keep its center and position intact during the working of the valve. We added this mechanism only on the right side since the differential pressure will be applied from the left side and the spring on the right-side acts as the main spring whereas the spring on the left side acts more for support. An image of the spool with the spring is added below.

We also modified our design to create male and female ends of flanges to reduce leakage possibility and within the flanges controlling the flow paths from the inlet, we press fitted orifice plates instead of the traditional way. This not only served our purpose of dropping pressure but the plates press fitted also acts as a gasket to prevent leakage.

We used silicone sealants so that the valve could not leak.

Figure 19: Spool with O-rings

Figure 20: Spool With Spring Clipon

Servo motor specefications

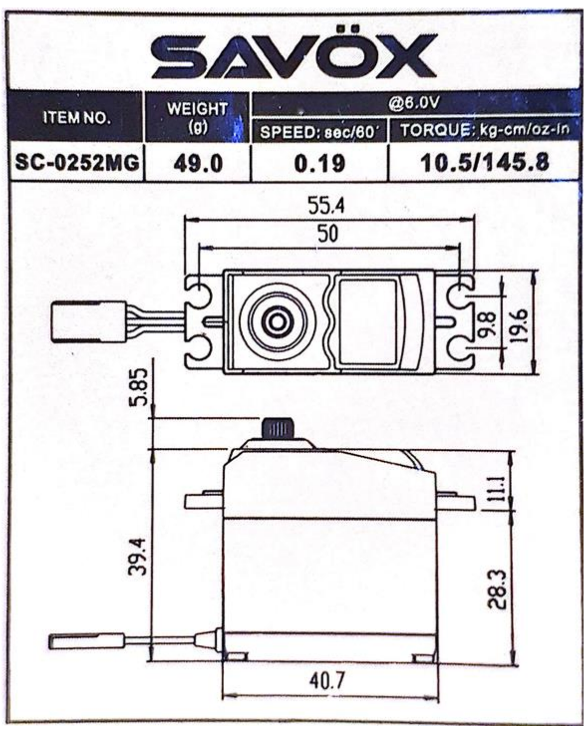

Figure 21: Servo Specs

Arduino

Arduino UNO was used to program the servo motor. Arduino Uno is a microcontroller board based on the ATmega328P (datasheet). It has 14 digital input/output pins (of which 6 can be used as PWM outputs), 6 analog inputs, a 16 MHz quartz crystal, a USB connection, a power jack, an ICSP header and a reset button. It contains everything needed to support the microcontroller; simply connect it to a computer with a USB cable or power it with a AC-to-DC adapter or battery to get started. [5]

Arduino code for servo motor operation is attached in the appendix.

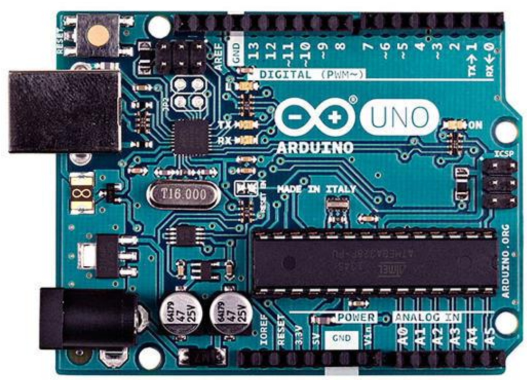

Figure 22: Arduino UNO

Flow meter

This sensor sits in line with your water line and contains a pinwheel sensor to measure how much liquid has moved through it. There's an integrated magnetic Hall Effect sensor that outputs an electrical pulse with every revolution. The Hall Effect sensor is sealed from the water pipe and allows the sensor to stay safe and dry.

The sensor comes with three wires: red (5-24VDC power), black (ground) and yellow (Hall Effect pulse output). By counting the pulses from the output of the sensor, you can easily calculate water flow. Each pulse is approximately 2.25 milliliters. Note this isn't a precision sensor, and the pulse rate does vary a bit depending on the flow rate, fluid pressure and sensor orientation. It will need careful calibration if better than 10% precision is required. However, its great for basic measurement tasks. [4]

Features:

- Model: YF-S201
- Sensor Type: Hall effect
- Working Voltage: 5 to 18V DC (min tested working voltage 4.5V)
- Max current draw: 15mA @ 5V
- Output Type: 5V TTL
- Working Flow Rate: 0.5 to 30 Liters/Minute
- Working Temperature range: -25 to +80℃
- Working Humidity Range: 35%-80% RH
- Accuracy: ±10%
- Maximum working pressure: 1.2 MPa
- Output duty cycle: 50% +-10%
- Output rise time: 0.04us
- Output fall time: 0.18us
- Flow rate pulse characteristics: Frequency (Hz) = 7.5 * Flow rate (L/min)
- Pulses per Liter: 450
- Durability: minimum 300,000 cycles
- Cable length: 15cm
- 1/2" nominal pipe connections, 0.78" outer diameter, 1/2" of thread
- Size: 2.5" x 1.4" x 1.4"

Connection details:

- Red wire : +5V
- Black wire : GND
- Yellow wire : PWM output

Figure 23: Flow Meter

User Interface

The user interface was created on LabView. The main logic behind the block diagram is reading values from arduino and then applying mathematical calculations on those values for converting them to the required units.

The front panel for the GUI is as follows:

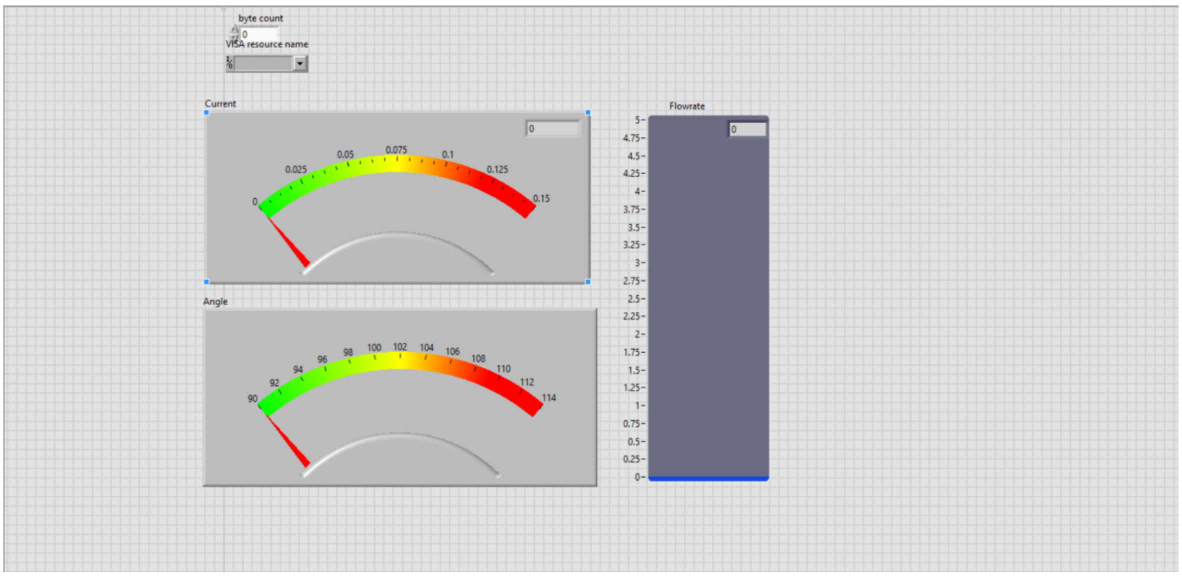

Figure 24: User Interface

The front panel shows the values for:

- Servo Motor Angle
- Current
- Flow Rate

Please find an attached image of the block diagram for the GUI on the following page.

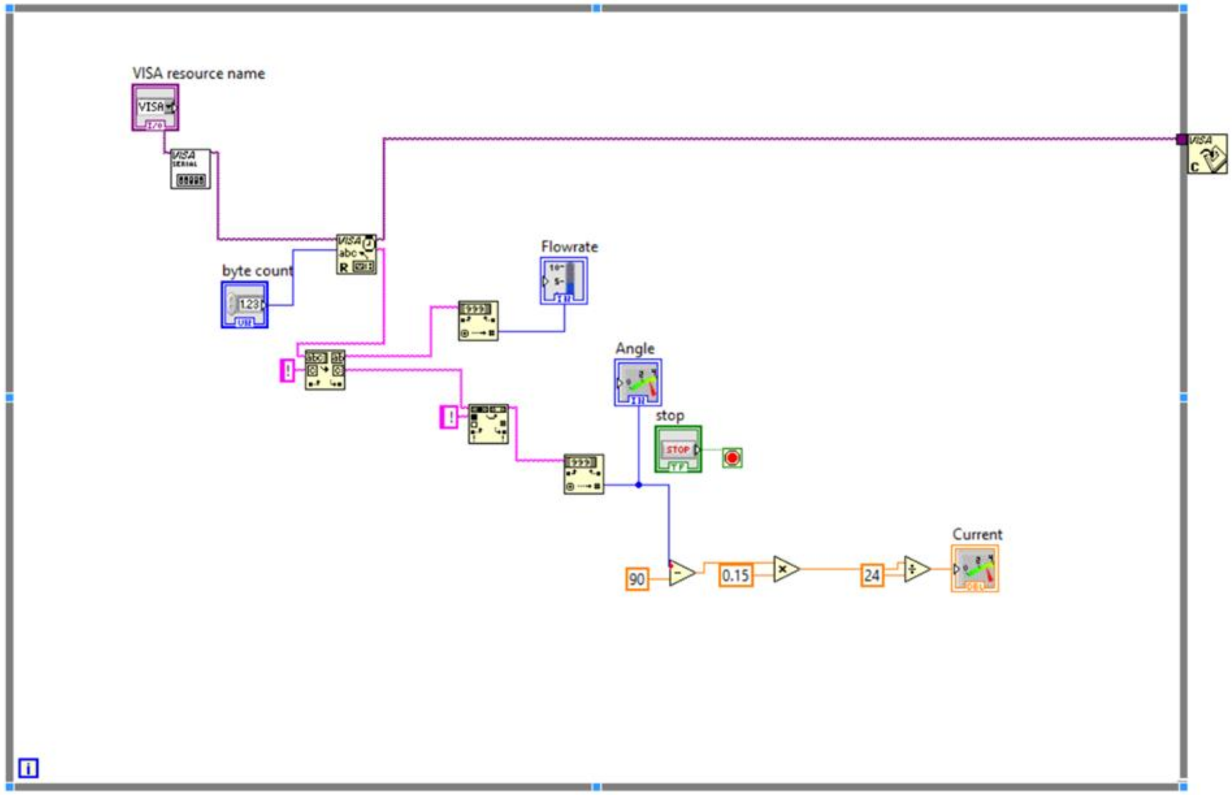

Figure 25: LABview Block Diagram

Rendered Model:

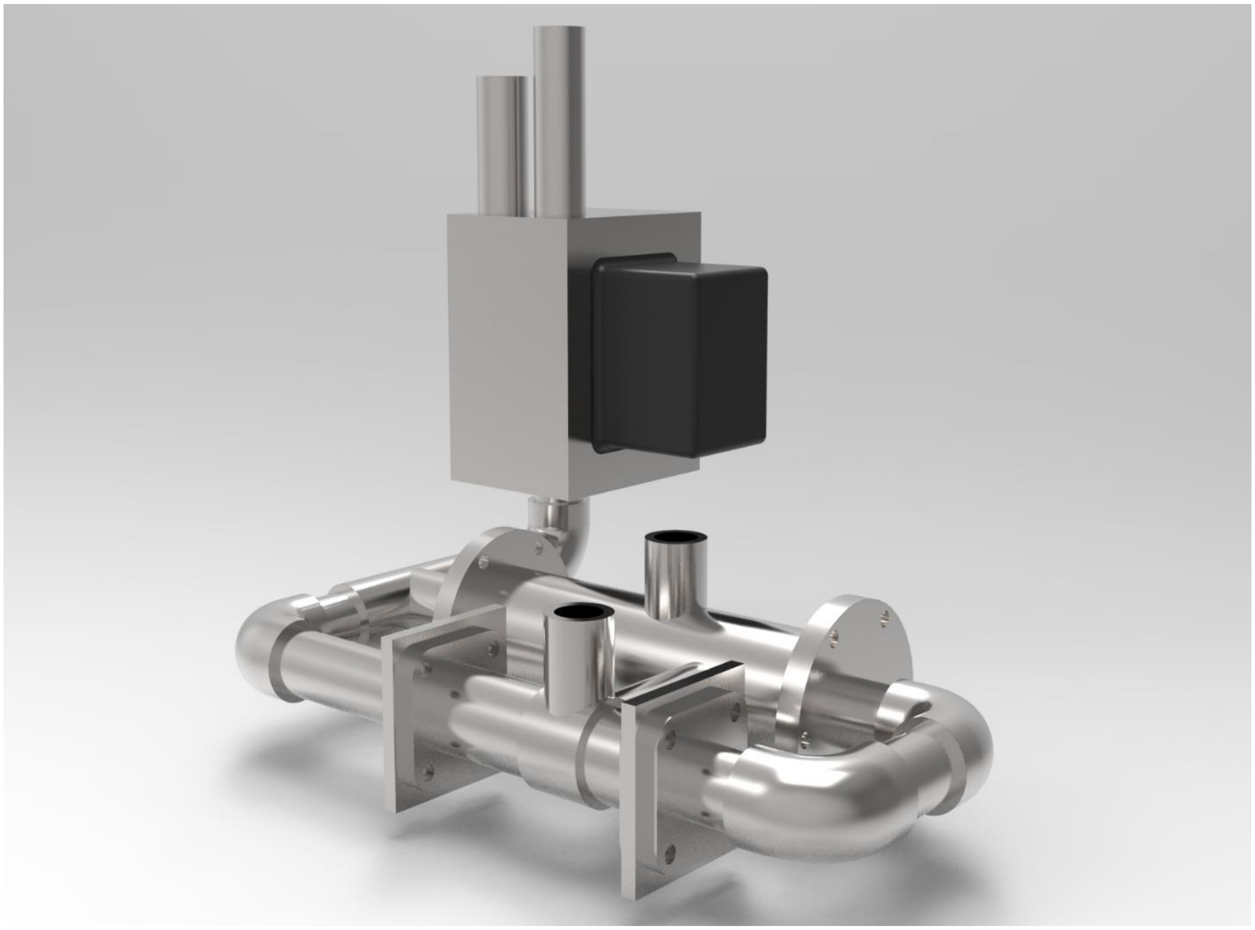

Figure 26: Rendered Model

Actual Valve:

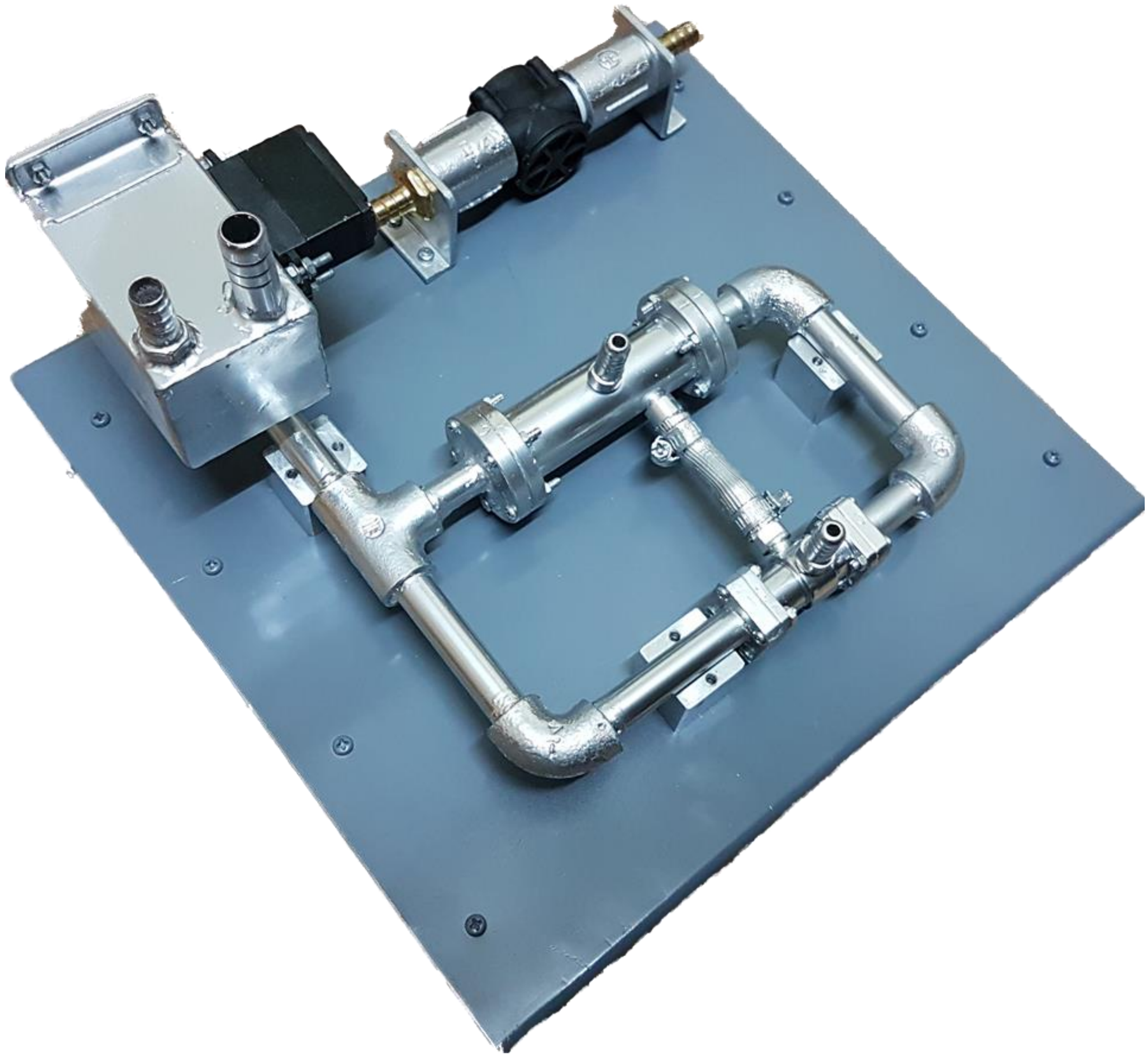

Figure 27: Actual Valve

CHAPTER 4

RESULTS

Simulation Results:

Simulation was done in ANSYS with the following settings and the following pressure contours were obtained:

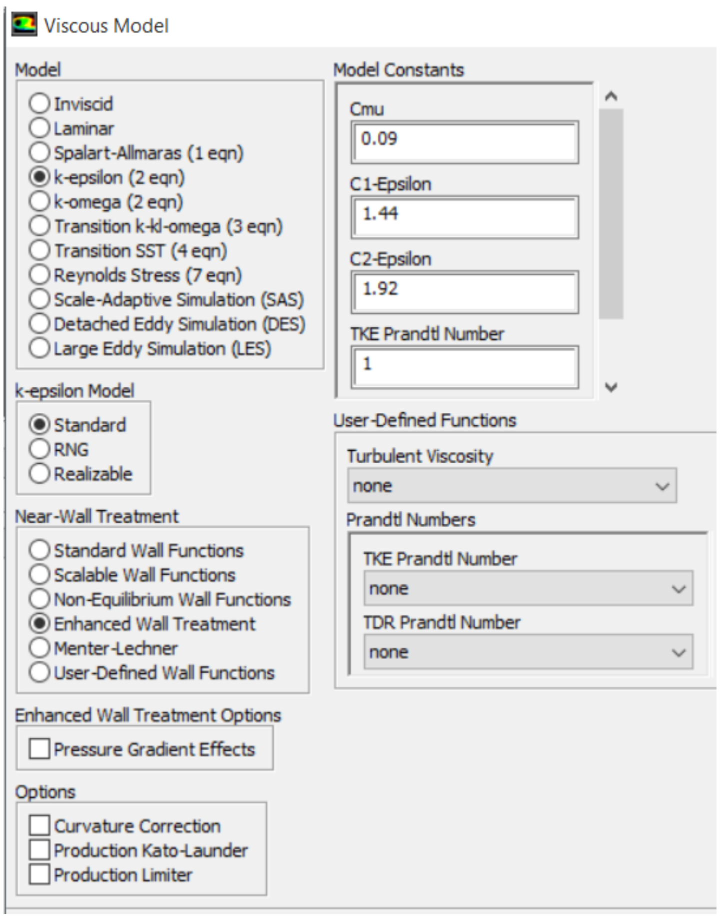

Figure 28: ANSYS Settings

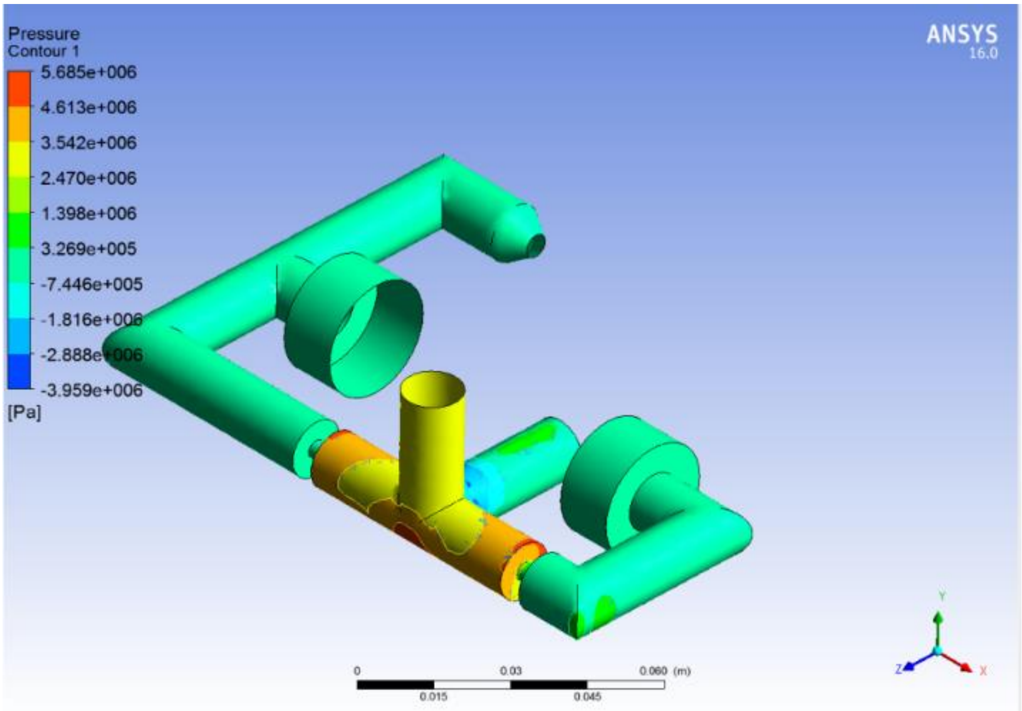

Figure 29: ANSYS Simulation at 60 bar

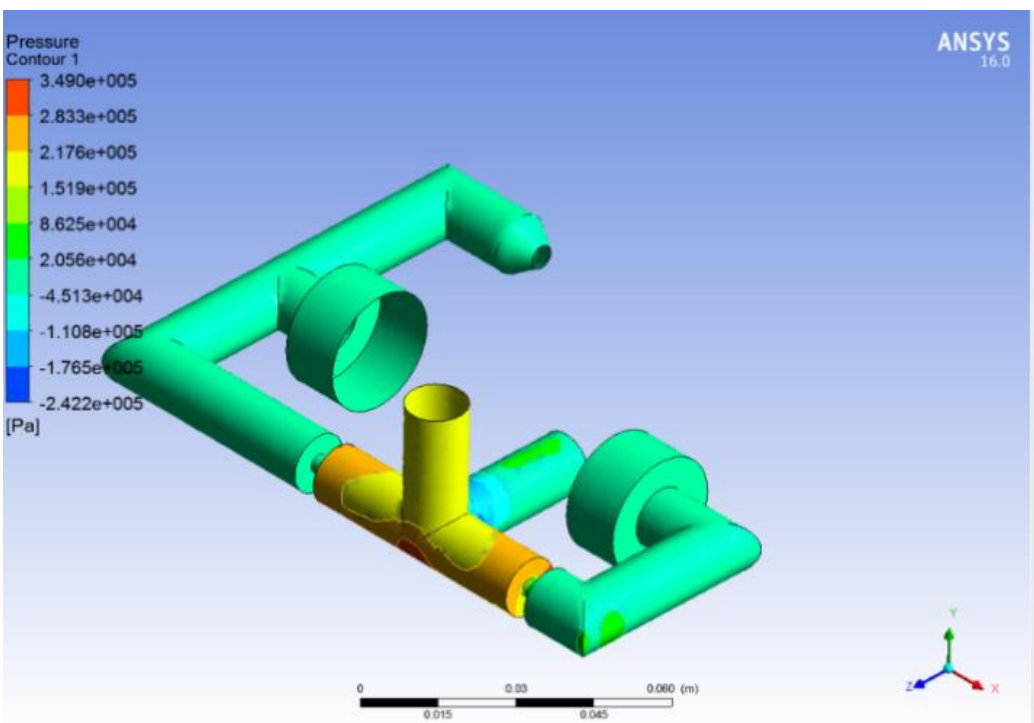

Figure 30: ANSYS Simulation at 3.68 bar

Flow Rate Results

We carried out our testing at 3.68 bars, with a 90ft head pump. The input current vs flowrate results were as follows:

Table 2: Current-Flowrate at 3.68 bar

Pressure 3.68 bar			
Current/ mA	Flow Rate L/hr	Current/ mA	Flow Rate L/hr
0	0	78	146
7	13	85	167
13	38	91	175
20	50	98	192
26	67	104	220
33	84	111	286
39	98	117	295
46	108	124	302
52	112	130	313
59	118	137	328
65	123	143	344
72	133	150	358

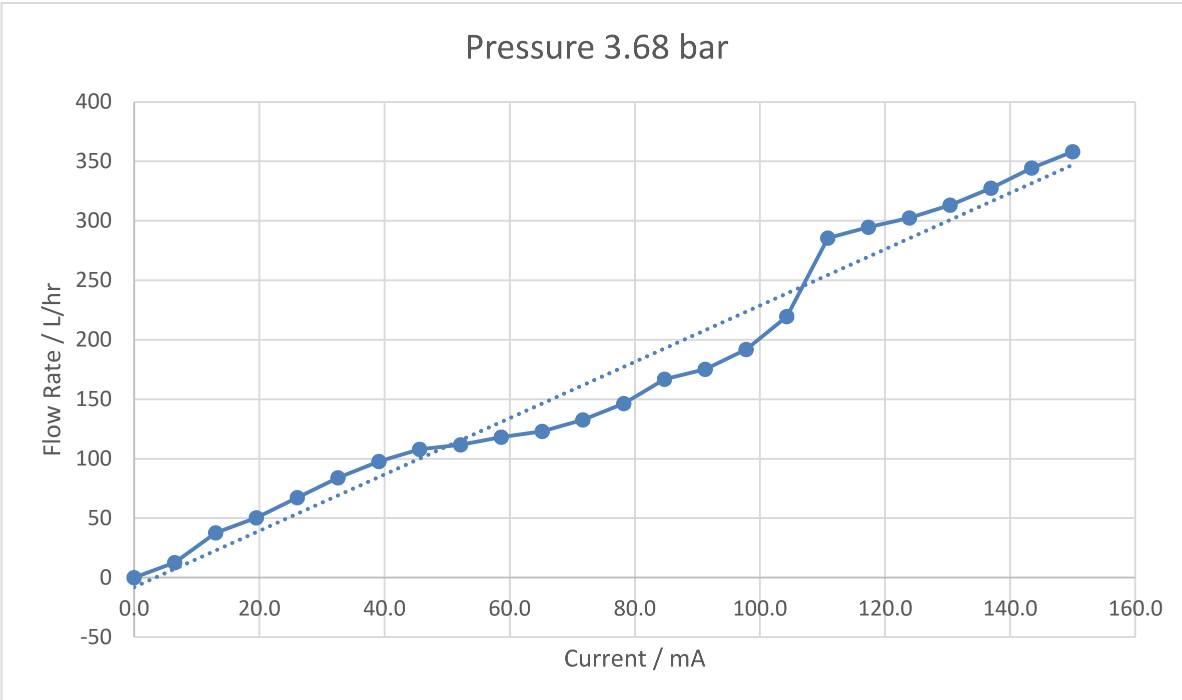

Figure 31: Current vs Flow Rate Curve at 3.68 bar

Extrapolated Results

We had to develop a relation between pressure and flowrate which we could use to extrapolate the results on 60 bar.

For that purpose, let us consider a situation in which we have a water tank and an outlet hole at the base of the tank. The equation of flowrate of water coming out of the hole is:

$$Q = AV$$

Where 'V' is the velocity, 'Q' is the flow rate and 'A' is the area of the hole. The equation of pressure of water inside the hole is:

$$PE_i + KE_i = PE_o + KE_o$$

$$P_i = P_o + KE_o$$

$$P_A + \rho gh = P_A + \rho v^2/2$$

$$\rho gh = \rho v^2/2$$

$$P_i = \rho v^2/2$$

$$P_i = \rho Q^2/2A^2$$

Where P_i is the pressure inside the water tank, P_o is the pressure outside the container, PE_o is the potential energy outside the hole and P_A is the pressure of the atmosphere.

So we can see that pressure inside the container is directly proportional to the square of flowrate through the hole, while keeping the area of hole constant.

So we will use this relation i.e $P \propto Q^2$ to extrapolate our results to 60 bar.

By doing so, we obtained the following results:

Table 3: Current-Flowrate at 60 bar

Pressure 60 bar			
Current/ mA	Flow Rate L/hr	Current/ mA	Flow Rate L/hr
0	0	78	591
7	51	85	673
13	153	91	707
20	203	98	775
26	271	104	887
33	339	111	1153
39	395	117	1189
46	436	124	1221
52	451	130	1264
59	477	137	1323
65	497	143	1390
72	535	150	1446

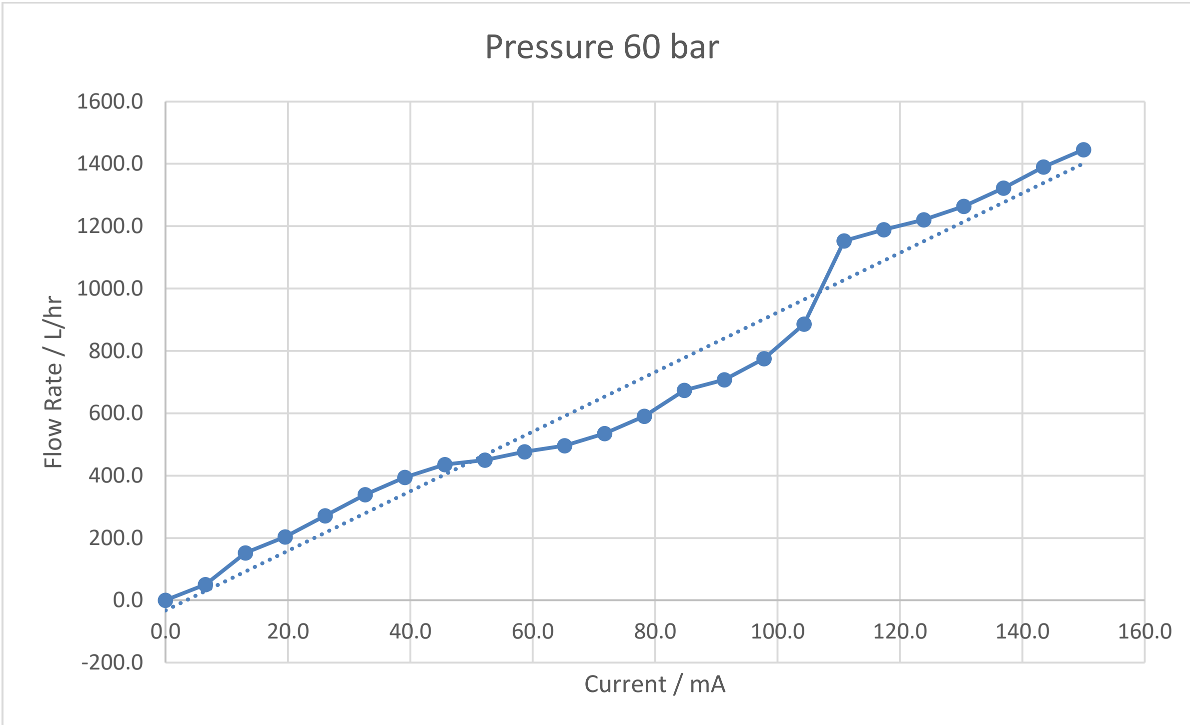

Figure 32: Current vs Flow Rate Curve at 60 bar

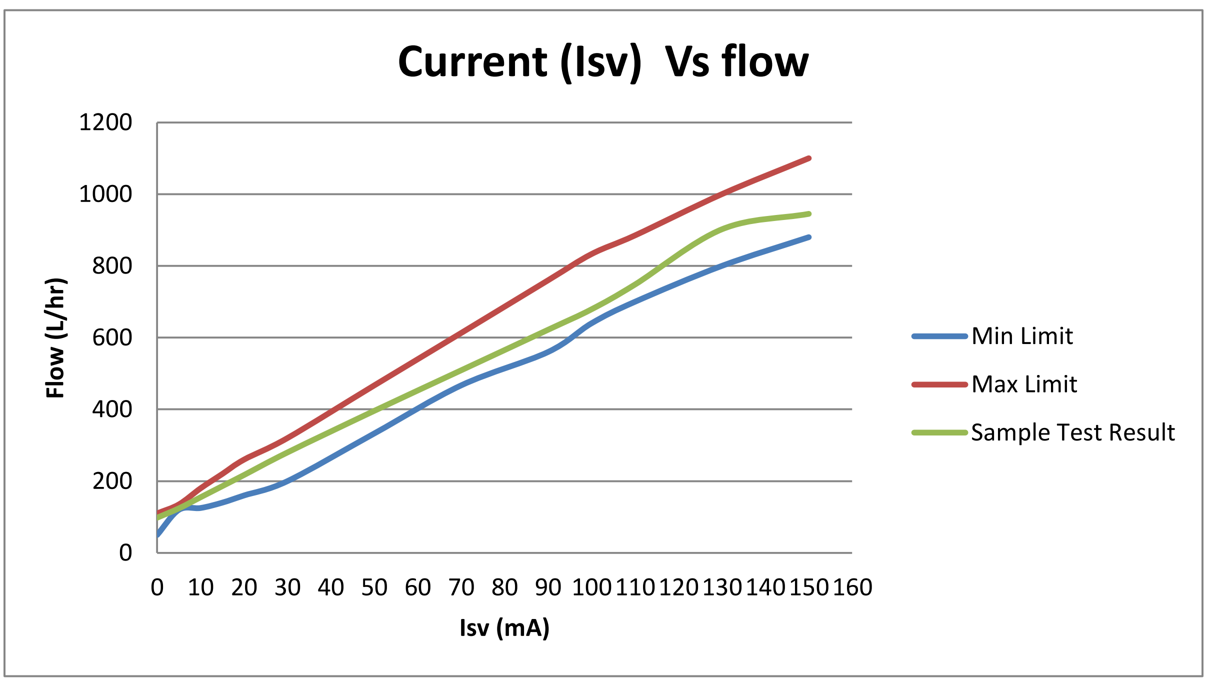

Figure 33: Current vs Flow Rate Curve of Industry Design

CHAPTER 5

CONCLUSION AND RECOMMENDATION

Conclusion

Subject to limited resources and design constraints, the valve was designed to operate at 15 bar gauge pressure and the results were to be extrapolated at 60 bar gauge pressure. However, the availability of pumps lowered our inlet pressure to 3.68 bar gauge pressure and all the flow calculations were done based on the above mentioned pressure condition with water as a working fluid.

It was concluded that water can be taken as a starting fluid to test an electro hydraulic servo valve, with friction and viscosity corrections, the results can be extrapolated to the required conditions and working fluid.

The experimental results were in line with the theoretical and simulation results with an error less than 5% meeting all the required deliverables.

Future Recommendations

- Stress and material testing at elevated temperatures to ensure the valve resilience at a working temperature of a combustor.
- Torque motor with a torque rating, designed specifically for this valve operation.
- To ensure maximum efficiency and reproducible product, the machining could be improved by employing high precision machining operations such as a CNC.
- Testing of the valve at maximum operating conditions in a dedicated testing facility.

WORKS CITED

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[3] History of Servo Valves. Retrieved from 10th International Fluid Power Conference:

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[4] Flow meter. Retrieved from Hall Effect Water Flowmeter:

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[5] Arduino Uno. Retrieved from Arduino Store

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[6] J.Wiley. (1967). *Electro-Hydraulic Servo Valve Construction, Models & Use.* Retrieved from https://www.google.com.pk/url?sa=t&rct=j&q=&esrc=s&source=web&cd=11&cad=rja&uact=8&ved=0ahUKEwj08qzl7ajUAhXHJlAKHSTcAW04ChAWCCQwAA&url=http%3A%2F%2Fume.gatech.edu%2Fmechatronics_course%2FIntroMech%2FHydraulic%2520Servo%2520Valve%2520Construction%2520plus%

[7] Li, P. Y. (2002). *Dynamic redesign of a flow control servovalve using a pressure control pilot*

[8] Plummer, P. A. (n.d.). *Electrohydraulic servovalves – past, present, and future.*

[9] Shuai Wu, Z. J.-Y. (June 2014). *Development of a Direct-Drive Servo Valve With High-Frequency Voice Coil Motor and Advanced Digital Controller.* EEE/ASME TRANSACTIONS ON MECHATRONICS.

APPENDIX I: ARDUINO CODE

```
#include <Servo.h>
Servo servo1;

int angle ;
byte statusLed    = 13;

byte sensorInterrupt = 0;  // 0 = digital pin 2
byte sensorPin       = 2;

// The hall-effect flow sensor outputs approximately 4.5 pulses per second
per
// litre/minute of flow.
float calibrationFactor = 6.5;
volatile byte pulseCount;

float flowRate;
unsigned int flowMilliLitres;
unsigned long totalMilliLitres;

unsigned long oldTime;

void setup()
{

  // Initialize a serial connection for reporting values to the host
  Serial.begin(9600);
  servo1.attach(9);

  // Set up the status LED line as an output
  pinMode(statusLed, OUTPUT);
  digitalWrite(statusLed, HIGH);  // We have an active-low LED attached

  pinMode(sensorPin, INPUT);
  digitalWrite(sensorPin, HIGH);
```

```
  pulseCount        = 0;
  flowRate          = 0.0;
  flowMilliLitres   = 0;
  totalMilliLitres  = 0;
  oldTime           = 0;

  // The Hall-effect sensor is connected to pin 2 which uses interrupt 0.
  // Configured to trigger on a FALLING state change (transition from HIGH
  // state to LOW state)
  attachInterrupt(sensorInterrupt, pulseCounter, FALLING);
}

/**
   Main program loop
*/
void loop()
{

  unsigned int x = 0;
  float AcsValue = 0.0, Samples = 0.0, AvgAcs = 0.0, AcsValueF = 0.0;

  for (int x = 0; x < 150; x++) { //Get 150 samples
    AcsValue = analogRead(A0);     //Read current sensor values
    Samples = Samples + AcsValue;  //Add samples together
    delay (3); // let ADC settle before next sample 3ms
  }
  AvgAcs = Samples / 150.0; //Taking Average of Samples

  //((AvgAcs * (5.0 / 1024.0)) is converitng the read voltage in 0-5 volts
  //2.5 is offset(I assumed that arduino is working on 5v so the viout at no
current comes
  //out to be 2.5 which is out offset. If your arduino is working on
different voltage than
  //you must change the offset according to the input voltage)
  //0.185v(185mV) is rise in output voltage when 1A current flows at input
  AcsValueF = (2.50 - (AvgAcs * (5.0 / 1024.0)) ) / 0.185;
```

```
  Serial.println(AcsValueF);
  AcsValueF = AcsValueF * 1000;
  AcsValueF = map(AcsValueF, 0, 3000, 90, 114);
  Serial.print('Current Value  ');
  Serial.println(AcsValueF);

  servo1.write(AcsValueF);

  delay(1000);
  if ((millis() - oldTime) > 1000)   // Only process counters once per second
  {
    // Disable the interrupt while calculating flow rate and sending the
value to
    // the host
    detachInterrupt(sensorInterrupt);

    // Because this loop may not complete in exactly 1 second intervals we
calculate
    // the number of milliseconds that have passed since the last execution
and use
    // that to scale the output. We also apply the calibrationFactor to scale
the output
    // based on the number of pulses per second per units of measure
(litres/minute in
    // this case) coming from the sensor.
    flowRate = ((1000.0 / (millis() - oldTime)) * pulseCount) /
calibrationFactor;

    // Note the time this processing pass was executed. Note that because
we've
    // disabled interrupts the millis() function won't actually be
incrementing right
    // at this point, but it will still return the value it was set to just
before
    // interrupts went away.
```

```
    // Divide the flow rate in litres/minute by 60 to determine how many
litres have
    // passed through the sensor in this 1 second interval, then multiply by
1000 to
    // convert to millilitres.
    flowMilliLitres = (flowRate / 60) * 1000;

    // Add the millilitres passed in this second to the cumulative total
    totalMilliLitres += flowMilliLitres;

    unsigned int frac;

    // Print the flow rate for this second in litres / minute
    //Serial.print("Flow rate: ");
    Serial.print('Flow Rate');
    Serial.print(int(flowRate));  // Print the integer part of the variable
    //Serial.print('!');
    //Serial.println(angle);

    //Serial.print("L/min");
    //Serial.print("\t");       // Print tab space

    // Print the cumulative total of litres flowed since starting
    //Serial.print("Output Liquid Quantity: ");
    //Serial.print(totalMilliLitres);
    //Serial.println("mL");
    //Serial.print("\t");       // Print tab space
    //Serial.print(totalMilliLitres/1000);
    //Serial.print("L");

    delay(1000);
    oldTime = millis();

    // Reset the pulse counter so we can start incrementing again
    pulseCount = 0;

    // Enable the interrupt again now that we've finished sending output
```

```
    attachInterrupt(sensorInterrupt, pulseCounter, FALLING);
  }
}


/*
  Insterrupt Service Routine
*/
void pulseCounter()
{
  // Increment the pulse counter
  pulseCount++;
}
```

APPENDIX II: CIRCUIT DIAGRAM

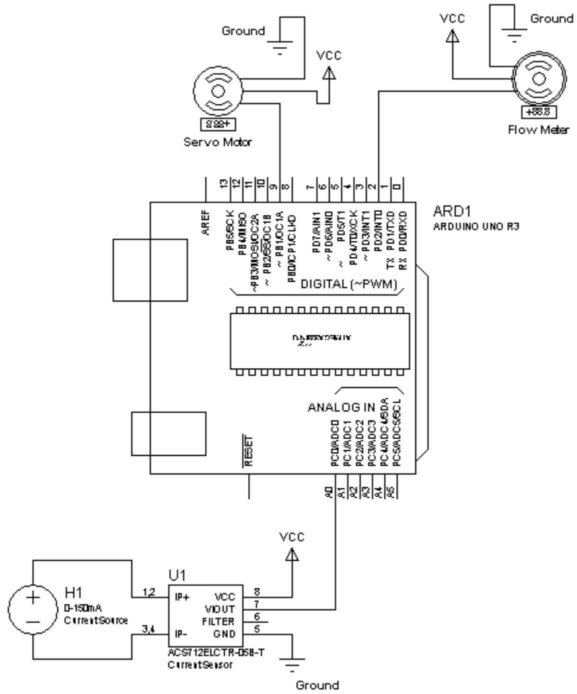

Figure 34: Circuit Diagram

APPENDIX III: MATLAB CODE

```
%MATLAB CALCULATIONS (EHSV)
%Given Parameters

p=800 %kg/m3 Density
L1=.0091 %m Length 1
L2= .2015 %m Length 2
L3= .0129 %m Length 3
neu=4.774*10^-6 %m2/s Kinematic Viscosity
V= 3.51 %m/s Assumed Inlet Velocity
D= 0.0127 %m Pipe Diameter
Roughness=0.045 %m
P1=368      %KPa Inlet Pressure
R_S=0.019    %m Radius of Spool
h_S=0.01   %m Thickbess of Spool
R_N= 0.003175 %m Radius of Nozzle

%Governing Equations
V_N=((3.14*0.00635^2)*(V/3))/(3.14*R_N^2); % Velocity After Nozzle

A_S= 3.14*R_S^2; % Spool Area

Re = (V*D)/neu; % Reynolds Number

f= (1/(-1.8*log(((Roughness/D)/3.7)^1.11 + (6.9/Re))))^2; % Frictional Factor

P5= P1-(((f*L1*p*(V/3)^2))/2*D)-2*((0.3*p*(V/3)^2)/2); % Pressure at position 5

P6= P1-(((f*L3*p*(V/3)^2))/2*D)-((0.3*p*(V/3)^2)/2);   % Pressure at position 6

P7= P1-(((f*L2*p*(V/3)^2))/2*D)-2*((0.3*p*(V/3)^2)/2); % Pressure at position 7

F5=P5*A_S; %Spring force on Side 5

F6=P6*A_S; %Spring force on Side 6

fprintf('Reynolds number is %d.\n',Re);
fprintf('Nozzle Velocity is %d m/s.\n',V_N);
fprintf('Pressure (Without Orifice) at Position 7 is %d KPa.\n',P7);
fprintf('Pressure (Without Orifice) at Position 6 is %d KPa.\n',P6);
fprintf('Pressure (Without Orifice) at Position 5 is %d KPa.\n',P5);
fprintf('Pressure (With Orifice) at Position 5 is %d KPa.\n',20.056);
fprintf('Pressure (With Orifice) at Position 6 is %d KPa.\n',20.056);
fprintf('Force at Position 5 is %d KN.\n',F5);
fprintf('Force at Position 6 is %d KN.\n',F6);
```