DEVELOPMENT OF A LINEAR CONTROL MODEL
FOR A CAMLESS VALVETRAIN

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DEVELOPMENT OF A LINEAR CONTROL MODEL FOR A CAMLESS VALVETRAIN

A THESIS

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By

Domenic R. Albert

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DEVELOPMENT OF A LINEAR CONTROL MODEL FOR A CAMLESS VALVETRAIN

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ABSTRACT

The disciplines of mechanical and electrical engineering have been merging slowly over the last few decades. More specifically, the automotive industry has reaped the benefits of using electronics to improve upon the economics and ergonomics of business as well as product. Although electronic systems have proved to be useful throughout consumer vehicles, the engine is arguably the area that benefits the most.

Essentially, efficiency and power output of an internal combustion engine can be improved by optimizing the airflow of the cylinders. Since this airflow is controlled by the valve train, their manipulation affects the overall performance of an engine. Additionally, mechanical power losses can be reduced by simplifying the components of the valve train. The focus of this experimentation is to prove that electronic motors can be utilized to mimic the function of timing and actuation devices such as belts, pulleys, and gears. Due to the absence of mechanical constraints, the potential for varying valve timing with motors, rather than a plethora of mechanical additions, is substantially greater. Coupled with the advantages of unconventional rotary valves, the technology of camless engines has new insight.
ACKNOWLEDGEMENTS

First and foremost, I would like to thank God for giving me the ability and opportunity to accomplish the things I have in my life. I would also like to thank my family, especially my mother, for presenting me with events that have undoubtedly shaped me into the man I am today.

"Commit your work to the Lord, then it will succeed."

-Proverbs 16:3
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CHAPTER I
BACKGROUND OF SYSTEM ELEMENTS

Introduction

Otto-cycle internal combustion engines are the most common engines found in automobiles. Put simply, the purpose an internal combustion engine is turn linear motion into rotational motion and produce useful power. The process of an Otto-cycle spark-ignition internal combustion engine is as follows:

1. An intake valve opens and the piston moves from top dead center (TDC) towards bottom dead center (BDC) causing an air-fuel mixture to be drawn into the cylinder.
2. When the cylinder reaches BDC, the intake valve(s) close and the piston begins to compress the air-fuel mixture by moving back toward TDC.
3. When TDC is reached, the compressed fluid is ignited by a spark plug and the piston is forced downward to BDC by the burning, expanding gas.
4. When BDC is reached, exhaust valve(s) are opened. The piston begins to move back towards TDC therefore forcing the burned air-fuel mixture to be evacuated from the cylinder.
5. Above steps are repeated in a continuous cycle.

The air-fuel mixture has a significant effect on the combustion process noted by the two variations of Otto-cycle engines; spark-ignition (SI) and compression-ignition (CI). Historically, SI has been the predominant process used in the automotive industry mainly because of performance and economic advantages. Recent development with CI engines has brought them to performance levels nearly equivalent to that of their
counterparts. The main difference between these processes is that the combustion process in CI engines is triggered by the injection of diesel fuel directly into the cylinder at the top of the compression stroke. Thermodynamic properties instantaneously cause the fuel to ignite therefore eliminating the need to mix air and fuel prior to the induction process. Since they do not suffer from factors such as flow losses in the carburetor and throttle, intake manifold heating, or the presence of fuel vapor, CI engines have a fuel economy significantly greater than comparable SI engines but often suffer in the areas such as acceleration. Nevertheless, in both cases the combustion process is optimized by the management of the air-fuel mixture introduced into the cylinder. Due to the continuing variance of environmental conditions, controlling the composition depends on variables such as the humidity of ambient air. Consequently, especially in diesel engines, controlling the airflow of the cylinder is the key to optimizing fuel efficiency.

In order for the cycle to function air and gas must be introduced and evacuated from a cylinder at the prescribed times. This process is the responsibility of a valve train. Throughout the history of internal combustion engines several variations of valves have been introduced; most notable are poppet and rotary valves. In terms of the poppet valve, recent innovation has included ways of variable actuation to alter performance. When looking to the future design of a system it is important to review the successes and failures of past designs.
Valves

An internal combustion engine is basically a glorified air pump, so obviously the best way to increase performance is to improve the way it processes air. Because valves are the gateways into and out of the combustion chamber, it is instinctive to concentrate on valves and valve timing in order to achieve optimum efficiency and performance.

Poppet Valves

Over the last century, poppet valves have dominated the world of valve trains for internal combustion engines. The actuation of poppet valves is most commonly controlled by a camshaft, which is basically a rod manufactured with multiple lobes, which is connected to a pulley rotated by a timing belt. As the camshaft rotates the lobes open and close the intake and exhaust valves relative to the position of the piston. Contact between the valves and camshaft is maintained through the use of springs (images may be found in Appendix V for clarity). The cam profile, or shape of the cam lobe, ultimately determines the distance the valve moves as well as the duration of time the valve is opened. Consequently, there is a direct relationship between the shape of the cam lobes and the way the engine performs through various frequencies.

There are a few major problems with this system. First, the maximum speed of the engine is restricted by the maximum speed at which the valves can be actuated. Since valve-cam contact is ensured by springs, their stiffness plays a significant role in the ability for an engine to rev high. Additionally, fluid flow is optimized by either increasing the area of the valve opening or increasing the maximum distance the valve opens. The former requires considerably more force to be applied by the cam simply
because the act of increasing the surface area of the valve creates more pressure against it inside the cylinder. The latter is accomplished by simply altering the cam profile but, unfortunately, every rpm range requires a certain cam profile in order to optimize performance. Nevertheless, both changes require an altered camshaft and an increase in the mechanical energy necessary to run the system.

Also, valve wear is near impossible to avoid because the valve head is continuously slamming into the port and the valve stem is continuously subjected to friction. Automakers have overcome the former drawback mainly through the use of proper springs and developing technology to allow soft seating. Valve train friction however can be combated in various ways. Although over-head cam engines have become extremely popular, especially in compact engines, it is not uncommon for engine designers to incorporate rocker arms with fulcrum needle bearings in an effort to reduce friction, rather than direct contact between the valve and the cam. By using roller followers the large frictional loss due to the cam/lifter interface can be improved despite the high loads over small contact areas. Although the same success can be had with the use of single cam pushrod designs, where the camshaft is located in the cylinder block, both approaches require the addition of mechanical devices between the essential components, the camshaft and valves, and ultimately increase overall cost. It is also important to note that the reduction of valve mass and spring load have a positive effect on frictional reduction. When dealing with these elements concessions must be made due to the effect that valve size, which affects weight, and spring stiffness have on engine
performance, as stated earlier. Obviously, lubricants are used to reduce friction on the valve stem but, due to the design of poppet valves, they inevitably make their way into the combustion chamber and add to emitted pollutants.

Ultimately, engine parameters are directly affected by valve train design or vice versa. For example, there is a direct relationship between cylinder bore size and the diameter of the poppet valve head. Engine designers must consider how many valves will be used per cylinder, the proportional size of intake to exhaust valves, combustion chamber geometry, and spark plug/fuel injector location. The selection of these parameters depends largely on the preferred combustion characteristics. It is common to make the valve heads as large as possible in order to maximize performance and minimize pumping losses. Essentially, both the combustion characteristics and valve head dimensioning depend on the cylinder head layout. Common layouts include disc, hemispherical, open, and bowl-in-piston designs. Consequently, when an engine designer selects a cylinder head layout it causes somewhat of a domino effect on all other parameters. Additionally, technologies such as variable displacement engines further restrict the displacement allocation of poppet valves due to the change in cylinder head geometry that takes place during operation. When it comes down to making final decisions for a given engine design it is these concessions which ultimately limit the ability for poppet valves overall efficiency and performance.
Rotary Valves

A rotary valve is like a camshaft with holes instead of lobes. Like an overhead cam, a rotary valve is positioned within what is called a cylinder head (named because it is positioned directly over the cylinders). Flow of gas is interrupted until the shaft is turned to the appropriate angle to expose an opening or hole. When an opening is exposed, fluid flows between the cylinder and the intake or exhaust manifolds in the same manner as conventional engines with poppet valves. Appendix V includes multiple illustrations of rotary valve variations.

The rotary valve has many advantages over the poppet valve. First, poppet exhaust valves get extremely hot from the passing combusted products and consequently require the use lower compression ratios. In rotary valve engines, the combustion flame is not in constant contact with the same valve surface. Heat from the propagating combustion flame is not allowed to accumulate because the rotary valve is a rotating shaft. As the shaft rotates, it travels out of the combustion area and onto a surface within the cylinder head where the heat is transferred to the cooling system. Because of this, the spark plug can detonate much closer to or just after TDC and higher compression ratios can be used. In essence, higher compression ratios increase power and thermal efficiency.

The breathing advantages of rotary valves are quite simple. The mass flow rate into and out of the cylinder is drastically increased since fluid is allowed to flow without any obstruction. Because there is no valve head, the choking phenomena that occurs with poppet valves at high engine speeds is avoided. This is very important because choking
is severely detrimental to volumetric efficiency. Additionally, fluid flow is less turbulent since fluid flow does not have to fight with the edges of the valve seat and head.

Yet another advantage of rotary valves is seen in the area of wear and noise. As described in the previous section, the contact between the camshaft and actuation device for the poppet valve is continuously exposed to friction. Also, when the valve returns to its closed position it must come into contact with the valve seat. Downfalls of such operation are revealed by wear and noise. With rotary valves however, there is no need to use rotary motion to actuate linear devices. Therefore, with designs utilizing bearings, the above issues are addressed and improved upon by eliminating the need for lubricants in the valve train. Ultimately, the result is an engine that runs smoother with a broader power band and quieter operation.

Lastly, power is partly wasted in the mechanical motion needed to run the engine. Other than the loss of energy due to friction of the piston and crankshaft, power is consumed by the belt which operates supporting engine components such as the alternator, fuel pump, and camshaft. Significant power is consumed by the motion of the camshaft, rocker arms, springs, and simply the inertial forces that contribute to controlling poppet valves. By reducing the number of required moving parts rotary valves are able to reduce this problem significantly. Cost savings result from manufacturing a cylinder head requiring fewer parts and eliminating the need for cooling and lubricating fluids to be pumped for operation. The simplicity of a rotary valve
assembly requires less maintenance and allows more opportunity for advanced control systems.

**Variable Valve Timing**

There are many factors that go into the design of a valve train that ultimately affect the performance of an engine. After the physical aspects of the valve train, such as dimensioning of the valve head, seat, and stem, have been finalized it is critical to schedule the time at which valve events happen. More specifically, each valve has a finite opening and closing time which effects when significant airflow through the valve-open area either starts or ceases. Since this open area is directly related to the geometric details of the valve train, there are separate stages to the development of flow into or out of the cylinder.

It is common for intake valve opening (IVO) to occur 10 to 25° before TDC in order to ensure that the cylinder pressure does not decline in the early stages of the intake stroke. Intake valve closing (IVC) usually begins 40 to 60° after BDC, to provide more time for cylinder filling under conditions where cylinder pressure is below the intake manifold pressure. Although engine performance is not severely affected by the prescribed point for IVO, IVC is one of the primary factors determining the volumetric efficiency at high engine speeds. Additionally it contributes to low-speed volumetric efficiency due to backflow into the intake system. Exhaust valve opening (EVO) starts 50 to 60° before BDC to allow a phenomenon called blowdown to assist in expelling the exhaust gases. Blowdown simply describes the suction of burned fuel through the
exhaust port as a result of the cylinder pressure being higher than that of the exhaust manifold. This action brings the pressure of the cylinder close to that of the manifold soon after BDC, where the upward movement of the piston works to expel remaining gases. The timing of EVO also determines the effective expansion ratio which is a contributor the efficiency of the cycle. Exhaust valve closing (EVC) occurs between 8° to 20° after TDC and is the key factor in determining valve overlap duration and the amount of exhaust which flows back into the chamber due to the influence of the vacuum created by the intake manifold. It is important that EVC occurs adequately after TDC to ensure that cylinder pressure does not rise near the end of the exhaust stroke [5].

Traditionally, during the onset of engine design timing points were determined based off the decision of whether the vehicle to be powered would be high performance or highly efficient. This decision had to be made because the orientation of valve events, as well as the valve lift, were “set in stone” for the most part and therefore could only be optimized on one end of the spectrum. For example, there is a ram effect that occurs at the end of induction which helps volumetric efficiency and power. Since this effect gets progressively greater as engine speed increases, late intake valve closing allows full advantage to be taken of this increased charging. However, at low engine speeds more backflow is created into the intake system if the intake valve is not closed soon after BDC and volumetric efficiency suffers. In fact, early IVC helps volumetric efficiency at low engine speeds because it reduces the amount of fresh charge lost in the cycle. Similar effects can be seen in the exhaust process. For instance, late EVC favors high power at
the expense of low-speed torque and idle combustion quality [5]. Figure A shows the effect of timing and lift over a given engine range.

Due to computing power it is possible to set up control systems that can eliminate some of the concessions previously inherent in valve train design. Ideally, automakers aim to develop systems which use Continuously Variable Valve Timing (CVVT) in order to utilize optimal airflow characteristics over the entire rpm range. Over the last few decades automakers have only utilized equipment in the development stages that are capable of camless CVVT to select the best cam profiles and phases for a specific application. This is primarily due to the fact that machines capable of such performance
have been too large and expensive to be used on consumer products. However, International Truck and Engine Corporation developed and is currently testing a camless CVVT system they plan to assimilate on all their new diesel engines by 2010. Otherwise, nearly all automotive companies utilize an electromechanical system with cams to vary valve timing and/or lift but have failed to incorporate truly CVVT systems in consumer vehicles.

Using advanced systems to alter the opening and closing of valves can produce engines which are cleaner and produce more power than their counterparts. Due to the radical change in an engine’s airflow dynamics between the 2,000 and 6,000 rpm range, it is impossible for traditional poppet valves to achieve optimal efficiency. However, by coming up with a way to alter valve timing between high and low rpms, automotive companies have gained the ability to tune valve operation for considerable improvement in performance and efficiency throughout the entire rev range.

One example of such technology is the Variable Valve Timing and Lift Electronic Control (VTEC) system used by Honda. As shown in Figure B, VTEC is comprised of two camshaft profiles in which one profile is designed for low to mid range rpm and the other for high range rpm. A computer-controlled electronic switch is used to shift to the specified profile at specific rpm in order to improve both horsepower and torque. Over time, Honda has developed various versions to include up to three different cam profiles. For the most part, however, the process of operation remains the same.
A VTEC engine when tuned for performance will allow a 150 horsepower four
cylinder engine without VVT achieve 190 horsepower or more with the technology. This
power is only realized in the upper rpm's when the cam profile is varied from fuel
efficient to performance level driving. Sometimes this switch in valve timing is
pronounced by the "VTEC growl", a louder engine sound when the car has changed how
it is distributing fuel and air to the engine.

Honda has been so successful with VTEC over the last decade that the technology
has spread from the Acura NSX to nearly every model. The success of Honda’s VVT
technology has also inspired Toyota join the bandwagon with a different approach.
Instead of the on/off system that VTEC employs, Toyota created a continuously variable
system to maximize valve timing throughout the rpm range. Named Variable Valve
Timing with intelligence (VVTi), Toyota uses a hydraulic rather than mechanical system
to alter the intake cam's phasing. The main difference between the two systems is that
VVTi maintains the same cam profile. Although Toyota found success by altering when
the valves open and close in relation to engine speed, the system is still limited to phases
specified by a few desired performance levels.
Perhaps the most complicate system to date is BMW’s VANOS (Variable Onckenwellen Steuerung). Although it uses a plethora of mechanical components, VANOS has helped BMW maintain its status as one of the best engine makers in the world. However, like most automakers of the western hemisphere, BMW utilizes a system that is theoretically closer to VVTi than VTEC.
CHAPTER II
LITERATURE REVIEW

The essence of this project is to design a system in which the variations in valve timing are truly changing continuously. As stated in the previous section, volumetric efficiency is improved at higher engine speeds if the intake valve is held open up to 60° after BDC. However, based off the premise of current systems, the initiation of valve closing is only optimized at a certain rpm. For instance, take an engine that redlines at 8000 rpm. Engine designers may choose an EVC point of 50° after BOC for high speed operation; yet, this may only be the optimal setting for the engine operating at 7500 rpm continuously. As illustrated by http://www.bmwworld.com/technology/vanos.htm, current variable valve timing systems change operation methods over only three different intervals:

VANOS operates on the intake camshaft in accordance with engine speed and accelerator pedal position. At the lower end of the engine-speed scale, the intake valves are opened later, which improves idling quality and smoothness. At moderate engine speeds, the intake valves open much earlier, which boosts torque and permits exhaust gas re-circulation inside the combustion chambers, reducing fuel consumption and exhaust emissions. Finally, at high engine speeds, intake valve opening is once again delayed, so that full power can be developed. [1]

Additionally, current systems are still predominately mechanical. The only electronics involved are related to collected and calculating data from sensors to send a
signal to pneumatic actuating devices. These actuating devices are still mechanical and are therefore governed by mechanical laws. Mechanically altered systems require additional components which make the valve train heavier and overly complicated. In addition to this being a downfall to initial cost, it also makes descriptions of system operation confusing without the use of detailed drawings or physical demonstration:

The crankshaft drives a sprocket on the exhaust cam, and the exhaust cam sprocket is bolted to the exhaust cam. A second set of teeth moves a second chain that goes across to the intake cam. The big sprocket on the intake cam is not bolted to the cam, for it has a big hole in the middle. Inside the hole is a helical set of teeth. On the end of the cam is a gear that is also helical on the outside, but it's too small to connect to the teeth on the inside of the big sprocket. There is a little cup of metal with helical teeth to match the cam on the inside and to match the sprocket on the outside. The V (Variable) in VANOS is due to the helical nature of the teeth. The cup gear is moved by a hydraulic mechanism that works on oil pressure controlled by the DME.

At idle, the cam timing is retarded. Just off idle, the DME energizes a solenoid which allows oil pressure to move that cup gear to advance the cam 12.5 degrees at midrange, and then at about 5000 rpm, it allows it to come back to the original position. The greater advance causes better
cylinder fill at mid rpms for better torque. The noise some people hear is the result of tolerances that make the sprocket wiggle a bit as the cup gear is moved in or out. [1]

When variable valve timing is used for intake and exhaust valves there are a few advantages and disadvantages. Most apparent to the consumer is the improvement in drivability which comes at the expense of the energy needed to make the system function properly.

Double-VANOS (double-variable camshaft control) significantly improves torque since valve timing on both the intake and outlet camshafts are adjusted to the power required from the engine as a function of gas pedal position and engine speed.

On most BMW engines that use a single VANOS, the timing of the intake cam is only changed at two distinct rpm points, while on the double-VANOS system, the timing of the intake and exhaust cams are continuously variable throughout the majority of the rpm range.

With double-VANOS, the opening period of the intake valves are extended by 12 degrees with an increase in valve lift by 0.9 mm.
Double-VANOS requires very high oil pressure in order to adjust the camshafts very quickly and accurately, ensuring better torque at low engine speeds and better power at high speeds. With the amount of unburnt residual gases being reduced, engine idle is improved. Special engine management control maps for the warm-up phase help the catalytic converter reach operating temperature sooner.

Double-VANOS improves low rpm power, flattens the torque curve, and widens the powerband for a given set of camshafts. The double-VANOS engine has a 450 rpm lower torque peak and a 200 rpm higher horsepower peak than single-VANOS, and the torque curve is improved between 1500 - 3800 rpm. At the same time, the torque does not fall off as fast past the horsepower peak. [1]

It is well known that adding parts to a mechanism has an adverse effect on the manufacturing processes associated with mass production. Put simply, addition of parts requires more time and money for assembly. It also means that there is a greater chance for something to go wrong. This has a negative effect on consumers because failure of a mechanism on their automobile not only increases their expenditures on repairs, but causes inconvenience by requiring them to function without their vehicle.
Simplifying mechanically actuated systems does allow cost and susceptibility to failure to decrease. More specifically, systems such as ones designed by Toyota and Porsche, focus on simply varying the phase angle of the camshaft rather than altering cam lobes in addition to cam position.

Cam-phasing VVT is the simplest, cheapest and most commonly used mechanism at this moment. However, its performance gain is also the least, a very fair compromise indeed.

Basically, it varies the valve timing by shifting the phase angle of camshafts. For example, at high speed, the inlet camshaft will be rotated in advance by 30° so to enable earlier intake. This movement is controlled by engine management system according to need, and actuated by hydraulic valve gears.

**High Speed - Toyota VVT-i**
Note that cam-phasing VVT cannot vary the duration of valve opening. It just allows earlier or later valve opening. Earlier open results in earlier close, of course. It also cannot vary the valve lift, unlike cam-changing VVT. However, cam-phasing VVT is the simplest and cheapest form of VVT because each camshaft needs only one hydraulic phasing actuator, unlike other systems that employ individual mechanism for every cylinder. [2]

Cost can be reduced by further simplifying system to have fewer operational settings.

Simpler cam-phasing VVT has just 2 or 3 fixed shift angle settings to choose from, such as either 0° or 30°. Better system has continuous variable shifting, say, any arbitrary value between 0° and 30°, depends on rpm. Obviously this provides the most suitable valve timing at any speed, thus greatly enhance engine flexibility. Moreover, the transition is so smooth that it’s hardly noticeable. [1]

There are other existing methods to vary valve timing and displacement that are not restricted by mechanical means. Termed electromagnetic control, camels systems have begun to develop over recent years. Because some electronic devices weigh significantly less than mechanical ones, weight savings from electromagnetic systems are generous. However, the actuation of electromagnetic devices requires greater computing power for control. This is not a major issue because many automakers are favoring an
In the standard power systems used in automobiles in order to support the increased demand for electronic components. Nevertheless, the cost of components necessary to run camless systems is still slightly high. Although the expense of computing systems has decreased over the years, units that are able to withstand the environment in the engine compartment are costly.

With the electromagnetic valve timing system, there are no camshafts, bucket tappets, timing chain or any other of the components previously required to drive the inlet and exhaust valves. All these components are replaced by 'actuators', which in this case are powerful, electronically controlled magnets with their armatures directly connected to the valves. The actuators operate the valves with the aid of valve and actuator springs. With this system, each individual valve can be opened or closed at any time on demand, either alone or simultaneously with others.

The system consists of one actuator per valve, so that the cylinder head of a modern 4-cylinder engine will have a total of 16 of these powerful actuators. Microcomputers on each cylinder bank control the actuators by regulating the power supply to the electromagnets. These microcomputers also control the ignition and fuel injection functions.
The free valve play technique significantly reduces engine weight and lubrication requirements, as no camshaft is required. Also it indicates a possibility of infinite timing variation. This technique leads to a reduction in weight of the engine and together with the advantages gained by the variable valve timing system it leads to a significant reduction in fuel consumption. However the microprocessor used to control the valve timing in this system increases the cost of the system and it can be incorporated only in luxury cars... [3]

It is logical to consider camless technology because it does not suffer from the inherent restrictions of camshafts. The versatility offered through processors allows engine designers to utilize knowledge of chemistry much more efficiently. This is an important notion because engine events are transient (for the most part) and the cycle of one cylinder has a direct effect on that of other cylinders.

Neither the masses of air inducted into the different cylinders of a multicylinder engine per cycle nor the masses of fuel which enter the different cylinders per cycle are exactly equal. In addition, mixing of fuel and air within each cylinder is not necessarily completely uniform. Thus the exhaust gas composition may correspond to a distribution in the fuel/air ration in the unburned mixture about the mean value. For
example, if the mean fuel/air ration is stoichiometric, extra oxygen will be contributed by any cylinders operating lean of the average and extra carbon monoxide by any cylinders rich of the average, so that the exhaust gas will have higher levels of O₂ and CO (and a lower level of CO₂) relative to an engine operating with identical fuel/air ratios in each cylinder [5].

Current technology allows engineers to take full advantage of scientific knowledge. Sensors are available that can collect data about atmospheric conditions and engine operation millions of times a second. Computers are available that can make calculations and send instructions faster than was ever fathomed when the first Otto-cycle engine was created. And, most importantly, scientific knowledge is available that can be readily employed by engineers. This research is aimed at developing a system with the ability to take these elements and optimize engine operation for the sake of conserving energy and protecting the environment without requiring consumers to sacrifice the automobile performance they have grown accustomed to.
CHAPTER III
METHODOLOGY

This project involves an approach for creating a camless valve train that contrasts completely from conventional systems. This is accomplished by using electronic motors to control the motion of the rotary valves instead of a timing belt and camshaft to actuate poppet valves. As represented in the schematic of Figure 1, the actions of an engine are simulated by a piston-cylinder device coupled with a rotary valve assembly. In the figure one motor is shown to actuate the intake valve while another motor is used to rotate the crankshaft of the piston-cylinder device. Control of a single valve is displayed in the schematic because it is assumed that the procedures used to model the intake valve will be identical for those needed for the exhaust. Obviously, the phase of the different valves will not be the same. Ultimately, this project focuses on the design and analysis of a control system to electronically manipulate the position of the rotary valves as a timing belt does the camshaft in conventional engines.

The process of the motor turning the crankshaft is assumed to be equivalent to the combustion process of an ICE. Ergo, increasing the current in order to rotate the crankshaft motor faster is considered equivalent to intensifying the combustion process in order to raise the engine speed. A dimmer switch could be used to control the amount of current flowing to the motor and hence would serve the same purpose of the gas pedal in a conventional vehicle. This approach is taken in order to eliminate the many complications encountered with the operation of an actual engine. Additionally, in order to assess the magnitude of improvement over traditional processes, it is necessary to take an in-depth look at the performance characteristics of a common engine equipped with
both technologies. Without fine-tuning the proposed approach on the theoretical level, it
would be wasteful to expose the proposed system to the intense environment of an ICE.
Therefore, the logical approach for a project of this magnitude is to:

1. Model, simulate, analyze, and optimize the system.
2. Create a physical model of the system and compare its performance with
   that estimated by simulations.
3. Implement tested system on an actual engine.

Due to the fact that this thesis only focuses on the first of the steps listed above, it
is best to model the system mathematically in a manner that represents the conceived
physical model. However, regardless of the process used to describe and simulate the
combustion process, the ultimate goal is to determine the location and angular velocity of
the crankshaft. This is because the instantaneous location of the valve is a function of
these factors. As seen in TABLE I, the expected location of the valve can be
approximated at anytime by the intersecting values of Crankshaft Speed and Crankshaft
Position. Control techniques, beyond those used for this project, can utilize such a table
to determine the desired valve position as an input value to the controller. Such an
approach of course would require the addition of a synchronizing pulse to ensure that the
relationship between the valve and crankshaft remain relative.

Since all the elements outside the feedback loop, signified by Box A, are a means
of producing a given signal to the closed-loop system, Figure 1 can be reduced to Figure
2 and the problem is then simplified to that of controlling the desired phase of the valve.
However, such a simplification eliminates the ability to ensure synchronization of the system. Consequently, as described above, the first step of developing this system must be on the theoretical level since synchronization must be ensured in order for an internal combustion engine to operate. The fine tuning of this element is an item that must be looked at in-depth and on its own. Additionally, it is an element that is beyond the scope of control theory used in this project. Any lack in the precision of this coordination can cause catastrophic damage to the engine through processes such as knock.

Controlling the phase of the valve consists of two elements: (a) generating an input signal of the desired phase and (b) controlling the frequency response of the closed loop. The former is a matter of deciphering the desired input in relation to the crankshaft’s actions, which is basically an issue related to the design of the Computer in Figure 2. As discussed previously, various control techniques, such as digital or fuzzy control, would be more sensible for the operation of the Computer than the analog methods used here. The aspect of controlling the frequency response is handled by focusing on the closed loop system. The performance of the system is ultimately a function of the PID controller, which is an analog controller. Although the frequency domain offers a means of analyzing the overall system performance, time response characteristics are of importance for the analysis of the valve and motor. By using both modes of analysis, a PID controller can be optimized for suitable system response.

In order to evaluate the performance of the system on either level, the component transfer functions had to be developed. If the sensor is assumed to provide unity
feedback its transfer function becomes the value of one. The transfer function of the motor is commonly used and is listed in Modern Control Systems as

\[
\frac{\theta(s)}{V_f(s)} = \frac{K_m}{s(Js + b)(L_f s + R_f)}
\]

where

- \(\theta(s)\) = Angular position of the rotor
- \(V_f(s)\) = Field voltage
- \(K_m\) = Motor constant
- \(b\) = Friction
- \(L_f\) = Inductance
- \(R_f\) = Resistance

Determining the transfer function of the rotary valve, however, is more in-depth. Since it is not commonly used and cannot be found in a table, the dynamics of the valve must be evaluated in order to model it mathematically. The best approach is to consider the torsional vibration of the valve in order to establish a relationship between the output position, which is ultimately considered the valve position, and the input position from the motor. Instantaneously, due to torsional vibration, these values will not be the same because there is some amount of deformation over the valve shaft. The valve can be modeled as shown in Figure 4(a) and the transfer function determined using the following equations of motion.
(1) \( J_2 \ddot{\theta}_2 + \eta (\ddot{\theta}_2 - \dot{\theta}_1) + k(\theta_2 - \theta_1) + k(\theta_2 - \theta_3) = 0 \)

(2) \( J_3 \ddot{\theta}_3 + k(\theta_3 - \theta_2) = 0 \)

where

\[ J = \text{Inertia of the masses of the shaft} \]

\[ \eta = \text{Frictional coefficient of the masses on the valve shaft} \]

\[ k = \text{Stiffness of the shaft} \]

It should be noted that the constants listed above will be set for a given setup.

The physical structure for a valve will not vary over time. Consequently, the time response of this system shows how well the structure physically withstands a change.

The find the transfer function, equations (1) and (2) must be simplified.

For equation (1)

\[
J_2 s^2 \ddot{\theta}_2 + \eta s \ddot{\theta}_2 - \eta s \dot{\theta}_1 + k \theta_2 - k \theta_1 + k \theta_2 - k \theta_3 = 0
\]

\[
\theta_2 \left( J_2 s^2 + \eta s + 2k \right) - \dot{\theta}_1 \left( \eta s + k \right) - k \theta_3 = 0
\]

setting

\[
G_1(s) = J_2 s^2 + \eta s + 2k
\]

\[
G_2(s) = \eta s + k
\]

gives

\[
(3) G_1(s) \dot{\theta}_2 - G_2(s) \dot{\theta}_1 - k \theta_3 = 0
\]

For equation (2)

\[
J_3 s^2 \ddot{\theta}_3 + k \theta_3 - k \theta_2 = 0
\]

\[
\theta_3 \left( J_3 s^2 + k \right) - k \theta_2 = 0
\]

setting
\[ G_3(s) = J_3 s^2 + k \]

gives

\[ (4) G_3(s) \theta_3 - k \theta_2 = 0 \]

Using equations (3) and (4), the relationship between the input position and the output position can be determined as follows

\[
\begin{align*}
\theta_2 &= \frac{G_3(s) \theta_3}{k} = \frac{k \theta_3 + G_2(s) \theta_1}{G_1(s)} \\
G_1(s) G_3(s) \theta_3 &= k^2 \theta_3 + kG_2(s) \theta_1 \\
\theta_3 \left[ G_1(s) G_3(s) - k^2 \right] &= kG_2(s) \theta_1 \\
\frac{\theta_3}{\theta_1} &= \frac{kG_2(s)}{G_1(s) G_3(s) - k^2} = \frac{k(\eta s + k)}{(J_2 s^2 + \eta s + 2k)(J_3 s^2 + k) - k^2}
\end{align*}
\]

The output position, \( \theta_3 \), represents the actual angular position of the valve while the input position, \( \theta_1 \), represents the angular position of the DC control motor. Unfortunately, the resulting equation creates a high level of complexity in the system analysis which leads to consideration of a tighter, lumped-mass approach. Modeling the system as in Figure 4(b) leads to the following derivation

\[
\begin{align*}
(A) & \quad J_1 \ddot{\theta}_1 + \eta (\dot{\theta}_1 - \dot{\theta}_2) + k(\theta_1 - \theta_2) = T_0 \sin \omega t \\
(B) & \quad J_2 \ddot{\theta}_2 + \eta (\dot{\theta}_2 - \dot{\theta}_1) + k(\theta_2 - \theta_1) = 0
\end{align*}
\]

For Equation (A)
\[J_1 \ddot{\theta}_1 + \eta \dot{\theta}_1 - \eta \dot{\theta}_2 + k \theta_1 - k \theta_2 = T_0 \sin \omega t\]

\[J_1 s \dot{\theta}_1 + \eta \dot{\theta}_1 - \eta \dot{\theta}_2 + k \theta_1 - k \theta_2 = \frac{T_0 \omega}{s^2 + \omega^2}\]

\[\dot{\theta}_1 \left( J_1 s + \eta + \frac{k}{s} \right) - \dot{\theta}_2 \left( \eta + \frac{k}{s} \right) = \frac{T_0 \omega s}{s^2 + \omega^2}\]

\[\dot{\theta}_1 \left( J_1 s^2 + \eta s + k \right) - \dot{\theta}_2 \left( \eta s + k \right) = \frac{T_0 \omega s}{s^2 + \omega^2}\]

\[\dot{\theta}_1 \left( J_2 s^2 \right) + \left( \dot{\theta}_2 - \dot{\theta}_1 \right) \eta s + k = \frac{T_0 \omega s}{s^2 + \omega^2}\]

For Equation (B)

\[J_2 \ddot{\theta}_2 + \eta \dot{\theta}_2 - \eta \dot{\theta}_1 + k \theta_2 - k \theta_1 = 0\]

\[J_2 s \dot{\theta}_2 + \eta \dot{\theta}_2 - \eta \dot{\theta}_1 + k \theta_2 - k \theta_1 = 0\]

\[\dot{\theta}_2 \left( J_2 s + \eta + \frac{k}{s} \right) - \dot{\theta}_1 \left( \eta + \frac{k}{s} \right) = 0\]

\[\dot{\theta}_2 \left( J_2 s^2 + \eta s + k \right) - \dot{\theta}_1 \left( \eta s + k \right) = 0\]

\[\dot{\theta}_2 \left( J_2 s^2 \right) + \left( \dot{\theta}_2 - \dot{\theta}_1 \right) \eta s + k = 0\]

(D) \[\dot{\theta}_2 G_1(s) + \left( \dot{\theta}_2 - \dot{\theta}_1 \right) G_2(s) = 0\]

Where

\[G_1(s) = J_2 s^2\]

\[G_2(s) = \eta s + k\]

\[G_3(s) = J_1 s^2\]

\[G_4(s) = \frac{T_0 \omega s}{s^2 + \omega^2}\]

Solving Equations (C) and (D) for \(\dot{\theta}_1\) & \(\dot{\theta}_2\) respectively
\[ (E) \quad \dot{\theta}_1 = -\frac{\dot{\theta}_1 - \dot{\theta}_2 G_2(s) + G_4(s)}{G_3(s)} \]

\[ (F) \quad \dot{\theta}_2 = \frac{\dot{\theta}_1 - \dot{\theta}_2 G_2(s)}{G_1(s)} \]

Dividing Equation (F) by (E)

\[
\frac{\dot{\theta}_2}{\dot{\theta}_1} = \frac{(\dot{\theta}_1 - \dot{\theta}_2 G_2(s) G_3(s))}{G_1(s) [- (\dot{\theta}_1 - \dot{\theta}_2) G_2(s) + G_4(s)]}
\]

\[ (G) \quad \frac{\dot{\theta}_2}{\dot{\theta}_1} = -\frac{(\dot{\theta}_1 - \dot{\theta}_2) G_2(s) G_1(s)}{(\dot{\theta}_1 - \dot{\theta}_2) G_1(s) G_2(s) - G_1(s) G_4(s)} \]

Subtracting Equation (F) from (E)

\[
\dot{\theta}_1 - \dot{\theta}_2 = -\frac{(\dot{\theta}_1 - \dot{\theta}_2) G_2(s) + G_4(s)}{G_3(s)} - \frac{(\dot{\theta}_1 - \dot{\theta}_2) G_2(s)}{G_1(s)}
\]

\[
(G_1(s) G_2(s)) (\dot{\theta}_1 - \dot{\theta}_2) = -(\dot{\theta}_1 - \dot{\theta}_2) G_1(s) G_2(s) + G_1(s) G_4(s) - (\dot{\theta}_1 - \dot{\theta}_2) G_2(s) G_3(s)
\]

\[
(G_1(s) G_2(s)) (\dot{\theta}_1 - \dot{\theta}_2) = -(\dot{\theta}_1 - \dot{\theta}_2) G_1(s) G_2(s) + G_2(s) G_3(s) + G_1(s) G_4(s)
\]

\[
(\dot{\theta}_1 - \dot{\theta}_2) (G_1(s) G_3(s) + G_1(s) G_2(s) + G_2(s) G_3(s)) = G_1(s) G_4(s)
\]

\[
(H) \quad (\dot{\theta}_1 - \dot{\theta}_2) = \frac{G_1(s) G_4(s)}{G_1(s) G_3(s) + G_1(s) G_2(s) + G_2(s) G_3(s)}
\]

Substitution Equation (H) into (G)
As shown, with the output position now denoted by \( \Theta_2 \), the equation describing the dynamics of vibration for the valve is much simpler. Coefficients will not lead to astronomical figures in the denominator. The analysis section will discuss how the constants affect the time response of the valve.

The PID (Proportional plus Integral plus Derivative) controller is an analog device that is commonly used due to its versatility. Its transfer function is commonly described as

\[
TF = \frac{K_D s^2 + K_P s + K_I}{s}
\]

where

- \( K_D \) = Derivative constant (gain or coefficient)
- \( K_P \) = Proportional constant (gain or coefficient)
- \( K_I \) = Integral constant (gain or coefficient)
Due to the fact that the dynamics of the valve and motor do not change unless the items themselves change, the sensitivity of the system rests solely upon the selection of the PID controller gains. This partially explains why the use of feedback is necessary. Ultimately, feedback benefits the performance because it allows the effect of the variation of the parameters of the process to be reduced, enables the system to be adjusted to yield the desired response by adjusting the feedback loop parameters, introduces control and partial elimination of the effect of disturbance signals, and provides the ability to adjust the transient response. Placing the obligation of controlling performance on the PID controller alleviates sensitivity of other parameters while utilizing the benefits of feedback.
CHAPTER IV
ANALYSIS

In control system analysis it is common to initially look at a system’s response in the time domain. Some cases justify the need for analysis in the frequency domain in order to take into consideration the affects of an oscillating input. As described previously, this case deals with the actuation of a valve to a given position. The use of frequency domain does not prove to be beneficial in this case simply because the input does not oscillate in a sinusoidal manner. The true problem lies within the ability for the valve to get to a given position. Therefore, analysis provided in this section will be related to the response of the system to a unit step response. This, physically, is equivalent to sending a signal to the valve actuation motor causing rotation to occur towards a set position.

First, it is important to look at the time response of the individual components. As displayed by Figure 5, the vibration caused in the rotary valve, when the torque is applied, causes distortion for a few fractions of a second. Since a shaft is stable until it reaches the limits of its strength, stability is assumed. Additionally, it is not surprising that overshoot is not an issue, 0.19%, because the body is fairly rigid preventing any oscillation. Similarly, Figure 6 shows that the valve actuation motor reaches a stable state and, with an overshoot of 4.32%, has potential to work for this application.

Due to the satisfactory performance of the valve and motor, it is safe to establish the overall system transfer function, as shown in Figure 3, and evaluate the root-locus plot, Figure 8, in order to get an idea of what coefficients need to be used for the PID controller. Since the focus is to minimize the overshoot and oscillation, careful selection
of controller’s gains is critical to system optimization. *Figure 7* gives a representation of the two previously described responses with the overall system response. In the plot, the gains have been adjusted, according to the root-locus, in order to achieve optimal performance. Although the overall system shows oscillation, due to the PID, the overshoot remains lower than 40% while the settling time is in the neighborhood of 0.2 seconds. In plain terms, the results show that using the purposed settings will result in the valve moving up to 40% past its desired position and taking up to 0.2 seconds to arrive within 20% of said position. The fact that the valve moves so far past the desired position is not disastrous because there can be some compensation made by adjusting the size of the hole in the valve. Additionally, because the opening will last for up to 90° of the valves duration, 39% overshoot will not completely restrict the airflow. One can estimate that the result would be airflow characteristics similar to that of a poppet valve for a brief moment of time.

It is important to note that this analysis does not cover the affects of engine speed on system performance. The response of the system is quite impressive compared to a purely P, PI, or PD controlled system; however, it is not fast enough to run an engine. Considering that the controller would consist of electronic circuitry, it is inevitable that thermal and power issues would arise which would prevent the controller from maintaining its highest level of performance. Regardless, the environment of high engine revolutions that rotary valves were invented for are unattainable with such a configuration.
The complications experienced by the approach used here provide a clear indication that analog control is not the best approach for such a technology. Camless engine technology revolves around removing one of the critical components in conventional valvetrains - the timing belt. Eliminating this component places the responsibility of maintaining the synchronization (provided by the timing belt) between the valve and the crankshaft on the control system. As described in the methodology, the assumption was made that synchronization was ensured so the focus could be placed upon the response of the closed loop to that synchronized input. Such a restriction, however, is not practical when considering implementation.

By utilizing a lookup table, such as Table 1, one can ensure that a specific valve value is related to instantaneous crankshaft position. However, there are a few issues related to providing values to the lookup table: (a) it is necessary to provide a reference value for crankshaft position and (b) some sort of logic must be used in order to decipher whether the crankshaft is on the first or second rotation of its cycle. The method used here does not allow these issues to be solved with ease.

Additionally, the feedback loop was designed and analyzed with the assumption that a linear control problem was being addressed. Truly, this is not an adequate assumption. For example, normal driving conditions involve a high level of frequency changes. With the ultimate goal being set toward varying valve timing and duration through multiple frequencies, it is necessary for the acceleration of the valve to be different during its open and closed cycles. This is apparent through evaluation of Table 1.
The fact that there is a direct relationship between the control of valve timing and engine performance is proven, as reviewed in the background. A camless system of this sort attempts to deal with the challenge of finding the correct approach to increasing that controllability. Obviously, the approach must allow parameters to be altered with ease while keeping implementation attainable. Since an engine demands precision while its components operate in a high speed environment, a control system that can handle a rapidly changing environment is required. Although this may be possible with a more in-depth look at the procedures used here, digital control techniques may prove to be more beneficial. Using state-space techniques can ultimately expound upon this method for a smooth transition into digital control. However, it is safe to state that the inability to responding to anything other than a step input, without loosing stability, has given a decent amount of insight into the problem at hand.

The onus for the control system to respond within milliseconds is nonnegotiable. Once this issue is resolved, efforts can be concentrated on maintaining the synchronization that is lost by disconnecting the timing belt. With the computing power available, coupled with the ability to use 42-volt power systems in automobiles, the only limiting factor is detailed construction of the control system layout. Consequently, the growing interest in camels systems will continue to push automotive technology and control system theory.
APPENDIX A

BLOCK DIAGRAMS
MODIFIED BLOCK DIAGRAM

FIGURE 2

\[ \theta_{\text{motor}} = \frac{200}{s^2 + 20s + 200} \]

Valve Actuation Motor

\[ \frac{100s + 2500}{25s^4 + 10s^3 + 50s^2 + 10s + 25} \]

Intake Rotary Valve
OVERALL SYSTEM TRANSFER FUNCTION

\[
\theta_{\text{Desired}} \rightarrow \frac{34500s^5 + 461500s^4 + 4.16e006s^3 + 180000s^2}{10s^7 + 430s^6 + 41110s^5 + 507700s^4 + 4.162e006s^3 + 180000s^2} \rightarrow \theta_{\text{value}}
\]

FIGURE 3
APPENDIX B

DYNAMIC MODELS
APPENDIX C

GRAPHS
Step Response for Actuation Motor

Figure 6

% Overshoot = 4.3214
Step Responses for Entire System

- Overall System: % Overshoot = 39.5422
- Control Motor: % Overshoot = 4.3214
- Rotary Valve: % Overshoot = 0.19597

**FIGURE 7**
APPENDIX D

CRANKSHAFT LOOKUP TABLE
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### TABLE I
INTAKE VALVE LOOKUP

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APPENDIX E

VALVETRAIN IMAGES
POPPET VALVE CONFIGURATION

FIGURE 9 (a) Shows a sketch of a valve, as it is positioned over a cylinder, with the sections labeled by their common names. (b) Shows a sketch of a single-cam arrangement. Here the valve is actuated by the camshaft actuating a push rod which moves the rocker arm which ultimately controls the valve. Air flow is controlled by the valve obstructing or revealing the valve port. [10]
FIGURE 10 CADD Model of a valvetrain with a dual overhead cam (DOHC) setup. This setup has gained popularity because it reduces components and is easier to maintain [11].
FIGURE 11 Sketch of a DOHC engine with 6 valves per cylinder. Original equipment manufacturers (OEMs) have found much success in improving airflow by increasing the number of valves per cylinder. This image shows how intake and exhaust flow takes place through the valvetrain [12].
ROLAT BY VALVE CONFIGURATION

PERFORMANCE

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WITH SAME UNBLOWN ENGINE

Smoth Power
High Performance
Reliability and Long Life
Silent Operation
Self Adjustment
Extreme Economy
Highly Commercial

Can use HIGH COMPRESSION because there are NO HOT SPOTS
GOOD PORTING – COMPACT COMBUSTION CHAMBER

FIGURE 12 Roland Cross was the great British exponent of the horizontal rotary valve. He began work on rotary valve engines in 1920. Unlike the Aspin, the Cross valve was always driven at constant speed from the crankshaft. The “reaction bridge” absorbs the upward forces on the horizontal valve assembly, and is supposed to have reduced the gas forces on the actual valve [9].
FIGURE 13 The Aspin Rotary Valve concept uses a vertical valve rotating above the cylinder. The basic idea was introduced as far back as 1911 by Vallilee (Patent 983328) who used a rotating disc. Several other versions by other people were patented later. Aspin spent decades developing his concept which basically consists of a cone-shaped valve in which air is introduced from the side and flows through the bottom [7].
ROTARY VALVE CONFIGURATION

Figure 14 (a) The Norton Rotary Valve Engine was developed for two and a half years during the 1960s by a development engineer named Joe Craig Norton. This is a horizontal configuration proposed by Laurie Bond who closely followed Cross' technology. There were the usual problems with lubrication, sealing, and plug fouling, and the resulting engine was less powerful than the standard version. The cylindrical valve at the top is driven by bevel gears from a vertical shaft, which is driven from another vertical shaft via two small pinions. The second vertical shaft is driven by bevel gears from the crankshaft. Why there are two vertical shafts is currently unknown to me, it may just have been a matter of alignment, allowing the bottom end of an existing engine to be used. (b) The Lotus Rotary Valve was the result of several manufacturers, in the 1990s, investigating supercharged two-stroke engines. Since a two-stroke has one firing stroke per crankshaft revolution as opposed to one every other revolution, the power-to-weight ratio is potentially much better. The engineering challenges involved in making an engine that can compete with the conventional four-stroke are, however, severe [9].
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Date