Hydraulic Variable Valve Timing Testing and Validation

by

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AUTHOR'S DECLARATION

I hereby declare that I am the sole author of this thesis. This is a true copy of the thesis, including any required final revisions, as accepted by my examiners.

I understand that my thesis may be made electronically available to the public.

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Abstract

This thesis documents the development of a fully continuous, hydraulics-based variable valve timing system. This hydraulics based variable valve timing system is capable of controlling an engine valves lift height and infinitely varying the engine valves lift profile. Along with full valve controllability during normal operation, the variable valve timing system is capable of providing the same operation as a classic cam shaft under engine power loss conditions. This is possible due to the rotating hydraulic spool valves coupled to the engines crank shaft, which are used to actuate the engine poppet valves.

The main focus of this thesis is to investigate, alter and implement a new iteration of the hydraulic variable valve timing system on a standalone test bench to validate the systems operating principles. The test bench utilizes servo motors to act as an engines crank shaft which runs the rotating hydraulic spool valves and hydraulic pump. This serves as an intermediate step to full engine implementation of the variable valve timing system.

The research begins by analyzing the current mechanical spool valve and hydraulic cylinder design for any potential problems that may occur either during assembly or full operation. The basic system equations are presented to give a glimpse into the working principles of the rotary valves. The mechanical, electrical, and hydraulic subsystems are discussed in terms of what was considered during the design and implementation process. Then design changes that were performed on the rotary valve system to overcome any failures. Lastly the resulting data is presented from the current variable valve timing design to verify proper system functionality.

Acknowledgements

I would like to thank my supervisor Professor Amir Khajepour for allowing me the opportunity to learn in such a knowledge rich environment that has taught me so much. It was an invaluable experience to complete my master's studies under his guidance.

I would also like to thank Jeff Graansma, Jeremy Reddekopp, Jason Benninger and Robert Wagner for all of their support in everything from machining to electrical design during the life of this project.

Lastly I would like to thank Yangtao Li for offering his insight during the thesis writing process.

Dedication

I would like to dedicate this thesis to my loving wife, Katherine, who has always encouraged me to pursue my interests and to my new daughter, Freya.

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Chapter 1 Introduction

1.1 Variable Valve Applications

As engine technology becomes increasingly advanced, there are three aspects that push the continuous innovation in engine design and control. The aspects that govern the majority of the engine design process are maximizing torque output, minimizing engine emissions and maximizing fuel economy.

The engine emission constraints are restrictions imposed on vehicle manufacturers through provincial and federal emission regulations. These regulations are some of the hard rules that govern the majority of engine design decisions built into consumer vehicles.

The fuel economy constraint is subjected to the vehicle manufacturers by the end consumer that is going to be purchasing the vehicle. This is not a hard rule like the emission standards since consumers look for different factors in a vehicle. A person buying a muscle car for instance, will not put as much consideration into the fuel economy as someone who wants to get the most mileage for their money.

Lastly, the engine performance in terms of torque output is one of the goals of all vehicle manufacturers, as someone who is purchasing a vehicle for its exceptional fuel economy does not expect a super car, but still want some measure of performance to go with it.

The need for variable valve timing has risen from these three requirements since maximizing torque is inversely related to maximizing fuel economy and minimizing emissions. With a conventional cam shaft engine the designers are able to either emphasize performance or fuel economy, or most likely a tradeoff between them, but not both since the valves will follow a discrete profile set by the cam shaft.

As soon as the valves are able to be controlled, however, an engine is no longer limited to fulfilling one role. An engine can switch from conserving fuel while cruising on the highway over to a more performance focused mode which provides the driver with more engine torque on demand. This would be valued in situations when there is a need to pass safely without being underpowered, or more importantly to avoid an obstacle.

Another benefit of valve control is reducing emissions by recirculating exhaust gasses causing any unutilized fuel to go through the combustion process again. This will allow vehicle manufacturers to have more freedom in engine design while still meeting the emission standards set forth by the government.

1.2 Previous Work

This project is a continuation of the thesis completed by M. Pournazeri entitled Development of a New Fully Flexible Hydraulic Variable Valve Actuation System [1].

M. Pournazeri has introduced a method of variable valve actuation though the use of a hydraulic cylinder that is actuated and returned by two rotating spool valves connected to the engine crank shaft through a pulley and phase shifting system. This allows the valves to infinitely change their angular relation to the engine crank angle and fully control the valve lift height.

The full system proposed by M. Pournazeri is illustrated in Figure 1.

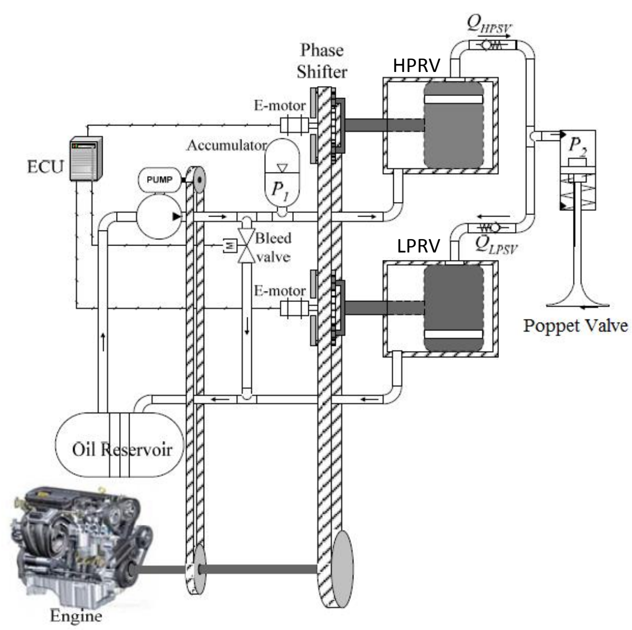

Figure 1 M. Pournazeri Variable Valve System [1]

The high pressure rotary valve (HPRV) and low pressure rotary valve (LPRV) design can be seen in Figure 2. The operating principles of this rotary valve is that hydraulic fluid is introduced from the accumulator/hydraulic pump to the input port pressurizing the casing cavity. With the spool shaft spinning at a 1:2 ratio to the crank shaft the spool port will align with the casing port allowing hydraulic fluid to flow through the high pressure check valve and pressurize the poppet valve cylinder. The low pressure spool port is trailing the high pressure spool port by a certain controllable angle, and therefore blocks the hydraulic fluid flow causing the poppet valve cylinder to extend.

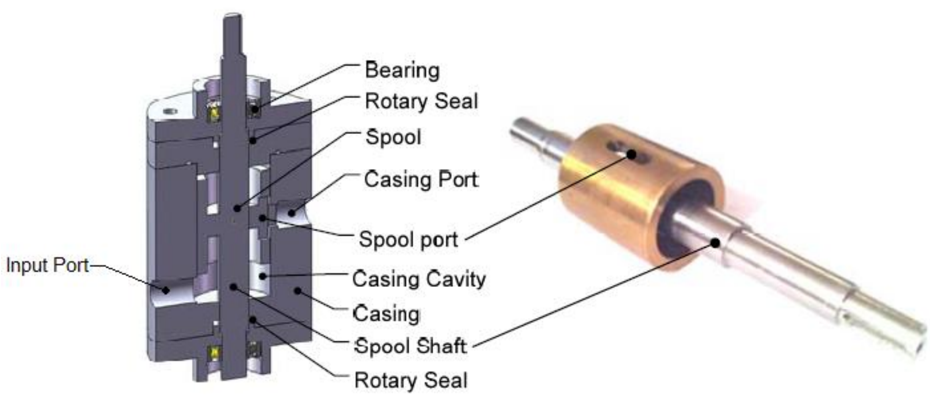

Figure 2 M. Pournazeri Spool Valve [1]

As the crank shaft continues imposing rotation onto the spool valves, the high pressure spool port continues past the high pressure casing port trapping the hydraulic fluid in the poppet valve cylinder.

The crank rotates further aligning the low pressure spool port with the low pressure casing port allowing the poppet valve cylinder to vent the fluid through the low pressure spool valve and into the oil reservoir causing the poppet valve to close through the force of the poppet valve spring.

The opening and closing time is controlled by shifting the phase between the crank and the spool valves allowing them to open sooner or later depending on the phase shift.

The valve lift is then controlled by changing the pressure stored in the accumulator. This is done by venting fluid from the hydraulic pump once certain pressure set point is reached using an analog controlled pressure relief valve.

M. Pournazeris experimental results proved successful in creating the desired valve profile and lift control up to his maximum tested engine speed of 1000rpm.

M. Pournazeris thesis served as a starting point for exchange student M.W.H. Bergmans who extended the rotary valve concept to control a 4 cylinder engine. The spool is designed with the same concept of an input port that can flow freely to the hollow cavity inside the spool as illustrated in Figure 3. The output ports are then configured 90^o axially and shifted by 30mm along the shaft from one another.

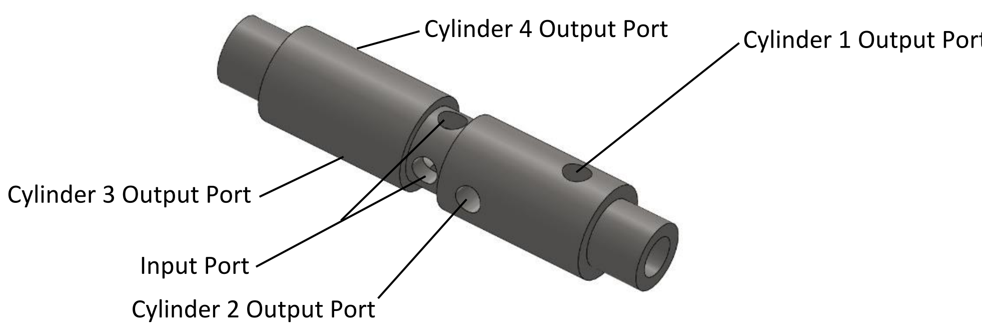

Figure 3 Four Cylinder Spool Design [2]

The housing for the spool is illustrated in Figure 4. As can be seen with the housing ports all aligned each port will allow hydraulic oil to flow to its respective cylinder with 90^o rotation of the spool which is in turn 180^o rotation of the crank shaft.

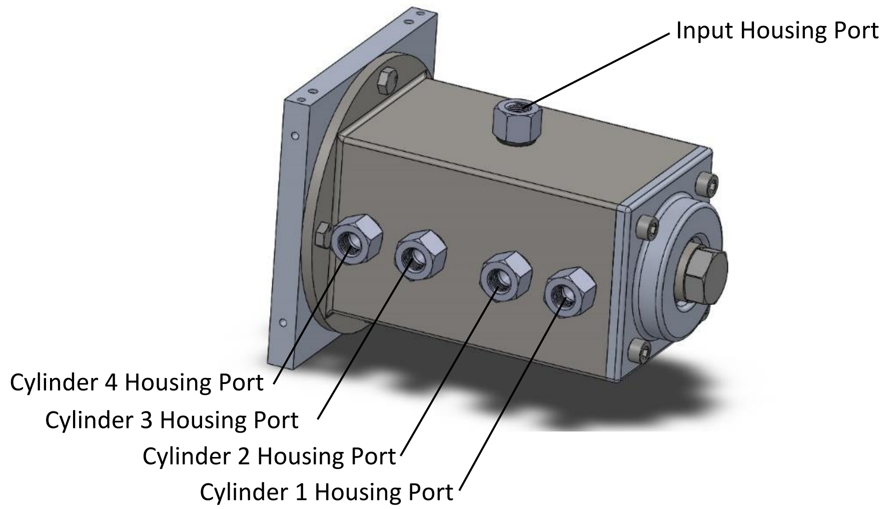

Figure 4 Four Cylinder Spool Housing [2]

The phase shifting gear train was also designed to be manufactured and shown in Figure 5.

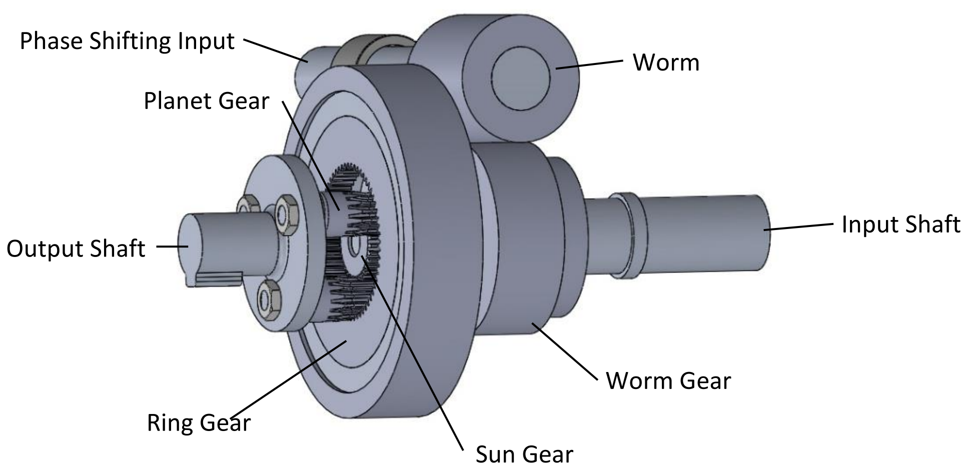

Figure 5 Planetary Phase Shifter [2]

The input shaft is powered by the crank shaft which then turns the sun gear and in turn transmits torque through the planet gears to the output shaft at a ratio of 3:1. This configuration will require a timing belt pulley ratio of 2:3 between the crank shaft and the input to ensure the overall 2:1 ratio of crank shaft to spool rotation. The ring gear is then controlled through a stepper motor attached to the phase shifting input. This transmits torque through the worm/worm gear set which is rigidly attached to the ring gear resulting in rotation of the ring gear. Control over the planetary ring gear allows the spool to advanced or retard its relation to the angle of the crank shaft.

1.3 Thesis scope and Structure

The goals, for this stage in the project, start with reviewing the design of the current full rotary valve and cylinder assembly to ensure ease of manufacturing and assembly for when the parts are made. Upon the completion of the design review and as the components are being fabricated, the test bench assembly is designed to allow implementation of the variable valve timing system. Included in the test bench design is the hydraulic system design, mechanical structure layout, and electrical system design seen in Sections 4.1.1, 4.1.2, and 4.1.3, respectively. Once the test bench is complete, with the manufactured variable valve timing components assembled, a controller is chosen and implemented to allow system parameterization and control seen in Section 4.2. Simulink is then utilized as the programming framework to tie all of the electrical and mechanical component control into the functioning variable valve timing test system described in Section 4.3. The completed test bench is then run to validate the variable valve operation supported by data retrieved from the sensors detailed in 4.1.3. The second stage of the project is to review the first revision of the system to make any design changes which improve performance, ensure reliability, or increase robustness.

Chapter 2 Literature Review

2.1 Conventional Valve Timing

Camshaft based valve timing engines are the vast majority of those currently in use today. They are mechanically very simple in their operation. Internal combustion engines require some type of fuel to operate. The fuel is introduced into an engine cylinder through a port opened and closed by an engine valve, which is controlled by a profile machined onto the rotating cam shaft. Once the fuel is introduced into engine cylinder and the valve is closed, the fuel mixture is ignited causing the piston to transfer the energy through a slider-crank mechanism and eccentric connection to the crank shaft resulting in rotation of the crank shaft. The burnt fuel mixture then leaves the cylinder though a separate exhaust valve also controlled by the cam shaft. A simplified illustration of this system can be seen in Figure 6.

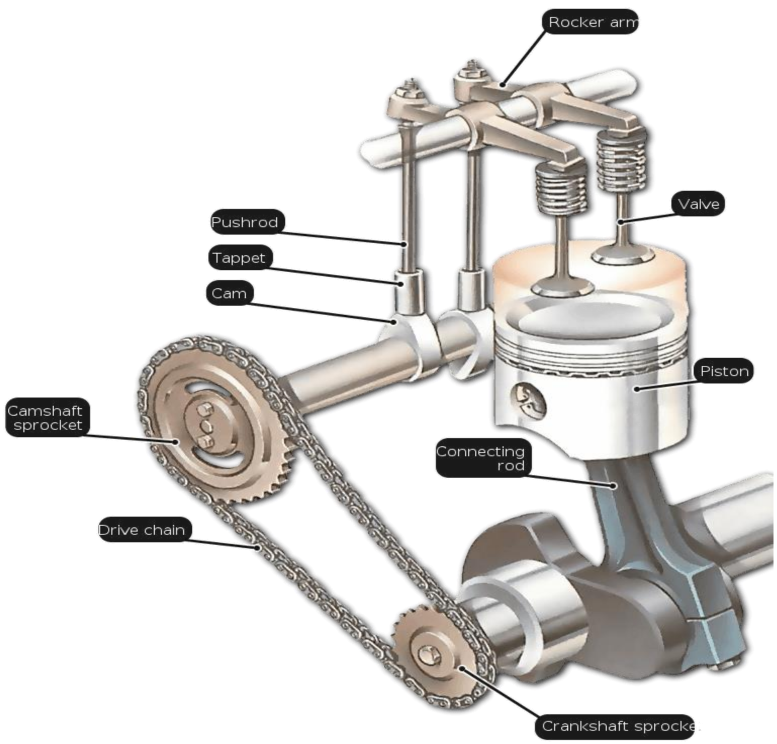

Figure 6 Classic Engine Operation [3]

Since there is mechanical contact between the valve and camshaft, as seen in Figure 6, the engine valves have one discrete profile in which they are capable of following. There is an option to change the camshaft for another with different profiles to achieve different engine behavior, but this requires a lengthy disassembly of the engine.

As discussed previously, control over valve phasing and valve lift height is the next logical step for automotive manufacturers to both meet environmental constraint and maximize engine power when it is needed. This field of engine research is advancing rapidly because of these aspects. Engine valve control solutions have been done but there is a high cost if failure occurs. Therefore, not only must the valve timing system be robust in its capabilities, but also reliable with redundancy in the case of failure. With failure protection of a variable valve timing system, the cost of such an incident changes from potential catastrophe in the case of an engine valve staying extended and striking the piston to the vehicle reverting back to classic passive valve operation.

Conventional cam based valve control may not disappear overnight or even over the next 5 years, but it is a dated technology that will not be able to meet the higher standards for emission requirements set forth by the government. Therefore, a look at the current systems on the market which offer some form of valve lift and timing control is required to address any caveats in their design.

2.2 Valve Lift Control

Variable Valve Timing and Lift Electronic Control (VTEC)

The most commercially recognized form of valve control is Honda's VTEC which is a technology used in many of their current vehicles and has proven to be quite effective and reliable. VTEC engines are different from conventional systems in that they have two cam lobe profiles for each set of intake and exhaust valves as seen in Figure 7. For low and midrange engine rpm, the engine valve rockers run on the two outer cam lobes, while the inner rocker passively follows the inner lobe. Once the engine reaches a certain engine speed, and the outer and inner rockers are aligned, a pin connects all three together. The outer rockers are then rigidly connected to the inner rocker and force the valves to follow the inner cam profile. The inner cam lobe is capable of providing a higher valve lift and slightly alter valve opening duration, but only through the profile of the second cam lobe, therefore, there are two discrete valve profile options.

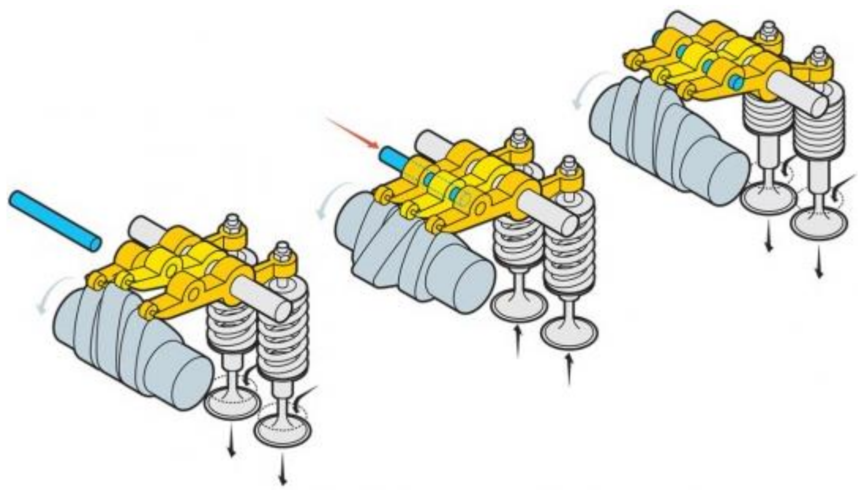

Figure 7 VTEC Operation [4]

The first VTEC systems used this double cam profile technique and, therefore, was able to alter the valve lift but had no means of changing the angular relationship between the cam and crankshaft in real time. However, this technology was added to the VTEC lineup in the form of a hydraulic phaser called Variable Timing Control (VTC).

Valvelift

Another multi cam lobe based strategy of controlling valve lift is the Audi Valvelift system which shares some similarities with the Honda VTEC method in that it has two discrete cam lobes for high and low rpm. The two methods differ, however, in where Honda VTEC tandems the valve rockers together, while Audi mechanically shifts the entire cam shaft axially so that the valve rocker transitions between the two profiles when the valve is in the closed position. An overview of the principles of the Audi Valvelift system can be seen in Figure 8.

Figure 8 Audi Valvelift System [5]

The limitation in this method again is the lack of variability since there are only two distinct valve profiles possible with no option of cam phasing.

<u>Electromechanical Variable Valve Train (EMAVVT)</u>

The EMAVVT solution of valve control is another cam based lift control method in which the cam runs against a hydraulic master cylinder, which transfers hydraulic pressure through a proportional pressure relief valve to a piston that extends the engine poppet valve. The valve lift is then controlled by adjusting the proportional pressure relief valve via the ECU [6]. The diagram of this system can be seen in Figure 9.

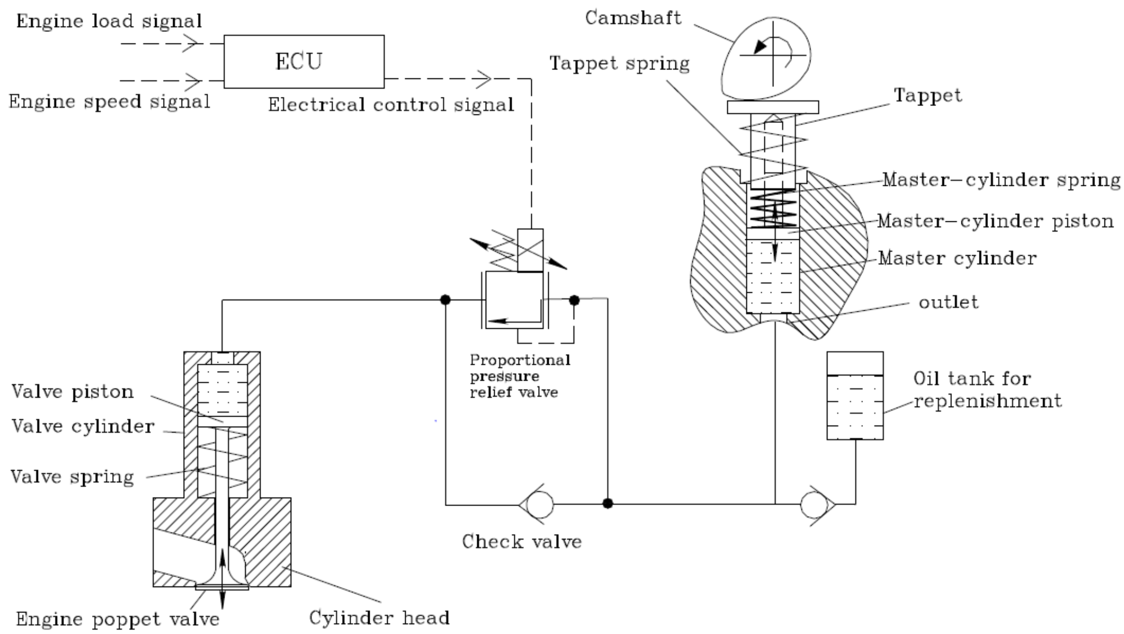

Figure 9 EMAVVT Valve Control System

2.3 Valve Timing Control

VTC

Honda accomplished Variable Timing Control by using hydraulics to advance or retract the cam shafts angle compared to the crank shaft as seen in Figure 10.

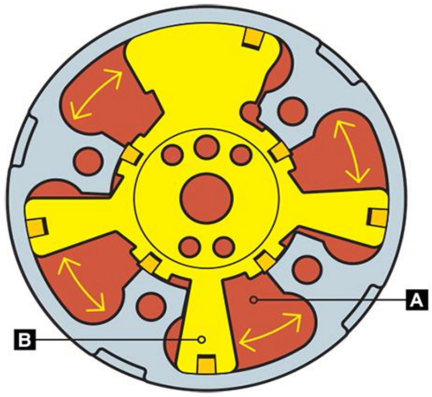

Figure 10 Valve Timing Control [4]

The crank shaft is attached to the outer housing of the phasing unit while the cam shaft is connected to the internal rotor (B). As the cavity (A) is filled with hydraulic oil the relationship between the two can be adjusted. As can be seen in the figure however, the relationship between the two has a maximum angle that it can be phased which is 30^o in either direction.

<u>VarioCam</u>

A similar technique to accomplish a limited degree of phase shifting is the Porsche VarioCam system which is very similar to the BMW VANOS method of using a hydraulic plunger that pushes against meshing helical gears resulting in a change in angle of the cam shaft based on the plunger displacement as seen in Figure 11.

Late valve timing

Early valve timing

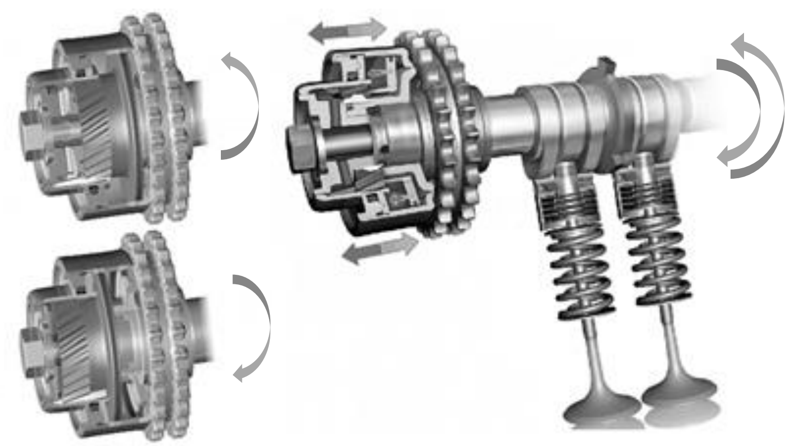

Figure 11 VarioCam Timing System [7]

Porsche also employs a separate method for controlling the lift height of the engine valves, but it is similar to the dual cam lobe profiles discussed previously, and is therefore, not worth discussing, as it is a variation on the same two level valve lift height result.

<u>Electromagnetic Control</u>

The electromagnetic versions of valve actuators proposed usually consist of two springs which keep the valve in a neutral center position while the electromagnets are not active

[8]. The upper coil is energized when the engine is turned on to seat the engine valve in the cylinder head and compress the upper spring. To extend the engine valve the upper coil is de-energized allowing the upper spring to force the valve downwards where the bottom core is energized to capture and hold the valve in the open position. A major issue that is present when using electromagnets to actuate the engine valve is that there is no complete resolution to the valve impacting the cylinder head under normal operation. This results in engine wear and excessive engine noise. The electromagnetic control method also does not offer any ability to alter the engine valves lift height.

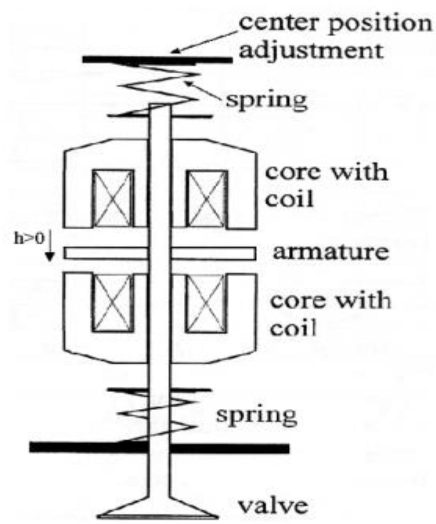

Figure 12 Electromagnetic Valve Actuator [8]

2.4 Combination Valve Control

Methods discussed thus far have been addressing the engine valve lift height and valve phasing as two separate problems. However, to truly achieve complete control over the valve lift height and the valve profile, there must be some form of variable actuation technique to actuate the engine's valves that has either a flexible mechanical relation to the crank or completely decoupled from the crank shaft all together.

Electro-hydraulic Control

The logical alternative to ridged cam based systems is to utilize some form of actuator that can open and close the engine valve. The first actuator is a two way hydraulic valve proposed by J. Allen [9] utilizing a high speed servo valve to control the opening and return strokes as illustrated in Figure 13. This is advantageous over the cam based system in that using hydraulics allows full continuous control over the valves lift height, as well as completely independent timing from the crank shaft. This type of application, however, requires a specialized high speed servo valve responsible for switching three times, initially to lift the engine valve, then to block the ports keeping the engine valve extended, and then again to retract the engine valve. A servo valve capable of accomplishing this level of timing allows a lot of flexibility from a control standpoint, but would be an extremely costly component when considering adoption of the technology and potential mass manufacturing.

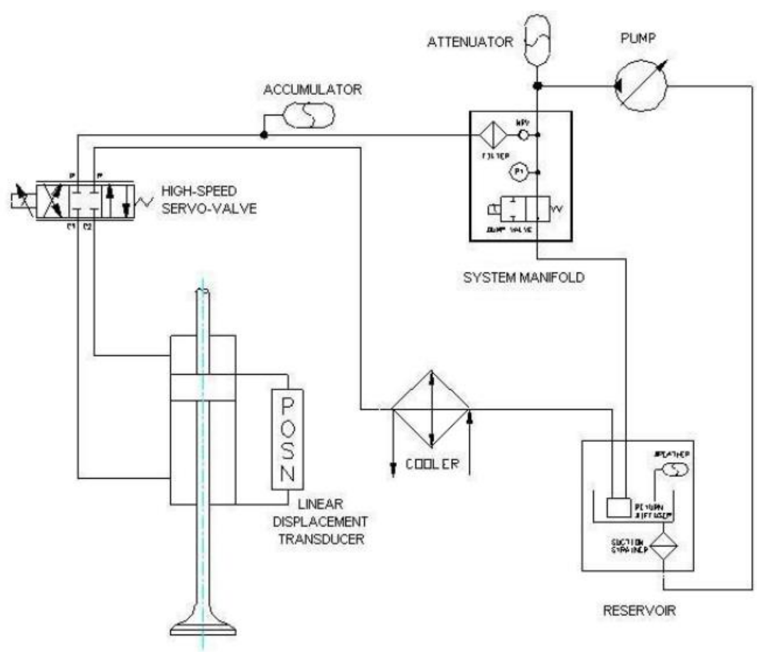

Figure 13 Electro-Hydraulic control principle for one valve [9]

Electro-pneumatic Control

A system called Free Valve produced by Cargine and Koenigsegg is the most well known in terms of electro-pneumatic engine valve actuation due to it being the most developed and currently in use on a SAAB car engine. The electro-pneumatic system used in the Free Valve system also employs an aspect of electro-hydraulic control to allow the position of the valve to be held and decelerated by metering hydraulic fluid flow through a small port internal to the valve lift mechanism. The entire system can be seen in Figure 14.

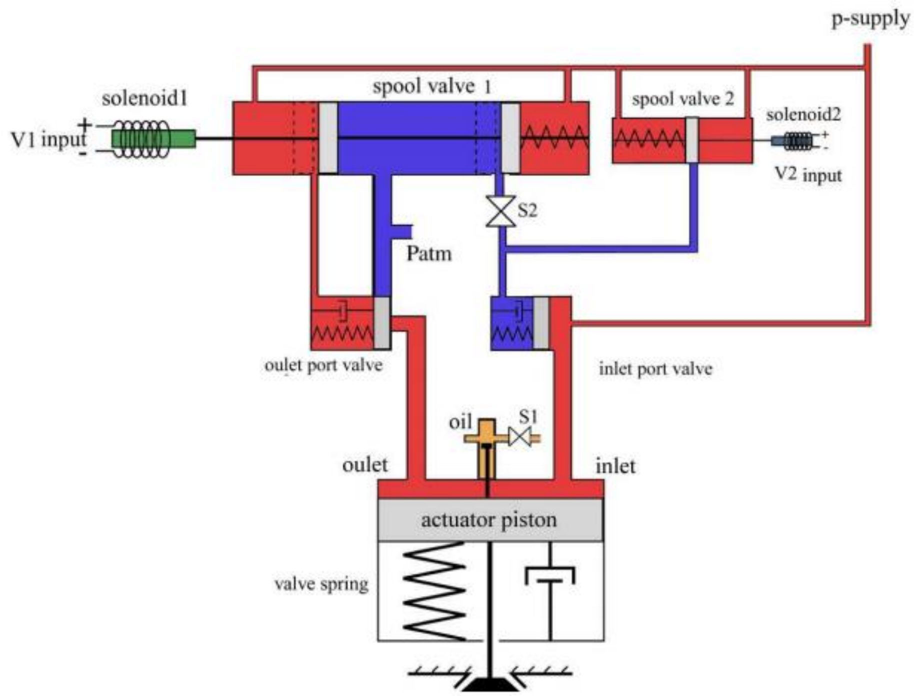

Figure 14 Electro-pneumatic variable valve control [10]

The above system operates in three stages. Solenoid 1 is energized allowing pressurized air to bypass the inlet port since the rear of the port is open to atmospheric pressure. The pressurized air is also introduced to the rear of the outlet port ensuring that the outlet port is closed. The air pressure in the actuator piston cylinder then causes the piston to extend the engine poppet valve. As the valve extends the S1 valve acts as a check valve allowing oil to flow through it and into the small oil piston reservoir. When the desired valve lift is reached, stage two begins by energizing solenoid 2 to pressurize the rear section of the inlet port and block the S2 valve effectively stopping air flow into or out of the actuator piston cylinder. Since S1 is acting as a check valve the oil pressure is at the same pressure as the actuator piston cylinder and is unable to flow back through the S1 port creating a hydraulic lock against the valves return. When the valve is required to return, the third stage occurs by de-energizing both solenoids and opening port S2, then metering port S1. This allows pressurized air to fill the rear section of the inlet port valve closing it and introducing atmospheric pressure to the rear section of the outlet port valve letting the pressurized air from the actuator piston cylinder to vent into the atmosphere. The valve spring retracts the engine poppet valve to its seated position. The flow control of the hydraulic fluid through S1 provides a deceleration to avoid valve impact with the cylinder head. The damper shown in the system is their method of modeling the approximate energy dissipation due to the friction in the valve mechanism.

This system works but has some short comings such as delayed response time of controlling solenoids which in this system is remedied by modern control system techniques. Another issue that arises with using pneumatics, is the nature of airs compressibility making the valve prone to issues with unpredicted engine cylinder pressure variations.

2.5 Summary

Two large hurdles that lay in the way of widespread adoption of many of these fully continuous valve controls are price and fault tolerance. If there is any specialized equipment required by the system then the technology suffers, as with any industry the cost of manufacturing plays a crucial role in what is and is not successful. Fault tolerance is also an issue as with a lot of the cam-less valve control technologies, if there is one of any number of electronics errors the engine will cease to operate which can range in consequences from having to pull over or more seriously causing a car accident.

Chapter 3 Variable Valve Design

3.1 Initial Design Review

The variable valve design at the beginning of this stage in the project was inherited from an exchange student, M.W.H. Bergmans [2] whose work was outlined in Section 1.2.
The variable valve system has two assemblies that required manufacturing. The first is the rotary valve assembly consisting of the spool valve portion and the planetary phase shifting portion displayed in Figure 15. The second being the cylinder head replacement assembly consisting of the cushion portion and the cylinder cover replacement shown in Figure 16.

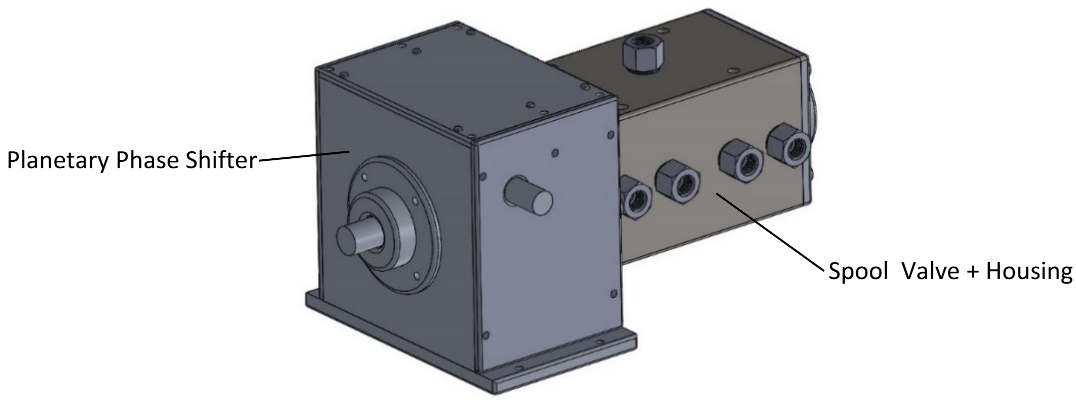

Figure 15 M.W.H. Bergmans Rotary Valve Assembly [2]

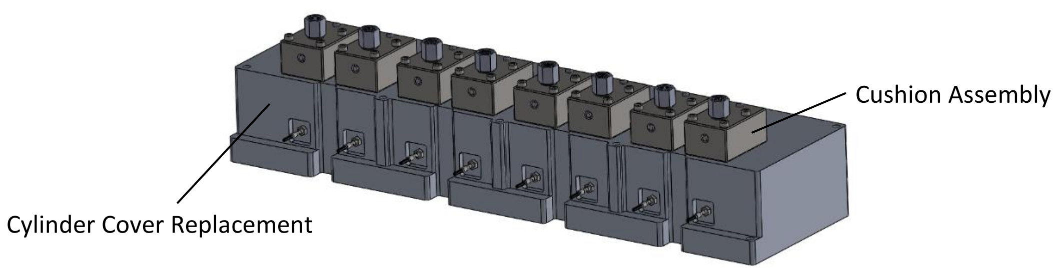

Figure 16 M.W.H. Bergmans Cylinder Assembly [2]

The last main component in the variable valve design is a custom designed cushion assembly that seats against the back end of the engine valve stem which is responsible for actually extending the engine cylinder valve as seen in a section view in Figure 17.

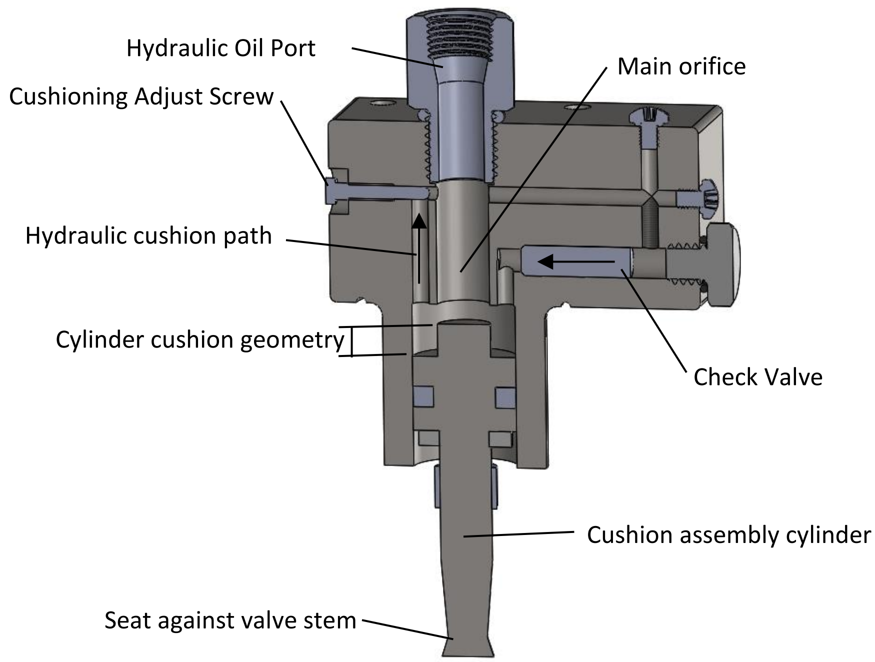

Figure 17 Valve Cylinder Section View [2]

This cylinder differs from M. Pournazeris design by adding cushioning geometry to the hydraulic cylinder to ensure that when the engine valve is closing it will decelerate shortly before it fully seats back into the cylinder head. This is not a problem with cam based designs because the engine valve is directly controlled by the cam lobe profile and does not run the risk to excess force of impacting with the cylinder head when traveling from an open to closed position. Cam-less valve control systems, however, must take this possibility into account as not doing so causes wear and eventual failure of the engine valve or improper sealing between the cylinder head and the valve. M. Pournazeri addresses this problem by using a second cylinder in his test system to dampen the main cylinder for the last 4mm before the valve seats. M.W.H. Bergmans, however, designs this functionality into the main cylinder as seen in Figure 17. When the cylinder is retracting the hydraulic oil leaves through the hydraulic oil port until the cylinder cushion geometry enters the main orifice. At this point the oil trapped by the cushion geometry cannot travel through the check valve and is, therefore, forced to travel through the hydraulic cushion path. The fluid flow through this path can be controlled by the cushion adjust screw to allow for different levels of cylinder deceleration. When the cylinder extends, however, the fluid is able to flow through the check valve freely so there is no hindrance to the speed in which the cylinder can extend.

The advantage of having the valve cylinder shown in Figure 17 as a standalone component is the ability to design an engine valve/camshaft cover specific to one model of engine. In these covers there can be a simple bolting interface above the engines valve stem which can accept the valve cylinder unit pictured in Figure 17, which will control the engines poppet valve. What this means is that current engines in the market can then have their valve cover and cam shaft removed then fitted with a valve cover designed to work with the variable valve timing system. The new valve cover can then have the generic valve cylinder attached to it.

3.2 Initial Design Changes

The changes required for the rotary valve assembly aside from minor bolt hole misalignments and tolerance changes can be seen in Figure 18.

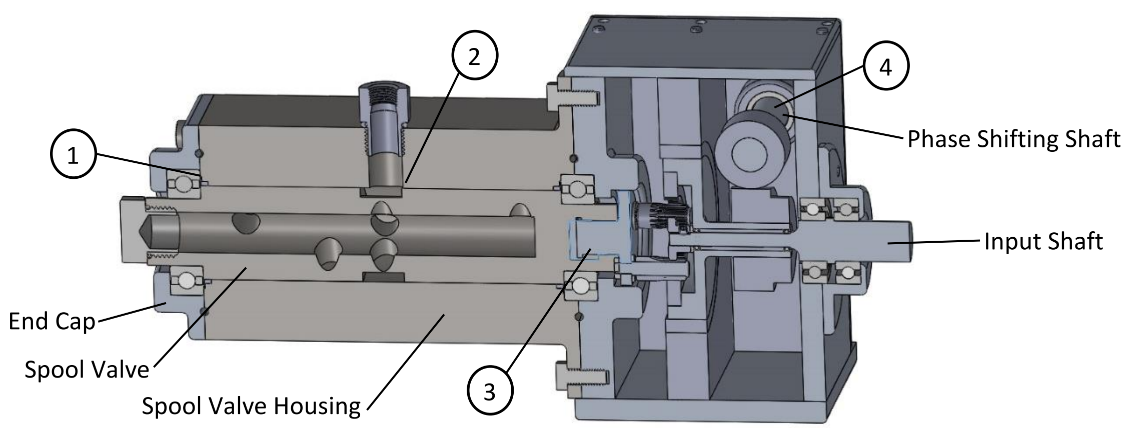

Figure 18 Rotary Valve Section View [2]

1. In M.W.H. Bergmans design the bearing sitting on the end of the spool valve provides no aligning geometry aside from what can be accomplished by the bolts holding the end cap onto the housing. This was addressed by recessing the

bearing partially into the spool valve housing. This ensures that the spool valve is perfectly aligned axially with the valve housing reducing the risk of wear between the two surfaces and ensuring minimal oil leakage between ports.

2. As can be seen in Figure 18, the spool valve intake port is not properly aligned with the housing port which happened to be the case for all the output ports as well. The spool valve geometry was therefore altered to match that of the spool housing ensuring correctly alignment when both end bearings are seated, holding the spool in place.
3. The output shaft connecting the planetary gear phase shifter to the spool valve originally consists of a keyed bore in the end of the spool valve with a matching keyed shaft on the phase shifter. Since this is a blind bottoming keyway, broaching would be difficult, therefore, the connection interface was changed to a simple three bolt flange with a small shaft used for locating, as seen in Figure 19.

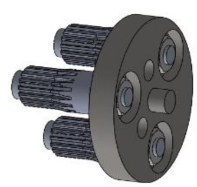

Figure 19 Planet Gear Assembly

4. The last significant alteration was the addition of a double bearing system for the phase shifting shaft, and an interface between the shaft and a stepper motor. The switch to double bearings would allow the shaft to have a shoulder that would fit between the bearings to eliminate axial movement. This is also meant to decrease the shafts deflection since it is only secured on one end.

The cylinder replacement assembly on the other hand was largely unchanged aside from a few bolt misalignments.

3.3 Mathematical Model

Hydraulic Cylinder:

The mathematical model that describes the motion of the hydraulic cylinder as stated by M. Pournazeri is:

$$m\ddot{x} = -Kx - F_{prelaod} - sign(\dot{x})F_{friction} - F_{gas} + A_p P_2 \tag{3.1}$$

where m is the combined mass of the engine valve and hydraulic piston, x (mm) is the displacement of the hydraulic cylinder piston, K (kN/mm) is the spring stiffness, A_p (mm^2) is the area of the piston in which the pressure is acting on to extend the engine valve, P_2 (kPa) is the cylinder pressure, $F_{friction}$ (kN) is the friction between the piston seal and the piston cylinder, $F_{preload}$ (kN) is the amount of preload force on the valve spring when the piston is fully retracted, and F_{gas} (kN) is the gas pressure subject to the engine valve from inside the engine cylinder which in the testing cases is zero. An illustration of the parameters in one valve cylinder model can be seen in Figure 20.

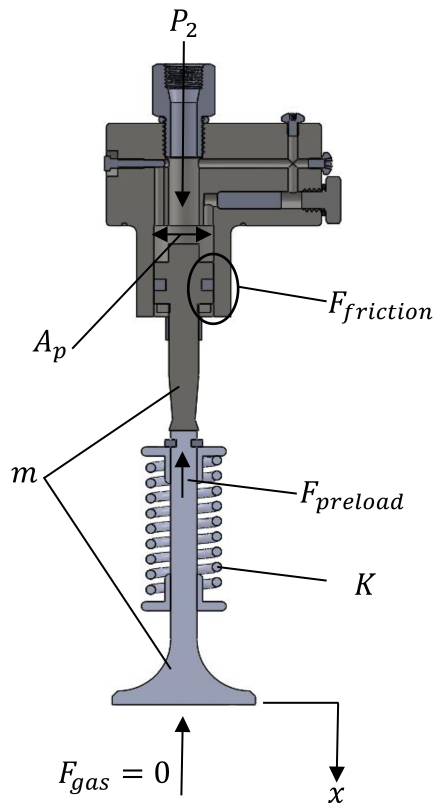

Figure 20 Hydraulic Cylinder Model

The friction force used in this model is an analytical coulomb friction based on o-ring material and deformation between the piston and housing proposed by Al-Ghathian [11] as:

$$F_{friction} = 2\pi\mu D r_o E\left(1-\frac{D-d}{4r_o}\right)\sqrt{1-\frac{(D-d)^2}{16r_o^2}} \quad (3.2)$$

where μ is the friction coefficient of the o-ring on steel, D (mm) is the piston housing diameter, r_o (mm) is the o-ring cross sectional radius, and E (kPa) is the Young's modulus of the o-ring. The general formulation that follows is based on N.D. Manring's Hydraulic Control Systems textbook which supplies the basis of the rotary valve operational fluid characteristics [12].

The cylinder pressure P_2 (kPa) of the hydraulic cylinder model can more accurately be referred to as the hydraulic cylinders pressure-rise rate. This parameter is not a constant value and subject to change based on aspects such as the flow rate through the high pressure rotary valve which is supplying the fluid, the flow rate removing the fluid through the low pressure rotary valve, the changing hydraulic cylinder volume while the piston extends or retracts, and lastly the fluid itself. This relationship is governed based on:

$$P_2 = \frac{dP}{dt} = \frac{\beta}{V_t}\left(\frac{Q_{HPRV}}{1e6} - \frac{Q_{LPRV}}{1e6} - \frac{dV}{dt}\right) = \frac{\beta}{V_i + A_p x}\left(\frac{Q_{HPRV}}{1e6} - \frac{Q_{LPRV}}{1e6} - A_p \dot{x}\right)$$

(3.3)

where β (kPa) is the bulk modulus of the hydraulic oil used in the system, which is a parameter that describes the elasticity of a fluid during changes in pressure, V_i (mm^3) is the initial volume of the hydraulic cylinder, Q_{HPRV} (L/min) is the flow rate through the high pressure rotary valve and Q_{LPRV} (L/min) is the flow rate through the low pressure rotary valve.

The hydraulic fluid bulk modulus can be parameterized as:

$$\beta = K_1\left(1 + \frac{(K_2 - 1)P_2}{K_1}\right)\left(1 + \frac{K_2 P_2}{K_1}\right) \tag{3.4}$$

where K_1 and K_2 are experimentally retrieved properties that describe individual fluids. K_2 can be considered constant between 0°C-100°C of fluid temperature, but K_1 is temperature dependent and can therefore be retrieved from a look up table. The oil used in the test bench, therefore, has a K_2 value of 5.6 with the values in Table 1 for K_1.

Table 1 Fluid Bulk Modulus Properties [12]

Temperature, °C	0	10	20	30	40	50	60	70	80	90	100
K_1, kbar	20.7	19.8	19	18	17.3	16.4	15.6	14.7	13.9	13	12.2

High & Low Pressure Rotary Valves:

The high and low pressure rotary valves operate on the principle of fluid ports aligning based on the angle of the spool shaft in relation to the outer housing. The opening area increases from 0 mm^2 to a maximum of the area of the spool port πr^2 mm^2 when the spool and housing ports are aligned, then decreases back to 0. The rotary valve opening area (A_{RV}) can be calculated through the use of the spool and housing's geometry, and the angle of the spool in relation to the house at any time. The opening area differs from that used by M. Pournazeri due to the new designs ports being circular rather than machined slots.

A method of calculating circle to circle intersection was used to approximate the actual port opening area. This calculated area requires the distance d between the housing port center point and the center point of a circle with an equivalent radius of the spool port as seen in Figure 21.

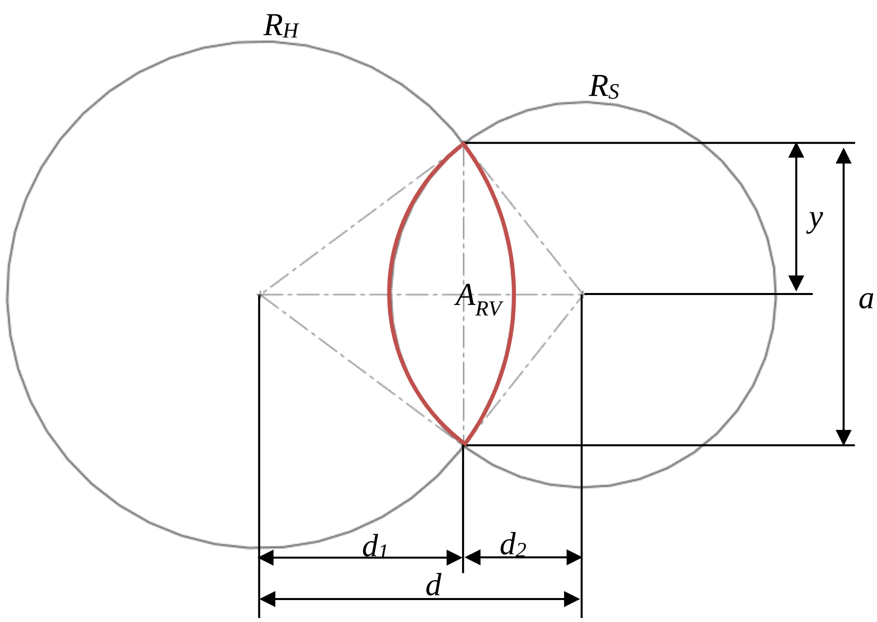

Figure 21 Circle-Circle Interception Area

To calculate the area between curves, d_1 (mm), d_2 (mm) and a (mm) must be retrieved using R_H (mm), R_S (mm), and d (mm). The triangle equations created by the intersecting circles are as follows:

$$d_1^2 + y^2 = R_H^2 \tag{3.5}$$

$$d_2^2 + y^2 = R_s^2 \Rightarrow \left(d - d_1\right)^2 + y^2 = R_S^2 \qquad (3.6)$$

Combining Eq.(3.5) and Eq.(3.6) will give the length d_1.

$$\left(d - d_1\right)^2 + R_H^2 - d_1^2 = R_S^2 \Rightarrow d_1 = \frac{d^2 + R_H^2 - R_S^2}{2d} \qquad (3.7)$$

Which can then be used to write d_2 in terms of the center distance d as well.

$$d_2 = d - d_1 \Rightarrow d - \frac{d^2 + R_H^2 - R_S^2}{2d} = \frac{d^2 - R_H^2 + R_S^2}{2d} \qquad (3.8)$$

Combining Eq.(3.7) with Eq.(3.5) results in the length y and chord length a.

$$\left(\frac{d^2 + R_H^2 - R_S^2}{2d}\right)^2 + y^2 = R_H^2 \Rightarrow y = \frac{1}{2d}\sqrt{4d^2R_H^2 - \left(d^2 + R_H^2 - R_S^2\right)^2} \qquad (3.9)$$

$$a = 2y = \frac{1}{d}\sqrt{4d^2R_H^2 - \left(d^2 + R_H^2 - R_S^2\right)^2} \qquad (3.10)$$

Calculating the area in terms of the center distance d, and radius of the two ports is then achieved in Eq.(3.11).

$$\begin{aligned}
A_{RV} &= A_{R_H} + A_{R_S} = R_H^2\cos^{-1}\left(\frac{d_1}{R_H}\right) - \frac{ad_1}{2} + R_S^2\cos^{-1}\left(\frac{d_2}{R_S}\right) - \frac{ad_2}{2} \\
&= R_H^2\cos^{-1}\left(\frac{d^2 + R_H^2 - R_S^2}{2dR_H}\right) - \left(\frac{d^2 + R_H^2 - R_S^2}{4d^2}\sqrt{4d^2R_H^2 - \left(d^2 + R_H^2 - R_S^2\right)^2}\right) \\
&+ R_S^2\cos^{-1}\left(\frac{d^2 - R_H^2 + R_S^2}{2dR_S}\right) - \left(\frac{d^2 - R_H^2 + R_S^2}{4d^2}\sqrt{4d^2R_H^2 - \left(d^2 + R_H^2 - R_S^2\right)^2}\right)
\end{aligned}$$

(3.11)

The distance between circle centers (d) is required to retrieve the flow area. The center distance is a function of the spools port angle in relation to the angle of the port in the spool housing, as well as the spool port radius. Figure 22 displays the geometry necessary to relate the spool angle (θ_S) to the center distance between ports (d).

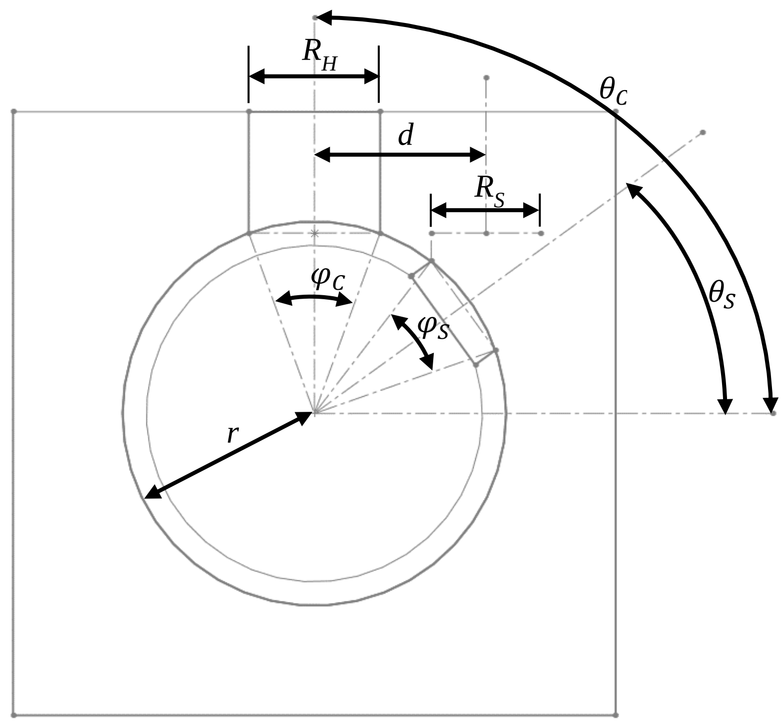

Figure 22 Rotary Valve Geometry

The center distance can be retrieved using the spool port angle to create a projection of the spool port along the housing/spool intersection plane. The distance formula will change slightly as the spool port angle θ_S (rad) bypasses the housing port angle θ_C (rad). The center distance between ports will be specified as d_1 and d_2. These will portray the center distances d_1 when $\theta_S < \theta_C$ and d_2 when $\theta_S > \theta_C$.

$$\mathrm{d}_1 = \sin\left(\theta_C - \left(\theta_S + \frac{\varphi_S}{2}\right)\right) * r + R_S \tag{3.12}$$

$$\mathrm{d}_2 = \sin\left(-\theta_C + \left(\theta_S - \frac{\varphi_S}{2}\right)\right) * r + R_S \tag{3.13}$$

Simplifying Eq. (3.11) results in the following area

$$A_{RV} = R_H^2 \cos^{-1}\left(\frac{d^2 + R_H^2 - R_S^2}{2dR_H}\right) + R_S^2 \cos^{-1}\left(\frac{d^2 - R_H^2 + R_S^2}{2dR_S}\right) - \left(\frac{1}{2}\sqrt{4d^2 R_H^2 - \left(d^2 + R_H^2 - R_S^2\right)^2}\right)$$

(3.14)

The intersecting circle area is used until the spool port is completely within the confines of the housing port area, in which case the full area of the spool port allows fluid passage. Therefore, the final governing area equation is:

$$A_{RV} = \begin{cases} 0 & \theta_S < \left(\theta_C - \frac{\varphi_C + \varphi_S}{2}\right) \\ \min\left(\pi R_s^2, \begin{array}{c} \cos^{-1}\left(\frac{d_1{}^2 + R_H^2 - R_S^2}{2d_1 R_H}\right) + R_S^2 \cos^{-1}\left(\frac{d_1{}^2 - R_H^2 + R_S^2}{2d_1 R_S}\right) \\ -\left(\frac{1}{2}\sqrt{4d_1{}^2 R_H^2 - \left(d_1{}^2 + R_H^2 - R_S^2\right)^2}\right) \end{array}\right) & \left(\theta_C + \frac{\varphi_C - \varphi_S}{2}\right) > \theta_S > \left(\theta_C - \frac{\varphi_C + \varphi_S}{2}\right) \\ \min\left(\pi R_s^2, \begin{array}{c} \cos^{-1}\left(\frac{d_2{}^2 + R_H^2 - R_S^2}{2d_2 R_H}\right) + R_S^2 \cos^{-1}\left(\frac{d_2{}^2 - R_H^2 + R_S^2}{2d_2 R_S}\right) \\ -\left(\frac{1}{2}\sqrt{4d_2{}^2 R_H^2 - \left(d_2{}^2 + R_H^2 - R_S^2\right)^2}\right) \end{array}\right) & \left(\theta_C + \frac{\varphi_C + \varphi_S}{2}\right) > \theta_S > \left(\theta_C + \frac{\varphi_C - \varphi_S}{2}\right) \\ 0 & \theta_S > \left(\theta_C + \frac{\varphi_C + \varphi_S}{2}\right) \end{cases}$$

(3.15)

where d_1 and d_2 are the center distances before and after the spool port angle passes the housing port angle, as specified in (3.12) and (3.13), respectively.

Utilizing the rotary valve flow areas, a flow rate equation can be specified in relation to the spool port angle. The flow rates through the high pressure and low pressure rotary valves calculated using the classic orifice equation based on the Bernoulli equation are:

$$Q_{HPRV} = \begin{cases} A_{HPRV} C_d \sqrt{\frac{2}{\rho}\left(P_{HPRV,in} - P_{HPRV,out}\right)} & P_{HPRV,in} > P_{HPRV,out} \; and \; A_{HPRV} \neq 0 \\ 0 & else \end{cases}$$

(3.16)

$$Q_{LPRV} = \begin{cases} A_{LPRV} C_d \sqrt{\frac{2}{\rho}\left(P_{LPRV,in} - P_{LPRV,out}\right)} & P_{LPRV,in} > P_{LPRV,out} \; and \; A_{LPRV} \neq 0 \\ 0 & else \end{cases}$$

(3.17)

where C_d is the discharge coefficient, ρ (kg/mm^3) is the fluid density, and P is the pressure entering and exiting the rotary valve. The discharge coefficient C_d is calculated as follows [12].

$$C_d = \frac{C_c}{\sqrt{1 - C_c^2 \left(\frac{A_0}{A_1}\right)^2}} \tag{3.18}$$

The discharge coefficient equation is calculated using a contraction coefficient C_d, which is derived from the geometry of the orifice the fluid is flowing through. As where A_0 (mm^2) and A_1 (mm^2) are the areas of the initial and final ports depending on which direction the fluid is traveling through the rotary valve.

The fluid density ρ also changes as a function of the fluid pressure. This change is stated as:

$$\frac{\Delta\rho}{\rho} = \exp\left(\frac{\Delta P}{\beta}\right) - 1 \tag{3.19}$$

Chapter 4 Manufacturing and Experimental System

4.1 Test Bench Design and Assembly

The test bench shown in Figure 23 was created to allow functionality testing of the variable valve timing system before implementation onto a commercial diesel engine.

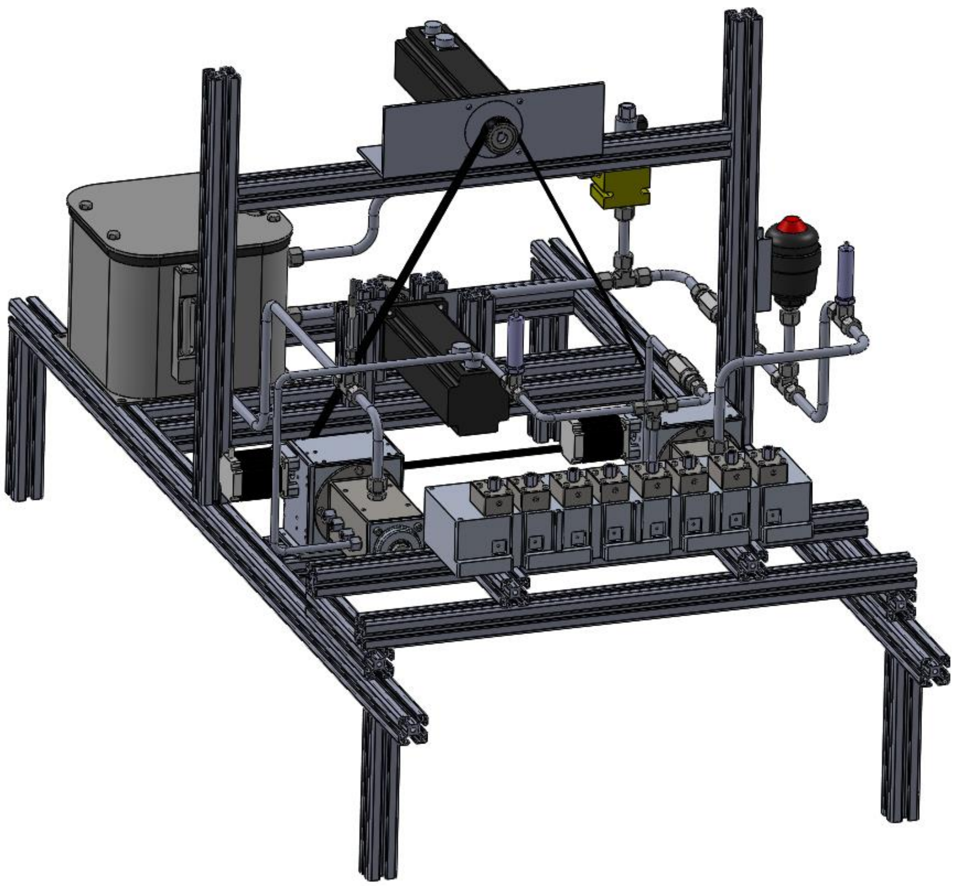

Figure 23 Test Bench Model

This allowed four aspects in particular to be validated.

1. Complete 720^{o} independent phase shifting between each of the rotary valves and the simulated engine crank shaft which in the test bench is a servo motor. What this means for the valve profile can be seen in Figure 24. The high pressure and low pressure rotary valves are able to be shifted independently to the crank angle. This

in turn allows control over the duration in which the engine poppet valve is in the open position. Alternatively, if the high pressure and low pressure ports are aligned then the hydraulic oil is not able to actuate the valve cylinder resulting in the engine valve to not open at all.

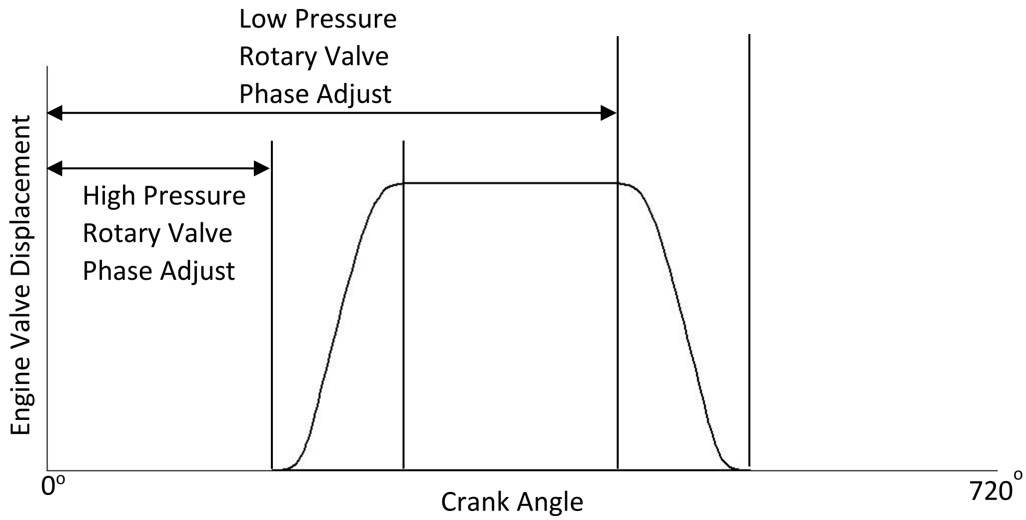

Figure 24 Valve Profile Phasing Ability

2. Change in the system pressure to control the total displacement of the valve which in turn will control the amount of the fuel/air mixture allowed to enter the engine cylinder. An increase in system pressure will result in increased pressure acting on the valve cylinder to counteract the valve spring force causing an increase in valve displacement. This change is illustrated in Figure 25.

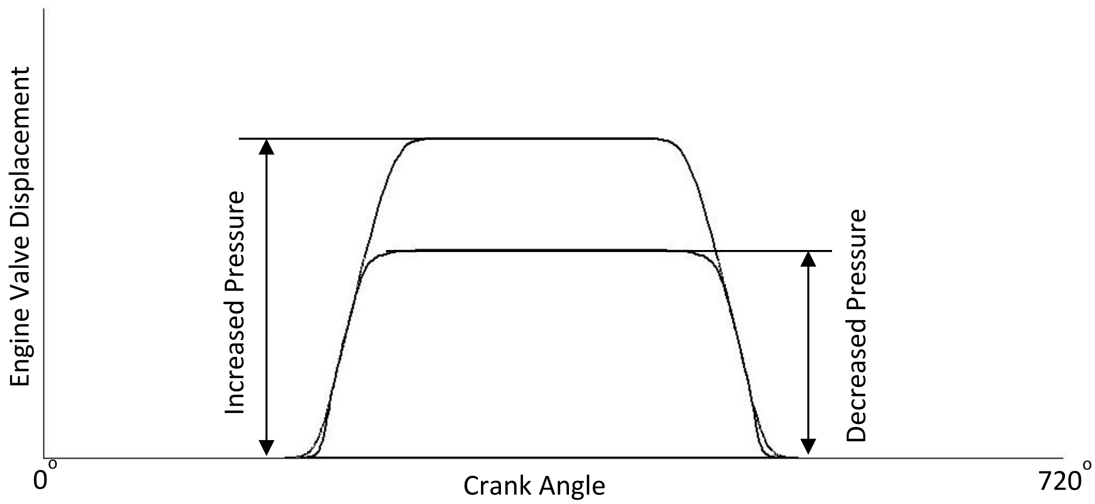

Figure 25 Valve Displacement Adjustability

3. Change in the valve cushion adjustment screw to subject the valve cylinder to hydraulic cushioning during the last 5mm of travel while the valve is closing resulting in a deceleration effect. The expected effect on the valve displacement profile is illustrated in Figure 26, where slope becomes more gradual as the valve displacement goes to 0.

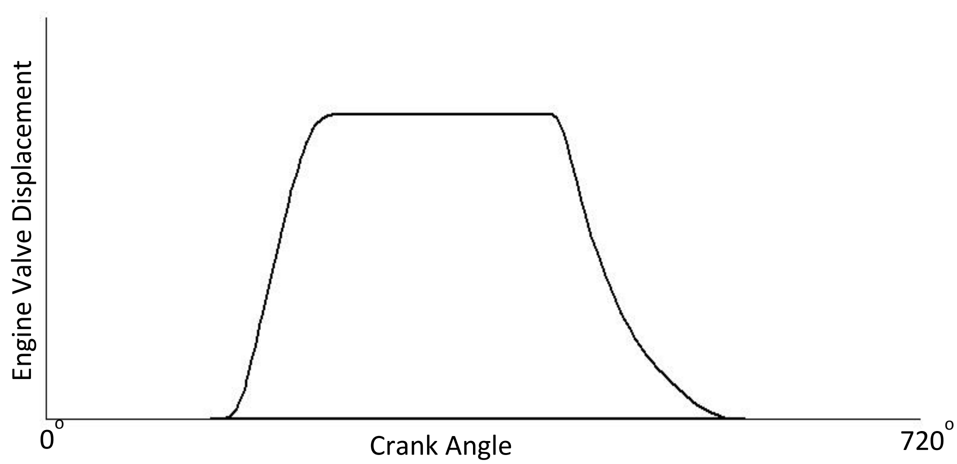

Figure 26 Valve Cushioning Functionality

4. The last aspect that must be validated during test stand operation is the robustness

of the mechanical design. During the next stage of the variable valve project the components will be implemented onto a standalone diesel engine which has a maximum engine speed of roughly 3600rpm. This means that the valves must be tested on the bench at 1800rpm for extended periods of time.

4.1.1 Hydraulic design

An overview of the hydraulic system made to operate the variable valve timing system is shown in Figure 27.

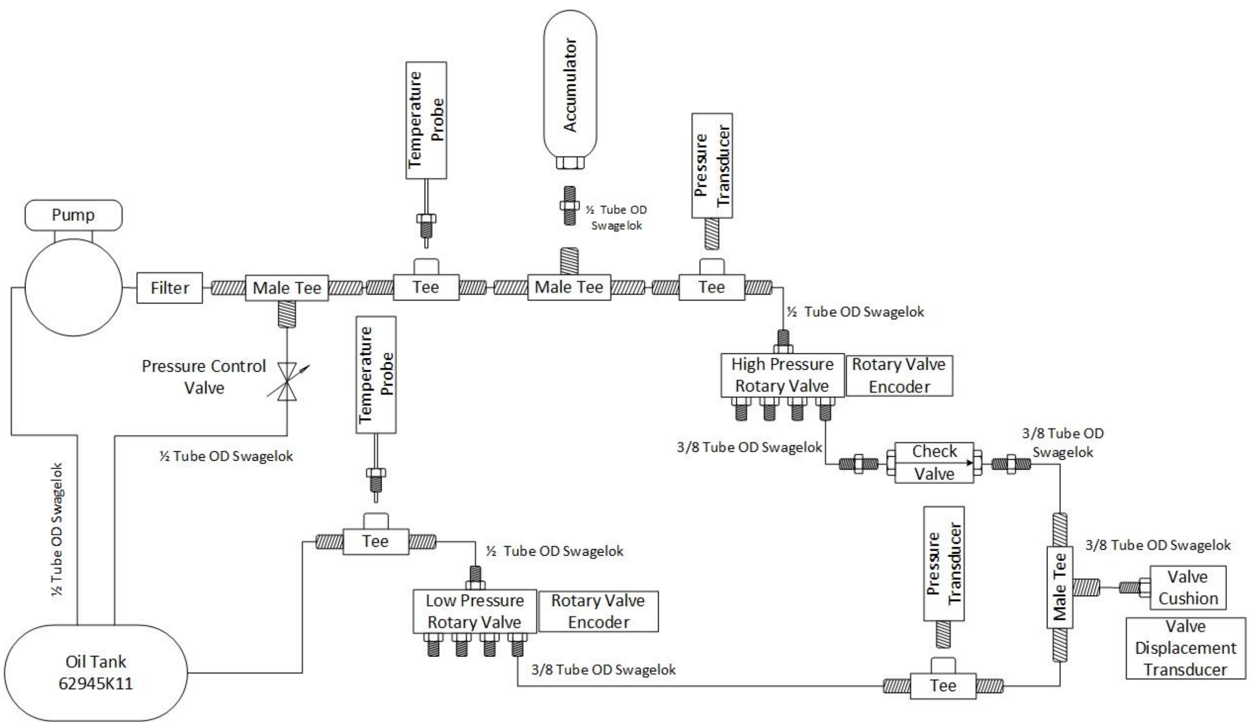

Figure 27 Hydraulic System Layout

The major components required for the proper operation of the hydraulic system are listed in Table 2.

Table 2 Hydraulic Component Parameters

Component	Notable Parameter 1	Notable Parameter 2
Pump	Max Flow Rate 5.27 gpm	Max Pressure 2,500 psi

Pressure Control Valve	Max Flow Rate 15 gpm	Max Pressure 1,000 psi
Diaphragm Accumulator	Capacity 0.0264 gal	Precharge 250 psi

The hydraulic system was designed around a maximum operating pressure of 1,000 psi which was used in M. Pournazeris thesis to achieve a valve lift height of 10mm [1]. This will govern all hydraulic components selected and act as the minimum pressure that the parts must be able withstand.

Pump Selection

Aside from the pressure requirements the pump must be capable of supplying the oil flow required by each valve cylinder to achieve the desired lift height. The maximum valve lift height required in this application is 10mm and will be used to determine the required pump flow rate. The dimensions of the valve cylinder can be seen in Figure 28.

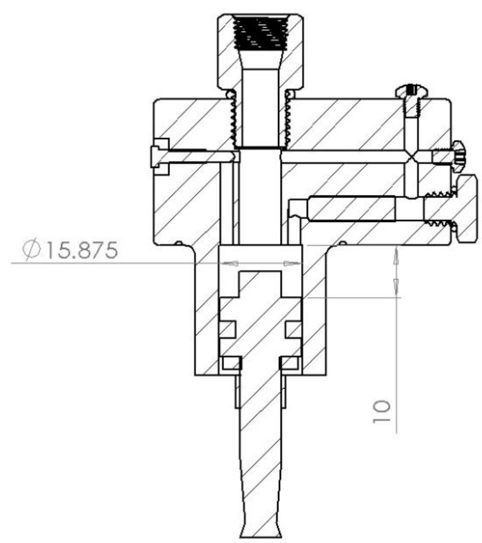

Figure 28 Valve Cylinder Chamber Dimensions

The volume required for each valve is therefore:

$$h\pi\left(\frac{d}{2}\right)^2 = 10\pi\left(\frac{15.875}{2}\right)^2 = 1979.3261mm^3 = 1.9793*10^{-3}l = 5.2288*10^{-4}gal$$

(4.1)

The time in which the spool housing port is aligned with the spool valve port is retrieved using the port geometry illustrated in Figure 29 and rotary valve speed.

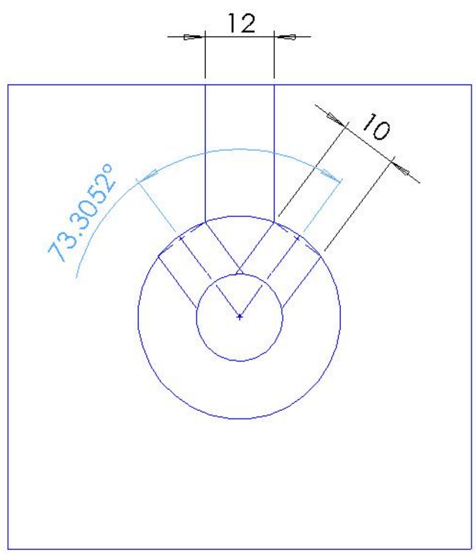

Figure 29 Rotary Valve Port Alignment

For an engine speed of 3600rpm and therefore a rotary valve speed of 1800rpm the total time that fluid can flow through the spool valve port is:

$$1800\frac{rev}{min} = 648000\frac{deg}{min}, \therefore \frac{73.3052\,deg}{648000\frac{deg}{min}} = 1.1313*10^{-4}min = 6.7875*10^{-3}sec$$

(4.2)

The flow rate required to actuate the valve cylinder 10mm at 1800rpm is therefore,

$$\frac{1.9793*10^{-3} l}{1.1313*10^{-4} min} = 17.4968 \frac{l}{min} \, or \, 4.6222 \frac{gal}{min} \tag{4.3}$$

This, however, is the flow rate if only a pump is used to run the system. If an accumulator is implemented, then there is time while none of the rotary valve ports are aligned and no oil is flowing to the valve cylinder. This time could be utilized charging an accumulator to take a lot of strain off of the pump to feed the system. To calculate the required pump flow rate while utilizing an accumulator, the ratio of ports allowing fluid flow to ports blocking fluid flow is required. This can be found simply by multiplying the degrees in which a port is aligned by the number of ports and dividing by 360 degrees.

$$\frac{73.3052^{o} * 4 \, ports}{360^{o}} = 0.8145 \tag{4.4}$$

This means a smaller pump can be utilized with an average flow rate of:

$$17.4968 \frac{l}{min} \left(0.8145\right) = 14.2511 \frac{l}{min} = 3.7648 \frac{gal}{min} \tag{4.5}$$

With this information the pump that meets these displacement requirements is an SAE-AA with a flow rate of 5.27gpm, max pressure of 2,500psi and max rpm of 4,000rpm.

Pressure Control Valve Selection

The major factors that required consideration were that the pressure control valve must be capable of diverting the entirety of the pumps fluid back to the tank, it must withstand the maximum designed system pressure, and lastly it must have some form of analog control to allow adjustment of the system pressure.

The valve selected was the ERV1-10 with a max pressure of 1000 psi and a max flow rate capability of 15gpm. This valve is proportionally controlled by a 12DC PWM signal which controls a solenoid position varying the amount of fluid allowed to pass to the oil tank. The valve operation is displayed in Figure 30 .

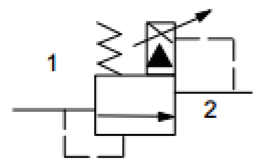

Figure 30 Proportional Pressure Relief Valve [13]

This valve also has the added ability of manually adjusting the load on the relief spring. What this means for our system is that the valve is able to be adjusted to provide a certain pressure to the system in the case of a controller malfunction or power loss to the solenoid. This provides a failsafe in which the system pressure and subsequently engine valve lift height will revert to a minimum value providing redundancy in the system.

<u>Accumulator Selection</u>

The accumulator was selected to give a minimum of 10 lift cycles for each of 4 engine valves at 1000psi before draining the accumulator of its oil supply. The amount of oil displaced by each valve can be seen in Eq.(4.6) which means the minimum accumulator capacity required is:

$$\left(1.9793*10^{-3}l\right)*4*10=7.9173*10^{-2}l=2.0915*10^{-2}gal \qquad (4.6)$$

Therefore, the 0.0264gal capacity accumulator was chosen and the precharge level was chosen as one quarter of the maximum operational system pressure.

4.1.2 Mechanical Design

The test bench was designed using aluminum t-slot extrusion as the main supporting structure for the variable valve timing test system. This allowed the structure to be ordered and assembled quickly without the need for any fabrication work. Another advantage to using t-slot extrusion on a test system is evident due to the nature of prototyping and experimental work where components and the systems layout can change drastically to solve certain problems or accommodate unseen changes that must be made. An additional decision made from a mechanical standpoint was to use hard stainless steel tubing for the hydraulic lines. This took more time initially to measure and bend the tubing to accommodate for the test bench layout, but saved time when troubleshooting and implementing additional pressure and temperature sensors.

4.1.3 Electrical Design

An overview of the electrical components utilized in the test bench can be seen in Figure 31.

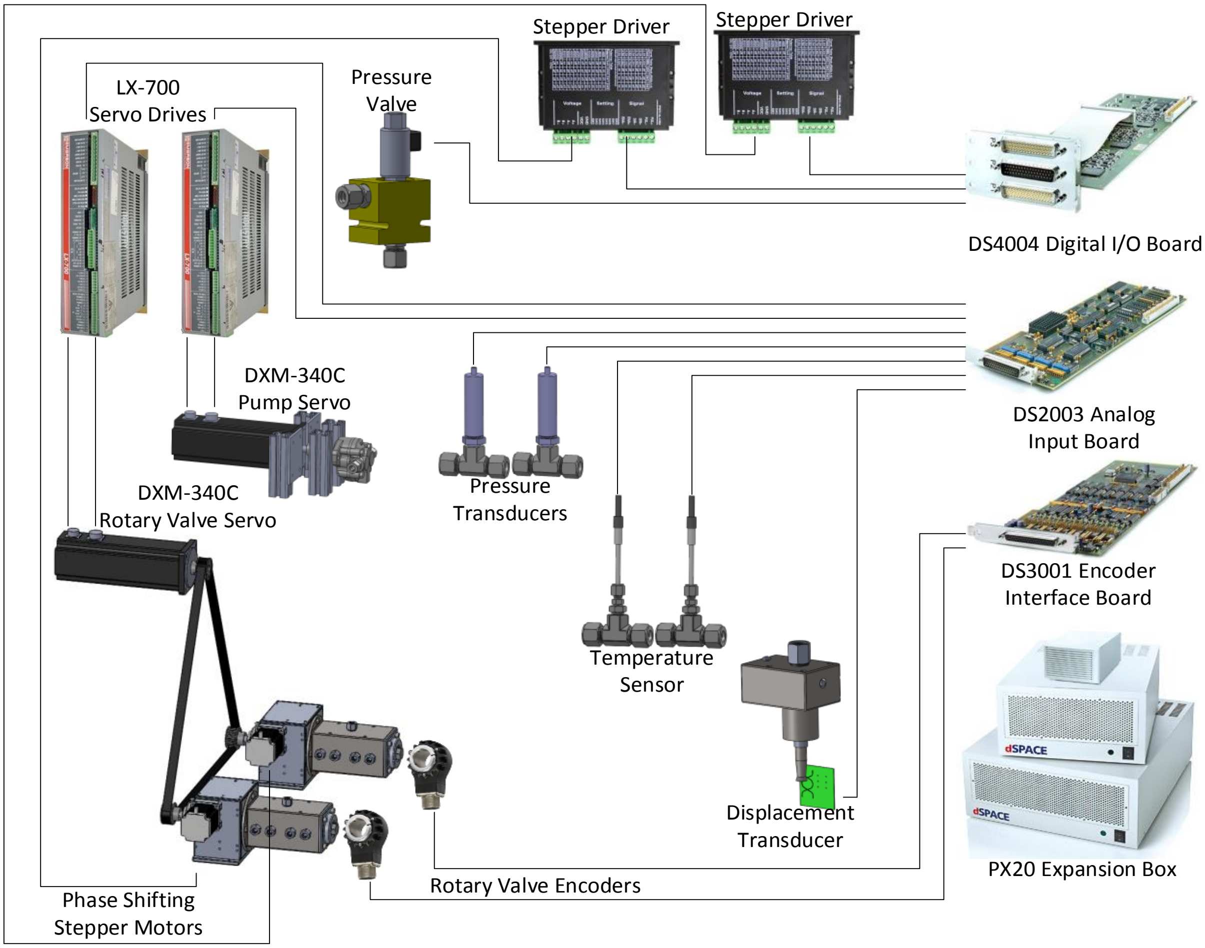

Figure 31 Test Bench Electrical Diagram

All controllable input or output devices that are required to both ensure the proper operation of the variable valve timing system and quality control over the hydraulic system can be found in Table 3.

Table 3 Electrical Inputs/Outputs

Temperature Sensor	The Sensor Connection	TCA-MS-K-1	EGT Probe Amplifier Signal Conditioner Module, 0-5VDC Analog Output

	The Sensor Connection	EGT-DP-XXX-S411-SS-N	EGT Probe - 4" Straight - 3/16" Diameter
Pressure Transducers	Transducers Direct	TD1000DDG3000003Q001M	0-3000 psi pressure transducer
Valve Displacement Transducer	KSR International	N/A	Custom made 15mm, 0-5VDC, 2kHz induction displacement sensor
Rotary Valve Encoder	Encoder Products Company	25T-25SE-1200NV1RPP-SMH	1200 cycles/rev, Quadrature A&B with index, IP50 sealed rotary encoder
Pressure Control Valve	Wajax	ERV11010.00012DG	12VDC PWM controlled pressure control valve
Pump Servo Rotary Valve Servo	Emerson	DXM-340C	240V, 6.5A, 3000RPM, 1.54Hp Servo Motor
Pump Servo Drive Rotary Valve Servo Drive	Emerson	LX-700	96-264 VAC 50/60 Hz
Phase Shifting Stepper Motor	OMC Stepperonline	23HS41-1804S	Nema 23 CNC Stepper Motor 2.4Nm(340oz.in) 1.8A
Phase Shifting Stepper Motor Drive	OMC Stepperonline	ST-6600	Bipolar Stepper Motor Driver Max 4A Current 40VDC Input 16 Subdivision

4.2 Controller Implementation

The three most prevalent controller systems in our research lab are Dspace , Beckhoff, and Motohawk. The Dspace and Beckhoff have a clear advantage over the Motohawk for this stage of the project, which is the ability to get a real time interface up and running quickly. The Motohawk would be able to fulfill the project needs, but would be better suited for a commercial product since it is more along the lines of an embedded system. The Dspace and Beckhoff, however, each offers unique benefits for the testing stage. The Beckhoff CX2000 series utilizes an embedded PC running windows embedded so a screen, keyboard, and mouse can be connected for the controller to run as a standalone unit. The programming is then done on the controller in a proprietary programming environment. The Beckhoff controller can be interfaced with Simulink, but this is a relatively recent development, therefore, Beckhoff does not have the same level of integration of Simulink libraries as other hardware. Dspace on the other hand has a commercially available set of Simulink libraries to utilize all of their available expansion cards. Since Simulink is the main development framework for the variable valve timing control, Dspace was the logical choice for its ease of compiling a Simulink program for the Dspace real time processor. This gives access to all input and output data through the Controldesk software interface illustrated in Figure 32.

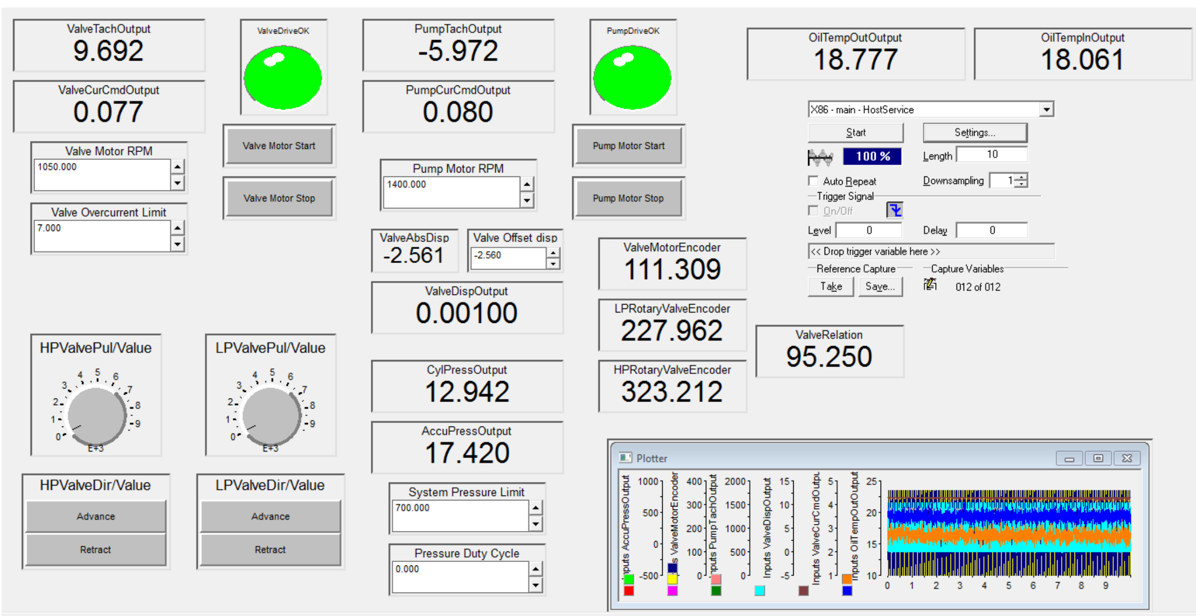

Figure 32 Controldesk Software Interface

4.3 Simulink Programming

The Simulink program used to govern the test bench is divided into two subsystems displayed in Figure 33. These are dedicated to interpreting the input signals coming from the systems sensors then controlling the outputs used to run all the pumps, rotary valves, phasing and pressure.

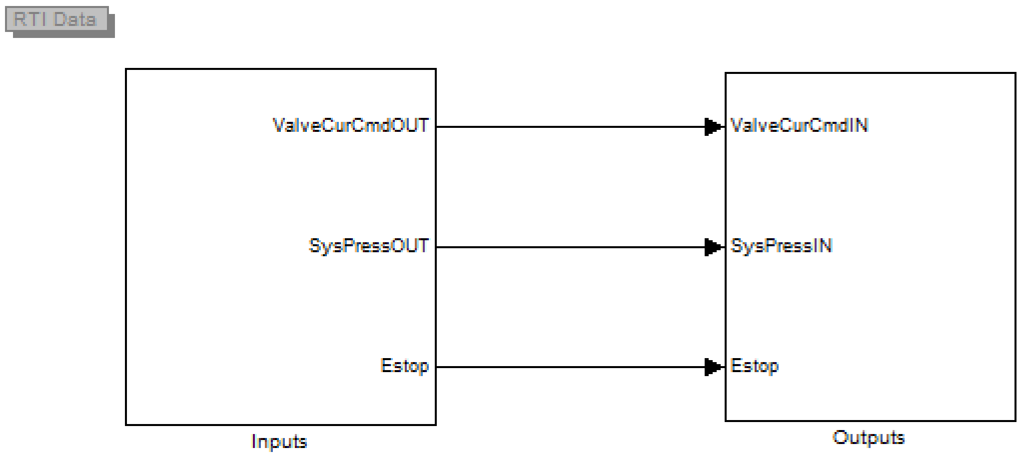

Figure 33 Top Level Simulink System

A snapshot of a few of the analog inputs can be seen in Figure 34. The inputs illustrated are the current command of the servo acting as the crank shaft of the engine to run the rotary valves, the pressure transducer at the accumulator, and the tach coming from the servo running the pump motor.

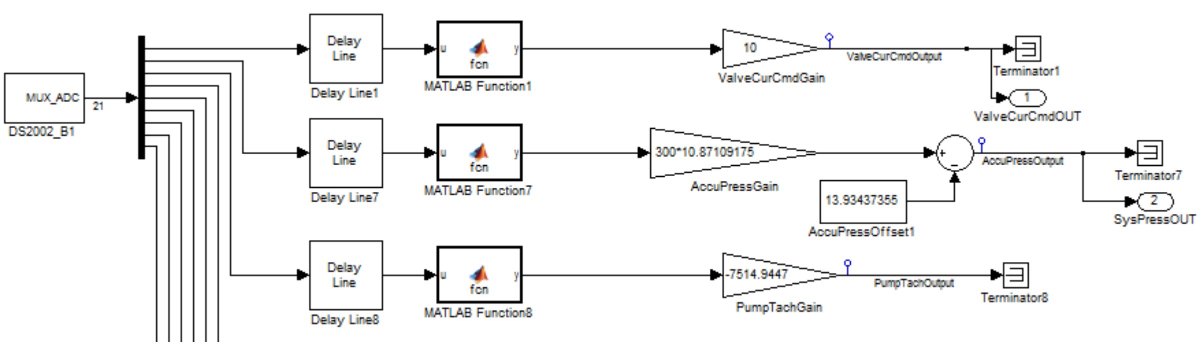

Figure 34 Analog Input Snapshot

The delay line block and matlab function blocks are used to obtain ten samples from the specific analog sensor signal then take the mean value of the samples obtained to minimize any noise present on the signal lines. This can be done because of the 3.9µs sample speed the DSpace hardware is capable of. Each of these analog inputs required calibration with some form of measurement device for validation to relate the signals to real world values such as psi or rpm. To make this conversion Simulink gains and added constant offsets are used before the signal is fed to the DSpace control desk environment for data collection. Digital (DS4004BIT) and encoder (DS3001ENC) DSpace inputs were used to retrieve the servo drives operational status along with the absolute position of the servo motor acting as the crank shaft and each of the rotary valves. The Simulink model responsible for retrieving this information can be seen in Figure 35. A matlab function block was used on each of the encoder inputs along with an offset that would allow a simple input to be used when syncing each encoder signal with a known point on the spool such as one of the fluid ports. Another piece of information that is required, is the angular relationship between the two rotary valves. This is retrieved through another matlab function that compares the two encoder positions and does some post processing to account for when each encoder rolls over from 360^{o} to 0^{o}.

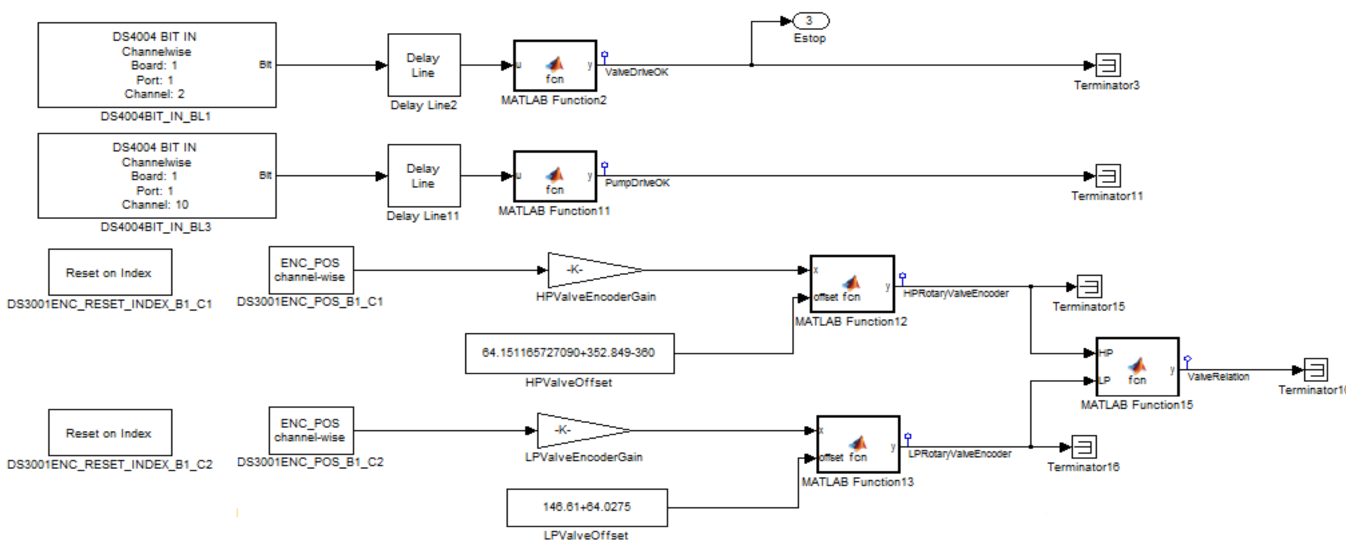

Figure 35 Digital + Encoder Inputs

In the output subsystem partially displayed in Figure 36 some analog output signals from 0-10VDC are written by the DSpace control desk environment then sent through the compiled

Simulink code such as the valve and pump motor speed which make use of the DSpace analog output Simulink blocks to control the servo motor drives. The digital outputs supported by the DS4004 card can take on several different forms. They can be a simple high/low signal, a PWM signal with controllable frequency and duty cycle, or a frequency output. Each of these types were used in the test bench application. The high/low signals were used to enable and disable motor drives and change the stepper motors rotation directions. The PWM signal was used to control the pressure in the system through a solenoid on the pressure relief valve. Lastly, the frequency output was used to control the speed of the stepper motors used for phase shifting.

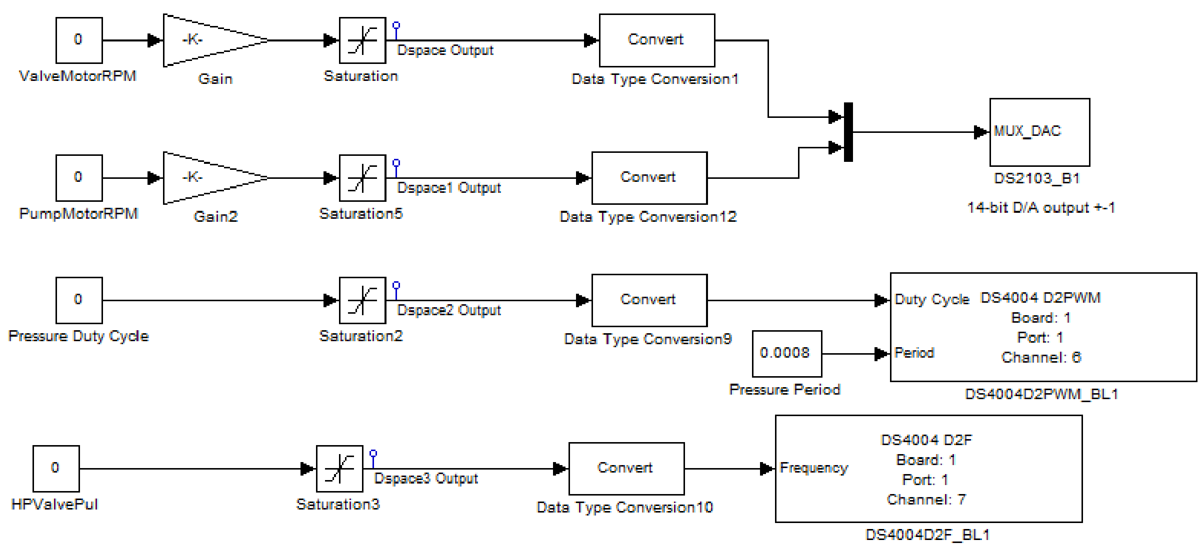

Figure 36 Analog + Digital Output

Several safety overrides are programmed into Simulink with thresholds on the pump motor current, valve motor current and accumulator pressure that are able to be set in Control Desk. These will shut down all the motors and change the system pressure set-point to zero, if any of the thresholds are passed. The safety overrides are achieved by using Simulink set reset flip flops witch control the servo drive enable pins as seen in Figure 37.

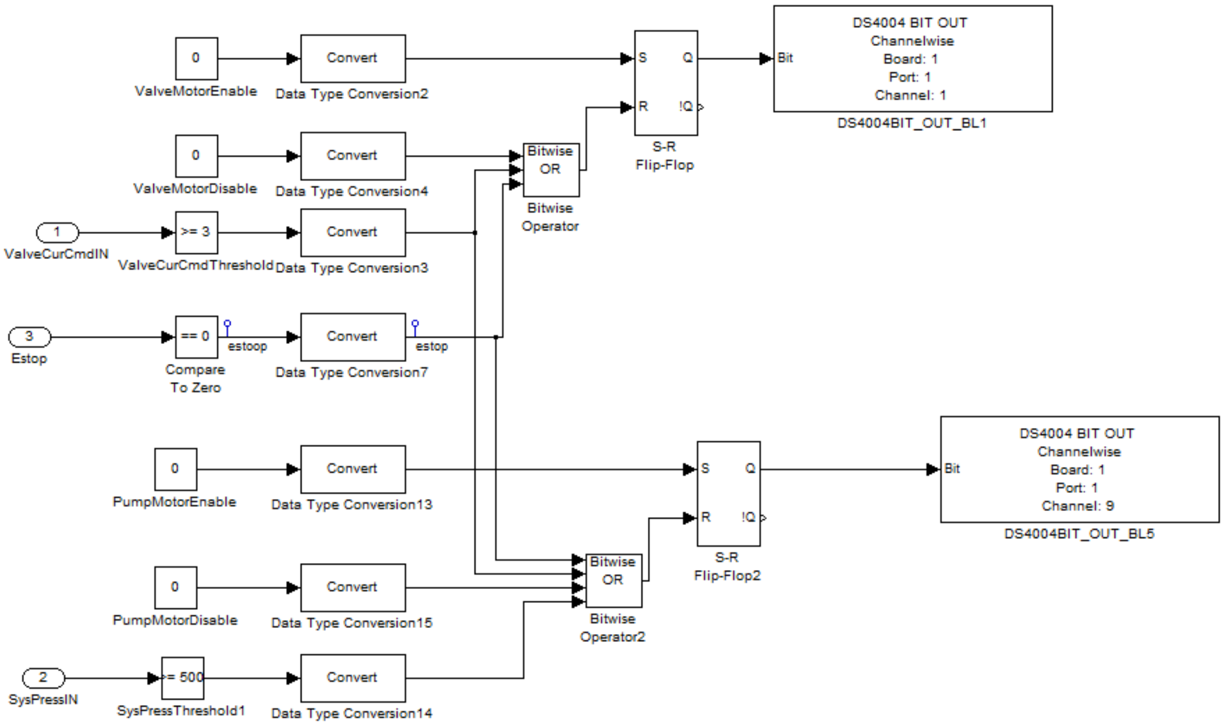

Figure 37 Safety Motor Stops

4.4 Test Bench Spring Characterization

The valve spring used from M. Pournazeris experiments was the same spring used on this test bench, however, the spring characteristics were not recorded from the previous project. Therefore, the spring rate k had to be characterized to provide insight into how much force and, therefore, how much pressure was required to achieve the desired 10mm valve lift height. This was achieved through the setup seen in Figure 38. The spring was measure at the un-sprung position then more weight was added incrementally measuring the spring’s displacement.

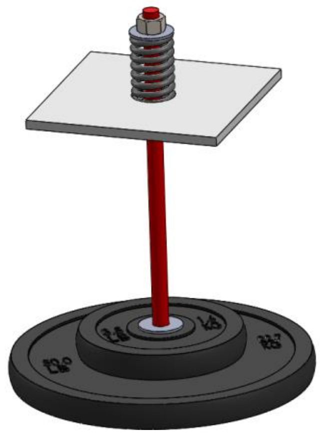

Figure 38 Spring Characterization Setup

The spring characterization results can be seen in Table 4.

Table 4 Spring Characteristics

Displacement (mm)	Weight (lb)	Force (N)	Spring stiffness (N/m)
0	0	0	0
2.3	25.002	111.13893	48.32127409
4.78	50.005	222.282306	46.50257448
6.71	75.007	333.421236	49.69019917
8.65	100.01	444.564612	51.39475283
10.64	125.012	555.703542	52.22777654
12.61	150.015	666.846918	52.88238842

With this information, the force required by our valve cylinder can be calculated to achieve 10mm of valve lift in a static environment. Since it is known that the valve cylinder piston radius is 7.9375mm (0.3125in), the piston area is calculated to be 1979.3261mm^2 (0.3068in^2). Using linear interpolations the force required to achieve 10mm of valve lift is:

$$\frac{10-8.65}{F-444.56}=\frac{10.64-8.65}{555.70-444.56}\therefore F\cong 512\,N\cong 117\,lb \qquad (4.7)$$

Considering the area of the valve cylinder piston, the required system pressure to achieve 10mm of valve lift is:

$$\frac{116.9712\,lb}{0.3068\ \text{in}^2} \cong 381\,psi \tag{4.8}$$

Note that this is the pressure required at the valve cylinder piston after pressure losses from each of the fittings, rotary valve and the sensors in the system, prior to the piston. Another consideration is that along with the pressure drops from components in the hydraulic system, the accumulator pressure set point must also account for the increased flow rate required to extend the engine valve at higher engine rpm.

4.5 Sun Shaft Failure

During intermittent testing the rotary valve system speed was being ramped up to achieve the maximum engine speed of 3600rpm. At an engine speed of 1000rpm, a failure occurred in the input shaft of the phase shifter at the location shown in Figure 39. The failure occurred exactly at the step of the shaft that had no fillet, resulting in location of high stress concentration. Upon inspection of the break the classic signs or fatigue failure could be seen, meaning an axial misalignment had occurred between the ring gear carrier and the double bearing support of the input shaft.

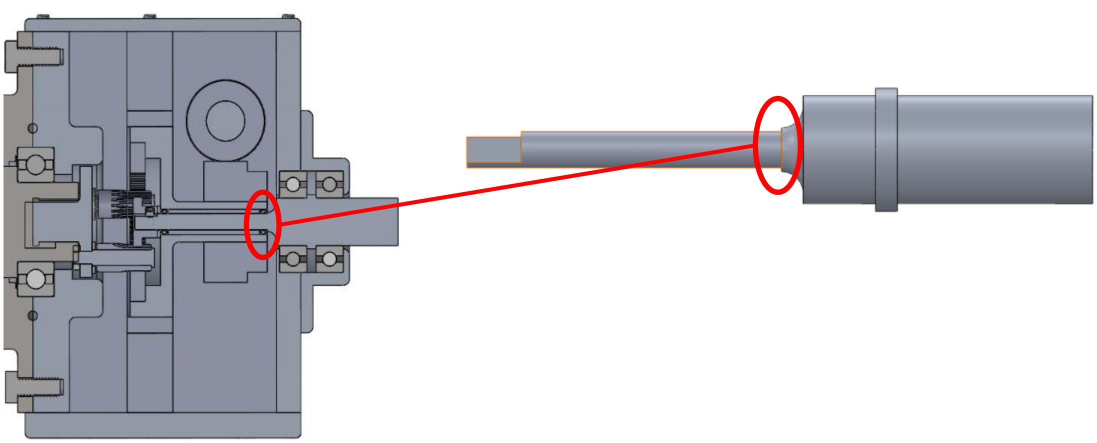

Figure 39 Input Shaft Step

This led to the conclusion that there was a flaw in the planetary gear design and this specific setup was not a viable long term solution for this project since there were too many variables in the chosen manufacturing process that could only be overcome with a more robust system. The most valid alternative to a custom made phase shifting solution was to find a complete commercial product which would serve the same purpose of providing control over the relationship between an input and output shaft.

Chapter 5 Phase Shifter Redesign

5.1 Design Change Options

The commercial solutions to achieving phase shifting are a planetary gear system with a controllable ring gear, a planetary gear reducer, or an open style differential shown respectively in Figure 40.

Figure 40 Phase Shifting Options [14] [15]

It was chosen to use the open style differential since it is both the most cost effective and lightest weight when compared to the other two solutions.

5.2 Design Selection and Modeling

The rotary valve phase shifter was therefore redesigned to utilize the open differential as a means of controlling the engine valve timing profile. The main objectives for the redesign of the phase shifter were to:

- Allow the differential phasing to be done through a worm gear and stepper motor
- Design to minimize manufacturing time
- Maximize purchasable components
- Minimize material usage by repurposing current phase shifter material
- Keep the spool valve assembly unaltered
- Ensure package size is no larger than the previous design

The resulting design with the previously mentioned constraints is illustrated in Figure 41. Where the red parts are purchased, the green parts are fabricated, and the yellow parts are purchased then altered in some way.

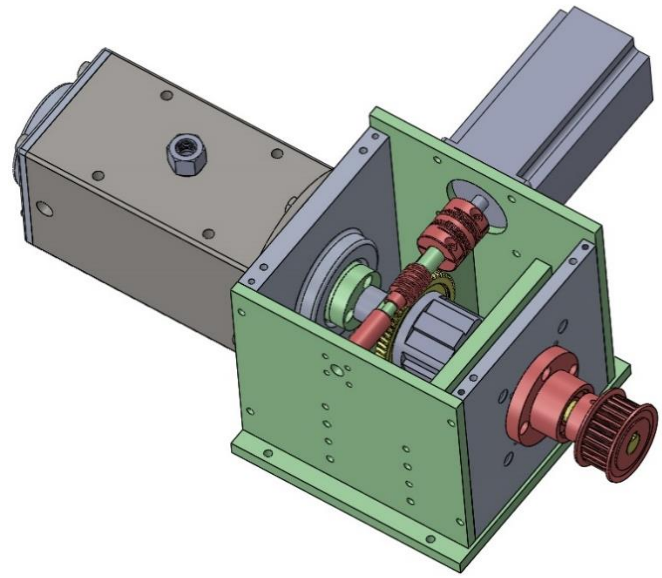

Figure 41 Phase Shifter Rev 2

To minimize downtime the differential received is used almost as is aside from changing the bevel gear to a worm gear. This ensures that no phasing can be done in the event of a power failure of the phase shifting motor due to the locking nature of worm gears. The alteration can be seen in Figure 42 as a purchased worm gear with a hub is machined into a replacement differential case cover replacing the bevel gear seen in Figure 40.

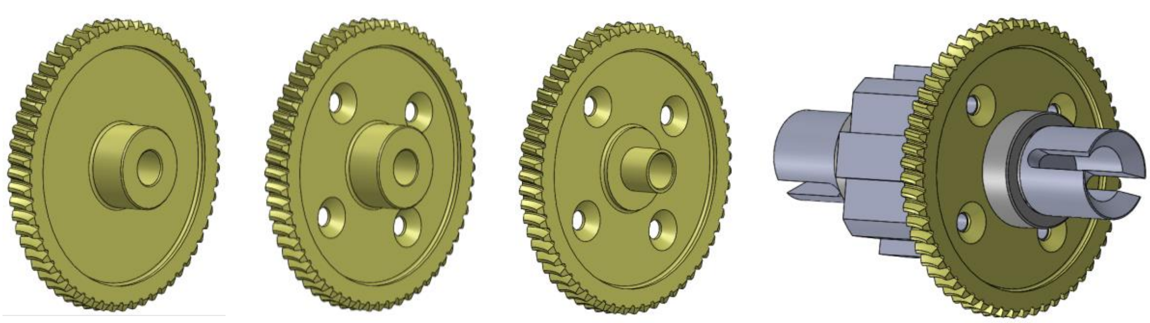

Figure 42 Differential Alterations

To interface with the differential, a simple pinned shaft was used to keep things as simple as possible, so if any problems were to occur with the differential another could easily be placed into the phase shifter with minimal down time. To keep the spool valve assembly unchanged a simple bolt adapter had to be fabricated with a pinned shaft that provides engagement with the differential illustrated in more detail in Figure 43.

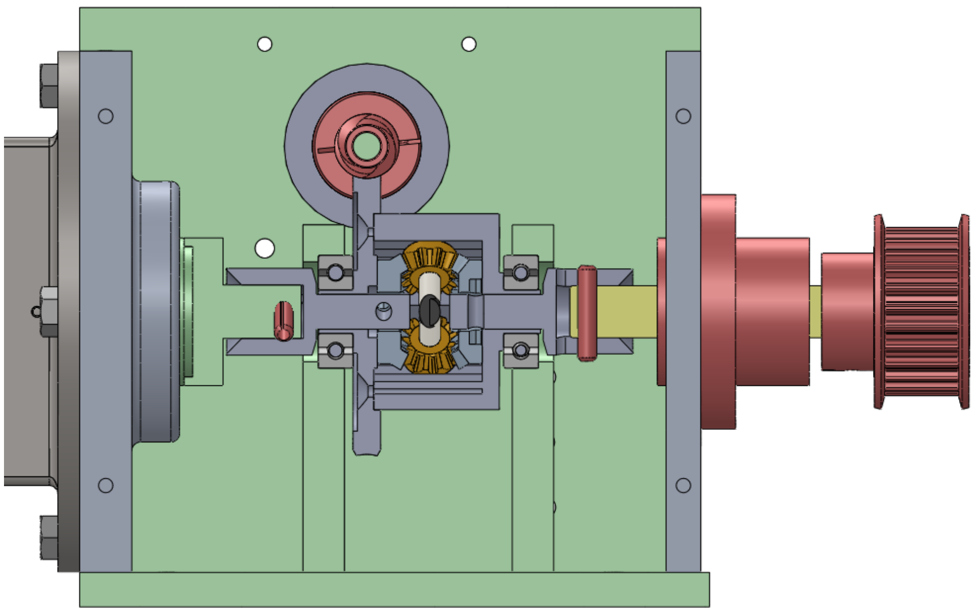

Figure 43 Phase Shifter Rev 2 Breakout

This off the shelf differential is designed to operate in much more demanding situations than our previous custom made planetary gear phase shifter and should, therefore, have no problems with operating at the required 3600rpm engine speeds for the rotary valve operation.

Chapter 6 Experimental Results

6.1 Physical interpretation of collected data

The data collected from the test bench system includes accumulator pressure, valve cylinder pressure, high and low pressure rotary valve positions, rotary valve motor position, rotary valve motor velocity, poppet valve displacement, and oil pressure before and after the rotary valves.

The system is ran at 1000 rotary valve rpm and under, which is equivalent of 2000 engine rpm while altering both the system pressure and rotary valve phasing angle to achieve different valve profiles and valve lift heights. These effects can be seen through the collection and analysis of the aforementioned data.

The variable valve system was ran at the maximum engine speed of 3600 rpm to ensure the feasibility of the newly designed phase shifter, but data collection was done at a more conservative 2000 rpm engine speed to ensure system longevity.

All of the data collected was sampled at a rate of 10000 Hz which resulted in a very high resolution for our purposes. Sampling at 10000 Hz for a total engine poppet valve lift cycle lasting 163.3052° of the rotary valve, at an engine speed of 2000 rpm results in 272 data points collected for this poppet valve cycle. At a maximum valve speed of 1800 rpm, the number of data points i [16]s reduced to 151, however, this is more than adequate to give a valid representation of the engines poppet valve motion.

A sample of this data can be seen in Figure 44. This is a 1 second interval of data collected at the highest speed recorded of 2000 rpm engine speed and the highest system pressure of 800 psi.

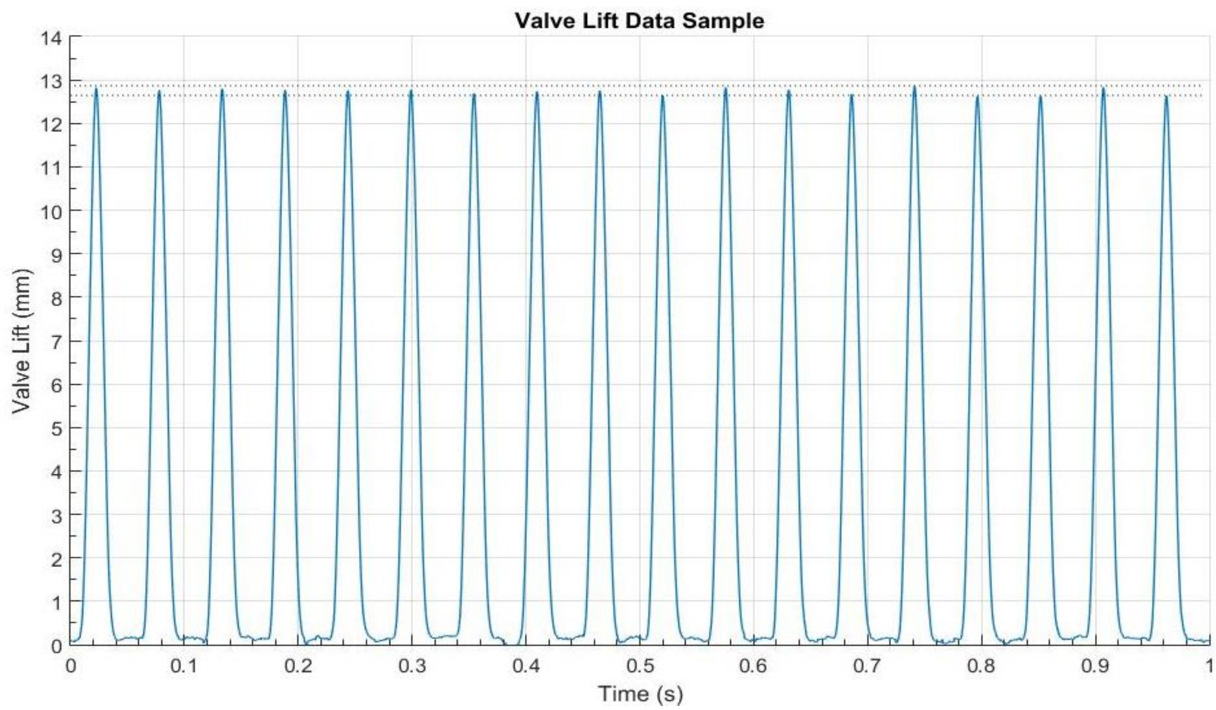

Figure 44 Valve Displacement Sample

As can be seen, the resolution of the data is very crisp from a visual standpoint with very little noise and consistently uniform valve displacement events. Another notable observation is the consistency in valve lift height. In these 18 occurrences of the engine poppet valve lift, the maximum deviation in lift height is less than 0.15 mm which demonstrates the reliability of the hydraulics based variable valve timing system. The only deviation occurs while the valve is fully seated with a minimal error of less than 0.2 mm. To demonstrate the repeatability in lift height, 80 consecutive valve lift cycles have been plotted together in Figure 45 at an engine speed of 2000 rpm and lift height of 12.75 mm.

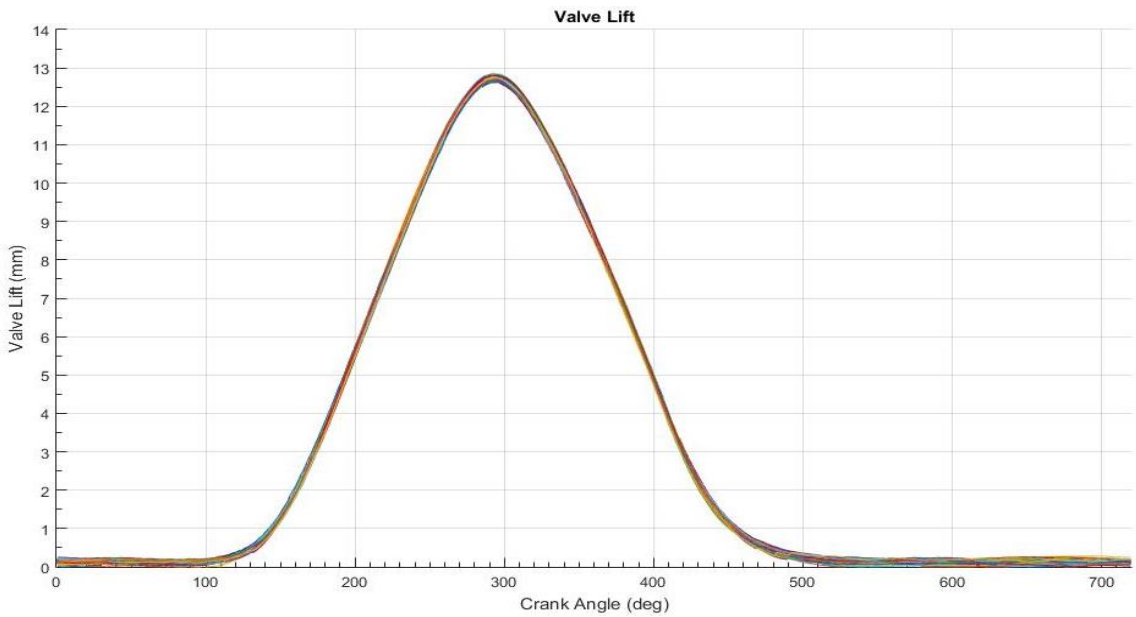

Figure 45 Valve Lift Profile for 80 Engine Cycles

As can be seen the valve profile is very consistent in terms of the system's ability to continuously achieve the desired lift height. This is further illustrated by comparing the maximum lift height for each cycle, which can be seen in Figure 46.

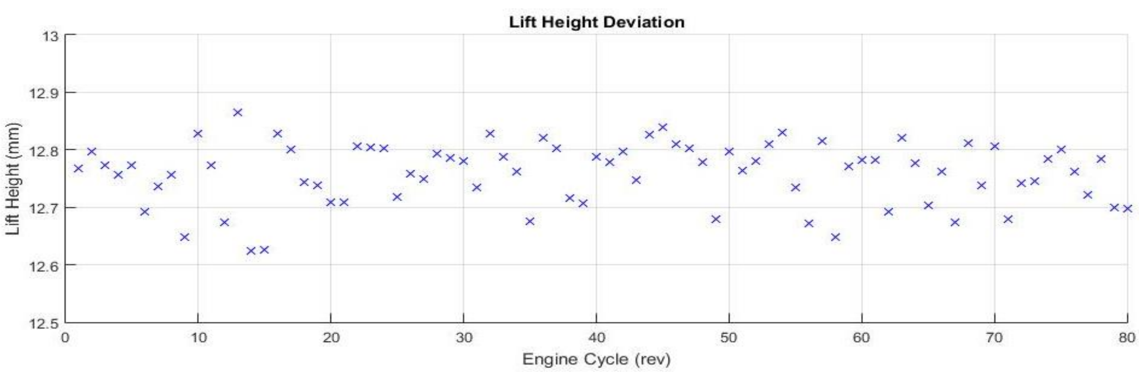

Figure 46 Lift Height Deviation

This, however, is the retrieved data at a valve speed of 1000 rpm. To get an accurate depiction of what the engine poppet valve is doing as the engine speed increases, a series of valve lift profiles have been compiled with a constant system pressure of 550psi shown in Figure 47. This deviation is acceptable considering the tolerance between cam lobes themselves are 0.13 mm. The hydraulics VVT's performance is even better when compared to the acceptable valve lift deviation between engine cylinders is 0.38 mm according to an engine building handbook [16].

Table 5 Test Parameters Variable Speed

Engine Speed	Accumulator Pressure	Valve Opening	Valve Closing	Sample Rate	Pump Speed
Variable	550 psi	75°CA	435°CA	10000 Hz	2000 rpm

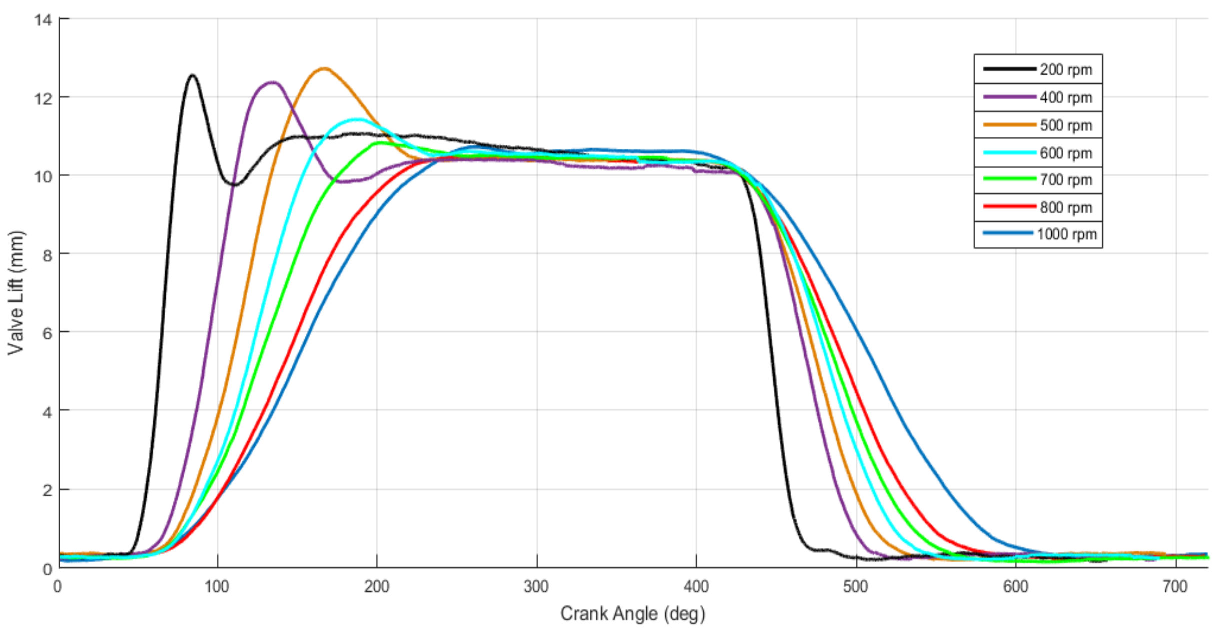

Figure 47 Speed Varying Valve Lift Profiles

Upon the initial review of the changes that occur by keeping a constant pressure/valve lift while changing the simulated engine speed, it was surprising to see that as the speed decreases the poppet valve starts to overshoot the desired valve lift height. What is also interesting is that the velocity in which the poppet valve extends also increases as engine speed decreases. Considering, however, what is occurring in the rotary valve this phenomenon can be explained. It is quite simple, in that with a decreased engine speed and subsequently a decreased rotary valve speed the time in which the ports are aligned is longer than at high engine speeds. This means that on a time scale the integral of the port area increases as engine speed decreases resulting in a higher fluid flow throughput. Therefore, this results in increased pressure rise rates and subsequently overshoot of the hydraulic cylinder used to open the engines poppet valve. A consideration that should be made for the next variable valve timing system is how to minimize this overshoot, as maintaining full control over the valve lift is paramount at all speeds.

6.2 Full Valve Control (demonstration and interpretation)

The three controllable aspects of the variable valve timing system are:

- The valve lift height by controlling the system pressure through the use of an analog controlled pressure relief valve. This allows full adjustability of the pressure acting against the poppet valve spring resulting in a controllable force equilibrium.
- The valve timing by controlling the angular relationship between the engine crank angle and the rotary valve angles. The opening and closing angle of the rotary valves are able to be infinitely adjusted in relation to the crank shaft angle. This allows the engine poppet valve to be opened and then closed at absolutely any point during the engine cycle.
- The valve cushioning by changing the cushioning adjust screw. This allows the seating velocity to be altered to ensure the engine poppet valves do not impact the cylinder with excess force which in turn ensures long engine life.

6.2.1 Valve Timing Profile Control

The valve timing profile controllability can be seen in Figure 48 as the opening angle was held constant at an arbitrary 0^{o} of the crank angle while the low pressure rotary valve relationship was altered to a relationship of 540^{o}, 360^{o} and 180^{o} of the crank angle trailing the high pressure rotary valve.

Table 6 Test Parameters Variable Closing Angle

Engine Speed	Accumulator Pressure	Valve Opening	Valve Closing	Sample Rate	Pump Speed
2000 rpm	560 psi	0^{o}CA	Variable	10000 Hz	1200 rpm

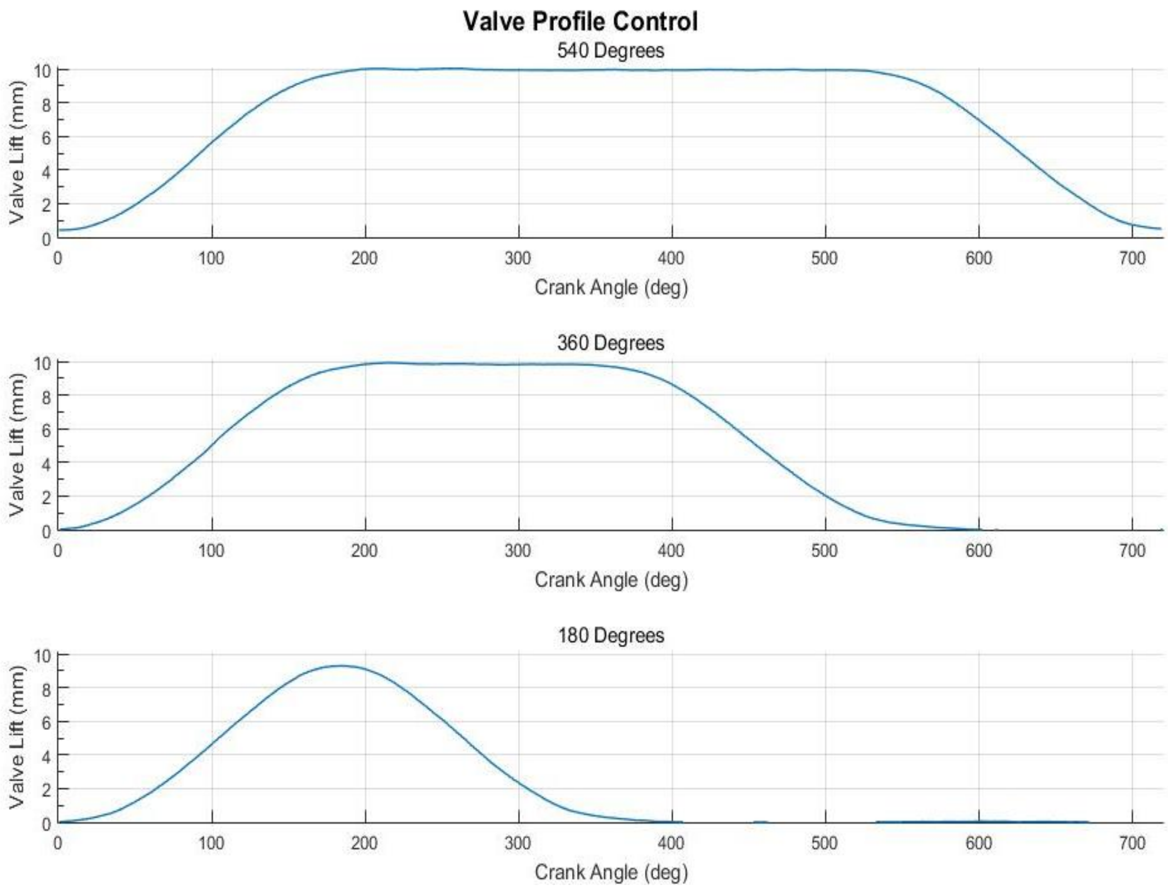

Figure 48 Valve Profile LP Control Data

As can be seen, the shift in the relationship between only the low pressure rotary valve and the crank angle results in the valve opening profile to extend for longer lengths of time resulting in the poppet valve staying open for longer, letting more air/fuel mixture into the cylinder. As there is no situation in which an engine poppet valve would be required to stay open for 540^{o}, or even 360^{o} of the engine crank angle these profiles themselves would not be used in the operation of the engine, but serve as demonstration of the fully continuous adjustment capability of the system. The clarity of this retrieved data should be noted as no overshoot and very little fluctuation is seen during the opening operation of the engine poppet valve.

Alternatively, if the low pressure rotary valve relationship is held constant while the high pressure rotary valve angle is advanced in relation to the crank angle the point in the crank angles cycle in which the valve is opened can be adjusted as seen in Figure 49.

Table 7 Test Parameters Variable Opening Angle

Engine Speed	Accumulator Pressure	Valve Opening	Valve Closing	Sample Rate	Pump Speed
2000 rpm	560 psi	Variable	525^{o}CA	10000 Hz	1200 rpm

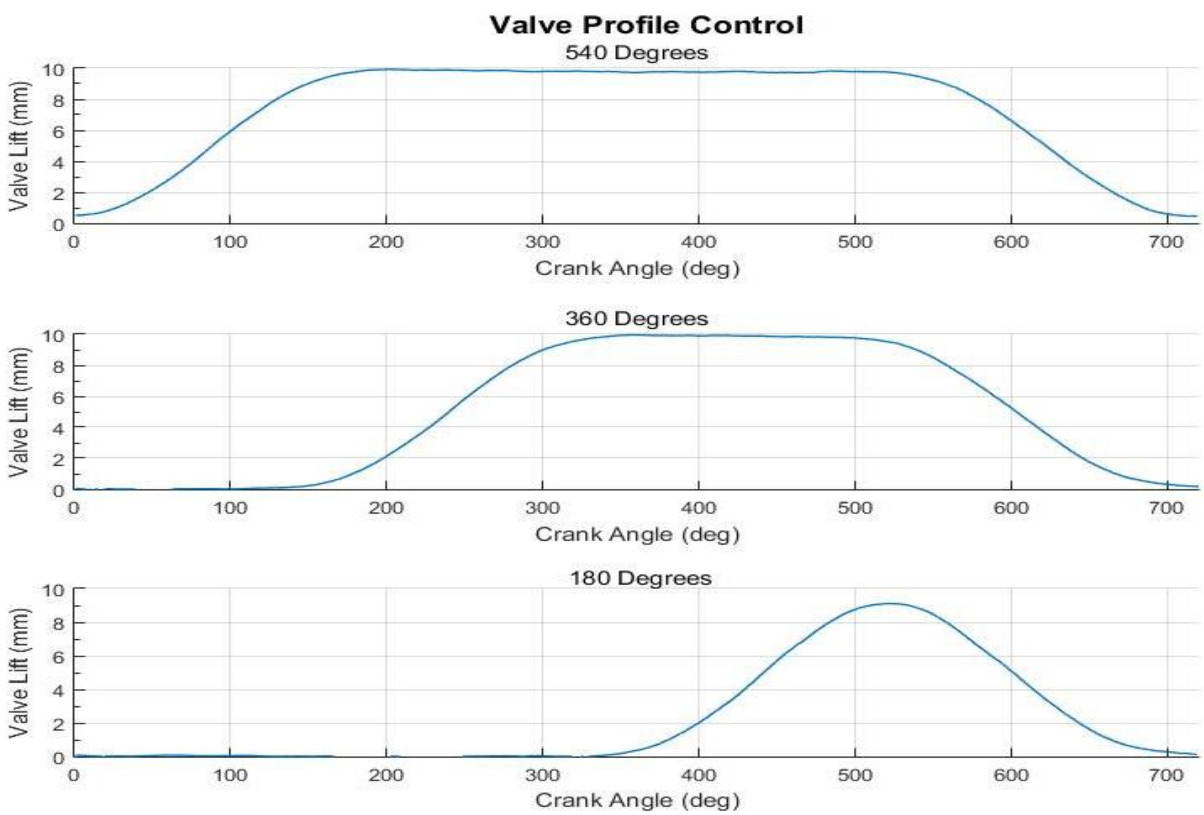

Figure 49 Valve Profile HP Control Data

Some of the benefits of having an infinitely adjustable valve profile are:

- The ability to close the intake valve later than normal resulting in the piston pushing a portion of the air/fuel mixture back into the intake manifold causing an increase in pressure in the manifold for the following intake stroke. Less force is then required during the next stroke to intake the air/fuel mixture into the cylinder. This results in up to 40% reduced pumping loses and a 24% reduction in emissions. [17]

- The ability to close the intake valve earlier resulting in a smaller volume of air/fuel mixture being taken into the valve cylinder. This is useful during times of low engine load such as idle or highway cruising. The reduction in air/fuel introduced to the cylinder results in up to a 40% reduction in pumping losses, a 7% reduction in fuel consumption, and a 24% reduction in emissions. [17]
- The ability to open the intake valve earlier resulting in overlap between the intake and exhaust valve opening cycles. This allows some exhaust to travel into the intake manifold of the engine before the exhaust valve is closed. The portion of exhaust in the intake is then introduced back into the cylinder along with the air/fuel mixture for internal exhaust gas recirculation. This results in a decrease in emissions since less exhaust is being expelled from the engine but increases the engines fuel consumption. [17]
- The ability to close the exhaust valve early or later than normal. Closing the exhaust earlier will result in a smaller amount of exhaust being recirculated in the engine, but will reduce fuel efficiency. As where closing the exhaust valve later, increases gas recirculation which is beneficial at higher engine speeds because it allows the scavenging of residual gas from the exhaust to be used to give a higher power output. [17]

6.2.2 **Valve Lift Control**

The valve lift controllability is illustrated in Figure 50, where the valve lift profile is held constant while the accumulator pressure is altered to achieve a valve lift height of 9.5mm at 575 psi, 7.8mm at 455 psi, and 5.5mm at 345 psi.

Table 8 Test Parameters Variable System Pressure

Engine Speed	Accumulator Pressure	Valve Opening	Valve Closing	Sample Rate	Pump Speed
2000 rpm	Variable	150^{o}CA	300^{o}CA	10000 Hz	1200 rpm

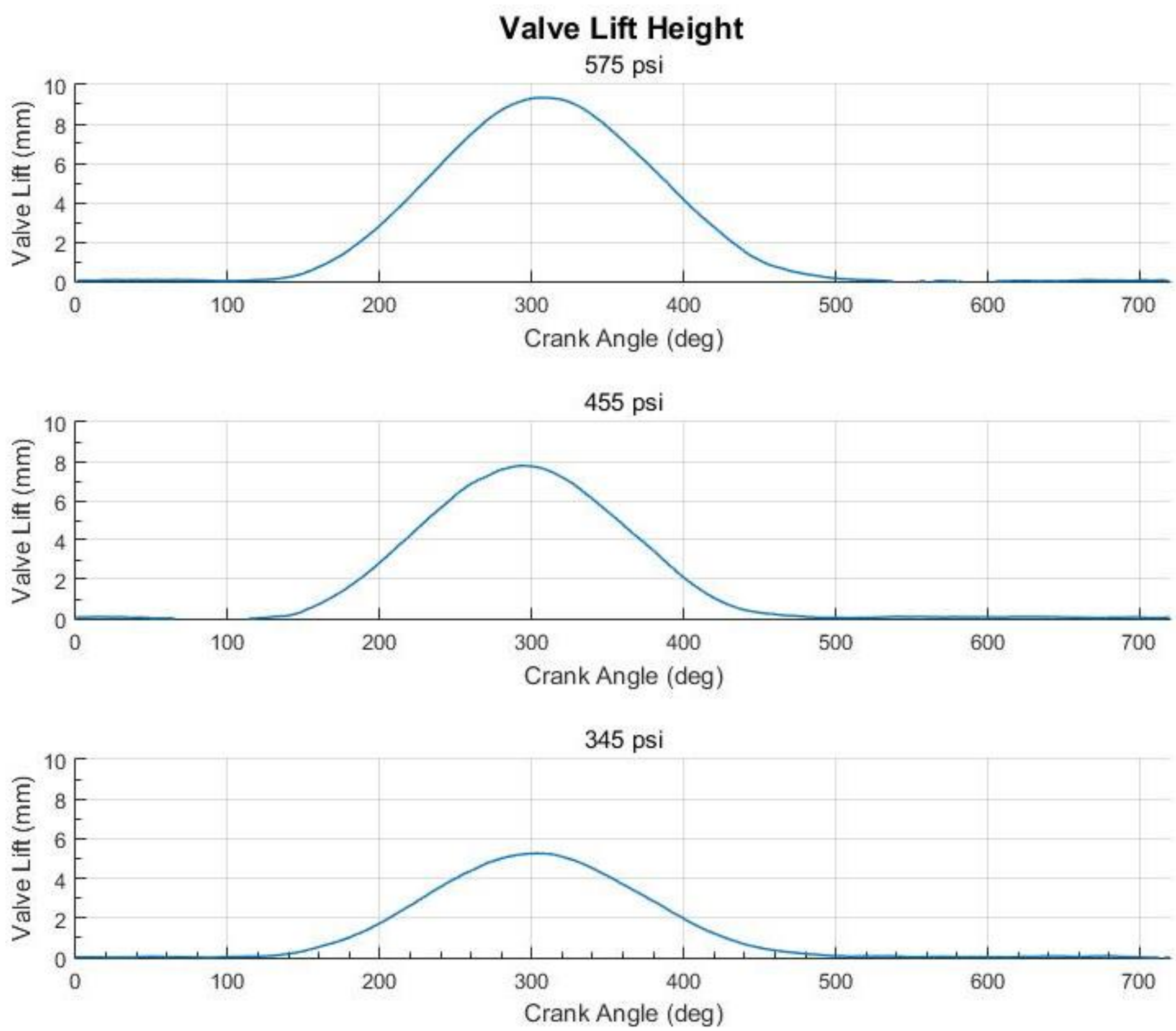

Figure 50 Valve Lift Height Data

The benefit of having continuously variable lift control is the ability to increase the valve lift height as the engines speed increases. As the engine speed increases, the amount of air/fuel required by the engine also increases, therefore, the ability to introduce more or less air/fuel as required by the engine can result in considerable performance gains. The amount of air/fuel mixture required by the engine is also dependent on the load of the engine. Therefore, during low power demand, such as engine idle or highway cruising, the engine valve lift height is able to be reduced, resulting in less emissions and lower fuel consumption. Alternatively, during acceleration the engine valve lift can be increased to allow for more air/fuel to be introduced to the cylinder and more power to be produced as a result.

6.2.3 Valve Lift Velocity

It can be noted that the time it takes to achieve the desired valve lift is roughly 150^{o} of the crank angle, which may not meet the speed required to act as a viable cam shaft replacement. To achieve a reasonable comparison two valve displacement profiles from our hydraulics based variable valve timing system running at 2000 rpm engine speed were scaled and overlaid onto a plot containing multiple different cam based valve lift profiles seen in Figure 51 [18].

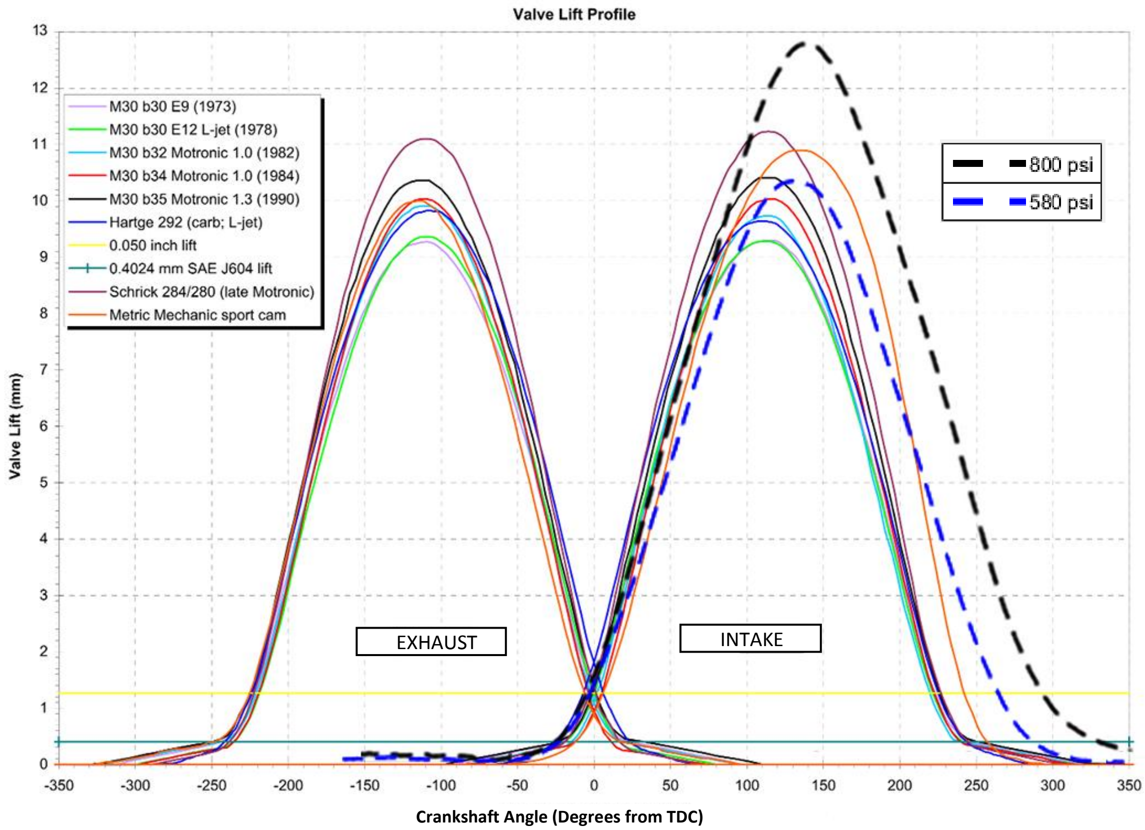

Figure 51 Valve Profile Comparison

While operating at 580 psi the poppet valve does not reach the desired valve lift height as quickly as the cam based systems, however, when the pressure is increased to 800 psi the poppet valve is very comparable to the came based systems in its lift velocity while reaching higher valve displacement. The closing in both cases does not align with the cam based systems simply because of the rotary valve offset of 90^{o} used during testing. The closing angle is capable of achieving the same results as the mechanical cams by reducing the rotary valve offset closer to 80^{o} but was not captured in this set of tests .A potential pitfall of needing to increase the systems pressure to achieve the desired valve lift velocity, is that the means of controlling the lift height is through the change in the systems pressure. This situation can be addressed in a couple ways, the first of which is by using the required system pressure to achieve the desired valve lift velocity, then using the rotary valve offset to regulate the lift height. This, however, would be situational since this strategy would only work if a quick poppet valve open then close profile was sought after as where an open and hold profile would not be possible. Another option for the next iteration of the hydraulic variable valve system is to adjust the rotary valve port size. Currently the rotary valve ports allow fluid flow for 73.3052^{o} of the rotary valve, which is 146.6104^{o} of the crank angle, which gives some insight into the time it takes for the valve to extend or retract. If the port size is reduced, the fluid would physically have less time in which it can flow into the hydraulic cylinder to extend the engine poppet valve but would require higher system pressure to ensure the desired valve lift height is reached. Alternatively, if the rotary valve port size is increased, the fluid would be allowed to flow for a greater period of time, however, there would be less restriction on the fluid flow resulting in the hydraulic cylinder pressure and the engine spring force to reach equilibrium faster potentially allowing the system to run at lower pressures. The middle ground and the best option between the rotary valve port sizing's would be to change the rotary valve ports in both the spool and the housing from simple circles back to slots similar to what M. Pournazeri used in the original rotary valve design. This would allow the crank angle in which fluid flow can occur to be controlled as well as creating a larger port area so that the flow is not restricted. Another consideration that must be made on the next iteration of the rotary valve design is that the engine in which the next design will be going onto has much softer valve springs than the one used on the test bench. Therefore, the system pressure required to extend the engine valve will be significantly lower.

6.2.4 **Valve Cushion Adjustment**

The valve displacement profile with the valve cushion adjust fully activated and fully deactivated can be seen in Figure 52.

Table 9 Test Parameters Hydraulic Cylinder Cushioning

Engine Speed	Accumulator Pressure	Valve Opening	Valve Closing	Sample Rate	Pump Speed
2000 rpm	600 psi	90°CA	270°CA	10000 Hz	1000 rpm

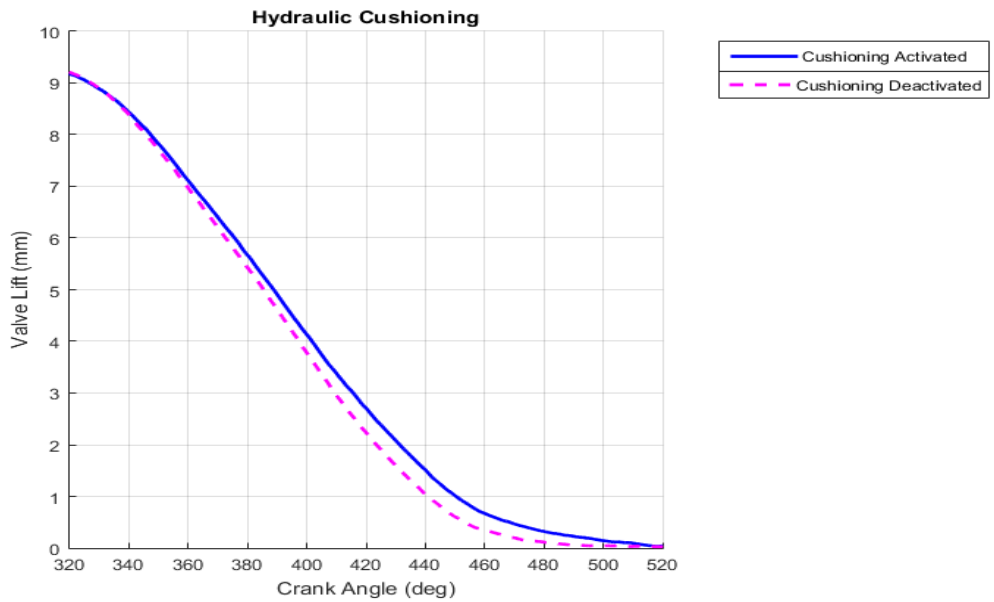

Figure 52 Valve Cushion Adjustment

The ability to control the seating speed of the engine valve is crucial to a cam-less variable valve timing system and though a difference can be seen in the valve displacement profiles a higher degree of controllability is required.

6.3 Controller Power Failure Operation

The rotary valve based valve timing system operates in such a way that any electrical power loss to the system will not have detrimental effects on the engines operation. Shown in Figure 53, the variable valve system is operating in a simulated power loss condition, in which case the engines valve functions normally, the only effect is that the valves timing cannot be altered.

Table 10 Test Parameters Power Loss Simulation

Engine Speed	Accumulator Pressure	Valve Opening	Valve Closing	Sample Rate	Pump Speed
2000 rpm	575 psi	150°CA	300°CA	10000 Hz	1200 rpm

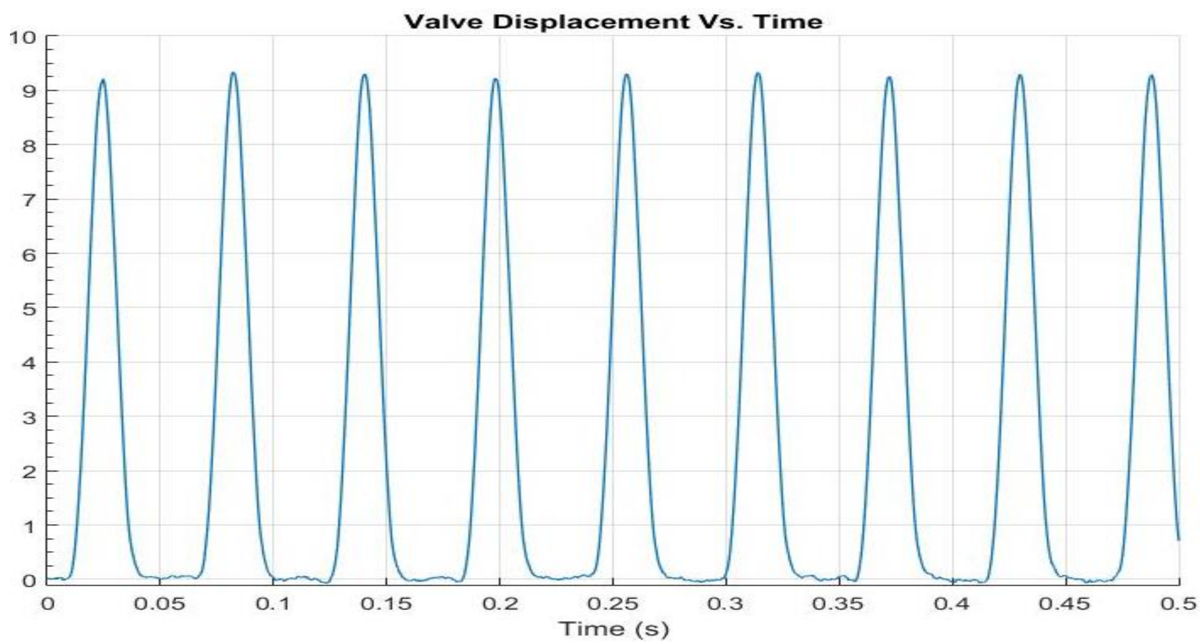

Figure 53 Power Loss Operation

6.4 Sensitivity

The previous experimental data was retrieved by using the system hydraulic pressure to overcome the engines valve spring force. During normal operation of an internal combustion engine, however, the spring is not the only force that must be overcome during normal operation. There also exists an in cylinder pressure that must be combated when first opening the engine poppet valve. Once the poppet valve is opened slightly, however, the pressure reaches equilibrium with the air pressure in the intake/exhaust manifold at which point only the valve spring force will be acting against the hydraulic cylinder. To give an idea of what effect an additional force would have on the system, a pneumatic cylinder was placed in conjunction to the valve spring on the test bench to add an additional disturbance to the poppet valves displacement profile. The pneumatic cylinder pressure was then set to 10 psi and 60 psi to show what changes would results in the poppet valve lift profile seen in Figure 54.

Table 11 Test Parameters Sensitivity

Engine Speed	Accumulator Pressure	Valve Opening	Valve Closing	Sample Rate	Pump Speed
1000 rpm	600 psi	90°CA	270°CA	10000 Hz	1000 rpm

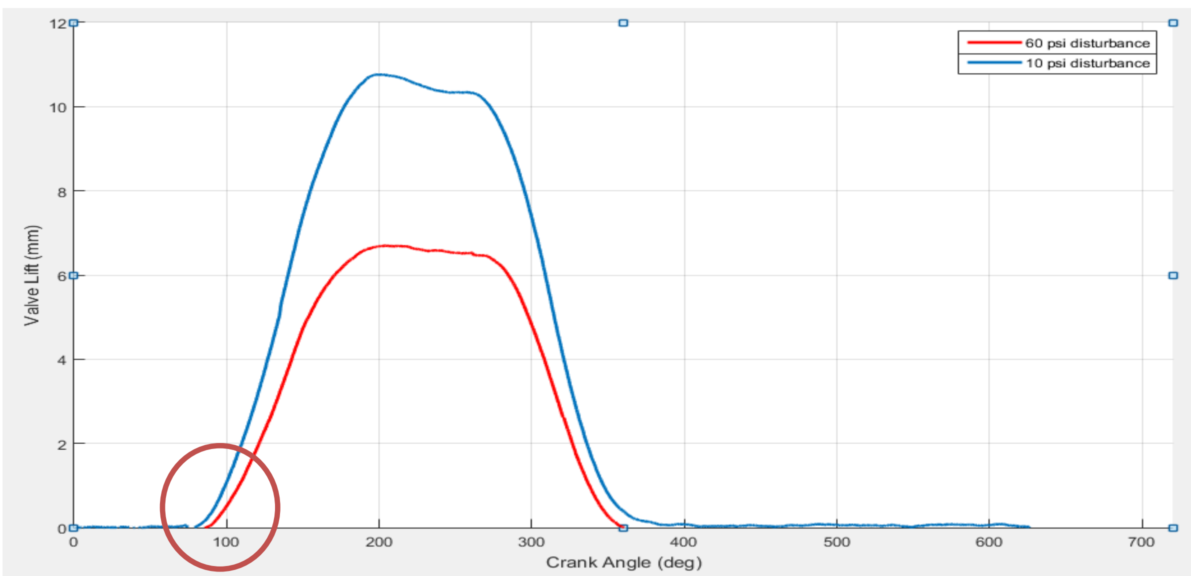

Figure 54 Simulated Sensitivity Analysis

The 60 psi pressure is approximately what can be expected for an engines in cylinder pressure. The most relevant section of the above plots is the ability for the hydraulic cylinder to overcome the initial pressure increase at the beginning of the poppet valves lift cycle. After the poppet valve has reach a height of 0.5 mm, the pressure from the engine cylinder should reach equilibrium with the engine manifold, and the valve lift profile will then resume its regular path since this additional force will effectively be gone.

Chapter 7 Conclusions and Future Work

7.1 Conclusions

Upon reviewing the current variable valve timing systems in the market each comes with its own unique set of drawbacks which include reliability, system complexity, robustness, and cost of large scale implementation. This system offers solutions to many of these drawbacks through the use of low cost components and rigidly coupled mechanical hydraulic valve design. The choice to use a mechanical rotary hydraulic valve completely eliminates the frequency response requirements that are a prevalent concern with solenoid based variable valve systems. The added benefit of using a rotary hydraulic valve system is the fully continuous control that it offers to extract the most power and fuel efficiency from commercially available engines with minimal hardware changes to the engine itself. The test bench design choices were discussed along with numerical justification when choosing the required sensor and hydraulic components while building the system.

As shown in the experimental results in Chapter 6, all of the desired functionality was achieved through the change in system pressure and altering the angular relationship between the crank and the high and low pressure rotary valves. Therefore the operating principles can be expanded upon for implementation onto a small 2 cylinder engine in the next stage of the variable valve system design.

Throughout this project the groundwork has been completed in terms of programming, mechanical, and hydraulic work which will allow the continuation of the project to focus on improving any mechanical requirements to improve system operating longevity and reliability.

7.2 Future Work

There are many aspects that can be improved upon in terms of mechanical, hydraulic, and programming design to ensure the variable valve timing systems success during real engine implementation. These will be split into their mechanical, hydraulic, and programming categories.

7.2.1 Mechanical

- Some space saving considerations can be implemented into the rotary valve design such as reducing the thickness of the spool valve housing as well as the overall dimensions of the phase shifter as there is a lot of unnecessary space in the design.
- The change from a custom made planetary gear reducer to a differential allowed the rotary valve to reach the required 1800 rpm valve speed, however, it resulted in the differential heating to above 100^{o}C in less than 5 minutes of operation. The differential could potentially be used in a production system, but would require the heating issue to be addressed through some form of active cooling. A more reliable solution would be to redesign the phase shifter to once again use a planetary gear set with a more robust sun shaft and a sturdier planet gear carrier. The planet gear carried should use hardened steel pins for the planet gears to run on along with oil impregnated bronze bushings pressed into the planet gears.
- The high and low pressure rotary valves should both be driven off of one input and configured into a “T” configuration shown in Figure 55. This will require only the one driven end of the spool valve to be sealed rather than two. The driven end of the spool valve can also have its diameter reduced to around 10mm to reduce the surface speed resulting in more rotary seal candidates.

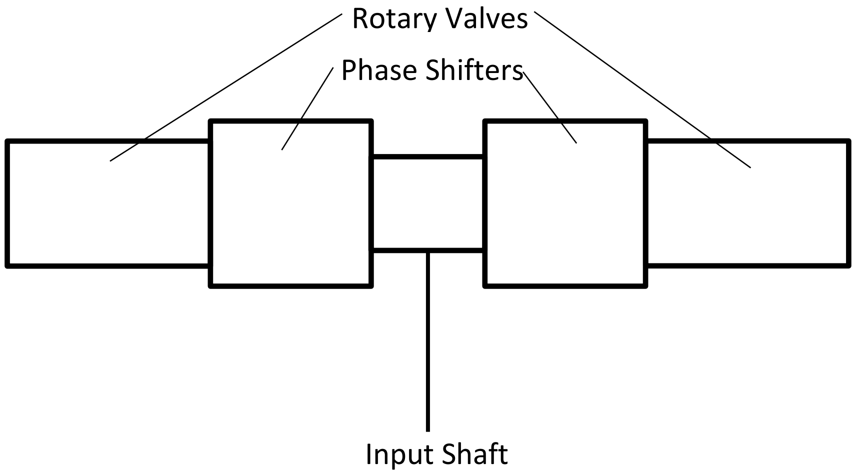

Figure 55 Single Input Rotary Valve Configuration

- Tighten tolerances on hydraulic valve cylinder by turning down the cushioning portion of the cushion assembly cylinder, then pressing a brass bushing onto the end to ensure proper tolerance between the main orifice and the cushion assembly cylinder. This will ensure proper cushioning action while the engine valve is seating.
- Recess the valve displacement sensor into the cylinder cover replacement block to protect the electronics from the harsh engine environment and ensure accurate valve displacement readings during operation.
- Revisit spool valve and spool valve housing port holes using M. Pournazeris flow calculations.
- Change spool valve housing from having four in line ports axially spaced along the spool valve to having four ports along the outside of the spool valve housing all in a plane perpendicular to the spool valves axis. This would allow the spool valve to have one outlet port to feed all four spool valve housing ports. If the four housing ports were spaced 45 degrees apart for half a the circle as seen in Figure 56 and a spool valve with two ports 180 degrees from one another then the spool valve could effectively turn at a quarter of the speed of the crank shaft.

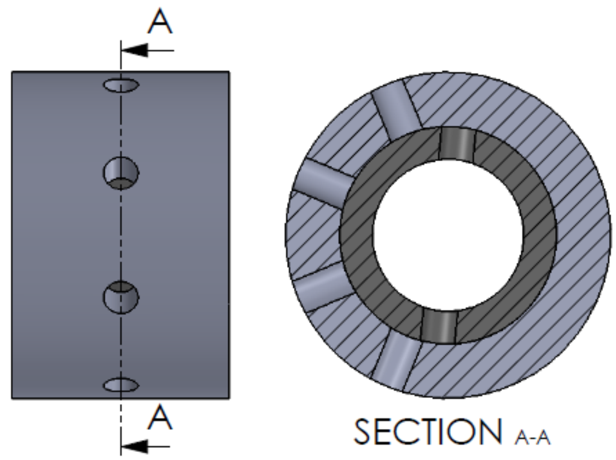

Figure 56 Spool Housing Redesign

- While operating at 1800 rpm valve speed the rotary valves heat up similar to the differential in the phase shifter. This can be addressed by ensuring proper spool valve and spool valve housing tolerances from manufacturers to ensure minimal contact between the two. The spool valve can be altered to utilize brass as the contact material. Cooling fins can be incorporated into the spool valve housing to allow for the maximum possible heat dissipation.
- A stepper motor with an encoder attachment should be implemented into the phase shifter so that a rotary encoder is not required on the spool valve.

7.2.2 Hydraulic

- The Parker Flexi Seals were the only option that met the requirements of the valve, however, the seals could not be reliably installed and, therefore, proper contact between the spool valve and the spool valve housing could not be guaranteed. Reducing the shaft diameter would allow more sealing options to meet the pressure and surface speed requirements.

- The seal used in the valve cylinder was a tight fit into the housing and could use slightly looser tolerances on the housing bore. Additionally the cylinder bore did not have a surface finish specification and, therefore, served as a poor sealing surface.

- It is recommended to lower the precharge of the hydraulic accumulator since the required system pressure to achieve 10mm of valve lift is far below the originally designed 1000psi. This would allow more reliable operation at lower pressures and more fluid delivery at higher pressures.

7.2.3 Programming

- A PID control could be implemented to ensure the angular relationship between the high pressure rotary valve setpoint and the crank angle, then another to ensure the angular relationship between the low pressure rotary valve setpoint and the crank angle.
- Control system techniques could also be employed to regulate the pressure according to the desired valve lift height rather than doing so manually.

Appendix A

Test Bench Parts List

Supplier	Part #	Product Description	Quantity	Package Size	Total Parts
McMaster Carr	9262K431	O ring 56mm ID, 3mm Wide	2	25	50
McMaster Carr	2337K611	Bearing 47mm OD, 25mm ID	4	Single	4
McMaster Carr	91290A320	Socket Head Cap Screw M6, 15mm Length	2	100	200
McMaster Carr	91280A325	Cap Screw M6, 14mm Length	2	100	200
McMaster Carr	51205K291	Pressure Plug 3/4-16 UN/UNF	2	Single	2
McMaster Carr	50925K231	Pressure Plug 1/8 NPTF	2	Single	2
McMaster Carr	50925K533	Pressure Fitting 1/8 NPTF Female, 9/16-18 UN/UNF Male	10	Single	10
McMaster Carr	50925K128	Pressure Fitting 1/8 NPTF Female, 1/8 NPTF Male	2	Single	2
McMaster Carr	92605A104	Set Screw M3, 10mm Length	1	50	50
McMaster Carr	5972K358	Bearing 32mm OD, 15mm ID	4	Single	4
McMaster Carr	90695A035	Thin Nut M4, 2.2mm High	1	100	100
McMaster Carr	5448T35	Bushing 70mm OD, 60mm ID, 50mm Length	2	Single	2
McMaster Carr	90965A160	Washer M5	1	100	100
McMaster Carr	7804K104	Bearing 8mm OD, 5mm ID	4	Single	4
McMaster Carr	6383K34	Bearing 1 1/8" OD, 1/2" ID	4	Single	4
McMaster Carr	57545K511	Worm Gear, 12 Pitch, 18 Teeth	2	Single	2
McMaster Carr	92871A213	Spacer M5 8mm OD, 5.3mm ID, 30mm Length	2	Single	2
McMaster Carr	91585A085	Dowel Pin 4mm, 14mm Length	1	50	50
McMaster Carr	57545K527	Worm 12 Pitch, 1/8"x1/16" Keyway	2	Single	2

McMaster Carr	92290A150	Socket Hear Cap Screw M4, 14mm Length	1	25	25
McMaster Carr	92290A161	Socket Hear Cap Screw M4, 18mm Length	1	25	25
McMaster Carr	91292A108	Socket Hear Cap Screw M4, 8mm Length	1	100	100
McMaster Carr	91292A027	Socket Hear Cap Screw M3, 14mm Length	1	100	100
McMaster Carr	91290A312	Socket Hear Cap Screw M6, 8mm Length	1	50	50
McMaster Carr	92605A098	Set Screw M3, 4mm Length	1	50	50
McMaster Carr	91292A140	Socket Hear Cap Screw M4, 50mm Length	2	25	50
McMaster Carr	92290A344	Socket Hear Cap Screw M6, 70mm Length	2	5	10
McMaster Carr	91292A212	Socket Hear Cap Screw M8, 80mm Length	1	10	10
McMaster Carr	9505K15	Seal 3/8" ID, 5/8" OD	8	Single	8
McMaster Carr	50925K172	Pressure Fitting 1/8 NPTF Female, 7/16-20 UN/UNF Male	8	Single	8
McMaster Carr	9262K649	O ring 29mm ID, 1.5mm Wide	1	100	100
McMaster Carr	90133A036	Sealing Washer 5/16" Screw Size	1	100	100
McMaster Carr	93627A404	Sealing Pan Head Machine Screw M3	2	10	20
McMaster Carr	92205A511	Sealing Cap Screw 1/4-20	8	Single	8
McMaster Carr	6381K425	Bushing 3/8" OD, 5/16" ID	8	Single	8
McMaster Carr	50925K464	Pressure Fitting 1/2"NPT female, 1-5/16"-12 Male UNF	2	Single	2
McMaster Carr	6296K37	Dispensing Pump 15.9gpm, 1.403 in^3/rev, 2800 rpm, 3200 psi	1	Single	1
McMaster Carr	7775K13	Quick-Opening Check Valve	2	Single	2
McMaster Carr	50925K459	Pressure Fitting 1/2"NPT female, 1-1/16"-12 Male UNF	1	Single	1
McMaster Carr	6435K14	1/2" Shaft Collar	2	Single	2

McMaster Carr	62945K11	2 Gallon Hydraulic Reservoir	1	Single	1
McMaster Carr	8682T23	Brass Through-Wall Fitting 3/8 NPT	2	Single	2
McMaster Carr	9800K56	10 Micron, 3000 psi, 22 pgm Hydraulic Oil Filter 3/4"-16	1	Single	1
McMaster Carr	9800K11	10 Micron Replacement Filter	1	Single	1
McMaster Carr	50925K415	Pressure Plug 9/16-18 UN/UNF	6	Single	6
McMaster Carr	4964K23	1/4" NPT Air Regulator for Double-Acting Cylinders	1	Single	1
McMaster Carr	4757K61	3 Port PVC Ball Valve	1	Single	1
McMaster Carr	5779K109	Push-to-Connect Fitting 1/4" NPT to 1/4" Tube	1	Single	3
McMaster Carr	3208K14	Acetal Check Valve 1/4" Tube	1	Single	1
McMaster Carr	6534K56	Industrial-Chape Hose Coupling 1/4" NPTF, 1/4" Coupling	1	Single	1
Mcmaster Carr	5967K81	5/8" Shaft 4-Bolt Square Flange Bearing	2	Single	2
Mcmaster Carr	2342K167	5/8" Shaft Diameter 1-3/8" OD Bearing	2	Single	2
Mcmaster Carr	6391K156	5/16" ID, 3/8" OD 1" long, Oil Impregnated Bronze Sleeve	8	Single	8
Mcmaster Carr	60355K502	Steel ball bearing 3/16" Shaft Diameter, 1/2" OD	4	Single	4
Mcmaster Carr	92383A746	1/4" Diameter, 7/8" length Spring Pin	1	50	50
Misumi	HTPA25S5M150-B-NUF	1/2" Bore Timing Belt Pully	2	Single	2
Misumi	HTPA25S5M150-B-N14	14mm Bore Timing Belt Pully	1	Single	1
Misumi	HTBN1500S5M-150	1500 mm circumference belt	1	Single	1
Misumi	TPBSN38-15	Idler Pulley 15mm belt width	1	Single	1
Misumi	GEABG0.5-15-8-K-5	Spur Gear 15 teeth, 5mm Shaft Bore	8	Single	8
Misumi	SCLBK5-18.5	Shoulder Bolt	6	Single	6
Misumi	GEABG0.5-30-3-B-5-KC90	Spur Gear 30 teeth, 5mm Shaft Bore	2	Single	2
Misumi	PCLBGNU5-18.5-FC6-TC2	Shoulder Bolt	6	Single	6

Misumi	MSRB8-1.0	Spacer 13mm OD, 8mm ID	8	Single	8
Swagelok	SS-810-1-6	SS Tube Fitting, 1/2" Tube OD, 3/8" Male NPT	2	Single	2
Swagelok	SS-810-1-8	SS Tube Fitting, 1/2" Tube OD, 1/2" Male NPT	2	Single	2
Swagelok	SS-810-1-12	SS Tube Fitting, 1/2" Tube OD, 3/4" Male NPT	2	Single	2
Digikey	277-1173-ND	Terminal block plug 14 position, 3.81 mm pitch	2	Single	2
Digikey	277-1172-ND	Terminal block plug 13 position, 3.81 mm pitch	2	Single	2
Digikey	277-1166-ND	Terminal block plug 7 position, 3.81 mm pitch	2	Single	2
Digikey	ED2776-ND	Terminal block plug 12 position, 5 mm pitch	2	Single	2
TTI inc	2315159	Terminal Block Interface Module DB50	8	Single	4
L-Com	CSMN50MF-5	5ft DB50 Male/Female D-Sub Cable	8	Single	4
Allied Electronics	70126169	D-Sub DB50 Female/Female Adapter	4	Single	2
OMC Stepperonline	23HS41-1804S	Nema 23 CNC Stepper Motor 2.4Nm(340oz.in) 1.8A	2	Single	2
OMC Stepperonline	ST-6600	Bipolar Stepper Motor Driver Max 4A Current 40VDC Input 16 Subdivision	2	Single	2
The Sensor Connection	TCA-MS-K-1	Single Channel EGT Probe Amplifier Signal Conditioner Module with 0 to 5 VDC Analog Output	1	Single	1
The Sensor Connection	CF-0XXX-18-NPT-IND	Compression Fitting Adapters 1/8" NPT Thread 316 SS for 3/16" (.187) Thermocouple Probe	2	Single	2
The Sensor Connection	EGT-DP-XXX-S411-SS-N	EGT Probe - Diesel - Commercial Industrial Duty - 4" Straight - 3/16" Diameter	2	Single	2
Vancouver Hobbies	57537	Nutech/Smartech 1/5 Center Differential	1	Single	1
Vancouver Hobbies	57541	Center Differential Bulkhear	1	Single	1
RC Planet	HPI1474	HPI Idle Adjustment Screw 0.21 BB	8	Single	8

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