Development of a Piezoelectric Controlled Hydraulic Actuator for a Camless Engine

by

John Steven Brader

Bachelor of Science
Boston University, 1995

Submitted in Partial Fulfillment of the

Requirements for the Degree of Master of Science in the

Department of Mechanical Engineering

College of Engineering and Information Technology

University of South Carolina

2001

Department of Mechanical Engineering Director of Thesis	Department of Mechanical Engineering Second Reader

Dean of the Graduate School

Dedication

This work is dedicated to the tireless effort and overwhelming support provided by my grandparents, Robert and Mary Brader. Throughout my entire education, they have encouraged me to succeed and shared in the joy of my accomplishments.

Acknowledgements

I would like to thank Dr. David Rocheleau for his assistance and guidance throughout the project. He has proven to be an excellent mentor and teacher, and I am thankful for his effort. I am especially appreciative of his willingness to listen and work with me to find the best solution, regardless of the challenge.

I also extend my sincere appreciation to Dr. Abdel Bayoumi for his valuable input into the project. Dr. Bayoumi continually strives to make students his priority, and he has shown me that he truly values my input and contribution to the college.

I would also like to thank Dr. Victor Giurgiutiu and Radu Pomirleanu for their help troubleshooting the electronics and piezoelectric systems. Appreciation is also extended to Richard Langdon for being the second pair of hands that I always needed.

Of greatest importance, I would like to thank my family for their support, especially my loving wife, Mary Claire. Her understanding of late nights and engineer ramblings made the project significantly easier.

Finally, I would like to thank the South Carolina Department of Commerce for their funding of this phase of the research.

Abstract

Presented within is a synopsis of the conceptual development, design, manufacture, and analysis of a piezoelectric controlled hydraulic actuator. This actuator was developed for use as a replacement for the camshaft in an internal combustion engine (ICE). Its development results in a new device; called, the camless engine (CLE).

The objective of the project was to design and manufacture a device that proved the concept of a CLE. More specifically, it is a electro/hydraulic device capable of producing engine valve displacement at typical automotive demands. The goals for maximum displacement and frequency are 10 mm and 50 Hz, respectively. In general, the unit must be capable of varying engine valve displacement and valve timing.

The system design utilized a customized piezoelectric stack and hydraulic spool valve combined with an in-house designed hydraulic amplifier. Control is facilitated by a function generator, and feedback is monitored with an oscilloscope.

The resulting system is capable of displacing an engine valve to nearly 11 mm, and frequencies up to 500 Hz have been obtained. The proof of concept can be considered successful, as it demonstrates the ability of piezoelectric control of hydraulics for use as an ICE valve actuator. Furthermore, the device has demonstrated potential areas of improvement that can be implemented in a second generation camless engine.

The overall project was divided into three phases. First, conceptual development and a review of existing technology was completed. Second, design, manufacture, and

assembly of the CLE was completed. Finally, testing and analysis was performed on the proof of concept device.

Table of Contents

List of Figures

Definitions and Abbreviations

The following definitions and abbreviations are often encountered in documents associated with internal combustion engines. Since their use is widely accepted, they will be used in this document.

BDC – Bottom Dead Center

CLE – Camless Engine

ECV – Electrohydraulic Camless Valvetrain

EGR – Exhaust Gas Recirculation

ICE – Internal Combustion Engine

IMEP – Indicated Mean Effective Pressure

TDC – Top Dead Center

VCT – Variable Camshaft Timing

VVT – Variable Valve Timing

Chapter One: Introduction

Automobile manufacturers have recognized the compromises associated with engines that are governed by the rotation of a camshaft. This rotation, the speed of which is proportional to the engine's speed, determines the timing of the engine valves. For this reason, automotive engineers must make a decision early in the design process that dictates the performance of the automobile. The engine will either have powerful performance or increased fuel economy, but with the existing technology it is difficult to achieve both simultaneously.

In response to the needs of improved engines, some manufacturers have designed mechanical devices to achieve some variable valve timing. These devices are essentially camshafts with multiple cam lobes or engines with multiple camshafts. For example, the Honda VTEC uses three lobes, low, mid, and high to create a broader power band. This does represent an increased level of sophistication, but still limits the engine timing to a few discrete changes.

The concept of variable valve timing has existed for some time. Unfortunately, the ability to achieve truly variable valve timing has eluded automotive manufacturers. Most variable timing mechanisms were created as tools for the automotive engineer. Their use was limited to the laboratory as a means of testing multiple, "virtual" cam profiles. These early camless engines allowed for the designers to choose the best cams for the engine under scrutiny, but were less than energy efficient. Furthermore, they were laboratory machines and were not capable of being mass produced or utilized in an automobile.

There have been a few attempts at developing production models of camless engines, most notably by Ford, but the use of solenoids has impeded their implementation. Using solenoids to control hydraulic fluid and ultimately the opening and closing of the engine valves introduces its own limitations. The solenoids consume considerable energy and are a binary control device – they are either on or off. Therefore the hydraulic fluid, controlled by the solenoids, is either flowing or blocked. This design allows for some variance of valve timing, but is still limited by the response capabilities of the solenoids. Furthermore, it cannot directly address valve velocity or displacement changes.

The development of the camless engine at the University of South Carolina overcomes these limitations through the use of piezoelectric stacks, a spool valve, and a hydraulic amplifier instead of solenoids. This combination results in a device capable of nearly infinitely variable valve timing, altered valve displacement, and controllable valve velocity.

Engine valve actuation is achieved through the following procedure. An electric impulse from the control hardware will cause the piezoelectric stack to expand. This linear expansion will be transferred into movement of a hydraulic spool valve. The slight movement of the spool valve will divert hydraulic fluid and pressure to one side of a hydraulic amplifier. The sudden increase of pressure in the hydraulic amplifier will be transmitted into linear motion by means of a piston. The movement of the piston acts as the actuator and is directly attached to an engine valve.

The system outlined above is required to overcome the displacement limitations of the piezoelectric stack, while exploiting its efficiency and ease of accurate control. The piezoelectric will offer the needed response for precise rapid changes in direction, but it cannot deliver the force over the required displacement needed for use as an engine valve actuator. Therefore, hydraulics are introduced as a proven technology, capable of actuating the engine valves.

It is this combination of hydraulics and piezoelectric stacks that constitutes a leap in automotive engine technology. A working prototype to actuate a single valve has been completed, and testing has proven that the system is a viable alternative to a camshaft.

Testing continues, but to date, the system has run at frequencies well in excess of any automotive engine. Although the original objective of the project was to develop a system that could actuate an engine valve at 50 Hz, this system has reached 500 Hz. The 50 Hz target was selected because it is the equivalent of an engine speed of 6000 rpm.

The second aim of the project was to overcome the piezoelectric stacks limited displacement. The stacks used currently have a maximum displacement of 30 μm, but engine valve displacement must approach 10 mm. This task is accomplished through two distinct multipliers. First, the movement of the stack is multiplied by a 5:1 solid hinge and lever. This creates 150 μm of movement and is sufficient to actuate the hydraulic spool valve. The slight movement of the spool valve redirects hydraulic fluid, pressurized at 50 bar, to either the top or bottom of a piston. This sudden increase of pressure on the piston's surface causes it to displace the engine valve. The piston essentially acts as a hydraulic amplifier.

Variable valve timing is achieved by varying the input voltage signal to the piezoelectric stacks. This variance alters the speed of response and deflection of the stacks. Therefore, the movement of the spool valve is varied and alters the flow of hydraulic fluid. It is this combination that allows for the independent control of valves, their displacement, and their opening and closing velocity.

It is the ability to vary valve timing that will provide tremendous improvements to the next generation of internal combustion engines. An engine will be capable of providing increased power when needed, increased fuel efficiency when allowable, and overall reduced emissions. For example, when entering onto a busy expressway, the onboard computer will sense the need for greater power. This results in valve timing changes to alter the overlap between intake and exhaust valves. Doing so will momentarily sacrifice efficiency for power. Then, once the automobile is cruising on the expressway, the computer will alter the timing again to reduce power and increase fuel efficiency. Furthermore, the timing can be optimized for a more complete burn; therefore the engine will produce fewer emissions. Fuel economy can further be increased by shutting down unneeded cylinders. When an automobile is cruising at a constant speed, it does not require all cylinders to be operational. With this newly developed piezoelectric controlled technology, complete cylinders can be removed from the timing cycle.

The overall results of a complete camless engine will provide the consumer with a vehicle that performs to expectations, but facilitates increased fuel economy. This combination is essential, since evidence shows consumers are not prepared to compromise on performance, while at the same time fuel prices continue to escalate.

Chapter Two: Phase I

2.1 Conceptual Development

The University of South Carolina, Department of Mechanical Engineering was approached in 2000 to consider the development of a device that would be capable of replacing the camshaft in an internal combustion engine. The original idea was to use hydraulics as a means of actuating the intake and exhaust valves in an ICE. This was not the first attempt at producing such a device; however, the proposal to use the developing technology of piezoelectric stacks provided the project's distinctiveness. It was proposed that piezoelectric stacks could control the movement, either directly or indirectly, of an engine valve. Due to the limited displacement of existing piezoelectric stacks, it was decided that they would not be used to directly actuate an engine valve for this phase of the project.

Instead of direct actuation, the concept was to use piezoelectric stacks to provide the displacement of a hydraulic spool valve. The movement of the spool valve would control the flow of hydraulic fluid. To utilize the hydraulic fluid flow from the spool valve, a hydraulic amplifier would be required. This would create the needed force and displacement for actuating an ICE valve.

The original anticipation of design requirements presented during the conceptual development phase were stated as follows.

- ICE valve travel requires 8 mm. Design the system for 10 mm.
- Forces encountered will be due to internal pressure within the ICE cylinder and from the valve closure spring. Design for 8 bar acting on a valve head diameter of 28 mm and a spring rate of 35 N/mm.
- ICE speed of 6000 rpm. This equates to valve actuation of 50 Hz.
- Develop control for the system that can vary valve displacement velocity and timing.

As the ICE valve opens, the forces due to pressure reduce dramatically while the spring force increases linearly. This is shown pictorially in Figure 1.

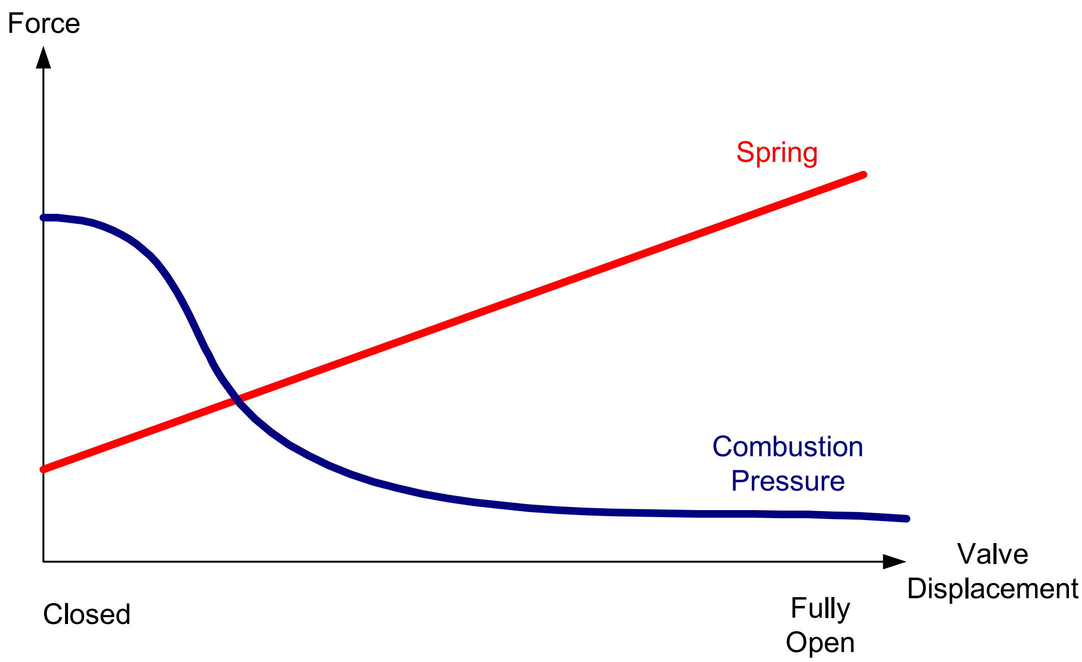

Figure 1 Resistive Engine Forces vs. Valve Displacement

The spring is designed to close the ICE valve when no force is being applied to open it. This is similar to existing engine valves, but ultimately may prove to be an obstacle to overcome. Even during the conceptual development, the replacement of spring-return with hydraulic-return was discussed. Figure 2 shows a schematic of the engine valve with a spring return. This is similar to the valve used on the test rig.

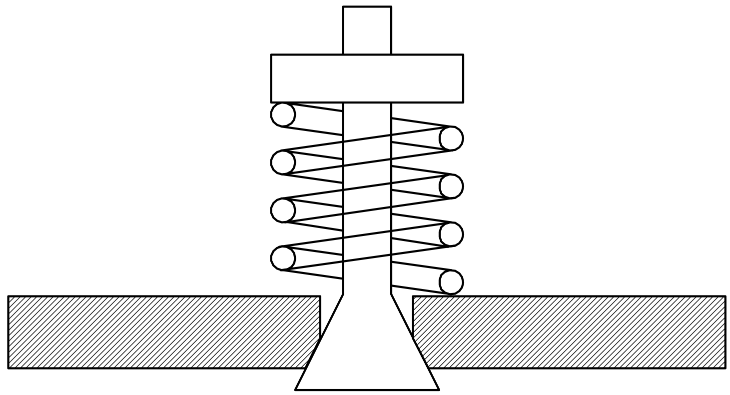

Figure 2 Engine Valve Schematic

The use of a compression spring, as shown in Figure 2, allows for thermal expansion of valve components while maintaining valve closure. If the spring is to be removed, the control system must be able to monitor the seal integrity and accommodate any displacement changes due to thermal expansion. During proof of concept testing, a spring return was maintained, as to not introduce further control complexity.

Aside from the provided spring return valve accompanying the test rig, the other major design constraint was the hook-up requirements for connecting to the spool valve. The existing spool valve had a four port interface based on ISO 4401: *Hydraulic Fluid Power – Four Port Directional Control Valves – Mounting Surfaces*. Figure 3 represents a schematic of the four port interface.

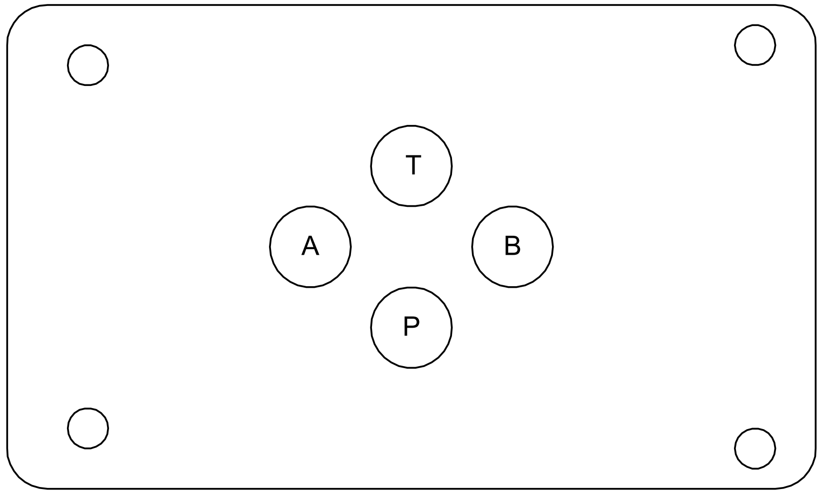

Figure 3 Four Port Directional Control Valve Mounting Surface

From Figure 3, it can be seen that there are four bolt holes and the four hydraulic ports labeled A, B, P, and T. These represent the following.

- Ports A and B are the output ports for hydraulic fluid. Fluid flow is directed to A or B depending on the position of the spool. For the camless engine application, ports A and B provide hydraulic pressure to the top and bottom of the hydraulic amplifier's piston, respectively.
- Port P is the input port. It is connected to the output of a hydraulic pump.
- Port T is the return port. It is connected to the input of a hydraulic fluid reservoir.

The concept was to develop a hydraulic actuator that would connect to ports A and B of the provided spool valve. By creating a hydraulic actuator based on a piston – cylinder arrangement, hydraulic fluid from ports A or B would cause displacement of the piston. This is shown schematically in Figure 4.

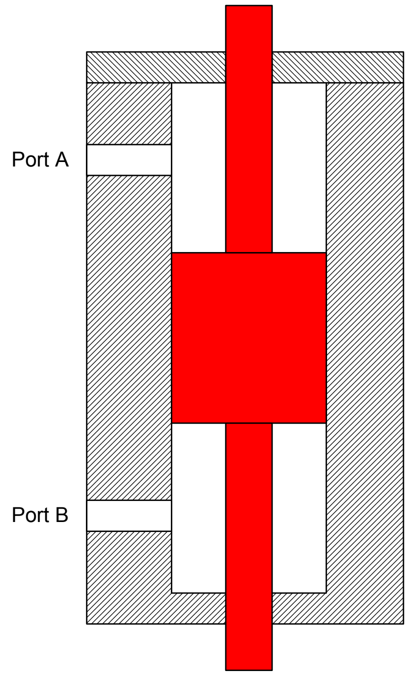

Figure 4 Hydraulic Actuator Schematic

As shown in Figure 4, if hydraulic pressure is introduced through port A from the spool valve, the piston will move down. Hydraulic pressure applied to port B will cause the piston to move up. Furthermore, hydraulic fluid must be able to drain out of the cylinder through the port opposite of that being pressurized. For example, as hydraulic pressure and fluid is applied through port A, the piston moves down. Since the hydraulic fluid is essentially incompressible, the fluid must be able to drain through port B.

Considering this concept, the ICE valve would simply be attached to the end of the piston. This would create linear actuation. Length of stroke would only be dependent on the piston surface area in contact with the hydraulic fluid, the pressure of the fluid, and the resistive forces associated with opening the ICE valve.

2.2 Existing Technology

2.2.1 Introduction to Camshaft Technology

Since the origination of the automobile, the internal combustion engine has evolved considerably. However, one constant has remained throughout the decades of ICE development. The camshaft has been the primary means of controlling the valve actuation and timing, and therefore, influencing the overall performance of the vehicle.

The camshaft is attached to the crankshaft of an ICE and rotates relative to the rotation of the crankshaft. Therefore, as the vehicle increases is velocity, the crankshaft must turn more quickly, and ultimately the camshaft rotates faster. This dependence on the rotational velocity of the crankshaft provides the primary limitation on the use of camshafts.

As the camshaft rotates, cam lobes, attached to the camshaft, interface with the engine's valves. This interface may take place via a mechanical linkage, but the result is, as the cam rotates it forces the valve open. The spring return closes the valve when the cam is no longer supplying the opening force. Figure 5 shows a schematic of a single valve and cam on a camshaft.

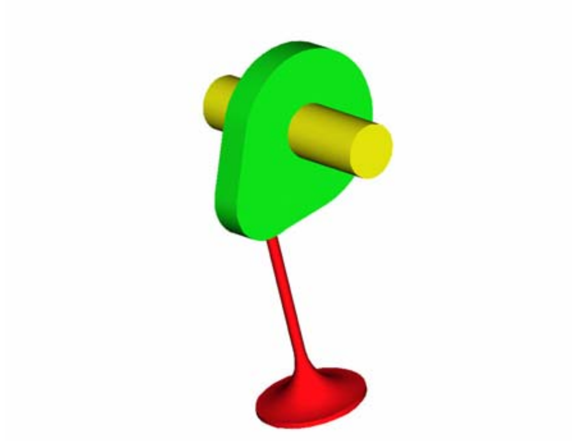

Figure 5 Single Cam and Valve

Since the timing of the engine is dependent on the shape of the cam lobes and the rotational velocity of the camshaft, engineers must make decisions early in the automobile development process that affect the engine's performance. The resulting design represents a compromise between fuel efficiency and engine power. Since maximum efficiency and maximum power require unique timing characteristics, the cam design must compromise between the two extremes.

This compromise is a prime consideration when consumers purchase automobiles. Some individuals value power and lean toward the purchase of a high performance sports car or towing capable trucks, while others value fuel economy and vehicles that will provide more miles per gallon.

Recognizing this compromise, automobile manufacturers have been attempting to provide vehicles capable of cylinder deactivation, variable valve timing (VVT), or variable camshaft timing (VCT). These new designs are mostly mechanical in nature. Although they do provide an increased level of sophistication, most are still limited to discrete valve timing changes over a limited range.

Early in the development of variable engines, Cadillac introduced its V-8-6-4 engine [1]. This 1981 engine was based on a 6.0 liter V-8, but was capable of operating as a 4.5 liter V-6 or a 3.0 liter V-4. The engine changes were made while running and were controlled by the on-board computer's determination of power requirements. The engine changed the number of active cylinders by adjusting the position of the rocker-arm fulcrum. To disable a cylinder, the fulcrum was moved via a hydraulic solenoid valve to the contact point of the rocker-arm and engine valve stem. This prevented the rotating camshaft from supplying enough force to open the engine valve. The computer then made adjustments to the fuel injection rates to compensate for the change in fuel requirements.

The Cadillac V-8-6-4 was the standard engine for all 1981 Cadillac models, but the engine experienced a short production run. Due to consumer complaints about the engine response and operation, especially when changing from one mode to another, Cadillac discontinued its variable engine.

As an update to the short lived Cadillac V-8-6-4, GM will introduce its "Displacement on Demand" engines in their 2004 models [2]. The concept is similar to the earlier Cadillac attempt, but this design limits the engine to operate either as a V-8 or a V-4. With the increase of computing power, GM states the design is more sophisticated, and they promise that the change from 8 to 4 to 8 cylinders will be virtually unnoticeable to the driver.

The new GM engine incorporates a special valve lifter, designed by Eaton Corporation. This lifter is a multi-shaft component capable of telescoping. A hydraulically actuated locking pin, when engaged, prevents the lifter from collapsing.

This allows the cam to open the engine valve. When the locking pin is hydraulically removed, the cam simply collapses the lifter and cannot actuate the engine valve.

Instead of the cylinder operating changes offered by the new GM engine, Honda has introduced its VTEC engines to address the need for greater levels of engine sophistication [3]. This design incorporates three cam lobes and three rocker-arms for each engine valve; see figure 6 for a schematic of the Honda VTEC. The unit locks the rocker-arms together as engine demands change. One rocker-arm is in contact with the engine valve stem and is directly responsible for the actuation of the engine valve. As engine demands change due to increased engine speed, the adjacent rocker-arms are linked, so the valve timing becomes a function of the second cam and rocker-arm pair. This process is repeated for higher rpm's, so the third cam controls the timing of the engine valve. A hydraulic spool valve connects the rocker arms together by driving a pin through the units. The three cam lobes are designated for low, mid, and high rpm requirements. There use generates a more consistent torque output and increase fuel efficiency by providing better valve timing at three different operating ranges.

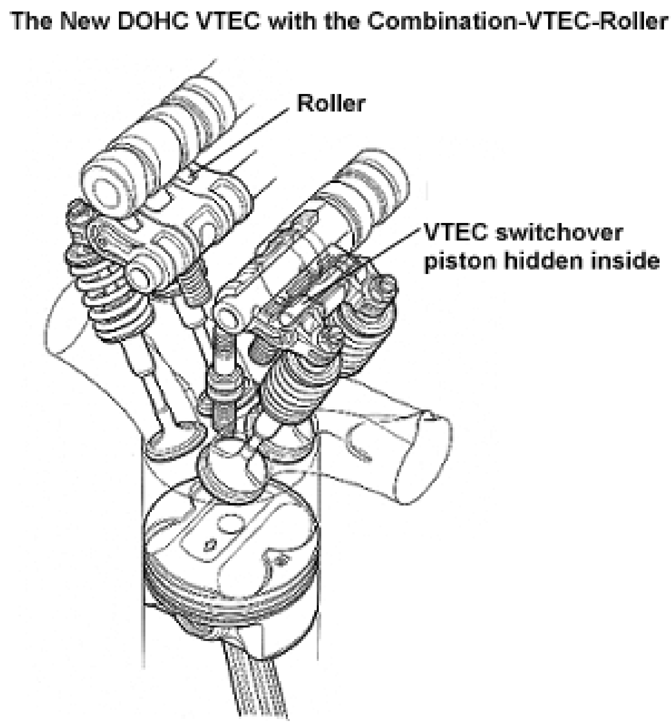

Figure 6 Honda VTEC Schematic [3]

As another approach to VVT, Lexus has developed a “variable valve timing, intelligent” (VVT,i) system for its engines. They claim to have produced the next level of sophistication by introducing continuously VVT [4]. The on-board computer monitors the engine demands and continuously adjusts the timing and overlap of the intake and exhaust valves.

Regardless of the VVT technology differences among the leading automotive manufacturers, the prime similarity of a camshaft remains. Therefore, limitations continue, since the timing is still a function of engine speed.

These limitations have initiated research into camless engine technology. The following section outlines some recent accomplishments of other researchers in an attempt to develop truly independent VVT.

2.2.2 Literature Review of Camless Engine Development

Originally, camless engines were developed for use as a design aide to automotive engine manufacturers. The use of a camless engine allowed the engineer to experiment with valve timing as a means of designing cam profiles. These early units were not limited by dimensional or power consumption restraints. Instead, they were solely developed for laboratory use as a design tool.

Aside from laboratory use, history shows that the idea of a camless internal combustion engine had its origins as early as 1899, when designs of variable valve timing surfaced [5]. It was suggested that independent control of valve actuation could result in increased engine power. More recently, however, the focus of increased power has broadened to include energy savings, pollution reduction, and reliability.

To provide the benefits listed above, researchers throughout the previous decade have been proposing, prototyping, and testing new versions of valve actuation for the internal combustion engine. Their designs have taken on a variety of forms, from electro-pneumatic to electro-hydraulic. These designs are based on electric solenoids opening and closing either pneumatic or hydraulic valves. The controlled fluid then actuates the engine valves.

Much of the remaining documentation deals with either the control of the solenoids or the computer modeling of such control systems. The research on the control of the solenoids is crucial since their precision and response is a limiting factor to the development of earlier camless valve actuators.

A comprehensive project using solenoid control of pneumatic actuators was completed in 1991 [5]. This research included the development of the actuators, a 16 bit microprocessor for control, and comparative testing between a standard Ford 1.9 liter, spark ignition, port fuel injected four cylinder engine and the same engine modified for camless actuation. Testing compared the unmodified engine to that of the same engine, altered to include eight pneumatic actuators in place of the standard camshaft.

The actuators used during the research required an off-engine power source because an engine mounted compressor was not feasible. The researches found that for engine operation at 1500 rpm, the eight actuators used a total of 2.5 kW of power. This compares very high to the 140 watts of power consumed by comparable production engines. As Gould et al. states, their work cannot be considered feasible for implementation due to the high power requirements of the actuator.

For their project, pneumatic actuators were chosen after running comparison tests among different methods. Pressurized air was chosen due to its low mass, allowing fast response and stability over a broad temperature range. The researchers found that hydraulic systems had sluggish response, especially at low temperatures.

The pressurized air was controlled by electromagnetic valves. All flow path distances were minimized to increase the response time of the actuator by reducing the volume of air required for actuation. The pressurized air opened the engine valves based on the timed electrical signal input to the “electromagnetic latch.” Residual air was compressed during valve seating and provided a means of slowing the valve for a soft seat.

The researchers concluded that the test engine produced approximately 11% greater torque at low engine speeds (below 2000 rpm) compared to a conventional engine. Furthermore, the camless engine was capable of reducing emission gasses, specifically "brake specific nitrous oxide emissions" (bsNOx), but only by degrading the combustion process [5].

In 1996 the next generation of camless engine was completed at the Ford Research Laboratory by, principally, Michael Schechter and Michael Levin. Ford's work has taken a detailed look at the plethora of parameters associated with consistent, reliable engine operation. The first half of the paper describing their work is focused on the base parameters of valve timing and overlap. This data will serve as beneficial information during the further development of the prototype at the University of South Carolina.

Beyond the basics, Schechter and Levin introduce a new concept of the hydraulic pendulum. It is stated that the use of a hydraulic pendulum decreases the system's energy consumption by converting the kinetic energy of a closing valve into potential energy stored in the pressurized fluid. This reduces the energy required for pumping the hydraulic fluid. Through this conversion of energy, the authors predict that a 16-valve, 2.0 L engine will consume about 125 W to operate at light loads.

The hydraulic pendulum also allows for the solenoid-based-system to slow valve velocity. This results in soft seating the valve and is a favorable attribute of the new system. Another benefit is the ability to vary the opening and closing velocity of the valve. This allows for increased variation to engine valve parameters.

A schematic of the hydraulic pendulum is shown in Figure 7. High and low pressure hydraulic reservoirs are connected to the engine valve's actuating piston. The control of this fluid is accomplished by means of two solenoids and two check valves. High pressure fluid is always in contact with the lower side of the piston, and either high or low pressure fluid is in contact with the upper side of the piston. The difference in pressure contact area is utilized in conjunction with the hydraulic pressure to vary the actuating forces.

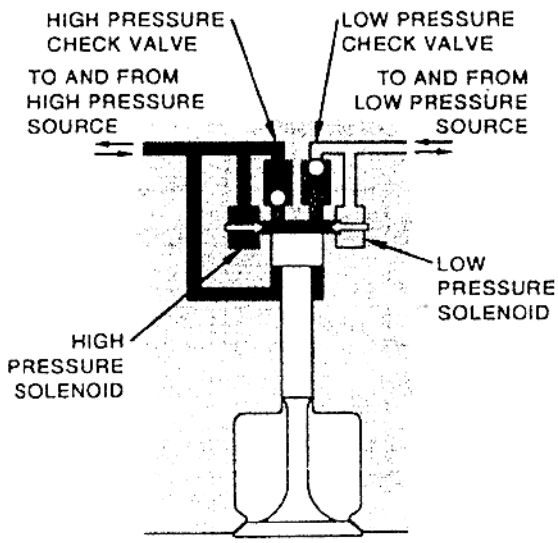

Figure 7 Hydraulic Pendulum Schematic [6]

The authors provide a detailed description of the valve actuation cycle. This is summarized as follows. To open the engine valve, the high pressure solenoid opens to allow high pressure hydraulic fluid into the upper chamber. Due to the difference in pressure contact area, the valve opens. Next, the high pressure solenoid closes, but the valve's momentum continues to open the engine further. This causes a reduction of pressure in the upper chamber and allows the low pressure check valve to open. The engine valve decelerates as it pumps the high pressure fluid from the lower cavity back to the high pressure reservoir. This process both slows the valve and recovers some energy by converting the kinetic energy of the engine valve into potential energy in the high pressure fluid. Once the upper cavity pressure equalizes with the low pressure reservoir, the check valve closes and the upper cavity fluid is static. This allows the engine valve to be held open.

Closing the valve is initiated by the opening of the low pressure solenoid valve. The engine valve accelerates toward its closed position based on the force differential between the high pressure lower cavity and the low pressure upper cavity. The upper cavity fluid is pumped back toward the low pressure reservoir. Energy is again recovered and the engine valve is soft-seated through a similar deceleration process. By closing the low pressure solenoid valve, the upward momentum of the engine valve pressurizes the upper cavity fluid. This increase in pressure opens the high pressure check valve and allows the upper cavity fluid to be pumped back to the high pressure reservoir. Again, energy is converted from kinetic to potential and the valve is decelerated.

The best timing of this process would allow for the kinetic energy of the engine valve to be exhausted exactly when it closes. However, the researchers provide an alternative to such precision. Instead, they suggest stopping the engine valve just prior to contact with the seat, and then briefly opening the high pressure solenoid to complete the cycle [6].

Through the use of a hydraulic pendulum, a complete four cylinder ICE was produced and found some success. However, the system is complicated and requires multiple components. The use of a hydraulic pendulum requires two solenoids and two check valves per engine valve and both a high pressure and low pressure hydraulic fluid supply. (Schechter et al. state that two solenoids can run a pair of valves as-long-as the pair are synchronized. However, this detracts from the concept of independent valve control.)

The camless engine developed by Ford and described above was then enhanced at the University of Illinois at Urbana-Champaign. The focus of the project was to advance the hydraulic-pendulum-based CLE actuator by developing an adaptive feedback control. Their research is focused on the electronics and algorithms of data acquisition and control and extends beyond the scope of the current phase of research here at the University of South Carolina. However, as a comparison, some of the results are presented here. The complete system was limited to operating at 3000 rpm. Valve lift greater than 5 mm could not be consistently controlled [7].

The authors raise concerns of component variation and its effect on the proper control of the engine valves. These issues rise mainly from the tolerances associated with the production of automotive components. Furthermore, the authors investigated the system changes due to fluctuations in hydraulic fluid temperature [7]. Both of these parameters will need to be addressed during USC's next phase of research.

Beyond the projects discussed within, little technical information exists on the development of the camless engine. More recent projects, including those at Siemens Automotive – Europe and International Truck and Engine Corporation represent the next phase of external CLE research.

In June, 2000, Sturman Engine Systems and International Truck and Engine Corporation completed their proprietary CLE project by being the first CLE based truck to reach the summit of Pikes Peak [8]. The International engine with Sturman hydraulic valves is a diesel system. One of the difference between diesel truck and gasoline passenger car demands is operating speed, and even experts familiar with the project have raised concerns about the electrohydraulic system capabilities at the higher speeds of passenger cars [9]. Although the system is proprietary, one available detail is the hydraulic fluid used to actuate the engine valves is controlled by a spool sandwiched between two electrical coils [9].

Concurrent to the International Truck and Engine and Sturman Engine Systems development, Siemens Automotive has been advancing their version of the CLE. Siemens claims their Electromechanical Valve Train (EVT) is nearing production capabilities [10]. Again, the system is proprietary, but some details are available. The Siemens' CLE is operating on the anticipated 42 volt bus and comes from an electrical distributor and crankshaft-mounted starter-generator. Special algorithms provide the current signal to the engine actuators and are capable of controlling valve velocity and soft-seating. Their actuator is based on a spring/mass system. Early results show a fuel savings of 10 percent. Siemens is currently working on the next phase, which will focus on sensorless actuator control [10].

The one common factor of all known CLE work is that none are using piezoelectric devices to control the engine valve actuation. It is this difference that makes the USC project unique and may prove to be the necessary element to provide the next level of engine sophistication.

----

[1] Lamm, J.; "V-8-6-4, Cadillac Tries Multi Displacement" Road & Track November, 1980

[2] General Motors – GM and the Environment. May 21, 2001. General Motors. June 13, 2001 <http://www.gm.com/company/gmability/environment/gm_and_the_env/releases/displacement_052101.htm>.

[3] Mori, Kaz. Honda's High-Output LEV Engine Home Page. Honda. June 13, 2001 <http://s2000.vtec.net/jdmhtml/high-output-lev.htm>.

[4] Lexus – Variable Valve Timing a First in an SUV. Autoworld. June 18, 2001 <http://www.autoworld.com/RX300.htm>.

[5] Gould, L; Richeson, W; and Erickson, F., 1991, "Performance Evaluation of a Camless Engine Using Valve Actuation with Programmable Timing," SAE Paper No. 910450.

[6] Schechter, M.; and Levin, M., 1998, "Camless Engine," SAE Paper No. 960581

[7] Anderson, M; Tsao, T-C; and Levin, M., 1998, "Adaptive Lift Control for a Camless Electrohydraulic Valvetrain," SAE Paper No. 981029

[8] Anonymous, "First Camless Truck Reaches the Summit of Pikes Peak!" Power Stroke Registry, Spring 2001

[9] Murray, C; Electronic Controls to Replace Camshafts in Diesel Engines. April 14, 2000. EETimes.com. July 7, 2000 http://www.eetimes.com/story/industry/system_and_software_news/OEG20000414S0050.htm>.

[10] Ladd, D; Camless Engine is Gaining Momentum. September 13, 1999. Siemens Automotive. July 4, 2000 <http://media.siemensauto.com/mediacenter2/queries/releasefull.phtml?prjob_num=1195>.

Chapter Three: Phase II

3.1 Design Process

This chapter summarizes the design process in, roughly, chronological order. Some designs were initiated and later altered or abandoned as more information about the project was discovered. The entire process is outlined as a historic reference and is written considering that the project will continue into a second design phase. Therefore, earlier work that did not appear in the proof-of-concept design may have significance in future development.

Initially, the boundary interface of the design had to be well defined. This is divided into three discrete components. First, the connections to the spool valve needed to be outlined. Next, the interface with the engine valve stem needed to be studied. Finally, the systems envelope needed to fit within the available space on the test stand.

The provided spool valve had standard four port control valve connections as dictated by ISO 4401. These dimensions are illustrated in Figure 8 and the table below.

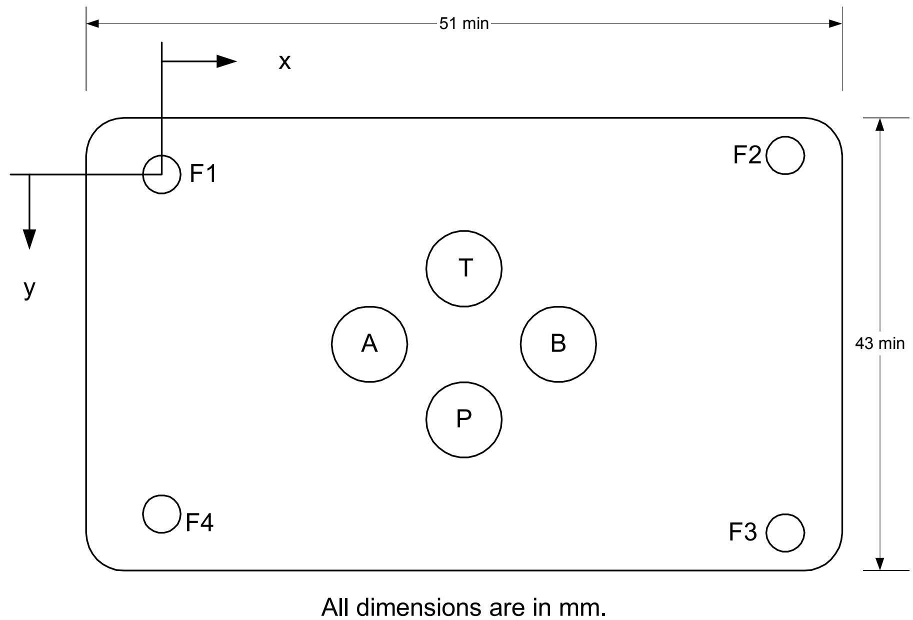

Figure 8 Four Port Mounting Surface Dimensional Scheme

Hole	X (mm)	Y (mm)
F1	0	0
F2	40.5	-0.75
F3	40.5	31.75
F4	0	31
P	21.5	25.8
A	12.7	15.5
T	21.5	5.1
B	30.2	15.5

Table 1

Note that the center of the bolt hole labeled F1 is the origin for the remaining dimensions.

Hydraulic ports P, A, T, and B all have maximum diameters of 7.5 mm. Bolt connections can either be M5 threaded connections or through holes for M5 bolts. For the development of the hydraulic amplifier, threaded connections are required to mate to the through holes of the spool valve housing. The standard, ISO 4401, indicates that for maximum valve interchangeability, thread depth should be a minimum of twice the bolt diameter plus 6 mm.

Having defined the connection requirements for mating to the spool valve, the next step was to determine how the forces of an engine valve could be simulated. Prior to receipt of the hydraulic test rig and cylinder-simulator-stand, it was decided that a piston/cylinder capable of providing resistive force to hydraulic-induced-displacement must be designed.

For the purpose of developing simple resistive force, the design shown in Figure 9 was proposed. The use of compression spring would provide resistance to the piston displacement and would loosely simulate the resistive forces on and ICE valve. As hydraulic fluid pressurized the cylinder through port A, the piston would displace against the resistive force of the spring. This would simulate opening the engine valve. The reverse is true for pressurization through port B.

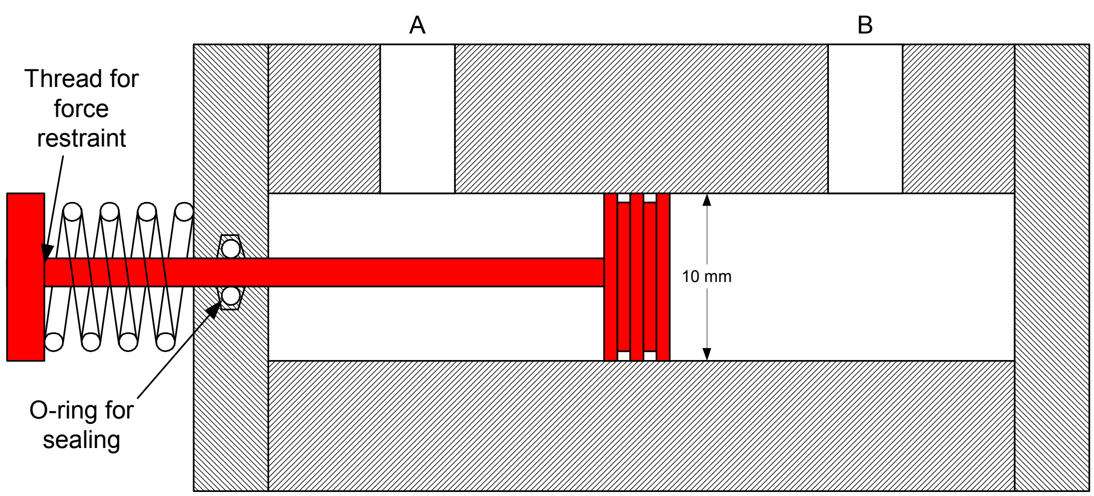

Figure 9 Resistive Force Simulator Schematic

The development of this first prototype design raised several new questions addressing materials and sealing concerns. As a starting point for material selection, the spool valve materials were examined. This was reasonable, since both units will be experiencing similar pressures and environments.

The materials of the spool valve are European (DIN) designations. For the body, GGG50, a form of ductile iron, is used. For the spool, 16MnCr5, a steel alloy, is used. These material designations were compared to United States standards to find an equivalent material.

The closest United States Standard materials with similar properties are as follows. In place of the GGG50 use ASTM A536 80-55-06, and for the 16MnCr5, use Grade 4120 steel.

Heat treatment and hardness concerns naturally arose from the choice of materials, and the following guidelines must be considered. For metal to metal contact inside the cylinder's bore and elsewhere, the metal containing the sealing (o-ring) groove should be the softer of the materials. Furthermore, in a piston/cylinder arrangement, the piston's hardness should be lower than the cylinder bore to minimize damage to the bore.

Sealing concerns were a primary issue from the beginning of the project. The unit would need to be able to actuate based on the provided hydraulic pressure. Therefore, seals would be needed to prevent the leakage of pressurized hydraulic fluid, but allow for the piston to reciprocate. For the purpose of sealing, o-rings were considered first. The piston design was modified to include standard o-ring gland dimensions based on the Parker O-ring Handbook. Figure 10 shows the gland design required for the 10 mm diameter piston. It is provided here as a reference for any future designs needing o-ring seals.

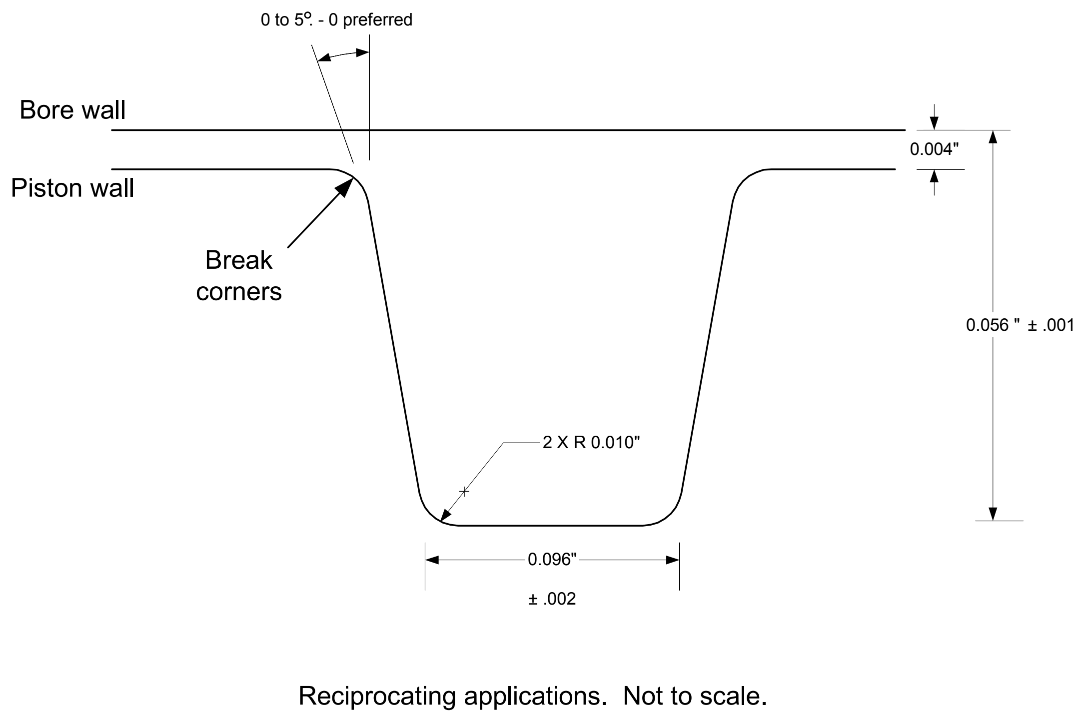

Figure 10 O-ring Groove Dimensional Example

Another design issue surfaced during work on the system shown in Figure 9. As the design shows, the piston had a rod only on one side. This was a concern since the piston was not balanced. Without a rod on both ends, the piston could waver from pure linear translation and become jammed within the cylinder bore. Furthermore, the addition of a second rod to balance the piston also would provide a natural instrumentation platform. The disadvantage of a two rod design is the need to seal a second reciprocating interface. However, it was decided the advantages overcame the shortcomings based on the addition of an instrumentation platform and the increased balance. Based on this reasoning, a balanced piston with a rod extending from each side was used for the remainder of the design iterations.

Although some piston details related to the pressure surface area and overall dimensions were not completed, attention turned to the development of the cylinder. The design of the cylinder needed to incorporate the connection requirements for attaching to the spool valve and would need to provide the proper size bore for piston displacement.

During this phase of the design, the internal routing of the hydraulic fluid within the spool valve was not well defined. Therefore, a hypothesis of fluid flow was formed based on the limited drawings of the spool valve. From this, Figure 11 was proposed as a design solution for the cylinder housing.

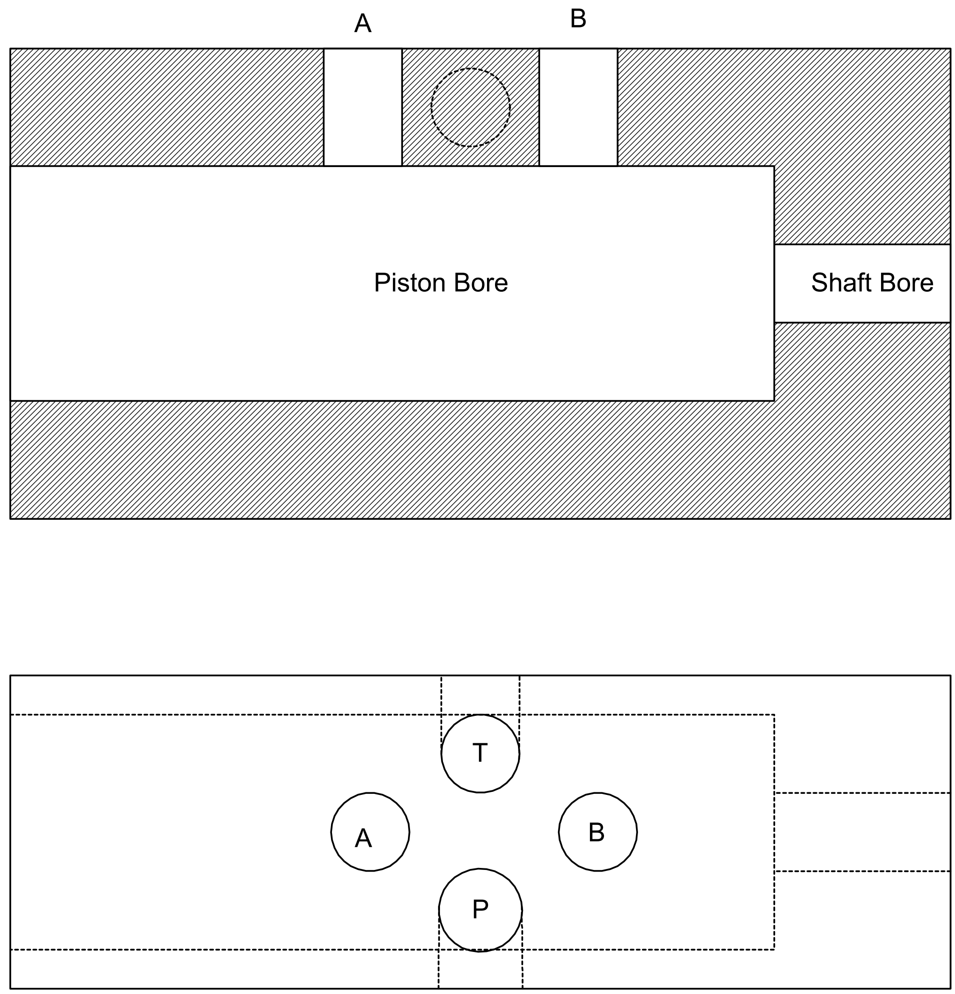

Figure 11 Cylinder Block Concept

This design of the cylinder incorporated the four hydraulic connections to the spool valve, a large diameter bore to house a piston, and a smaller bore for one end of the piston's shaft. One end was left open for installation of the piston. Following installation, this end was to be closed with a plate containing a small diameter hole for the piston shaft. All sealing, based on the use of o-rings would be integrated into the shaft bores. The single open-ended design was proposed as a means of limiting the number of face seals; therefore, reducing the number of leak paths.

As seen in Figure 11, the four hydraulic ports were integrated into the cylinder design. Ports A and B allowed hydraulic fluid to pass from the spool valve directly into the piston bore. Connections for ports P and T made a 90° bend and connected the spool valve to openings in the side of the cylinder block. These openings could be threaded as necessary to connect hydraulic fluid supply and return to ports P and T respectively.

These connection were based on the hypothesis that hydraulic fluid must be supplied to port P of the spool valve. In turn, the spool valve, depending on the position of the spool, would direct the hydraulic fluid into port A or B. Furthermore, the fluid from the port not being pressurized would be routed through the spool valve to port T and ultimately to the hydraulic fluid reservoir. This hypothesis was based on the limited spool valve drawing information and basic hydraulic knowledge. The fluid routing through the spool valve was later confirmed by Peter Deuschle (Hydraulic Control Systems, Inc., Stuttgart, Germany.)

Unfortunately, this design had one significant problem to overcome. The distance between ports A and B were determined by the same distance on the spool valve. This was a standard set forth by ISO 4401. Therefore, the distance between ports A and B centerlines was 17.5 mm and the distance between the closest edges was 10 mm. This limited distance was in direct conflict with the design specification stating that the valve, and therefore, the piston had to translate 10 mm.

It became clear that the holes would need to be further apart in order to accommodate the proper translation with a reasonably sized piston and without covering the ports. If the ports became covered by the translating piston, the entire system would become hydraulically locked. Therefore, the next evolution of the design needed to increase the distance between ports.

A second problem with the design proposed in Figure 11 was the change in bore diameter from that needed to house the piston to enclosing the rod. Based on the expected tolerances associated with the design, it was considered that machining a diameter step-change like that introduced in Figure 11 would be very difficult and expensive.

Two design alternatives were generated to address the problems encountered in the first cylinder design. They are shown in Figures 12 and 13. Both are based on securing an adapter block between the spool valve and the cylinder block. Furthermore, the design of the cylinder block was updated to a through hole equal to the size of the piston bore.

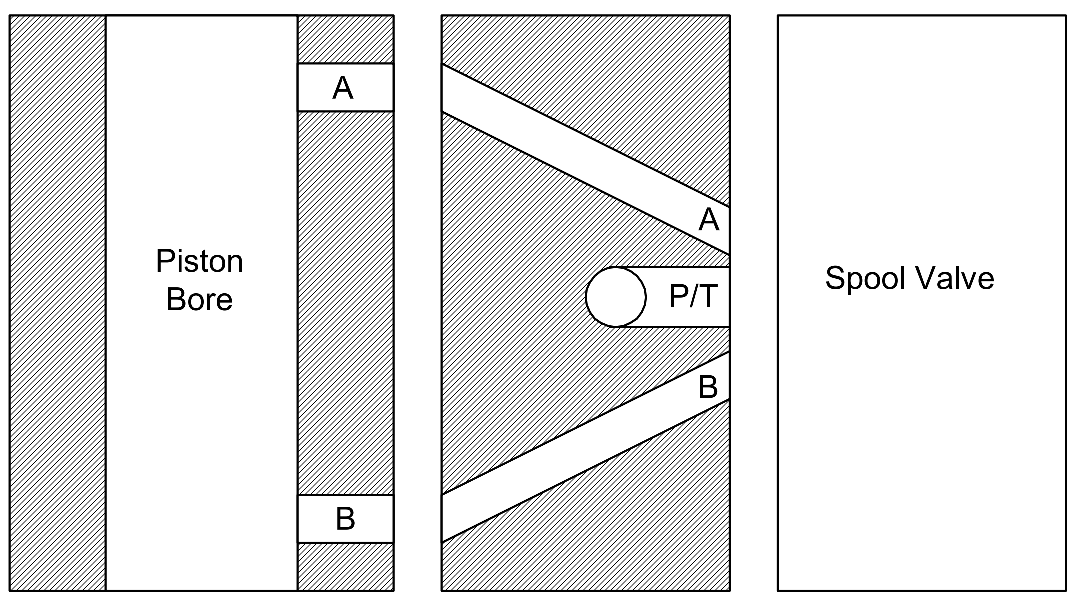

Figure 12 Cylinder Block Adapter Plate Concept

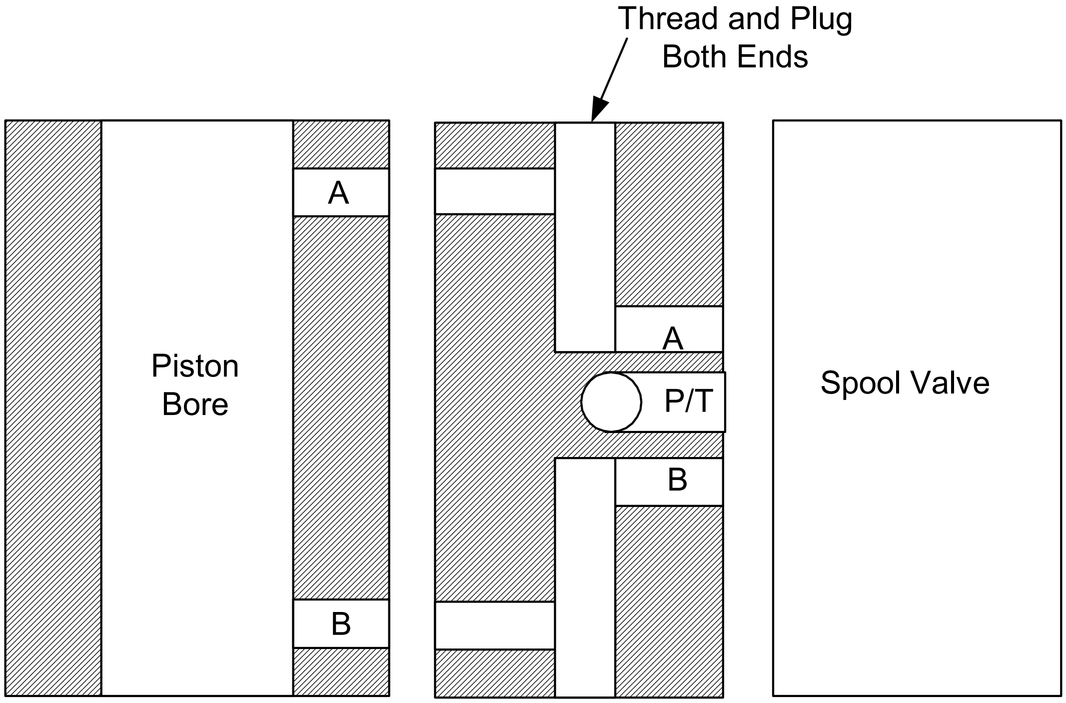

Figure 13 Cylinder Block Adapter Plate Concept II

Both designs, schematically shown if Figures 12 and 13 successfully address the need to move ports A and B further apart along the piston bore. Doing so would allow for the needed displacement of 10 mm while preventing the unit from hydraulic lock. It was decided at this phase to move forward with the design in Figure 13 due to ease of manufacture.

However, as detail design work began, it was clear that this design introduced significantly greater volume that the hydraulic fluid would need to fill. This became a concern if a compressible fluid, such as air, ever became entrapped in the fluid: the greater the fluid volume, the greater the losses due to any compression introduced by entrapped gases.

The other concern generated by the Figure 13 design was the introduction of flow losses due to the 90° bends in the adapter block. Since fluid flow was going to be a major governing factor for the successful operation of the camless engine, it was decided to minimize flow length, losses, and volumes associated with the hydraulic fluid paths. Therefore, a new design that utilized the angled holes in Figure 12, but integrated them into the cylinder block was initiated. From this, the final version of the cylinder block began to form. This is shown schematically in Figure 14.

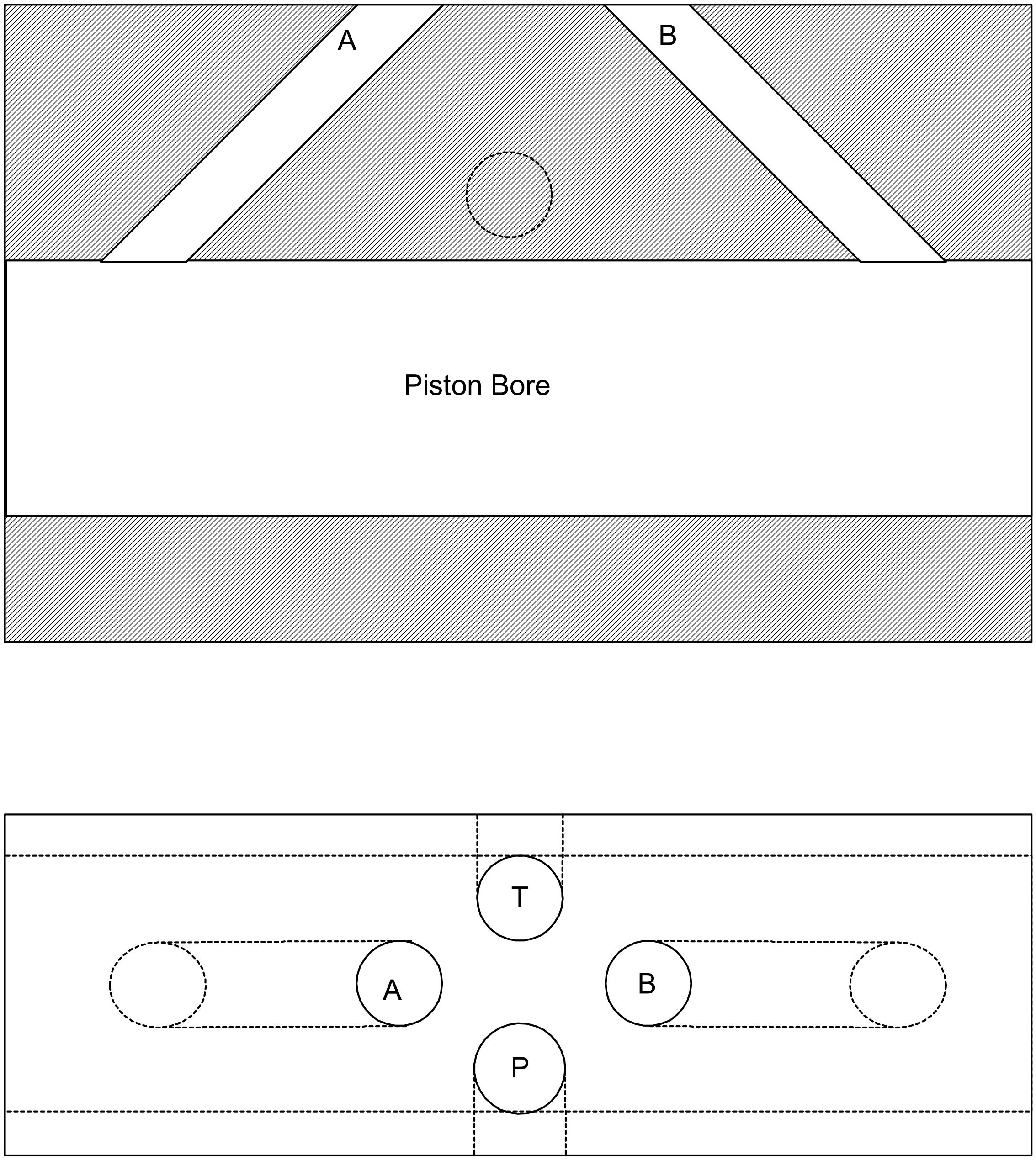

Figure 14 Angled Hole Cylinder Block Schematic

Once the design had evolved to this point, concern turned to the details of its development. The primary issues to be addressed were sealing, fasteners, dimensions, and hook-ups to the recently received pneumatic test fixture. On loan from Siemens

Automotive, Inc., Regensburg Germany, a test fixture designed to mount their earlier camless engine attempt arrived for the purpose of testing the USC prototype.

The test fixture was comprised of three key components. First, an engine valve with spring return was mounted vertically. Second, surrounding the lower portion of the valve was its valve seat integrated into a pneumatic cylinder. Finally, surrounding the upper portion of the valve was a mounting block. It was this mounting block that dictated some of the detailed dimensions of the new cylinder block and piston. Based on measurements taken from the mounting block, Figure 15 was developed.

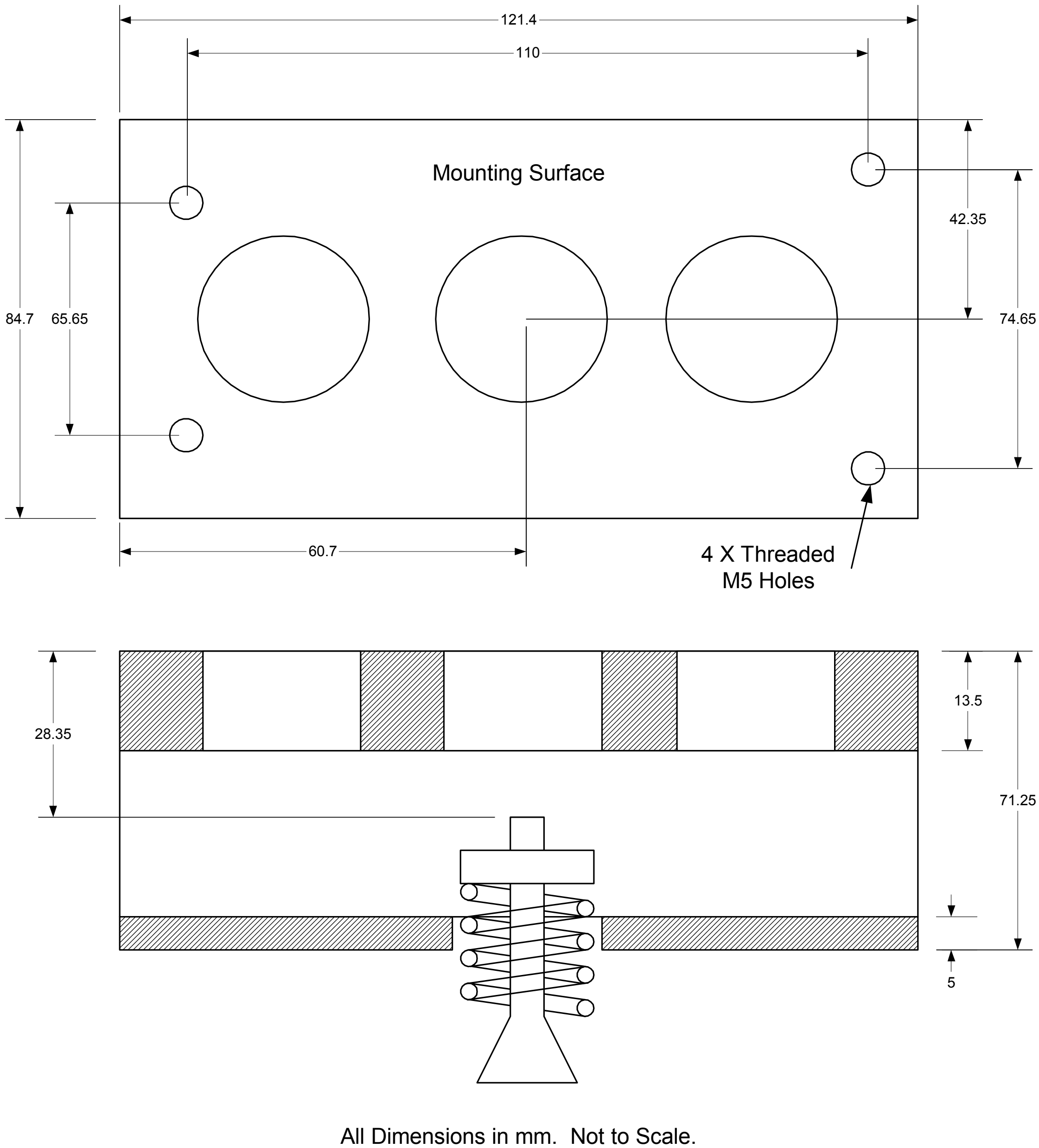

Figure 15 Pneumatic Test Stand Mounting Surface

The critical dimensions are outlined in Figure 15. The 28.35 mm is the dimension from the top of the mounting block to the top of the valve stem when the mounting block is secured to the remainder of the test fixture and the valve is in its closed position. Since the piston makes direct contact with the top of the valve stem and the piston must translate 10 mm, this 28.35 mm dimension dictates much of the dimensional layout of both the cylinder block and the piston. Also of prime importance are the four bolt holes and the bolt pattern. This pattern must be duplicated on the camless engine prototype to enable assembly.

Since the test fixture's dimensions need to match, an assembly concept was produced to drive the detail dimensions of the piston, cylinder block and the bore plates used to seal the cylinder block. This assembly schematic can be seen in Figure 16.

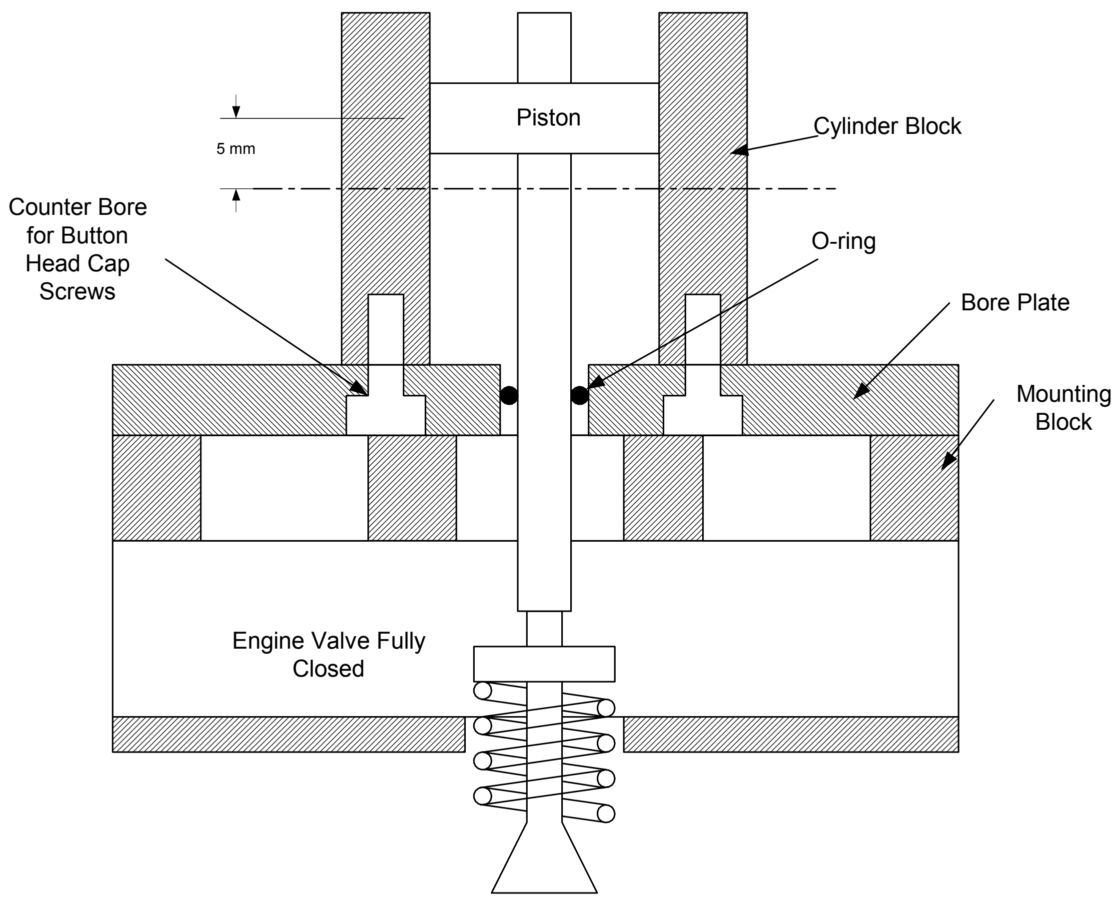

Figure 16 Cylinder Block Mounting Schematic

The design shown schematically in Figure 16 demonstrates some key design elements. First, button head cap screws will be used and mounted into a counter bore. This is necessary to obtain a flush fit with the mounting block. Any other connection is undesirable because only a face-to-face fit ensures that the valve stem will be a specific distance from the centerline of the cylinder block.

Second, a seal is required to prevent pressurized hydraulic fluid from leaking past the bore plate. This seal represents another problem because it must not only prevent leakage, it must do so without introducing high levels of friction on the reciprocating shaft.

Finally, an important dimension results from the assembly. Regardless of piston, cylinder block, and bore plate design, the center line of the piston must offset 5 mm from the centerline of the cylinder block when the engine valve is closed. This maintains symmetry within the system and allows for the piston to translate 10 mm, 5 mm on either side of center, and actuate the valve the required 10 mm.

With dimensional constraints beginning to emerge, other design parameters were chosen arbitrarily as a starting point for detail development of the piston and bore. From these originally chosen dimensions, equations were established to outline the relationship between critical dimensions. Of greatest concern was the design of the piston and how its dimensions affected the interface location of ports A and B with the bore. This critical concern stems from the need to avoid hydraulic lock while maintaining the required 10 mm valve translation. The goal was to maintain full stroke without interfering with the port/bore interface.

To establish the location of ports A and B along the length of the bore, the following simple equation was derived.

$$L = \tfrac{1}{2} \cdot (P + S + D) \qquad \text{Eq. 1}$$

Where,

P ≡ Length of Piston

S ≡ Total Length of Valve Stroke Required

D ≡ Diameter of Port

L ≡ Minimum Length from Bore Center Line to Port Center Line

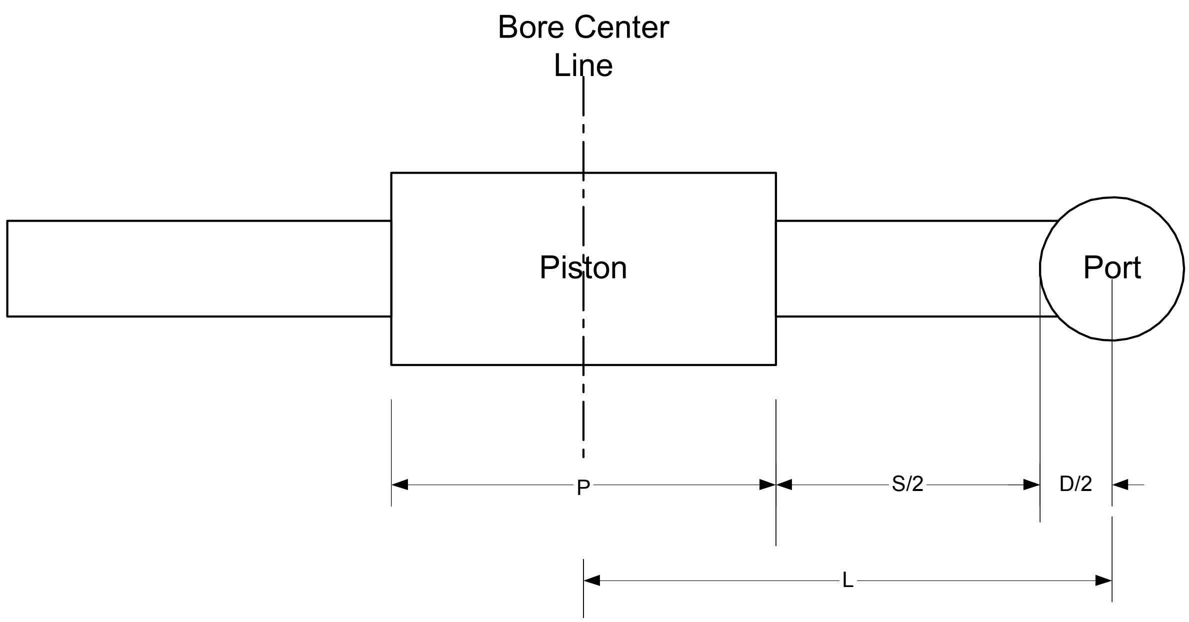

Figure 17 Piston and Port Relationship

Based on the value of L, calculated above, and the given port configuration from ISO 4401, the angle of the port A and B holes are defined as shown schematically in Figure 14. Having defined the basic parameters of the ports based on arbitrary dimensions, focus turned to the detail design of the piston.

The piston went through several design iterations to develop both the overall dimensions and satisfy the means of manufacture. Due to the close clearance needed between the piston and the bore, manufacturing capabilities were scrutinized to develop a design that could act as a metal-to-metal reciprocating seal and could be aligned with multiple components (cylinder block, two bore plates, and the valve stem). Of greatest concern was the ability to center the shafts of the piston through the bore plates and maintain the minimal clearance within the bore. This obstacle was directly related to the ability of the manufacturer to maintain a concentricity tolerance between the piston diameter and the shaft diameter.

The first proposed solution was to develop a three part piston assembly. This assembly would be made of a male rod, female rod, and piston. With ample clearance between the rods and a hole through the piston, the unit could be assembled with, essentially, two separate centerlines. One centerline could pass through the rods, while the other centerline passed through the cylinder. This would allow for the rods to align to the center of the bore plate holes and the piston to align to the center of the bore, even if the bore plate and bore centers were out of alignment. Shown schematically in Figure 18, this design went through several iterations before ultimately being abandoned for a simpler one-piece piston.

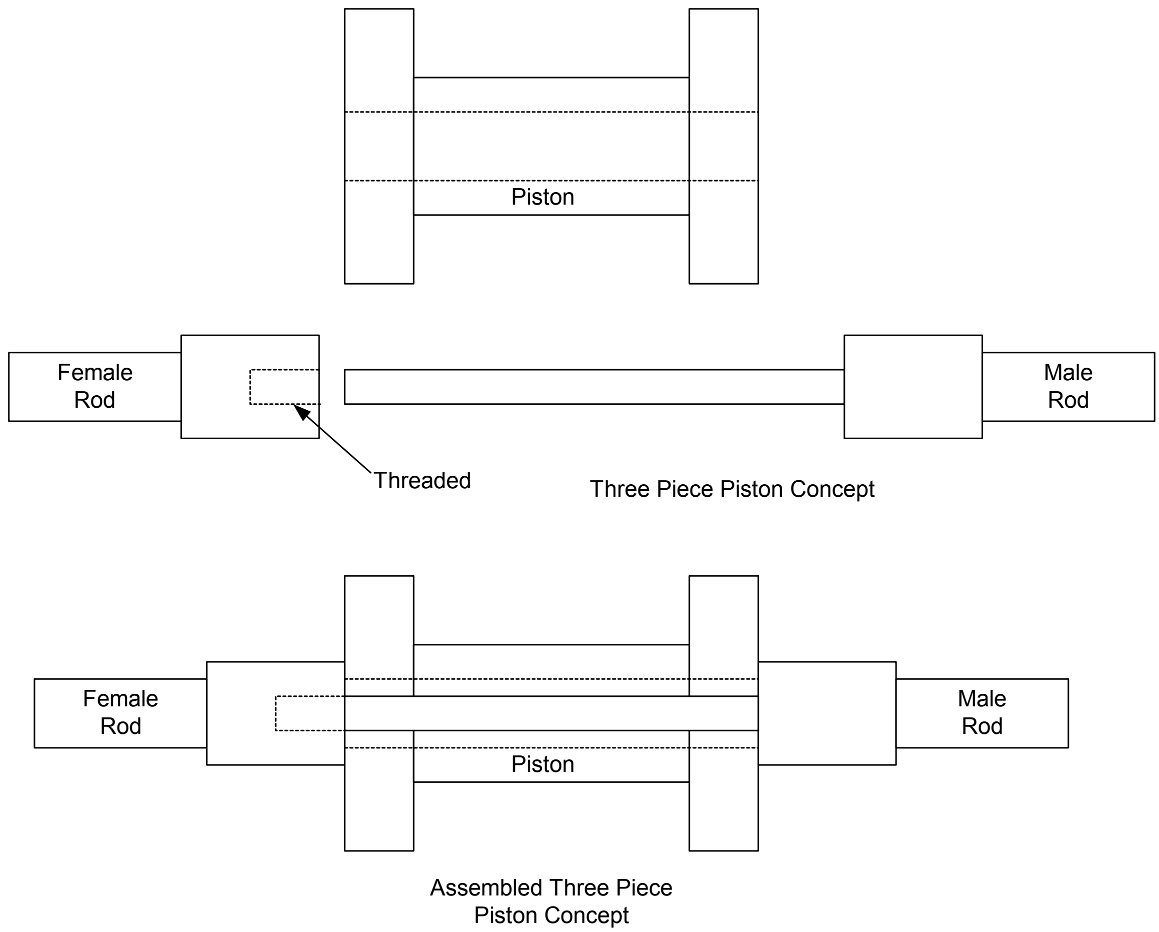

Figure 18 Three Piece Piston Concept

It was decided with a high quality manufacturer, a one piece piston could be developed that would have adequate geometric tolerances. Combining a well manufactured one-piece piston with a position tolerant bore plate design would be adequate. This change results in fewer components and a simpler design.

Similar to the three-piece piston design, the one-piece piston went through several design iterations of both size and geometry. Following a consultation with the manufacturing group at Siemens Diesel Systems Technology (DST), Blythewood, SC, the final piston design was completed. This design created the basis for the development of the remaining components, including the bore plate and the cylinder block. Below are some of the critical factors associated with the piston. See Figure 19.

- The piston diameter and tolerance shall be $15.000^{+0.000}_{-0.003}$ mm.
- The bore diameter and tolerance shall be 15.003 ± 0.001 mm.
- Rough machining must leave approximately 0.03 mm for the grinding operation.
- Both ends of the rod shall be chamfered to allow for ease of manufacture and maintenance of concentricity tolerance.
- An undercut is needed at the piston and rod interface to ease grinding.

Siemens DST indicated that post heat-treat grinding would clean-up any concentricity problems. This realization made the one-piece piston design a feasible approach.

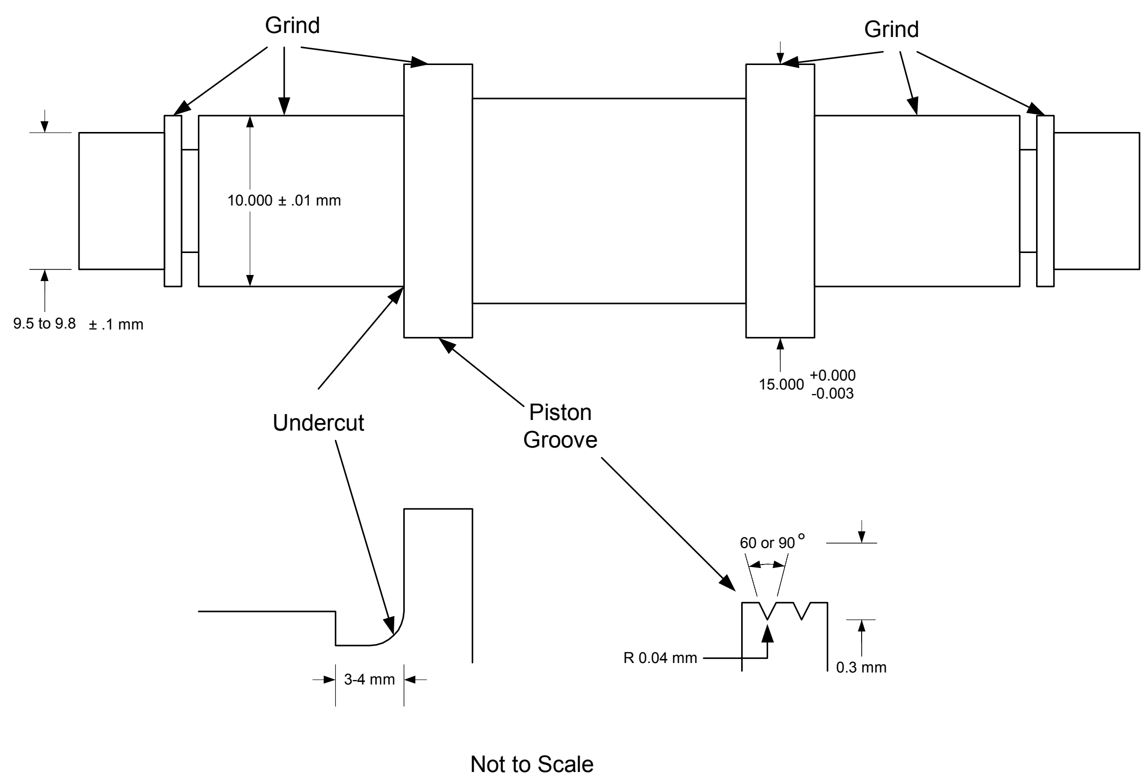

Figure 19 Piston Manufacturing Critical Concerns

Based on these diameters, calculations were developed to understand the output force of the unit based on the input pressure. The following example demonstrates the force calculations for an input pressure of 50 bar. Conversions are included for ease of understanding.

$$A_r = \pi \cdot (r_r)^2 \quad \text{Area of the rod} \qquad \text{Eq. 2}$$

$$A_p = \pi \cdot (r_p)^2 \quad \text{Area of the piston} \qquad \text{Eq. 3}$$

$$A = A_p - A_r \quad \text{Area of pressure contact} \qquad \text{Eq. 4}$$

$$F = P \cdot A \quad \text{Force generated by pressure} \qquad \text{Eq. 5}$$

For example, r_r = 5 mm, r_p = 7.5 mm, and P = 50 bar = 5 N/mm^2. This results in a force of 490.875 N or 110.4 lbf.

Based on equations 2 – 5, the force output can be calculated for any pressure input. These equations can be used to determine the needed hydraulic pressure based on the resistive forces of opening the engine valve.

With the overall diameter dimensions and the general design of the piston completed, the remaining details were surface finish and length requirements. First, surface finish was addressed. The manufacturing group at Siemens DST provided some insight into the surface finish needs of a reciprocating metal-to-metal seal, and they outlined the manufacturing limitations. For the critical piston surface in contact with the cylinder bore, a surface designation of Rz1 was applied. This is a nearly polished surface. For the contact surface of the rod and seal, Rz4 was used. The meaning of the surface finish designation is shown below in Figure 20.

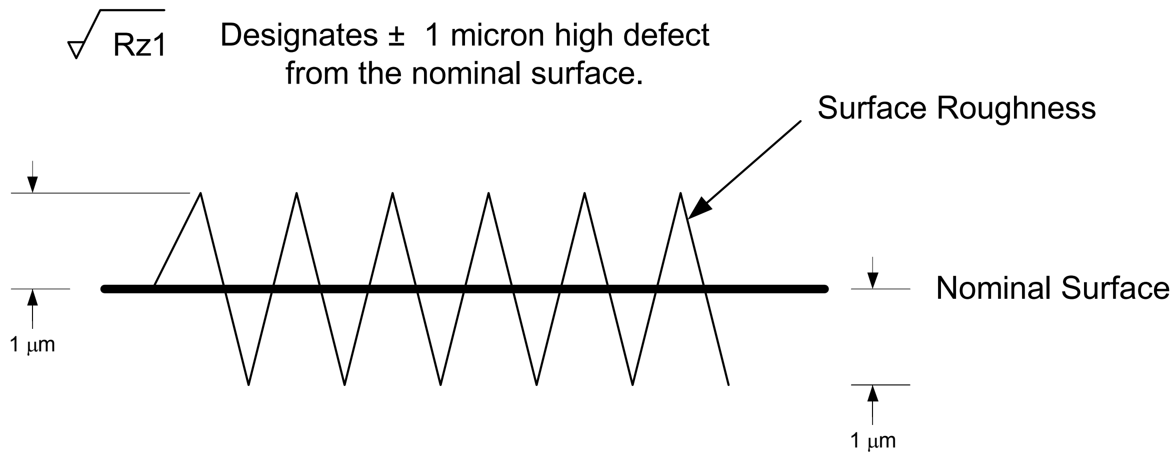

Figure 20 Surface Roughness Example

Development of the piston rod length was a less clear procedure. It is at this point in the design process that several factors became interrelated and the process became an exclusively iterative procedure. Design iteration was necessary because of the relationships between component dimensions. Listed below are some of the primary interactions that had to be confronted. It should be noted that based on these dimensional interactions, solid models were developed. The solid models were continuously updated to reflect the needed changes and provided an effective tool for virtual assembly and component design. The end result addresses the concerns and creates a functional system.

- The length of the piston rod must be such that the piston is offset 5 mm from the center of the cylinder bore. This distance is defined by the distance between the mounting surface and the closed engine valve, which is a constant 28.35 mm, the thickness of the bore plates, and the distance between the hydraulic ports A and B within the cylinder block.

- The distance that the ports are separated is based on several parameters. Of primary concern is the requirement that the piston must actuate a minimum of 10 mm without covering the hydraulic ports. Second, they must be as close together as possible to minimize the fluid volume within the bore.
- The bore must be physically separated from any other machining operations to avoid distortion. The result of this can be seen as the bore being offset from the threaded connections needed to assemble to the spool valve.
- The piston, itself, must be of reasonable length so that it maintains its orientation while translating. A piston that is too short has a greater chance of tilting slightly and becoming jammed within the cylinder bore.
- The piston surface area in contact with the cylinder bore should be small to reduce the friction between surfaces.

The results of this design process are shown schematically below in Figure 21. As shown in Figure 21, some of the components appear oversized. However, this assembly looks odd only because of the mating requirements to the other components.

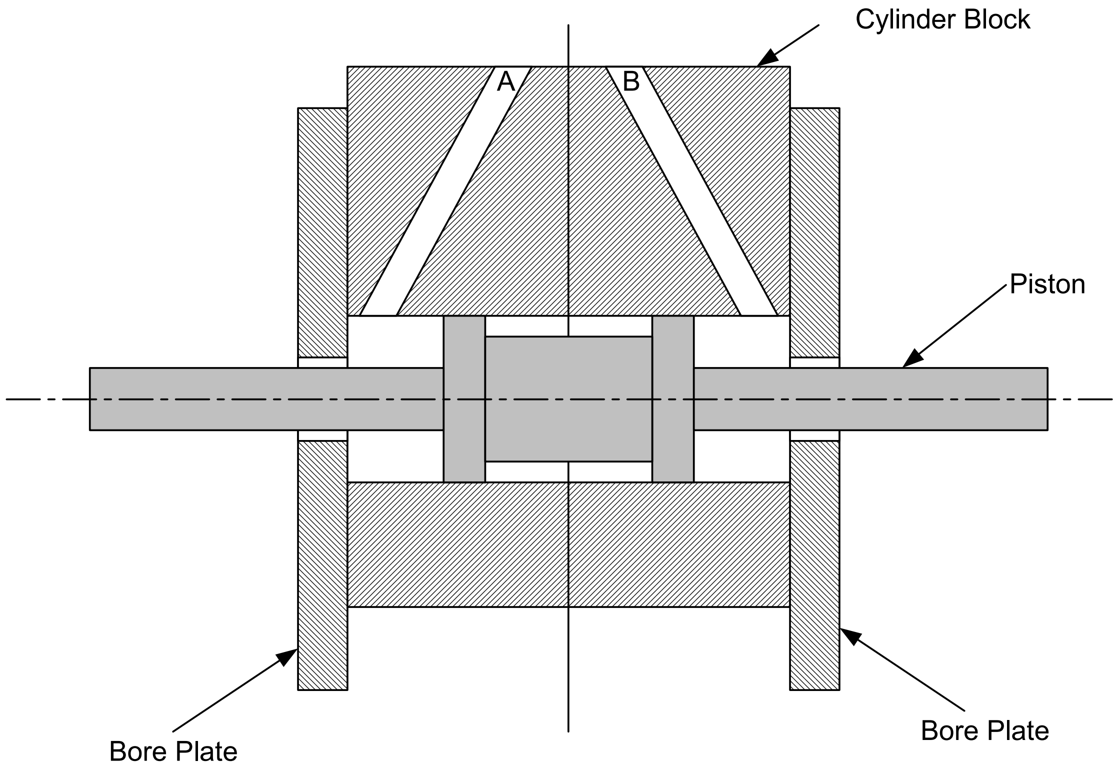

Figure 21 Final Piston – Cylinder Concept

Sealing was the final design aspect to be addressed following the iterative process needed to produce the components shown in Figure 21. It was already decided that o-rings on any of the reciprocating components may introduce too much friction and be detrimental to the design. Therefore, polytetrafluoro-ethylene (PTFE) lip seals were located and integrated into the design of the bore plates.

The lip seals used are a stock item from Fluorocarbon Company Limited, Hertford Herts, England, UK. Therefore, there is a manufacturer recommended design for the gland that was utilized. The manufacturer gland design was added to the bore plate drawings. This seal would prevent hydraulic fluid leaking past the reciprocating interface between the piston rod and the bore plates.

A second static seal was required to prevent leakage from between the bore plates and the cylinder block. This seal was accomplished through the use of an o-ring, and again, its gland dimensions were based on the manufacturer's recommendation.

Finally sealing between the spool valve and the cylinder block was considered. Fortunately, the spool valve has the static o-ring glands integral to its design. Therefore, the design of the cylinder block only needed to have the hydraulic port holes smaller in diameter than the o-rings to affect a good seal.

With the sealing concerns addressed, and the detail drawings completed, the project's focus turned to the manufacture of the components.

3.2 Manufacturing

Due to the tight tolerances of the machining, the project team turned to Siemens DST for manufacturer recommendations. They provided the following list of contacts and stated that considering the microns clearance between the piston and the bore, a limited number of facilities could complete the project.

- C & A Tool, Churubusco, IN; contact Todd Rehrer (219) 693-2167
- Alpha Manufacturing, West Columbia, SC; contact Charlie Hicks (803) 739-4500
- Chucking Machine, Chicago, IL; contact Tim Merrigan (847) 678-1192

Each of the manufacturers were sent a request for quote package that included drawings of the three components and an assembly schematic, the requested lead time, heat treat requirements, and the total number of parts. For this phase, the project team requested three complete prototype units. This includes three cylinder blocks, three pistons, and six bore plates. The request for three was justified by three parameters. First, there was little cost increase between the production of one and that of three, and second, by obtaining three sets, it was more likely that a good working unit would be available. Finally, there was a concern that the learning process associated with the control of the piston translation may lead to damaged components.

Through the quoting process, Alpha Manufacturing had the best combination of price, manufacturing capability, and lead time. Therefore manufacturing of the components began at Alpha Manufacturing in December, 2000.

Due to the tolerances and clearances associated with the piston and cylinder bore, Alpha chose to match-grind components. This resulted in specific pistons that were matched to specific cylinder blocks. The bore plates had less critical dimensions and were interchangeable among the assemblies.

Components were completed in stages, but the first complete assembly was ready in late January, 2001. Upon its receipt, assembly procedures began.

3.3 Assembly

3.3.1 Assembly of the Camless Engine Actuator

The major components that make-up the camless engine actuator are two bore plates, one cylinder block, one piston, and the piezoelectric controlled spool valve. Additional elements include the fasteners, o-rings, and PTFE lip seals.

Assembly went very smoothly aside from the insertion of the lip seals. Because of their small diameter, they were very difficult to install. The manufacturer was questioned, and they confirmed that the gland design was correct. Furthermore, they provided some suggestions for installing the seals. Normally lip seals of this design are to be installed by placing part of the seal into the gland and working the remaining diameter into the assembly by symmetrically pressing with ones thumbs. With the small diameter this was impossible. Instead, the entire unit was deformed simultaneously and pressed into the gland. This was accomplished by two different methods. The first used a piston from the camless engine assembly to support the inside diameter of the seal and force the seal into the gland. Using the piston was abandoned for fear of damaging the unit; instead, the seals were forced by pressing the entire unit with one's thumb into the gland. After one failed attempt, this process proved to be very successful.

The result of the camless engine actuator assembly is shown below in Figure 22. This assembly was then connected to the pneumatic test stand via the mounting plate shown in Figure 15.

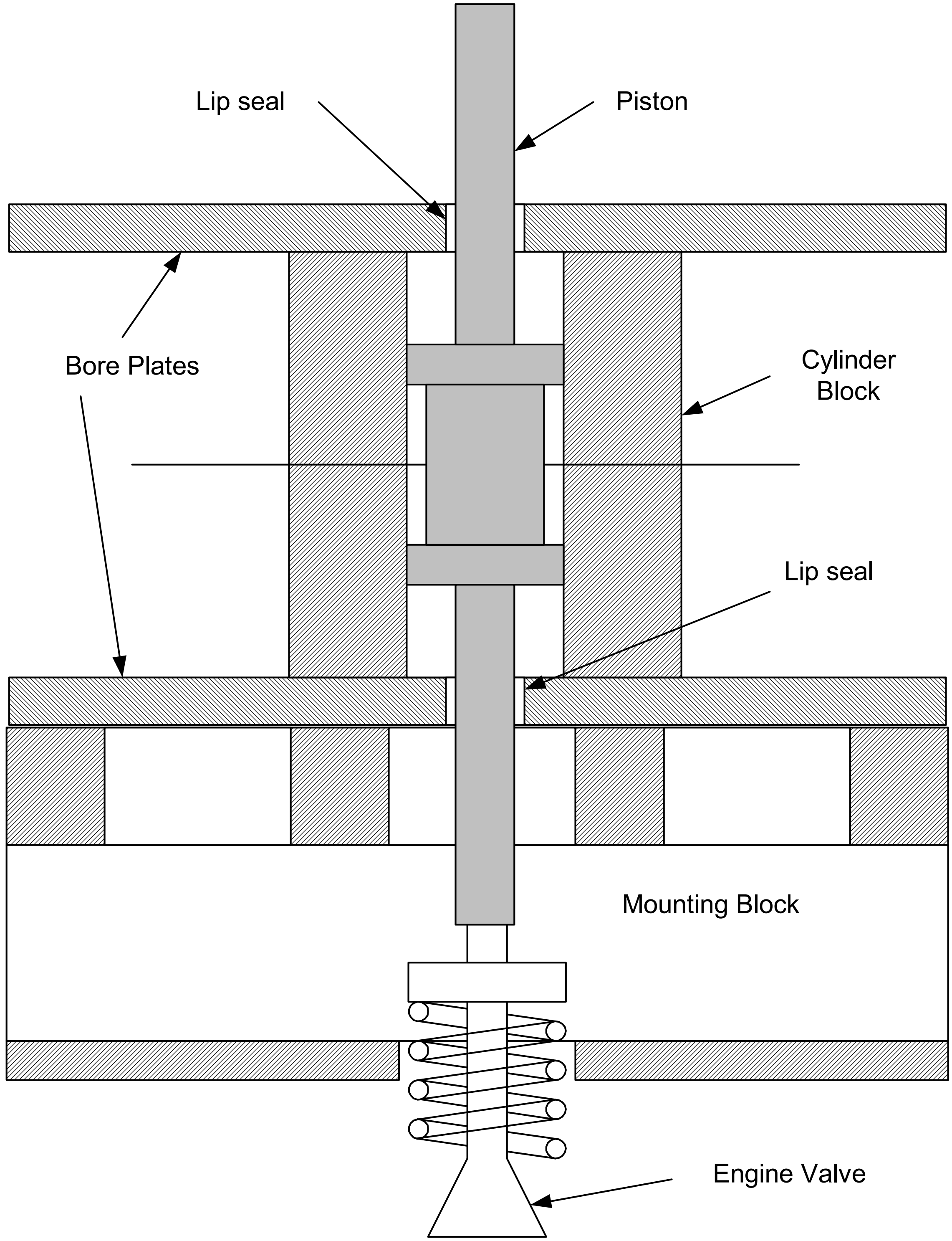

Figure 22 Hydraulic Actuator and Mounting Block Assembly

Figure 22 shows a cutaway view of the camless engine actuator assembly. Because it is cutaway, the spool valve and the ISO 4401 port connections are not visible. In this view, the spool valve would be coming out of the paper toward the reader. A schematic of the assembly from the side can be seen later in Figure 27.

The camless engine actuator assembly and mounting block were then attached to the hydraulic test bench. Hydraulic connections and layout are addressed in the next section.

3.3.2 Assembly of the Hydraulic System

The camless engine actuator assembly outlined in the previous section was mounted onto the hydraulic test stand. Hydraulic connections were made via the standard hydraulic threaded connection ¼ - 19 BSP (British Straight Pipe). The system flows hydraulic fluid from a pump and back to a reservoir and is a self contained scheme.

Hydraulic fluid is pumped through a ball valve and into the side port of the cylinder block. This connection is directly routed to the P port of the spool valve. From there, the position of the spool valve determines where the pressurized fluid goes. In the neutral position, the fluid is dead-headed, and aside from any leakage past the spool, the fluid is static. See Figure 23.

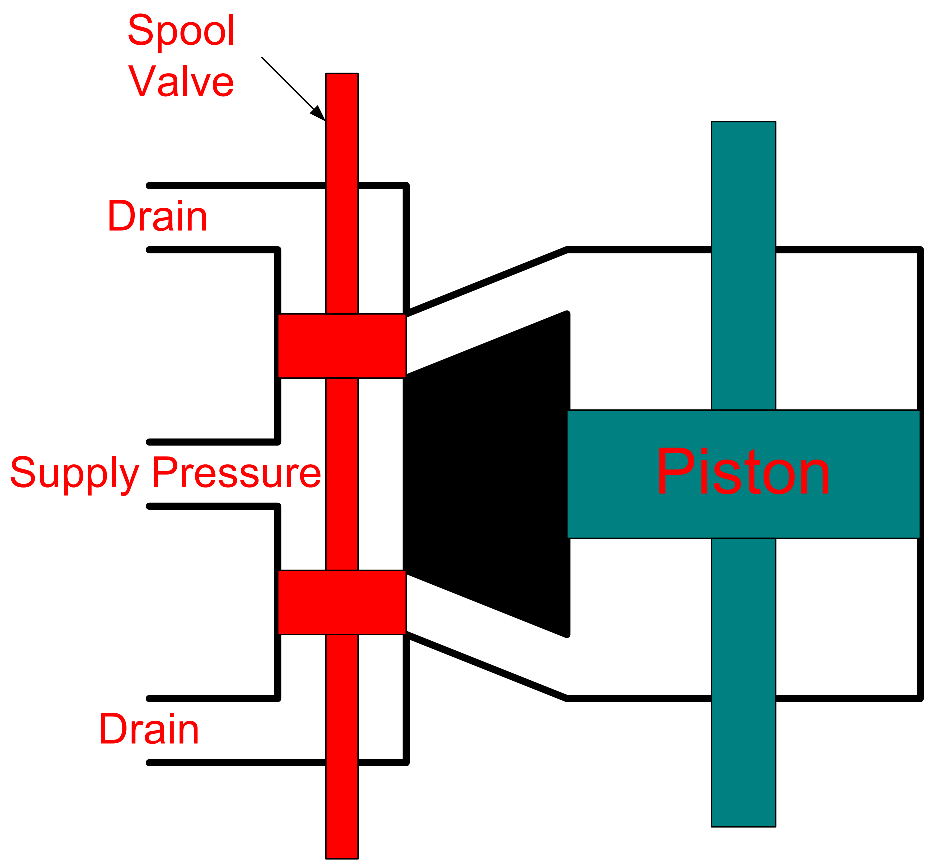

Figure 23 Hydraulic Amplifier Schematic

When the spool valve translates up, fluid flows through the B port and pressurizes the upper cavity of the cylinder block. This pressurization results in the downward translation of the piston. In turn, the engine valve is being opened as the piston translates down. This is shown in Figure 24.

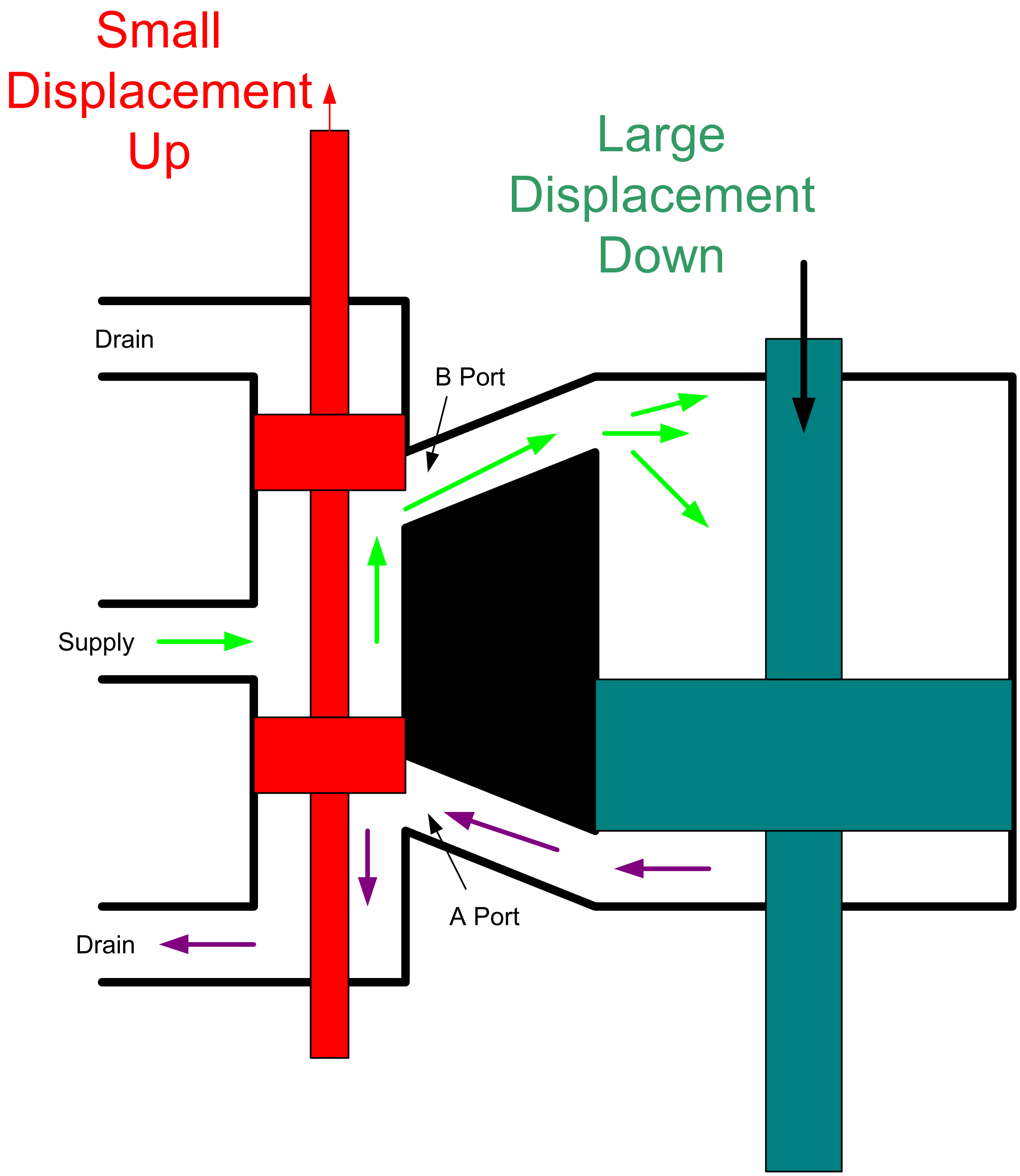

Figure 24 Hydraulic Amplifier – Spool Valve Up

The opposite occurs as the spool valve translates down. Fluid flows through the A port and pressurizes the lower cavity of the cylinder block. This pressure causes the piston to rise and allows the engine valve to close. See Figure 25.

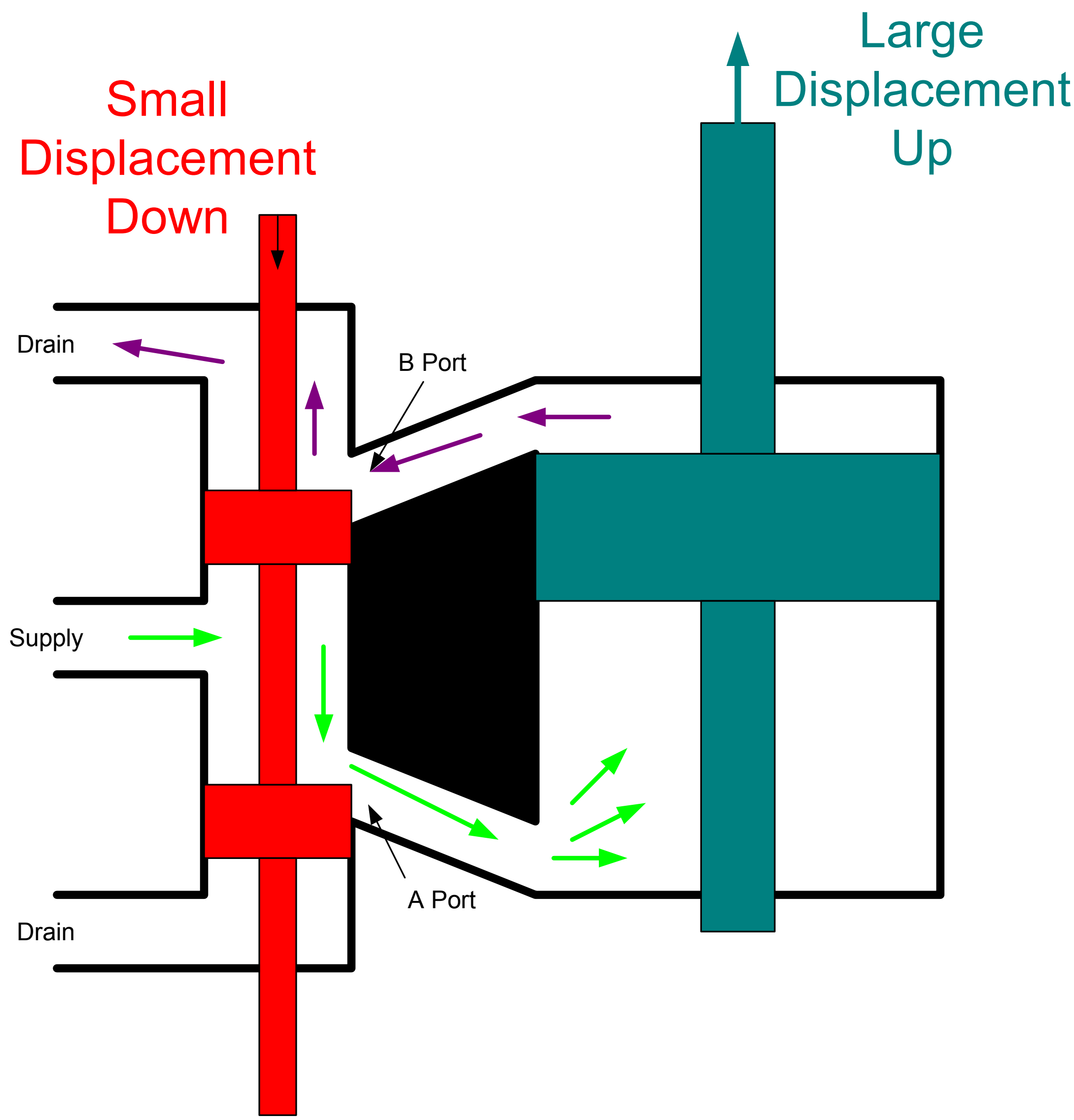

Figure 25 Hydraulic Amplifier – Spool Valve Down

Drainage of fluid from the cylinder block takes place through the A or B port, whichever is not being pressurized by the spool valve. As the piston translates toward the non-pressurized port, hydraulic fluid is forced back into the spool valve. This fluid is then routed directly to the T port (drain) and returns to the reservoir. From the reservoir, the fluid is pumped back into the system, and the process repeats.

In addition to the test rig described above, a second pump can be added in parallel to meet the demands of increased hydraulic flow. This additional hydraulic line has its own isolation ball valve. It is piped in parallel to the primary pump and joins the system just prior to the fluid connection into the supply side of the cylinder block, connected to port P (supply).

The hydraulic flow described above is based on the control signal generated by the electronics and supplied to the piezoelectric stacks. The development of the control system is discussed in the following section.

3.3.3 Assembly of the Control System

The control of the camless engine system is facilitated by a series of electrical components that ultimately supply a variable voltage to piezoelectric stacks. The applied voltage causes expansion of the stacks, and in-turn causes the displacement of the hydraulic spool valve.

Considerable time was spent troubleshooting and gaining an understanding of the high voltage (HV) amplifier[1] for the piezoelectric device. During initial trials, there was no observable voltage output from the amplifier, and the piezoelectric stacks would not respond. Upon disassembly it was observed that some of the electronics had broken free of their adhered connections and had fallen against other components. This caused a short and promptly blew a fuse. After repairing the unit and replacing the fuse, the HV amplifier provided an output to the piezoelectric stacks.

A second problem was observed with the HV amplifier box. Although an output voltage could be measured, it did not have the power needed to provide movement to the piezoelectric stacks. During troubleshooting, the output of the AC to DC converter[2] within the HV amplifier box was found to be outputting 15 volts. This was low compared to the expected 24 V needed to supply the amplifier. Without the 24 V feeding the voltage splitter and amplifier, there was not enough power to actuate the stacks.

Following more investigation, a clue was found printed on the AC to DC converter. It read that the expected input was 240 V AC for an output of 24 V DC. After some rewiring, a nominal 220 V AC was supplied to the unit. The actual output of the 220 V outlet was very high and averaged 250 volts. The input of 250 V resulted in an output of 32 V from the AC to DC converter. This combined with testing at a nominal 110 V input , actually 120 V, and an output of 15 V revealed a constant AC to DC conversion factor. This conversion factor was approximately 1 V DC output to 8 V AC input. Based on this conversion, the required input of 192 V AC should provide 24 V DC output. This was later found to be correct after more wiring changes.

The input requirements of the HV amplifier box was the key element to getting the controls to operate. Originally, the project team was told that the box could operate at both 110 or 220 V input. This was found to be incorrect. The set-up shown in Figure 26 demonstrates the wiring series used to obtain the 24 V DC from the AC to DC converter.

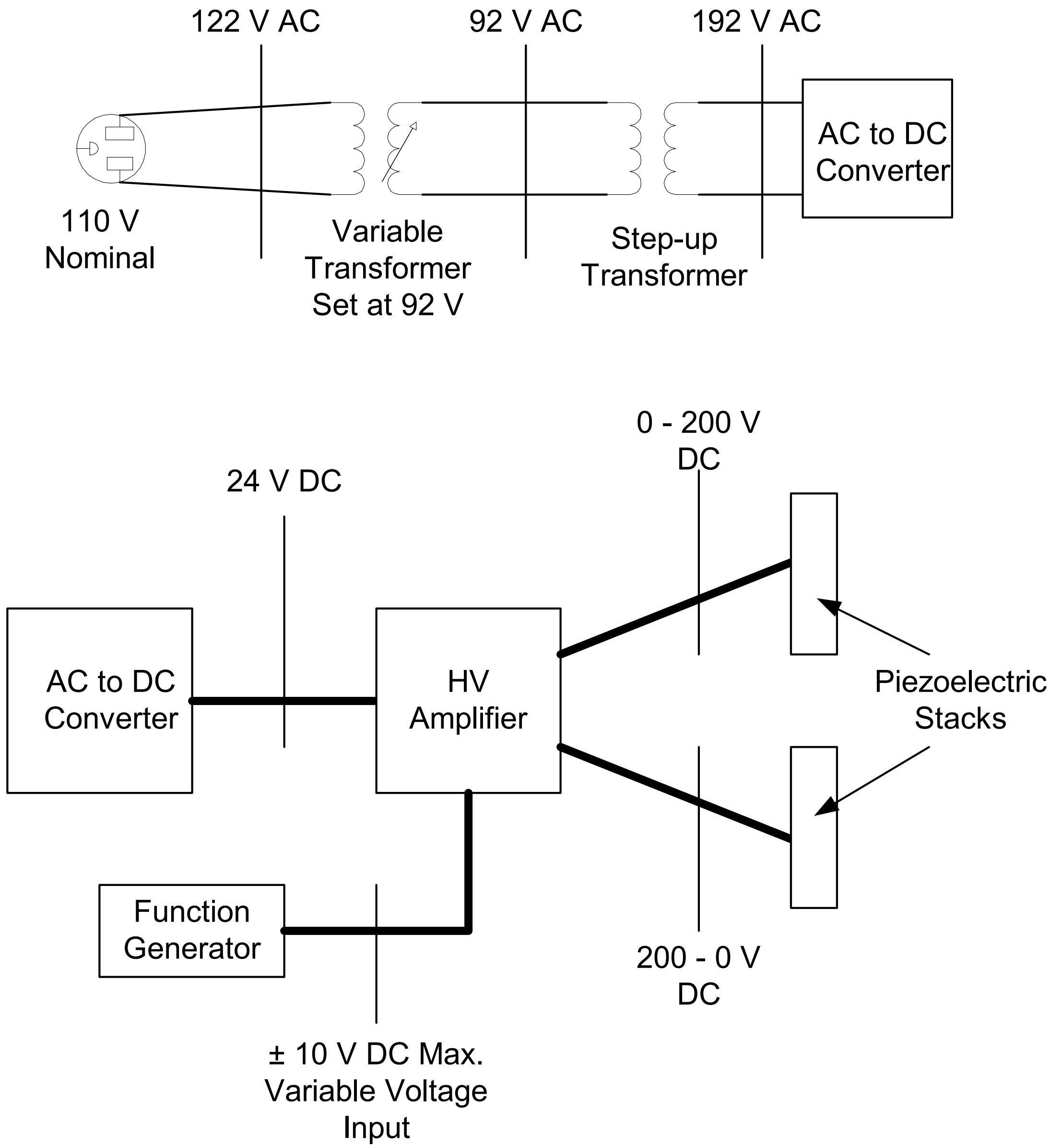

Figure 26 Wiring Schematic

More rewiring was required to supply the needed power to both the controls and the pump simultaneously. This involved making changes from the German (Shuko) plug to an American configuration. Once the rewiring was complete, the entire system was operational. The final form of the test rig wiring is as follows.

- Hydraulics: The pump is wired directly into a 3-pole locking 220 V outlet.
- Electronics: The electronic wiring advances from an outlet at 122 V AC to a variable output transformer to create 92 V AC. This 92 V AC is doubled plus a small offset through a step-up transformer. The resulting 192 V AC supplies the AC to DC converter in the HV amplifier box. The converter supplies 24 V DC to the splitter/amplifier.

  The splitter/amplifier outputs two voltage signals based on a separate input voltage wave. One output signal is 0 – 200 V DC and the other is 200 – 0 V DC. For example, both signals can be at 100 V DC simultaneously, but for one signal at 50 V DC, the other is at 150 V DC. Furthermore, the extreme occurs when one signal is 0 V DC and the other is 200 V DC. These two variable DC signals are based on a separate voltage input supplied by a function generator. The nominal input from the function generator to create the full range 0 – 200 V DC is ± 10 V DC with a 0 V DC offset at variable frequencies.

---

[1] Produced by Marco, Hermsdorf, Germany
[2] Produced by Hydraulik Ring, Nűrtingen, Germany

Chapter Four: Complete System Overview and Operation

Through the effort outlined in the previous chapters, a complete system was created. This system was capable of actuating a single engine valve and met the original parameters outlined for the project. This chapter reviews the hydraulics and controls introduced during the previous chapter and develops their usefulness as part of the complete system.

The goal of the design, troubleshooting, and development of the system during phase one was to actuate a single engine valve. This actuation needed to have a displacement of 10 mm with variable timing up to 50 Hz. Initial qualitative testing revealed several parameters that would require adjustments to reach the project's goal. These tests are outlined in the next chapter; however, some qualitative results are presented here to explain the working of the camless engine system.

A schematic of the actuator, inclusive of the spool valve and piezoelectric stacks is shown in Figure 27. From this schematic, one can see that the two piezoelectric stacks sit on either side of a lever's fulcrum. The expansion of one of the stacks causes the lever to move. For example, if the left piezoelectric stack in Figure 27 expands, the lever moves down. Expansion of the right piezoelectric stack causes the lever to move up. The movement of the lever causes the motion of the spool valve.

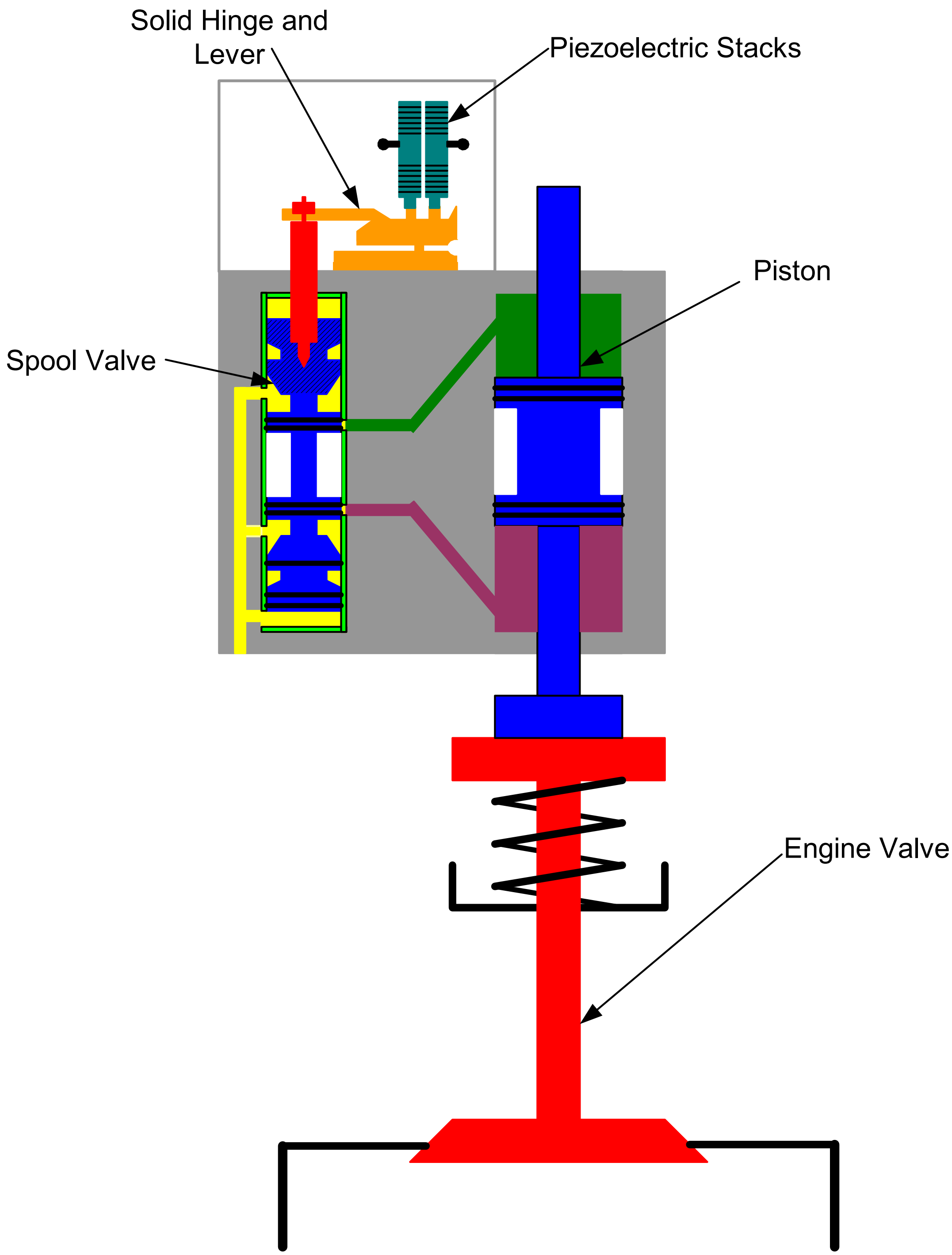

Figure 27 Complete System Schematic

The expansion of the piezoelectric stacks is generated by the input voltage from the HV amplifier Box. As discussed in the previous section, there are two input signals. One input ranges from 0 – 200 V DC, while the other is exactly opposite ranging from 200 – 0 V DC. One signal powers one stack, while the other powers the other stack. Using this arrangement, the two stacks never expand at the same time. The cycling of expansion from one stack to the other allows the lever to oscillate up and down, and therefore, oscillates the spool valve.

The oscillation of the spool valve directs the hydraulic flow path. Based on the spool's position, the fluid can either divert to the top of the adjacent piston or the bottom. This pressurization causes the piston to displace down or up, respectively. Since the piston is in direct contact with the engine valve, its movement is equal to the displacement of the engine valve.

Early qualitative tests showed the engine valve displaced 11 mm with 50 bar input pressure and low frequencies ($\leq$ 15 Hz). This early testing was not designed to provide detailed data; instead, it was part of the discovery process. Adjustments to input pressure, frequency and nominal spool valve position were continuously done until the unit was functioning well. It should be noted that during these early tests, the input voltage frequencies were limited to a simple sine wave with amplitudes varying from 1 to 10 V.

During the qualitative testing and system set-up, it was also observed that the engine valve displacement and/or stroke became smaller with increasing frequency. This led to questions about flow rate limitations. To check whether the limitations were due to the available hydraulic fluid, the input flow rate was doubled by bringing the second pump on-line. The added fluid was mixed into the flow prior to entering the cylinder block.

This additional fluid flow did not alter the engine valve displacement; therefore, it was concluded that the available input flow rate was sufficient. The next logical step was to observe the flow rate through the system. It was here, that formal testing began. The testing is outlined in the following chapter.

Chapter Five: System Testing

5.1 Flow Rate Experiments

Flow rate testing can be divided into multiple phases. For all testing, the experiments were divided into three distinct runs – both pumps, right pump only, and left pump only. Since the pumps themselves are together, the distinction of left and right is based on the location of their isolation ball valve.

Initially, flow rate testing was done to determine the flow output of the pump systems. These measurements were necessary to ascertain the maximum flow rate available to the camless engine system. For these measurements, flow was diverted from the camless system, just prior to its entrance into the cylinder block, into a graduated cylinder. Practice runs were made to determine the length of time to use for each run without overflowing the graduated cylinder. When running both pumps, a time limit of 5 seconds was used for each test. The tests were repeated a minimum of five times to establish consistency in volumetric readings. These readings were established for a variety of pressures. Pump pressures were set with the isolation valves closed.

The following graph (Figure 28) shows the output volumetric flow rate for a variety of input pressures. Except at pressures below 20 bar where the readings became unstable and inconsistent, the available flow rate is relatively constant regardless of input pressure. This is reasonable considering the input pressure could only be established in a static situation, but pumping into the graduated cylinder creates nearly zero pressure increases. The data indicates that the pumps could each generate an unrestricted flow of about 90 cm^3/sec and were additive when run together.

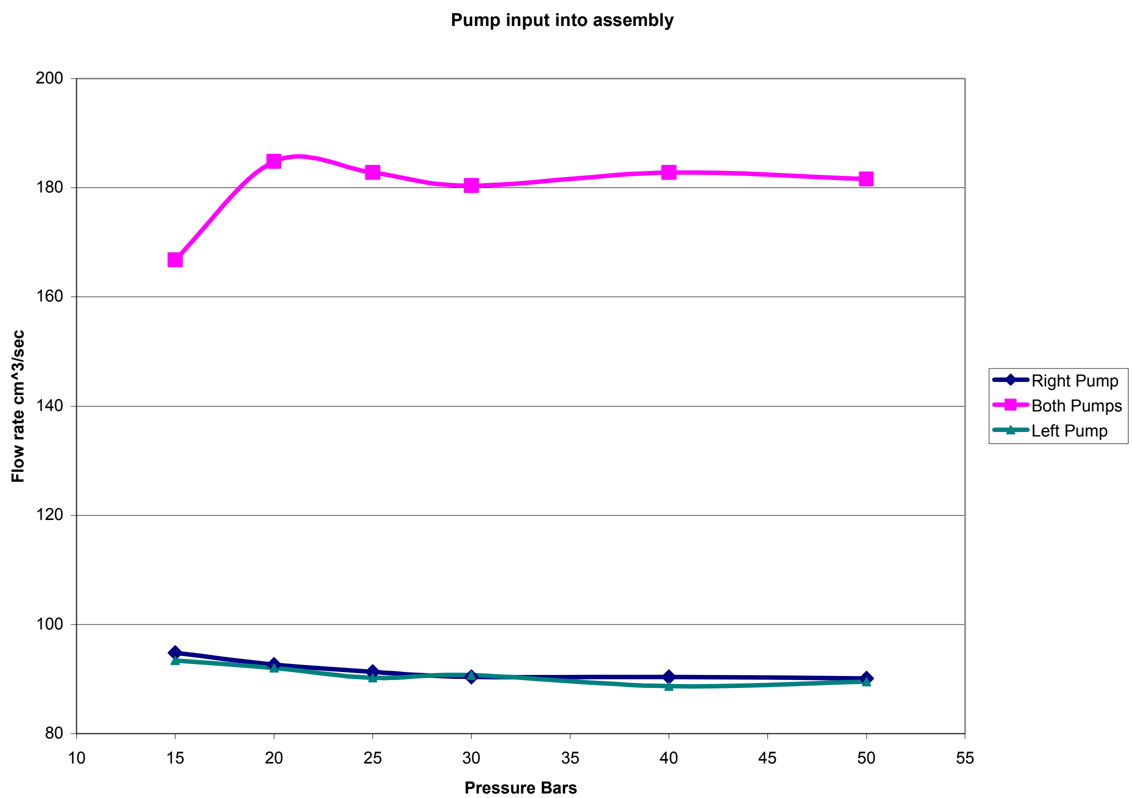

Figure 28 Available Flow Rate from Pumps

The next phase of flow testing gathered data from the output of the camless engine assembly. The graduated cylinder was piped directly to the T port outlet from the cylinder block. Again, fluid was collected for a specific length of time. Instead of running at multiple pressures, a constant anticipated operating pressure of 50 bars was used. In place of pressure variation was the change to piezoelectric input frequency. Therefore, this experiment monitored the output volumetric flow rate of the entire assembly compared to the input frequency of the spool valve control. These results are shown in Figure 29.

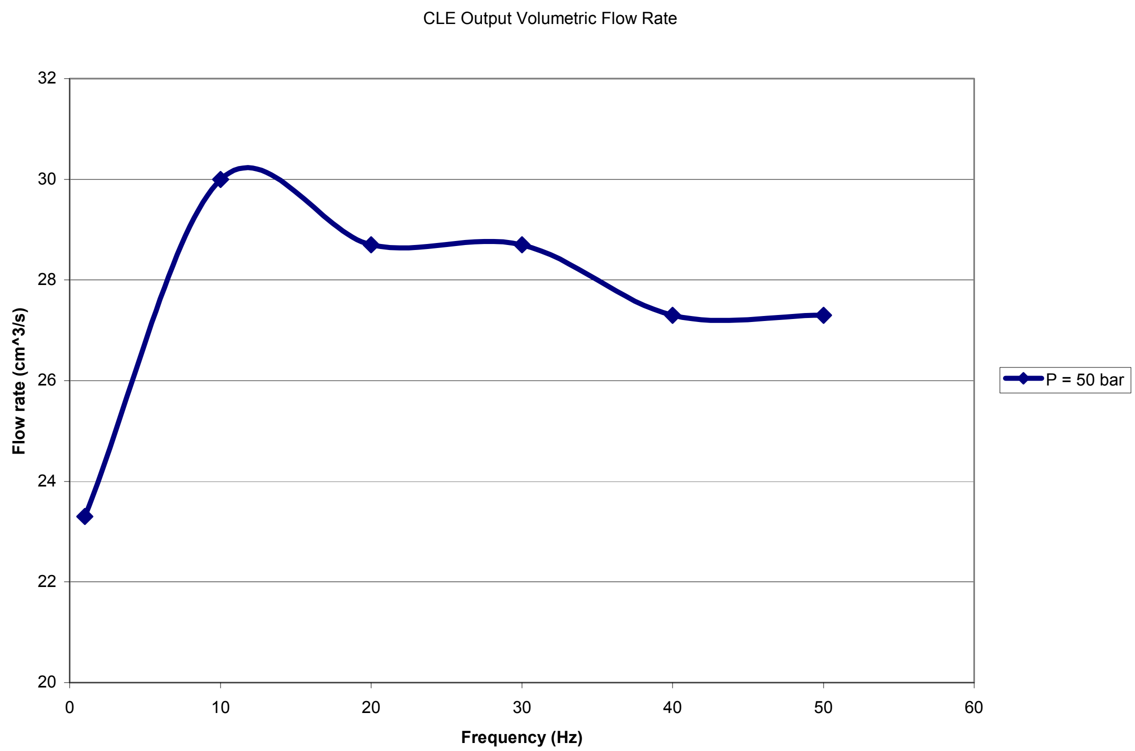

Figure 29 Camless Engine Return Flow Rate

Fluid returning to the reservoir does so at approximately 28 cm^3/sec for input frequencies within the anticipated operating range – those between 15 and 50 Hz. Considering this value, flow rate is obviously restricted by the assembly. This restriction in flow rate is hypothesized to be the reason for reduced engine valve displacement at higher frequencies. This return volumetric flow rate is approximately 1/3 of the available flow rate provided by one pump.

Based upon the findings in the output flow rate experiment, an examination into the volumetric requirements of actuating the engine valve ensued. The design of the cylinder block and piston, combined with the design requirement for valve stroke results in a constant fluid volume requirement. The pressure contact area found in Eq. 4 multiplied by the required stroke of 10 mm results in a volume of 982 mm^3 or 0.982 cm^3. This volume is equal to the volume of fluid required to actuate the engine valve 10 mm in one direction. Twice this volume is required to complete one cycle of both opening and closing the engine valve. Therefore, 1.964 cm^3 must be supplied and evacuated from the assembly during each stroke or period. From this requirement of 1.964 cm^3, volumetric flow rate requirements can be extrapolated for all anticipated frequencies. These data are shown graphically in Figure 30.

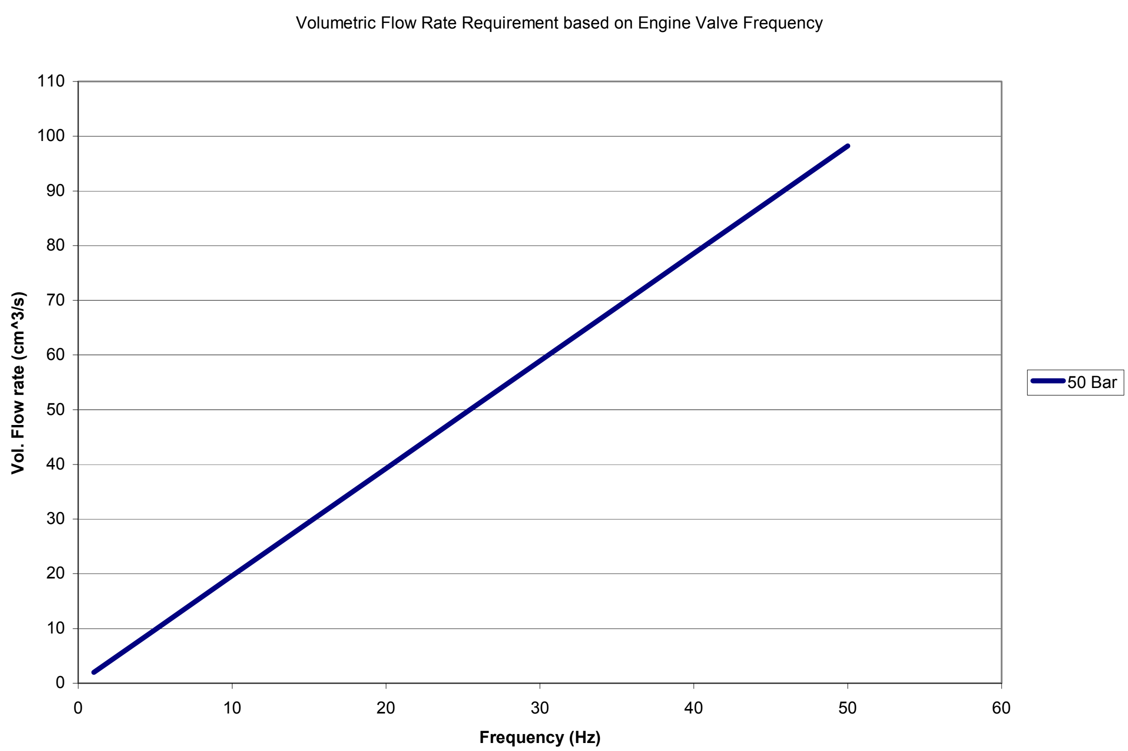

Figure 30 Theoretical Flow Rate Requirements vs. Input Frequency

Considering that the output flow rate through the CLE system was found to be 28 cm^3/sec, the theoretical maximum frequency that the engine valve can obtain full displacement is approximately 15 Hz. For frequencies greater than 15 Hz, the engine valve displacement would reduce. This is due to the conflict between constant flow rate input and higher volumetric flow rate demands due to increased cycles per second.

To verify the theoretical results discussed above, displacement testing was performed on the engine valve. These tests and their results are outlined in the next section.

5.2 Valve Displacement Testing

Displacement testing was designed to monitor the position and the extremes of engine valve movement. For these experiments, an LVDT was attached to the CLE piston. Output values from the LVDT in the form of voltages were monitored using an oscilloscope. The following results were based on monitoring the oscilloscope's output and determining the voltage for both the extreme open and extreme closed position of the engine valve. The voltages were then converted into position measurements based on the voltage vs. displacement calibration curve. From this data, the distance the valve opened and the distance the valve remained open when it was to be closed was determined. Furthermore, the difference of the two positions equals the valve stroke. The results of the 50 Bar test are shown in Figures 31 and 32.

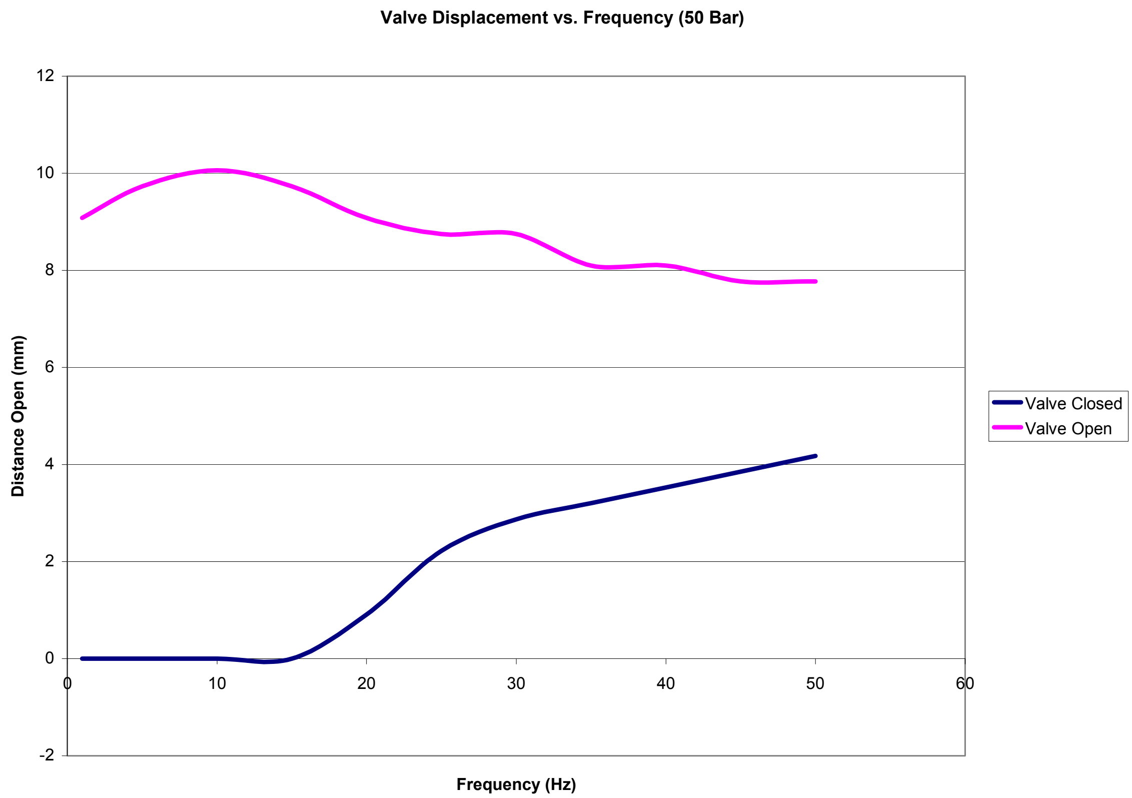

Figure 31 Valve Displacement vs. Input Frequency

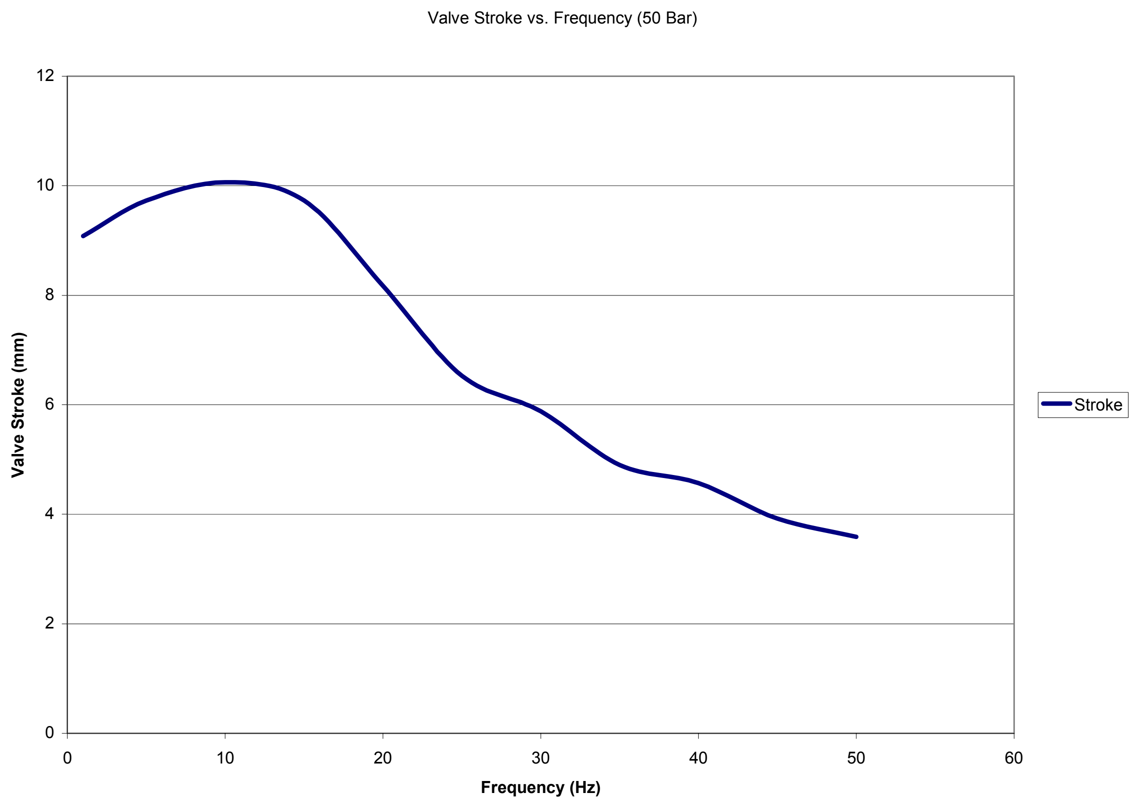

Figure 32 Valve Stroke vs. Input Frequency

Shown in Figure 31, the amount that the valve is open, when it is supposed to be open, decreases steadily with increasing frequencies above 14 Hz. The valve begins to remain open, when it is supposed to be shut, for frequencies above 15 Hz. The amount that the valve remains open also steadily increases with increasing frequency. The results of the changes in displacement for both the open and closed valve position can be seen in Figure 32. This shows that the valve stroke begins to decrease steadily for frequencies greater than 15 Hz.

This experiment supports the calculated maximum frequency for maximum valve stroke. Furthermore, the valve stroke decreases at a nearly linear rate between 15 and 50 Hz. This is also in support of the calculations showing that the volumetric fluid flow remains constant, but as the number of strokes/sec increase (i.e. frequency), the compensation for limited flow is reduced valve displacement.

It should be noted that all the previously discussed testing was done with a sinusoidal voltage input at ± 10 V DC with a 0 V DC offset. Other qualitative testing has demonstrated that changing from a sinusoidal wave to a square wave input dramatically affects the overall operation of the system.

5.3 Spool Valve Calibration

After completing the flow rate and valve displacement tests outlined in the previous sections, it was observed that the spool valve position could be calibrated using the visual output from the LVDT signal. At low frequencies, when the valve remained fully open and closed for a short period of time, the time the valve was fully open could be compared to the time it was fully closed. From this observation, it was clear that the engine valve was remaining fully open much longer compared to the time the valve was fully closed. A schematic of the oscilloscope output of the LVDT signal can be seen in Figure 33.

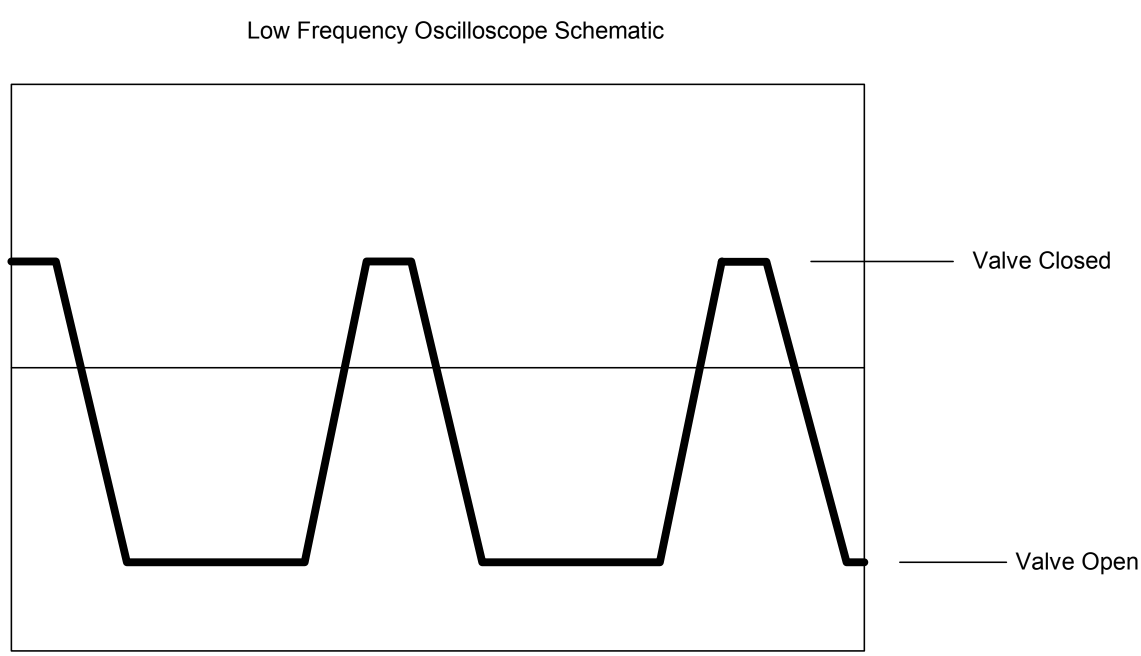

Figure 33 Oscilloscope Output from LVDT prior to Calibration

While monitoring the LVDT signal with the oscilloscope, the threaded rod controlling the position of the spool was manipulated. This process continued until the time fully closed and fully open became equal. The locknut was then tightened to secure the position of the spool. Immediately, the entire prototype showed considerable improvement.

Flow testing was repeated for the output flow rate of the system. These tests indicated a significant increase of volumetric flow rate. The output flow varied between 70 to 85 cm^3/sec based on the operating temperature of the hydraulic fluid.

At low operating temperatures, the displacement testing was repeated. This testing resulted in full valve displacement at 36 Hz using a sine wave input at $\pm$ 10 V DC. This value compares very favorably against the earlier results of 14 to 15 Hz. Furthermore, the values of 70 cm^3/sec and 36 Hz continue to prove the relationship between output flow rate and theoretical maximum full displacement frequency.

Based on the favorable results from the sine wave input, tests were repeated using a square wave input at $\pm$ 10 V DC. This voltage input results in full engine valve displacement through the entire 1 to 50 Hz operating range.

It was also observed that at low frequencies, lower voltage inputs continued to produce full engine valve displacement. These recent finding indicate the system is operating at greater capacity and upholds the theory that custom voltage inputs may create even greater benefits.

5.4 Qualitative Testing and System Observations

As mentioned in the preceding sections, other tests were conducted to observe the response of the system. From these qualitative experiments, some thoughts have been given to future experimentation to quantify these anomalies.

Of greatest interest is the system's response to a change of waveform input to the piezoelectric stacks. Sinusoidal experiments are well documented and outlined in the previous sections; however, the introduction of a square wave input radically changes the system. The square wave applies +10 V or – 10 V suddenly.

This sudden, binary application of voltage causes the spool valve to operate similarly to a solenoid-based system. The result on the system is an increase in displacement all the way to the piston's physical stops. These stops are positioned to prevent the piston from stroking more than 12 mm; therefore, the valve is stroking 12 mm. This impact with the physical stop can be heard, and its noise is similar to other solenoid based camless engine attempts.

The ability to hear the impact and resolve that the valve is stroking 12 mm allows for some audible observation. The noise of impact continues through the entire frequency range from 1 to 50 Hz.

Based on the audible observations, it reasons that the input voltage waveform to the piezoelectric stacks alters the flow characteristics of the system. This is reasonable, considering the input voltage directly affects the positioning of the spool valve.

Therefore, it reasons that the design of a custom waveform will allow for the quiet control associated with the sinusoidal input, while generating the full valve stroke over a greater range of frequencies.

Chapter Six: Discussion

This development has proven the concept of replacing an internal combustion engine's camshaft with a piezoelectric controlled, hydraulic actuator. Valve stroke requirements have been achieved, and, at some input variations, have been exceeded. The actuator has also performed at frequencies well in excess of 50 Hz. Testing has shown some limitations related to the flow rate and input waveforms. However, it is these limitations that have provided the inspiration for continued development and discovery.

Progress on the camless engine can continue into a second phase by investigating the following. First, a detailed look at the voltage input waveform's effect on the operation of the system is in order. From this investigation, a new custom waveform can be designed and generated by the existing equipment.

The two extremes of the waveforms have been already tested; namely the sine wave and the square wave. Each provide some beneficial operating parameters. Therefore, it is conceivable that the combination of the two will result in a beneficial blend of their advantages.

A second design phase may also include a new approach to the overall system. Of prime concern is the reduction of size. Thought has already been given to creating smaller components and utilizing the developing piezoelectric technology under the

"Thunder" model name. The initial concept is to sandwich a spool between two Thunder piezoelectric devices and integrate the hydraulic amplifier into the engine valve.

Of course, an obvious future step is to incorporate the piezoelectric/hydraulic system into a working engine. This will take the project far beyond its initial scope, as the analysis can shift to examining the impact of a camless valvetrain on an operating engine.

Regardless of the direction that the project takes, this phase can be considered successful. The working unit proves that camless engines based on piezoelectric control are feasible. This phase alone makes a broad leap into the automotive and actuator technology of the future. It stands apart from previous attempts and has shown a new way forward.

Appendix

References

Anderson, M; Tsao, T-C; and Levin, M., 1998, “Adaptive Lift Control for a Camless Electrohydraulic Valvetrain,” SAE Paper No. 981029

Anonymous, “First Camless Truck Reaches the Summit of Pikes Peak!” Power Stroke Registry, Spring 2001

Ashhab, M-S; and Stefanopoulou, A., 2000, “Control of a Camless Intake Process – Part II,” ASME Journal of Dynamic Systems, Measurement, and Control – March 2000

Ashhab, M-S; and Stefanopoulou, A., 2000, “Control-Oriented Model for Camless Intake Process – Part 1,” Transactions of the ASME Vol 122, March 2000

Dopson, et al “Lotus Variable Valve Train Systems for Production and Development.”

Dobson, N. and Muddell, G., 1993, “Active Valve Train System Promises to Eliminate Camshafts,” Automotive Engineer February/March 1993.

General Motors – GM and the Environment. May 21, 2001. General Motors. June 13, 2001 <http://www.gm.com/company/gmability/environment/gm_and_the_env/releases/displacement_052101.htm>.

Gould, L; Richeson, W; and Erickson, F., 1991, "Performance Evaluation of a Camless Engine Using Valve Actuation with Programmable Timing," SAE Paper No. 910450.

Kim, D; Anderson, M; Tsao, T-C; and Levin, M., 1997, "Dynamic Model of a Springless Electrohydraulic Valvetrain," SAE Paper No. 970248

Ladd, D; Camless Engine is Gaining Momentum. September 13, 1999. Siemens Automotive. July 4, 2000 <http://media.siemensauto.com/mediacenter2/queries/releasefull.phtml?prjob_num=1195>.

Lamm, J.; "V-8-6-4, Cadillac Tries Multi Displacement" Road & Track November, 1980

Lexus – Variable Valve Timing a First in an SUV. Autoworld. June 18, 2001 <http://www.autoworld.com/RX300.htm>.

Mori, Kaz. Honda's High-Output LEV Engine Home Page. Honda. June 13, 2001 <http://s2000.vtec.net/jdmhtml/high-output-lev.htm>.

Murray, C; Electronic Controls to Replace Camshafts in Diesel Engines. April 14, 2000. EETimes.com. July 7, 2000 http://www.eetimes.com/story/industry/system_and_software_news/OEG20000414S0050.htm>.

Richman et al "A Computer-Controlled Poppet-Valve Actuation System for Application on Research Engines."

Schechter, M.; and Levin, M., 1998, "Camless Engine," SAE Paper No. 960581