ELECTRONIC ENGINE CONTROL AND OPTIMIZATION

Results of Pneumatic Cylinder Construction and Data Collection by S. Chad Gibbs

INTRODUCTION AND PROJECT MOTIVATION

The goal of this project was to construct a lightweight electronically controlled steam engine to re-evaluate potential applications of steam engine technology in transportation and energy production. The motivation was to understand if the application of modern materials and control mechanisms could make steam engines competitive with technologies such as the internal combustion engine. Further motivation comes from the ability to create steam from renewable sources (biomass, and solar) meaning a modern steam engine could be run renewably. Because of time and safety considerations the project direction was shifted towards creating a single stroke cylinder that could run on both steam and compressed air. Furthermore, the application of modern lightweight materials was superseded by the desire to create a robust test rig that will allow for the testing and optimization of electronic control mechanisms. The efforts of the project centered on the design and construction of the test rig and the testing and analysis of the test rig operated using compressed air, actuated by a pair of on-off solenoid valves with a Labview controller.

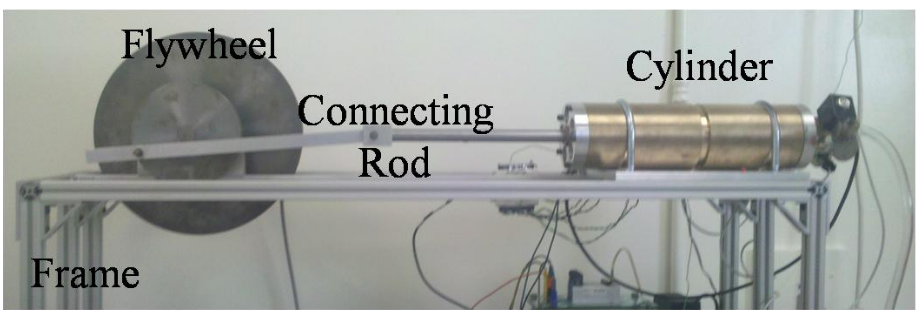

FIGURE 1: TEST RIG (COMPONENTS LABELED)

TEST RIG DESIGN AND CONSTRUCTION

A significant amount of the energy spent on the project was directed towards designing, analyzing and constructing a robust test rig which has valves that can be activated electronically. This design task was a departure from the traditional compressed air/steam cylinder which is mechanically activated. Motivation for electronic control was taken from recent research on performance gains that have been achieved for camless internal combustion engines using electronic valve control[1-3]. Peer and fellow mechanical engineer, Thomas Gallmeyer, provided design and construction assistance.

MECHANICAL SYSTEM ARCHITECTURE AND DESIGN

The first design task for the project was to define and construct the mechanical system. This design task for the mechanical system was segmented into four critical design components shown in Figure 1. The first design component was the engine cylinder. The first task in designing the cylinder was to size the diameter and stroke. Based on conventional wisdom for traditional steam engines the stroke of the cylinder was sized to be twice the diameter of the bore. The bore was set at 2.5” as a tradeoff between manageable and usable power production and piston head area when considering mounting sensors and electronic valves. The cylinder was constructed as a single acting system where energy is added to the system only one time per rotation. The level of precision that is required to seal a moving rod required for a dual action cylinder motivated the selection of single acting.

The material selection for the cylinder was driven by the desire for the system to be self lubricated. Traditional steam engines which used cast iron for the cylinders and had pistons which were sealed with metallic rings required the entering steam to be injected with oil or other lubricant. This process often made the outlet steam unusable. To avoid this traditional method for lubrication the cylinder was constructed using non-traditional materials. The cylinder was machined out of bronze, a material commonly used for bearings. This material provided a slick surface for a PTFE piston ring to slide on, even in the presence of the significant radial force which was required to maintain a pressure seal. More discussion of the success of this design will occur during the system characterization section. The factor of safety for the cylinder was calculated using a Solidworks FEA analysis with pressure above the nominal operation range. The initial design had more material taken off the cylinder to minimize weight, but this wasn’t realized to minimize construction time and cost. Factors of safety for the end caps and bolts were calculated analytically.

The next component was the piston head. The piston head was designed to incorporate the PTFE piston ring. This capability required the ability for the piston head to assemble around the ring and provide support for both the rigid PTFE ring and the rubber inner ring required to maintain the outward pressure on the PTFE ring. Next, the connecting rod and flywheel were designed to allow the cylinder to remain in motion even though the cylinder only applied energy to the system on the forward stroke. Finally the frame was constructed out of 80/20 extruded Aluminum, providing easy mounting for configuration modification. Parts were machined by John Goodfellow in the Duke University machine shop.

Activation Method Design

The activation method for the engine defines the amount of flexibility that the control scheme will have when controlling the system. The activation tasks for the engine require controlling the fluid in and out of the cylinder. Traditional steam engines were activated using a wide range of valves from slide valves, to roller valves. All of the valves share the common characteristic that they were activated mechanically by a cam attached to the rotating portion of the engine. Both for the scope of this project and for the usefulness of the test rig for future applications, the activation method for the project was required to be electronic. With this constraint the options that were explored were a servo activated ball valve, on-off solenoids and solenoid valves with position feedback.

Based on the available literature on camless engines, the solenoid valve with position feedback is the best option. This type of activation method allows the user to minimize impact velocity to limit the sound of valve opening and closing as well as gives the user the ability to have more fidelity over the valve opening and closing profile[1, 3-9]. A significant amount of research was conducted on the control schemes for this type of valve, because it was initially thought the control aspect for the engine would be the focus of the GWDD project. However, due to the complexity and cost of designing and building a solenoid valve with position feedback and the financial resources already used to construct the test rig, this valve choice was not viable. After exploring the performance of manually operated ball valves and predicting a less than optimal operation, the on-off solenoid was chosen as the activation method for both the inlet and exhaust ports. Specifically a 3/8" NPT direct acting solenoid valve driven by a 24VDC, 20W activation coil was used. The valve had a C_v of .6 which was adequate for the operating range of the system.

Instrumentation and DAC

Instrumentation for the device is largely driven by the type of data that is needed to apply a range of controls to the system and is derived from the mathematical model of the plant which will be discussed in more detail later in the paper. The sensors that were required were a 100 psi pressure transducer which records gauge pressure and produces an analog output between 0 and 5V. The second sensor was an inductance probe which measures distance and outputs a voltage between 0 and -18 volts which yields a measure of distance as well as conductivity of a material. This sensor was placed facing the flywheel and was used to determine position. The data was collected using the NI USB 6009 DAQ card and was processed on a laptop running Labview. The DAQ card is capable of 12 kS/sec and data was collected on 2 analog input channels and sent out on 2 analog output channels.

System Characterization

Once the system had been constructed, the pressure transducer was used to characterize the plant. System characterization is an important step in order to determine a mathematical model for the system. The mathematical model is extremely important when attempting to design advanced controllers for the plant that can optimize its performance.

Cylinder Characterization with No Flywheel

The first set of characterization tests was used to determine the static friction of the system as well as the flow rate through the inlet valve. Figure 2 shows a characteristic plot of the pressure vs. time for the tests that were run on the cylinder without the flywheel attached. The test was conducted with the piston head initially at back dead center. A 24VDC, 20W signal was applied across the inlet valve which is connected to the shop air through a regulator limiting the flow to 60 Psi. Pressure data was collected using the pressure transducer.

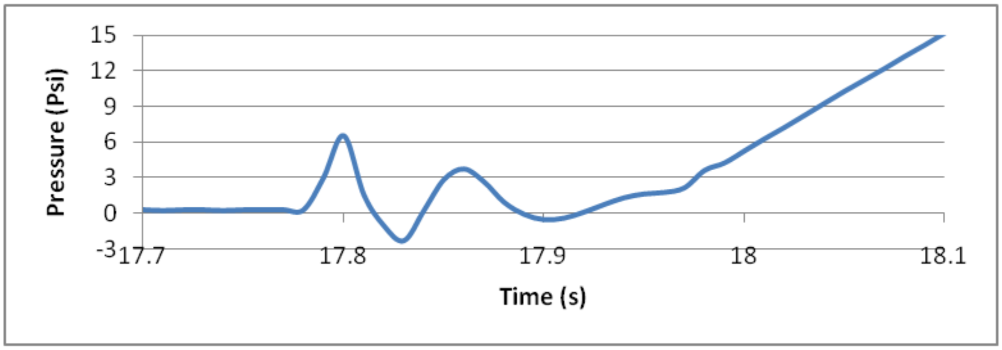

FIGURE 2: SYSTEM CHARACTERIZATION RESULTS

Based on the results shown in Figure 2 the first calculation that can be done is an estimate of the static friction. Based on observations of the motion of the piston head, the peak of the first hump represents the pressure that was required to overcome the static friction in the system. Looking at the data the peak pressure recorded was 6.54 psi gauge. Equation 1 uses the dimensions of the system to convert this to a force.

$$F = (Pressure)(Area) = \left(6.54\frac{lbf}{in^2}\right)\left(\pi * \left(\frac{2.5}{2}in\right)^2\right) = 32.08\, lbf \qquad \boxed{1}$$

This static friction loss places an upper limit on the amount of energy that can be lost by the system due to friction. Realistically because the dynamic fiction is most likely lower than static friction, the amount of energy that is lost through this mechanism is smaller. Taking the instantaneous loss and applying it to an entire revolution of the engine you get a loss of 26.73 ft-lbf as shown in Equation 2.

$$W = (F)(d) = (32.08\, lbf)\left(\frac{10}{12}ft\right) = 26.73\, ft - lbf(36.24\, J) \qquad \boxed{2}$$

Comparatively looking at the energy added to the system using the same set of calculations suggests that the power stroke adds over 100 ft-lbf to the system. Although this calculation suggests that nearly ¼ of the energy is lost to friction, it is important to remember that this was a worst case scenario calculation using the static coefficient of friction. Further work could go into optimizing this friction loss to pressure loss caused by decreasing the interference pressure between the cylinder walls and the piston ring.

The second result that can be calculated from the data collected is the mass flow through the valve. In order to do this calculation the last half of the data is used. At the point where the data begins to climb linearly the cylinder has reached front dead center. Assuming temperature is constant and using the knowledge of the location of position being fixed to dictate a constant volume the change in mass between two discrete data points can be calculated using the modified ideal gas law shown in Equation 3.

$$m_2 - m_1 = \frac{(P_2 - P_1)MV}{RT} \qquad \boxed{3}$$

Using the ambient temperature and the properties of air the mass flow rate between two points in the dataset was calculated to be .316 kg/sec. In order to put this number in perspective the mass of the air in the cylinder at 60 psi when the cylinder is at front dead center during operation is .060 kg so the valve is capable of filling the cylinder more than 5 times per second which suggests that operating the cylinder at the designed 100 RPM will not be limited by the valves.

Cylinder Characterization with Flywheel

The second set of system characterization tests was run with the piston rod attached to the fly wheel representing the final configuration for testing. This time the piston was started slightly after back dead center and then a 24VDC, 20 Watt signal was applied to the inlet solenoid allowing the 60 psi air to rush into the cylinder. The data shown in Figure 3 shows the pressure data versus time for this process plotted against the results that were observed for the system without the flywheel. Visually the motion of the flywheel associated with the presented pressure data is to fire from back dead center to front dead center and then past until the back pressure slows the motion and fires the flywheel back the other way. It continues this oscillation until it centers on front dead center.

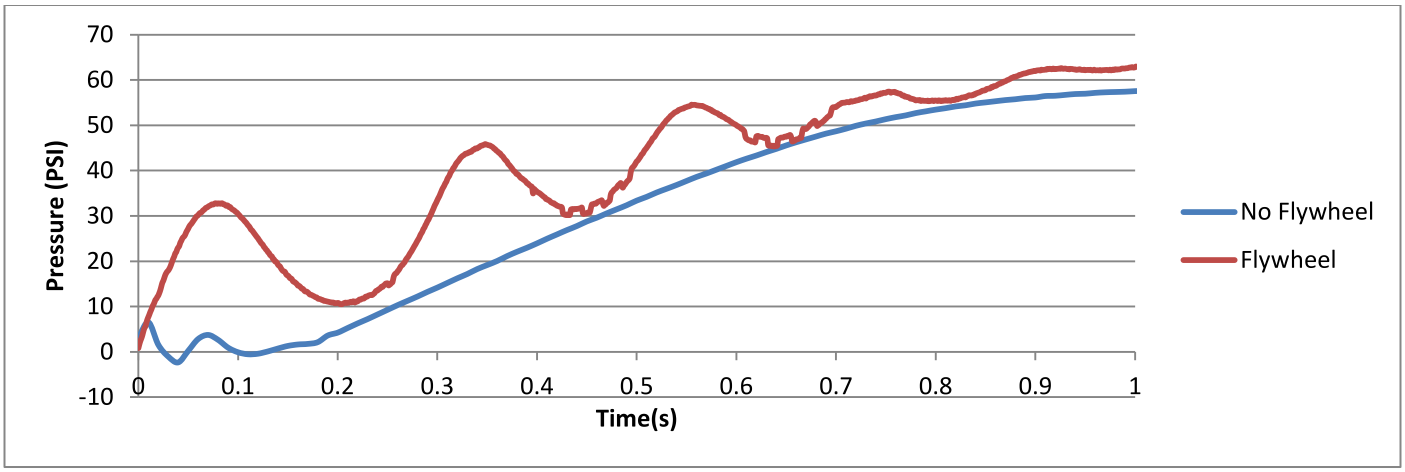

FIGURE 3: SYSTEM CHARACTERIZATION RESULTS 2

As expected the system requires much more force to begin moving with the flywheel attached. Using Equation 1 applied to the flywheel data suggests that it takes 160.2 lbf to get the flywheel moving. However this force is applied at a radius of .25" initially so only 37.5 lbf goes towards moving the flywheel. This suggests that the friction in the bearings on the flywheel and other flywheel system losses are minimal. Another system parameter that can be derived from this data is the angular momentum of the flywheel. This data can be derived by seeing how much increase in force is required to stop the flywheel. To interpret the data, the valley that is observed at .2 seconds correlates to the flywheel being at front dead center. Similarly, the peak at .35 seconds represents when the pressure inside the cylinder has reached a level high enough to stop the fly wheel and begin to move it back towards front dead center. The force required to stop the angular momentum of the flywheel must therefore be larger than the pressure difference between the peak in the fly wheel and the nominal data from the no flywheel case. This relation is shown in Equation 4.

$$Min\ Force = (P_{flywheel} - P_{no\ flywheel})(Area) \quad \boxed{4}$$

Using this equation and the data from the model yields that the amount of force required to stop the flywheel is greater than 127.5 lbf. This force is applied at a radius of about 2.5 inches which indicates that the minimum torque required to stop the flywheel is greater than 318.7 lbf-in. Equation 5 shows the relationship between torque and angular momentum.

$$L = \int \tau dt \quad \boxed{5}$$

Applying Equation 4 to a series of discrete points in the flywheel and no-flywheel data set allows the approximate integral to be calculated. The calculation yields an angular momentum of 23.64 lbf-in-s. Looking at just the flywheel and using the moment of inertia calculation shown in Equation 6 with a speed of 40 RPM yields a predicted angular momentum of 57.02 lbf-in-s.

$$L = I\omega \tag{6}$$

Although the moment of inertia calculation suggests a significantly higher value this difference can be explained by losses in the system due to friction which as, discussed previously, has not yet been optimized.

System Model

Using the results of the system characterization testing done on the test rig, a more accurate model can be created to model the dynamic performance of the plant. Applying a force balance to the piston head when there is no flywheel attached yields Equation 7.

$$M_{Moving}\ddot{x} + F_{fr} = (P - P_{atm})(Area) \tag{7}$$

As predicted this simple model suggests that you are able to create the largest amount of acceleration by increasing the amount of pressure difference while minimizing friction and the mass of the moving parts (Piston, Piston Rod, etc). The more interesting model is of the entire system attached to the flywheel. The mathematical model of this plant includes a force balance that is shown in Equations 8 and 9 and an energy balance inside of the pressurized chamber which is given in Equations 10 and 11.

$$M_{Moving}\ddot{x} + F_{fr} + F_{flywheel} = (P - P_{atm})(Area) \tag{8}$$

$$F_{flywheel} = \frac{\frac{d}{dt}[I\omega]}{r} = I\frac{5}{\pi}\frac{\ddot{x}}{r} \tag{9}$$

$$q_{in} - q_{out} + \ kc_v(\dot{m}_{in}T_{in} - \dot{m}_{out}T_{out}) - \dot{W} = \dot{U} \tag{10}$$

$$\dot{P} = \frac{RT}{V}(\dot{m}_{in} - \dot{m}_{out}) - \frac{P}{V}\dot{V} \tag{11}$$

Combining Equation 8 and 9 in addition to Equation 11 yields the equations that govern the operation of the plant. All of the parameters for the system except the mass of the moving mechanism were discussed in the system characterization portion of the paper. The non linear aspect of this mathematical model comes from the r term in Equation 9. This term varies from 0 at front and back dead center to 2.5” in the middle of the cylinder stroke. The r term can be defined as a function x and will have the form of a cycloid. Equation 11 applied to the system without a flywheel reduces to Equation 3 which shows good consistency between the mathematical models.

System Control and Results

Once the model of the system has been constructed a control scheme can be designed and implemented. The model suggests that the motion of the system largely relies on two variables, position, pressure and their derivatives (assuming mass flow is mostly a function of the valve used). This drove the efforts for instrumentation on the device. Pressure was collected, as alluded to in previous section with a pressure transducer that records gauge pressure.

Position Sensing

Position sensing is extremely important because it is vital for timing the inlet and exhaust valves. Traditional cylinders control the valve timing mechanically therefore "know" position through the mechanical connection between the cam on the wheel and the valve activators. While this achieves great accuracy on firing the valves at the same position every time, it also limits the system to a single timing for all operating conditions. In the compressed air cylinder this mechanical linkage is absent but the requirement for accurate position sensing for valve timing is still present.

The key requirements for the position sensing design is to have the ability to give good position and velocity readings around the valve firing locations and the ability to always be aligned with the actual position of the cylinder. Furthermore the data must be received and analyzed in real time so accurate control can be achieved. The sensor system that was used to get position was an inductance probe. The inductance probe gives a linear relationship between position and voltage for any given material. However, for different materials with different conductivities the scaling coefficients do not remain the same. After some initial testing it was determined that a strip of aluminum tape on the cast iron fly wheel was seen as a 6-8V drop in signal from the inductance sensor. This was a large enough change to consistently be recorded by the DAQ system and act as a trigger.

Method 1

The first method placed a 4 toothed piece of aluminum tape at the sensor location of front and back dead center. Each of the teeth had a width seen by the sensor of 10 degrees on the fly wheel with 10 degree spacing between strips. The idea of this scheme was to record the time between triggers of known angular distance apart to get accurate velocity readings near front and back dead center and use the space between the front and back dead center signals to act as a reset.

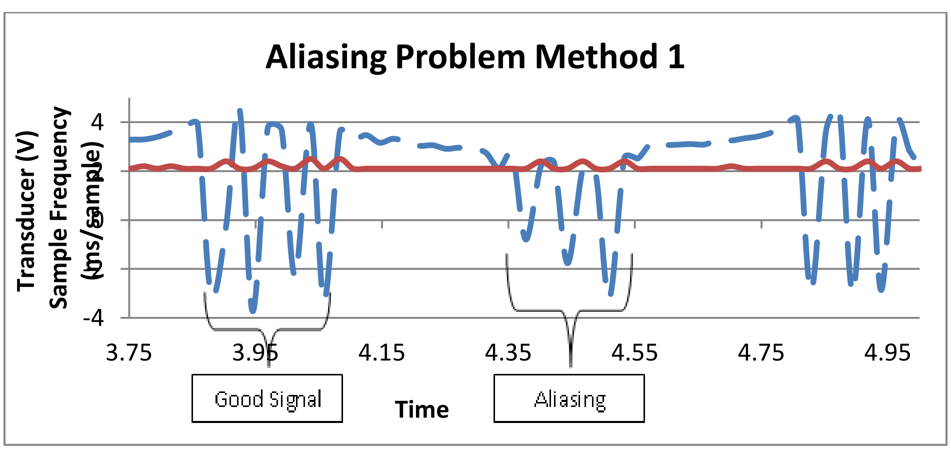

FIGURE 4: ALIASING PROBLEM METHOD 1

The dashed line in Figure 4 shows the signal that was collected as the system was rotated at around 60 RPM. As identified in the figure, even at this slow speed the sensor was not able to pick up every trigger. In order to diagnosis this problem the sampling rate for the data was also plotted. The solid line in the figure shows the time between samples in ms/sample. Although the DAQ was capable of 12ks/sec this data suggested that Labview was only able to access around 50 samples/second. Because of this limitation, method two was developed.

METHOD 2

Based on the observations from the first position measurement it was determined that more analysis needs to go into the spacing of the triggers. Based on a max operating speed of 120 RPM (4π rad/sec) Equation 12 suggests that the DAQ only collects a sample every 8.6 degrees.

$$4\pi\frac{rad}{sec}\times .024\frac{sec}{sample} = .151\frac{rad}{sample}\left(8.6\frac{deg}{sample}\right) \tag{12}$$

With this data it was not surprising that there were instances when the trigger was not seen by the sensor with only 10 degree spacing. In order to remedy the situation the size of each aluminum tape was widened to 20 degrees with 20 degree spacing. Because of this, In order to get 4 data points around each center it became necessary to have triggers throughout the entire rotation of the flywheel as shown in Figure 5.

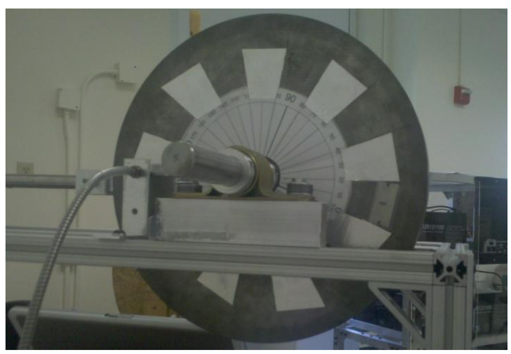

FIGURE 5: FLYWHEEL POSITION SENSING SETUP

With this setup it became necessary to have a hard reset once per revolution to ensure that the divergence from absolute position was not lost if a signal was missed. This was done using a razor blade which provided a 4V jump in the signal as shown in Figure 6.

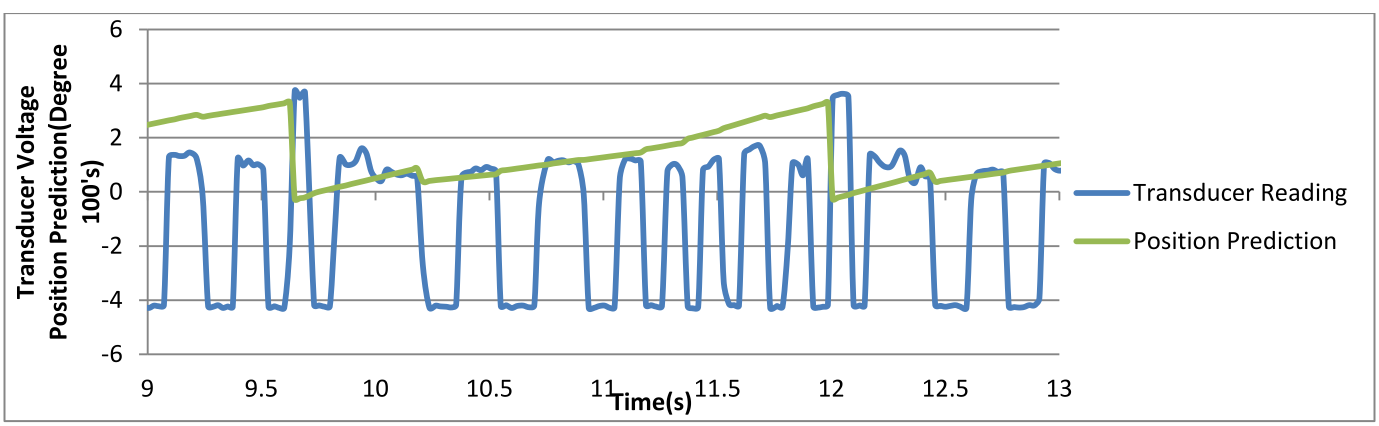

FIGURE 6: METHOD TWO POSITION SIGNAL

This method proved a good way to provide position and velocity data. Velocity was determined from the signal by dividing 40° by the time between triggers. The current distance from the last trigger was then calculated to be the time since the last trigger multiplied by rate from the previous two triggers. The data in Figure 6 also shows the predicted position using the method described in the system control section. Discontinuities in the position line are not good although the ability to model changes in rate is good. Overall the position prediction method was able to give enough position data to accurately activate the valves.

SYSTEM CONTROL

The control scheme for the steam cylinder was designed and run through Labview. The VI was responsible for reading the data from the NI USB 6009 DAQ card, analyzing the data to produce the pressure and position data and then storing the data collected for post analysis. Furthermore the VI is responsible for using the position data to control the solenoid valves. The code takes the data and passes the data through a pair of If statements to look for either high or low triggers. Each low trigger causes the system to calculate the angular velocity based on the past two data points collected and then increment an index to retain knowledge of where the system is. The high trigger causes the index to reset. Position is determined by taking the index multiplied by 40° then adding to the previous velocity times the difference between the current time and the time of the previous trigger. The position data is then used by two logical blocks that analyze if a given valve should be on or off. If the valve is triggered the VI tells the DAQ card to send the signal to the valves.

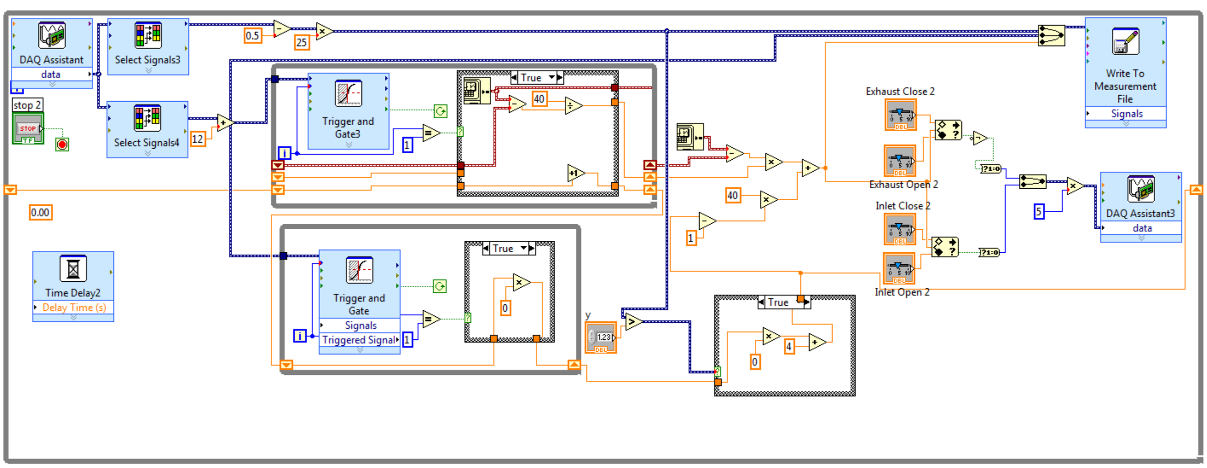

FIGURE 7: CONTRL VI

Because the valves requires 20 W to activate the signal, the DAQ card is actually used to trigger a relay which then sends the required power to the valve solenoids. Currently the control scheme for the system acts like a simple cam causing the valve to open and close at the same position every time. However, because the system is activated electronically the timing can be changed by moving a slider in Labview instead of having to re-machine a new part if the timing is not perfect.

Conclusions / Further Work

The work conducted for this project creates a foundation for future testing of electronically actuated pneumatic and steam piston control at Duke. The test rig and instrumentation that were developed were intentionally created in a manner that allows substitution and replacement. While there is a wide range of further research that will be able to use the test rig there are a number of avenues that could produce extremely valuable insights. The first avenue is developing an understanding of the different optimized timing configurations for different operating conditions for the cylinder such as speed and inlet pressure. Similarly the mathematical model developed though the system characterization testing can be used to design optimal or intelligent controllers. Secondly the rig could be deployed to rapidly test proposed cam designs without machining a part. By developing a model of the cam in Labview the performance could be simulated electronically. Finally the system could incorporate solenoid valves that have accurate position feedback which could allow for the exploration of different inlet and exhaust opening and closing profiles and the respective impact on performance.

In conclusion, the efforts on the project lead to the creation of an apparatus for evaluating electronic valve operation for a steam/compressed air system. While we were not able to directly evaluate whether a modern steam engine could be used for power generation and transportation good progress was made towards developing a manner to test and evaluate vital technologies for a modern steam/compressed air system. The project was a combination of mechanical design and system testing and analysis.

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