VARIABLE POSITION AND FORCE CONTROL OF A PNEUMATIC ACTUATION SYSTEM

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ABSTRACT

The implementation of a variable pneumatic actuation system with precise position and force control is both a costly and complicated endeavor due to the high cost of variable flow solenoids. Through the use of pulse width modulation, off-the-shelf standard components can be utilized to create a low cost pneumatic control system. This report demonstrates that such a system is capable of both the position control, movement response and force control required to make it a valuable asset in the field of robotics.

A series of tests are performed to closely examine a representative system with respect to position control, impulse response, and force control. The results show that angular transit speed functions in proportion to the programmed controller response, with the minimal overshoot transit time of less than one second. Impulse response tests show that impulse response is a factor of incident force and duration, with a 25lb test load causing a 14% deflection with a total response time of one and one half seconds. Force control trials demonstrate a system in which the force response is very closely linked to the user control (< 200ms), allowing for high level force compensation.

The results are further expanded upon through the listing of a noted limitations inherent to the system, particularly the necessity of a properly designed pneumatic system to minimize flow restrictions, as well as the benefits of implementing a predictive control system to compensate for variations in the compressibility of the medium.

1. INTRODUCTION

Modern pneumatic actuators consist of two general formats, standard (on-off) and variable flow. Standard configurations are simplistic in nature, with a single or double acting pneumatic cylinder coupled to at least a single solenoid valve. It is due to this simplicity that they are very economical, however their useful capacity is, by nature, extremely limited, as their range of motion is confined to only an extended and a retracted state.

Variable flow actuators act to supplant dual position actuators in situations where more than two states are required and while they are identical to standard actuators in theory, their design is significantly more complicated. The increased complication is primarily due to the necessity of a precision solenoid valve, pressure/flow rate sensors, and a closed loop feedback control scheme to compensate for the compressibility of the pneumatic medium.

This paper will demonstrate and experimentally verify that a variable flow pneumatic actuator system, constructed entirely from standard configuration, off-the-shelf components, can achieve the level of accuracy required to be utilized in situations where precision control is necessitated.

2. NOMENCLATURE

Plot Nomenclature:

Units are as specified on individual plots for each data set. Designation colors are not consistent throughout the individual plots. Datasets identified as possessing the unit of “Var” have been recorded in values native to the Arduino microcontroller. “Var” units have a range of 0 to 1024 for an input and 0 to 255 for an output.

Equation Variables:

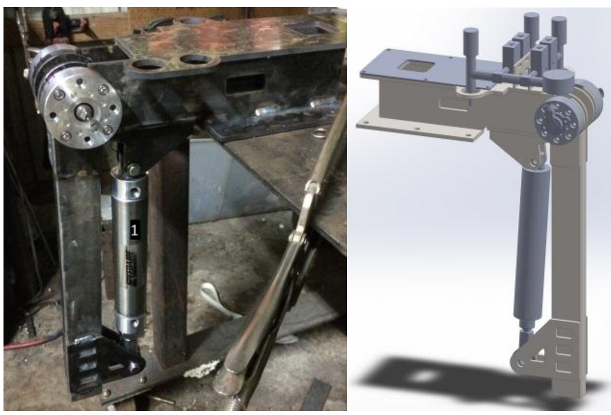
A_{in}	Targeted amplitude (deg)
A_{exp}	Experimentally obtained amplitude (deg)
E	PID error value
K_p	PID controller proportional constant
K_i	PID controller integration constant
K_d	PID controller derivative constant
t_{in}	Time (ms), measured from program start
t_{exp}	Time(ms), measured from program start

Θ_{target}	Targeted angle or pressure, depending on the instance of the PID control scheme.
$\Theta_{current}$	Current leg angle or pressure, depending on the instance of the PID control scheme.

3. METHODS

<u>Hardware Configuration</u>

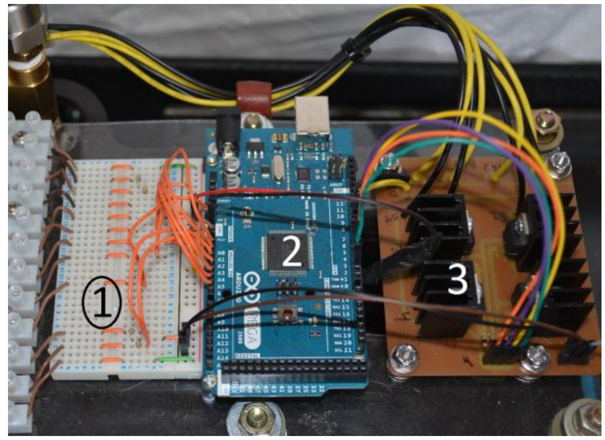
The experiment apparatus for this project centers on the Arduino Mega 2560, which is utilized for general computation, feedback analysis, and data collection. The physical apparatus consists of a pneumatic cylinder (Grainger #6CZZ8) attached to a customized steel framework. Completely designed in SolidWorks, the framework is oriented and sized so as to approximate the motions of a rudimentary human knee joint. All rotational motion is isolated from the base through the use of dual sealed bearings to reduce the effects of friction on the experiment.



1 - Pneumatic Cylinder

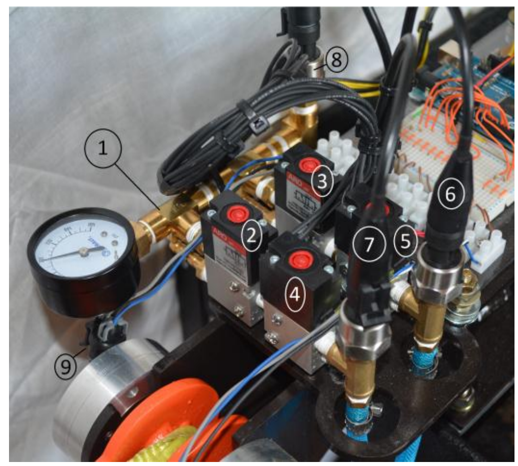
Figure 1: (Leg) Apparatus Framework, SolidWorks Model

Control of the pneumatic cylinder is achieved through the use of four ARO 2-position solenoid valves (#P251SS-012-D), oriented in a parallel series configuration so as to allow two valves for each port on the pneumatic cylinder. The valve connected to the air supply line acts to direct the flow, so as to either intake or exhaust, while the second valve functions solely to control the rate of flow. As the valves each require approximately 0.3 amperes to fully actuate, they are electrically linked to the Arduino Mega 2560 via four TIP-31C based power amplifiers, configured on the attached PCB board, right of the Arduino controller in Figure 2.



1 – "Solderless" breadboard, 2 – Arduino Mega 2560,
3 – Solenoid Control Power Amplifier

Figure 2: Electronic Control Assembly



1 - Supply Line, 2 – "A" Directional Valve,
3 – "B" Directional Valve, 4 – "A" State Valve,
5 – "B" State Valve, 6 – "B" Pressure Transducer,
7 – "A" Pres. Transducer, 8 - Supply Pressure Transducer,
9 - Angular Position Sensor

Figure 3: Pneumatic Control Assembly

The project sensor pack consists of three 150PSI #D4 pressure transducers and two MasterPro EC3048 throttle position sensors (TPS). Due to low amperage requirements, all sensor wire connections were established using a standard "solderless" breadboard to facilitate experimental modification. As the Angular position sensors are resistive, the connections are attached through a resistive voltage divider, so as to cause a change in voltage relative to angular position. The power required for the project is supplied by a standard 12 volt, 3 ampere power adaptor. The air supply line is charged by a portable, 5 gallon, 150PSI air compressor, regulated to 100PSI for all sections of this project except the last constant force trial, in which the operating pressure is increased to 125PSI to compensate for the increased weight applied to the testing apparatus. The complete electrical schematic is available in the appendix.

Data acquisition is accomplished through the use of the Parallax PLX-DAQ software suite, which facilitates the importation of up to twenty four channels of data directly into a Microsoft Excel spreadsheet. Data communication is accomplished via the onboard Arduino serial port interface operating at 128 kilobits per second.

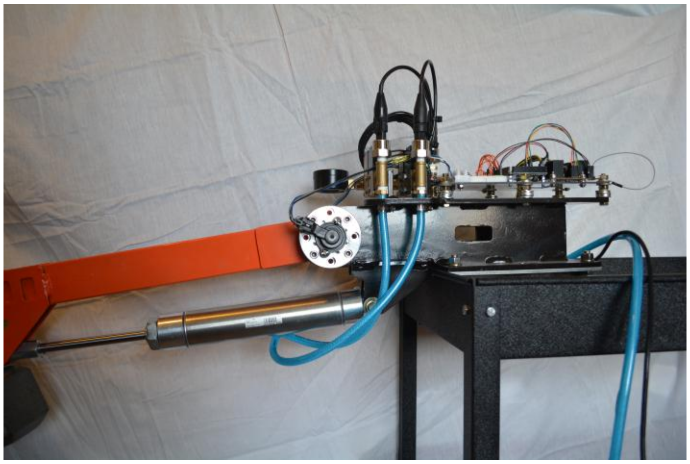


Figure 4: Completed Hardware Configuration

Pneumatic Configuration

All of the processes described in this report operate in a negative pressure configuration, or a configuration in which the medium is released from one side of the pneumatic cylinder to induce motion. Early attempts were made to operate in a positive pressure configuration, or a configuration in which medium is injected into one side of the cylinder to cause motion; However, it was quickly noted that at lower pressures the compressibility of the medium becomes much more pronounced. Almost no level of variable flow could actuate the un-loaded leg to a specific level without excessive overshoot and destructive oscillation.

Software Configuration

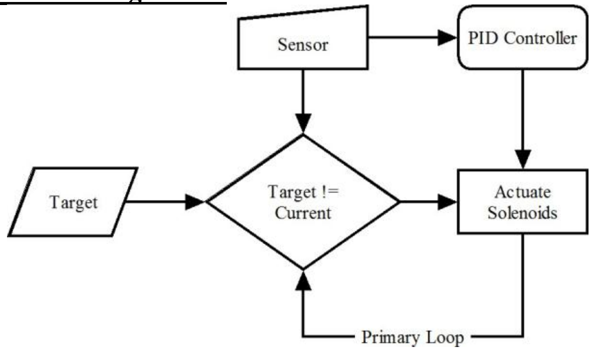


Figure 5 Primary Control Theory

The software configuration for this project is centered on a relatively simple primary command loop: check value and if the current value deviates from the target value, actuate the solenoid valves in accordance with the specified controller. Global variables are called to store data utilized by multiple subroutines, with write privileges for each global variable restricted to no more than one subroutine at any given time. The setup sequence contains the requisite input/output (I/O) calls and commands to reset the clock frequency on register 2 (digital I/O pins 9 & 10, Solenoid State valves) to 30 hertz, the optimal pulse width modulation (PWM) frequency for a standard pneumatic solenoid valve [1].

Lower level functions are delegated into a number of subroutines that are called by the main loop as required. These are divided into primary, secondary, and tertiary subroutines, depending on their level of use and importance. The primary subroutine contains the joint solenoid control scheme (JSC). The JSC subroutine accepts two simultaneous values, from 255 to -255, and utilizes these to both set the state for the directional solenoid valve and the PWM duty cycle for the state valve. The JSC performs the same function for both sets of solenoid valves. Input values for each set of solenoid valves is inverse to the other, such that a positive input value for both subroutine entries causes one set of solenoids to intake while causing the other set to exhaust, thereby causing positive linear motion in the pneumatic cylinder.

Secondary subroutines consist of subroutines that update the angular position and pressure sensors as well as the subroutines that contain the proportional-integral and proportional controllers for the angular position scheme and constant force scheme, respectively.

As the EC3048 angular position sensor is equipped with dual non-linear resistive elements of different value, the angular position subroutine is equipped with graphically formed equations that equate angular position to measured electromotive force (voltage). Within the angular positioning subroutine, all four angular inputs are averaged, so as to minimize any errors that may exist within any individual sensor. As the equipped pressure sensors output a single, proportional, voltage, the pressure update subroutine does not average values across the individual inputs. It does, however, in response to noise induced during PWM movement, average the pressure values for each pressure sensor input over a number of individual readings, so as reduce pressure sensor noise. It is noted that the averaging operation induces virtually no lag, as the current value is averaged with previous values, rather than future values.

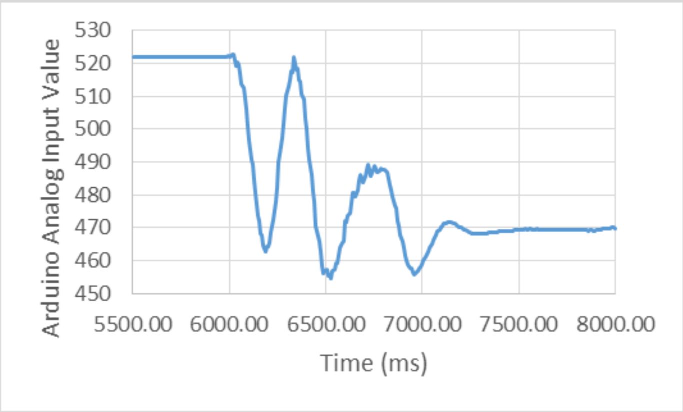


Figure 6 Pressure Variations due to PWM during Movement, After Smoothing Operation.

The angular location and pressure control scheme controllers function similar to a standard proportional-integral-derivative controller (PID). The error value, the difference between the current value and target value, is calculated (Equation 1). This value is then utilized to calculate the output value using equations 2. Both PID control schemes were manually tuned[2] to minimize overshoot at the expense of settling time. More aggressive control schemes may be utilized to decrease settling time, but those control schemes do not fall within the scope of this project.

Equation 1: PID Error

$$E = \theta_{target} - \theta_{current}$$

Equation 2: PID constant

$$PID = K_p E + K_i \int_{t_0}^{t} \tau dt + K_d \frac{dE}{dt}$$

A scale constant is applied to bring the resultant PID value into congruency with the requirements for the JSC subroutine. The resultant value directly controls the actions of the solenoids throughout the program. To avoid premature solenoid failure, a dead band or space about the target variable in which all PID function ceases, is utilized. For angular positioning, this dead band takes the form of a two degree tolerance about the target angle, while for pressure regulation, the dead band takes the form of a one half degree tolerance about the target pressure.

In both the angular positioning and pressure regulation subroutines, it is necessary to institute limits to prevent the over accumulation of the integral component of the PID controllers. To accomplish this, commands are issued within the dead band that causes the integral total to reset to a zero value, effectively causing the controllers to treat each move from the previous target as if it were the first move to have been made. While both subroutines contain the structure to be classified as PID controllers, the angular positioning and pressure regulation controllers utilize a derivative constant of zero. In addition, the pressure regulation controller has an experimentally determined integral constant of zero, due to the way in which the integral term tended to conflict with the long term operation of the subroutine. These values effectively limit the functionality of the angular positioning and pressure regulation control schemes to those of a PI and P controller, respectively.

Tertiary subroutines are considered those which are utilized only on an intermittent basis, generally for very specific tasks. They include subroutines that act to fulfill specific initialization requirements, calculate the static force at the current angle, record data to establish static force versus angle plots, detect user input and collect the data necessary to construct Bode and Phase Shift plots. The initialization subroutine, which in the current version of the program is considered defunct, is designed to move the leg to its extreme values, collecting pressure and angular measurements at each. This data would then be utilized by later processes as comparisons against current values to determine the position of the leg relative to its upper and lower sensor limits.

The static force subroutine, a more recent addition, is utilized solely by the pressure regulation subroutine. It possesses two distinct functions. First, utilizing the calculated pneumatic cylinder net force and the current angular position, the subroutine calculates the applied static load. Once the static load has been calculated, calls to the subroutine will return a value representative of the force required to hold the leg at a constant position for the current angle. This value is utilized to offset the gravitational forces on the leg as the main loop attempts to maintain a constant net cylinder pressure. The calculations performed within this subroutine utilize an equation relating static pressure, current angle, and load. This equation was derived experimentally (Figure 7) through the use of the pressure test subroutine described below.

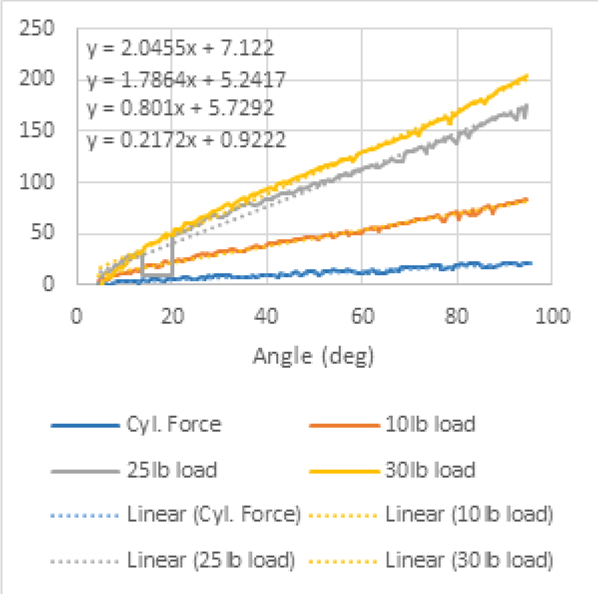


Figure 7: Static Pressure vs Angle for 0, 10, 25, 30lb Applied Weight

The subroutine utilized for the static pressure versus angular position and weight plots is simplistic by necessity. In order to negate its effect in the operational cycle speed of the program, it functions only to move the leg through a defined range of angles at a defined step increment, reporting values to the PLX-DAQ data acquisition software via serial port interface at the completion of each step. This subroutine effectively replaces the primary loop during operation, necessitating that all main calls for movement be minimized, lest operational errors occur.

Indirectly related to the primary goal of this project, a user input detection subroutine was designed to facilitate the switching of the main program from a constant angle state to a constant force state. This subroutine utilizes rudimentary data recognition to detect minute pressure variations in the net cylinder pressure value, corresponding with a user slightly lifting the leg for a defined period of time. To detect variations, a hash (or two dimensional array) maintains a log of the previous 100 pressure values (spanning approximately 700ms). An average of all 100 values is computed to establish a baseline value. When the current pressure value exceeds this baseline by a defined threshold value for a defined time period, a true state is triggered to switch the primary loop to its alternate operation state.

It should be noted that an earlier attempt at detecting user input in the form of a “double tap” on the leg is included in the main program in the form of defunct code. While the subroutine, itself, is functional, the pressure variations induced by solenoid valve PWM modulation are nearly identical to those expected during a “double tap”, causing almost constant false triggers. Attempts to filter PWM noise from the pressure sensors were unsuccessful. It was determined that a physical modification to the test platform would be required in order to bring PWM noise to an acceptable level, therefore the “double tap” detection scheme was abandoned in favor of the upward lift detection scheme described above.

Bode plot and phase shift diagram formation is accomplished through the use of an independent subroutine. Like the pressure test subroutine, the operation of the bode plot subroutine replaces the operation of the primary loop. It functions by generating a varying waveform output and passing the waveform to the constant angle and JSC subroutines to cause cyclic motion. Early attempts utilized the Arduino onboard sine wave function, but results were unacceptable as the sine wave possessed low resolution due to the 7ms operating cycle time. Revised attempts supplanted the sine function with a square wave function. Amplitude is held at a constant value while the frequency is varied from a defined minimum to a defined maximum.

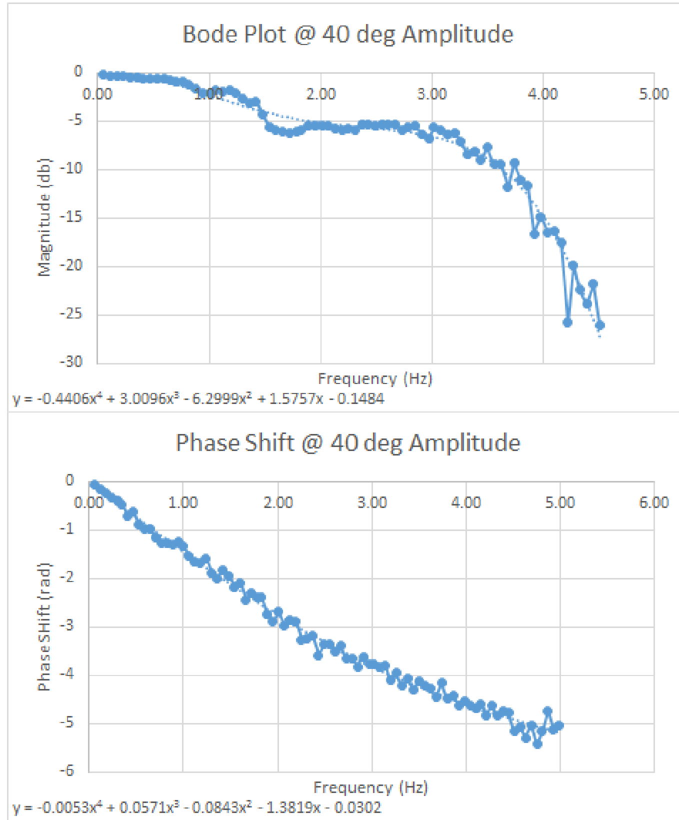


Figure 8: Bode and Phase Shift Plot at 40 Degree Test Amplitude

Variables recorded for formation of the bode plot include the targeted amplitude, the experimental amplitude (after a time of one half period has elapsed), and the frequency. Variables recorded for phase plot formation include the clock time at waveform initiation, the clock time at waveform peak and the targeted frequency. Data collection for the Bode plot subroutine is performed using the PLX-DAQ suite, with all data being recorded to an external computer for analysis. Data analysis is performed on the external computer to calculate the logarithmic power (decibels) and the phase shift (radians) based on the collected data, using equation 3 and 4. Results of the Bode and Phase plot tests are visible in figure 8. Note that the frequency response cutoff occurs at approximately 3 hertz. The phase shift may be improved by increasing the aggressiveness of the angular position PI control scheme, however, a corresponding increase in target overshoot will occur.

Equation 3: Phase Shift

$$Phase\,(rad) = \frac{t_{in} - t_{exp}}{t_{in}} 2\pi$$

Equation 4: Bode Plot Magnitude

$$Mag\,(db) = 20log_{10}\left(\frac{A_{exp}}{A_{in}}\right)$$

4. RESULTS

Defined Angular Target

For PID constants of 3, 2.7, and 0 for Kp, Ki, and Kd, respectively, a closed loop feedback control system with critical damping approximately equal to one was observed. As shown in figure 9, the positioning time for the arm with an attached five pound weight is approximately 1.2 seconds. It is important to note that slope of the line denoting the angular position verses time is not a smooth curve. This is due to the flow rate of air varying in proportion to the pressure difference existing across the actuation solenoid valves. In short, if the orifice (solenoid valve modulated via PWM) is held constant, the volumetric flow rate through the orifice will vary as the pressure difference approaches zero.

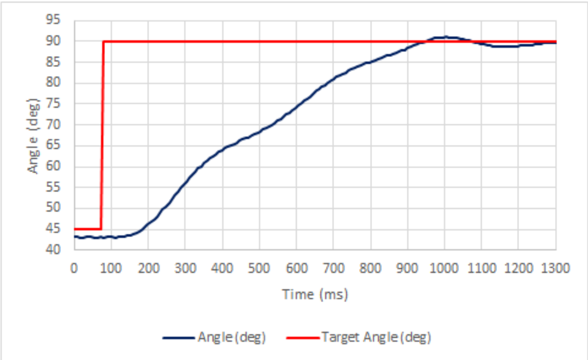


Figure 9: Defined Angular Position, Upward Stroke

Multiple variations of the PID control constants were experimentally examined, with the listed set corresponding to lowest overshoot while still reaching the target region within a reasonable duration. As previously noted, increasing the proportional constant will cause the angular displacement to approach the target in a much shorter time period, however it will overshoot and oscillate for an unacceptable period of time (greater than 10 times that required to initially reach the target). Note that between time 0ms and 80ms the current angle does not equal the target angle; This is due to the implementation of the dead band previously mentioned, which serves as a region for which all solenoid valves are closed. Without an implemented dead band, the PID control will bring the leg into congruency with the target angle, however long period, convergent, oscillations will occur as the PID continuously overshoots and corrects in the negative and positive directions. This effect is due to the compressibility of the medium (air) and will be further expanded upon in the discussion section.

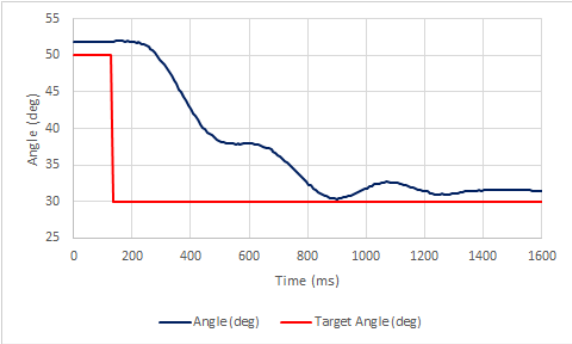


Figure 10: Defined Angular Position, Downward Stroke

Conversely, when observing the downward movement of the same PID configuration previously described, it is first noted that the actuation speed is significantly increased over that of the upward stroke. This is due to the fact that the downward stroke is being assisted by forced due to gravity, which are causing the exhausting side of the pneumatic cylinder on a down stroke to be under a higher pressure than that of the exhausting side on the upstroke. As previously noted, with a compressible medium volumetric flow rate is proportional to the pressure difference experienced. It is therefore concluded that the downward stroke experiences a greater volumetric flow rate than the upward stroke due to this additional force.

The second noteworthy point is that which occurs at the time of 900 milliseconds (Figure 10). While the angular position does not pass below the targeted angle, a rebound occurs. This is due to the solenoids terminating the flow momentarily before the inflection point is reached, effectively causing a mass – spring system to be formed. The resulting oscillations are the result of this, as the PID controller attempts to position the leg as it oscillates beyond the confines of the dead band.

When the leg apparatus, currently in a static position, is imparted with a force impulse (Figure 11, point A), the controller is capable of restoring the leg to the same position within approximately 1500ms. Such is the case when the leg is imparted with an impulse force in the form of a 25 pound weight. The leg apparatus experiences a deflection of 14% before the PID controller is able to compensate for the added mass. After a time period of one second the leg apparatus is restored to within 80% of its original position, with full restoration occurring within 1500ms. The pneumatic cylinder experiences a force increase on the order of 100 pounds, demonstrating the effects of the geometry of the leg and the force amplification factor it possesses. When the weight is removed (Figure 11, point B), the leg experiences a much less pronounced reaction. This is again due to the pressure differences that are experienced across the pneumatic valves.

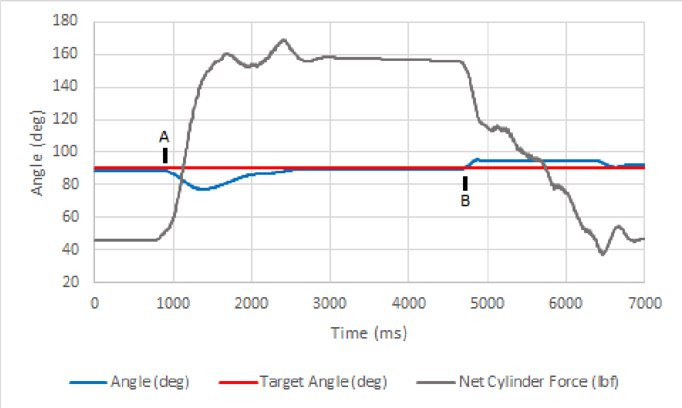


Figure 11: Effects of Impulse on the Static Apparatus

Defined Constant Force

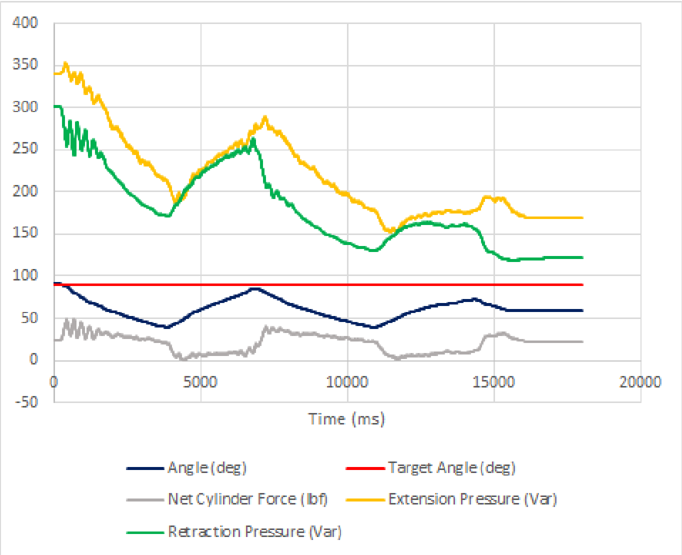


Figure 12: Defined Constant Force Compensation with User Controlled Motion

During testing in which the control system attempts to compensate for forces exerted on the leg apparatus, effectively rendering it "weight-less", all motion is controlled by user stimulus. The test cycle consisted of two moves over the majority of the apparatus's available travel range, with the first move performed under force, while the second move is performed with as little force as possible.

As is shown in figure 12, the control loop attempts to compensate by increasing or releasing pressure in each side of the pneumatic cylinder. The result of this is visible through direct comparison of the Extension Pressure and Retraction Pressure data sets on the associated plot. As they are approximately linear, this indicates a proportional relationship with a low resultant "Net Cylinder Force".

The results of this section are somewhat more evident through the examination of a constant force configuration which is loaded with a weight near the maximum that the system is capable of moving. The plot below is the same test, repeated under a 35lbm load, causing all pressure values to be exaggerated over those visible for the no load scenario.

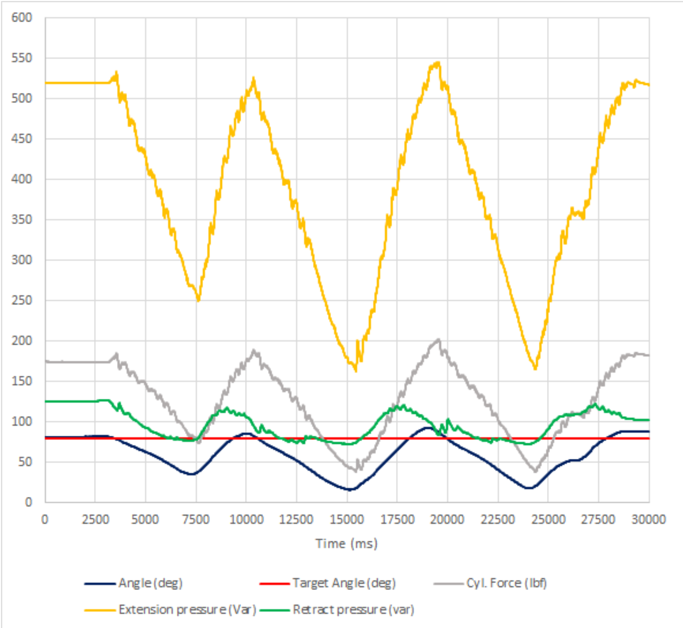


Figure 13: Defined Constant Force Compensation with User Controlled Motion (35lbm Load)

When the system is placed under a significant load, the constant pressure relationship is clearly visible, in the form of a linear relationship between the extension and retraction pressure values. The difference in slope between the extraction pressure and retraction pressures is due to the change in relative arm lengths with respect to gravity as the angle changes. Were the torque about the knee joint not dependent on leg angle, it would be reasonable to expect the extension pressure and retraction pressure slopes to be equal in magnitude, as their difference would hold constant to maintain the leg force at a constant magnitude. The required user force to induce motion in this trial is less than five pounds of force, with little to no discernable difference depending on direction of actuation. Lastly, it is important to note that the response of the constant force configuration is high, with little lag between input and reaction.

A variation of the constant force routine, that was specifically tuned to compensate for a 40lbm end load, was implemented. It utilizes an algorithm (Fig. 14) that equates the current angle of the leg with the required cylinder pressure differential to maintain a zero net resultant leg force. The input variable is the angular position of the leg, which is manually positioned by the user. A proportional controller then attempts to bring the net resultant leg force to zero by varying the PWM duty cycle of the control solenoids. The results of this variation are visible in Figure 15.

This variation provided a higher degree of compensation than that of previous versions, as the "Net Cyl. Pressure" value demonstrates. Fluctuations occur about the zero point, but do not exceed a relative value of 50 once startup transients have dissipated, indicating relatively little force is imparted on the leg by the system, beyond that required to maintain the current position. It is important to note that the weight utilized in this trial (40lb) is reaching the limit that this experimental apparatus is physically capable of controlling due to the constraints imposed by the operating pressure and diameter of the pneumatic cylinder. In fact, the retraction side (Ret Pavg) of the pneumatic cylinder is tending towards a fully actuated exhaust state, causing force compensation to be accomplished solely by the extension side of the cylinder (Ext Pavg).

The results of this variation are promising, with leg position maintained indefinitely with no discernable movement. Error due to leakage is not problematic as the control system is able to compensate for the decreased pressures as they occur. It is important to note that under extreme pressure and force variations, it is possible for this configuration to enter a destructive feedback loop, in which pressure variations induced by leg motion are reinforced by the controller's attempts to equate the net force. This is especially true at angles less than 20 degrees, where the damping constraints imposed by the flow of air through the control solenoids at low pressure differentials are not present. To avoid such a situation, it is advisable to implement a software based separation scheme between the angular position input and the controller to avoid reinforcing destructive movements. A simple scheme would be to disable the controller when high amplitude oscillations occur, enabling it only once the oscillations have been reduced below acceptable values.

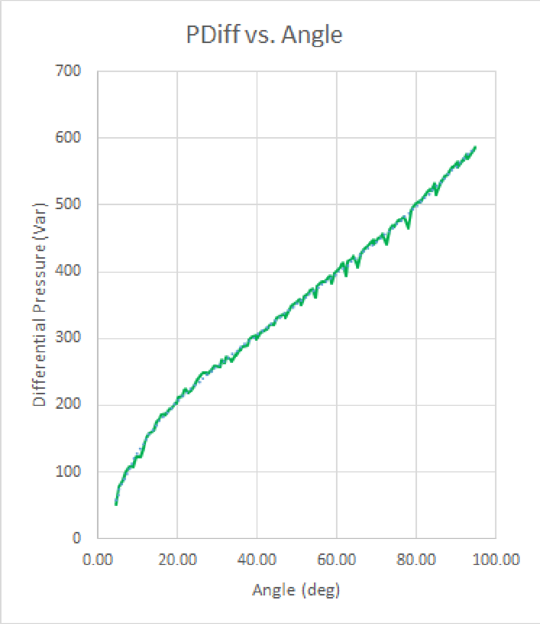


Figure 14: Constant Force, Differential (Static) Pressure vs. Leg Angle

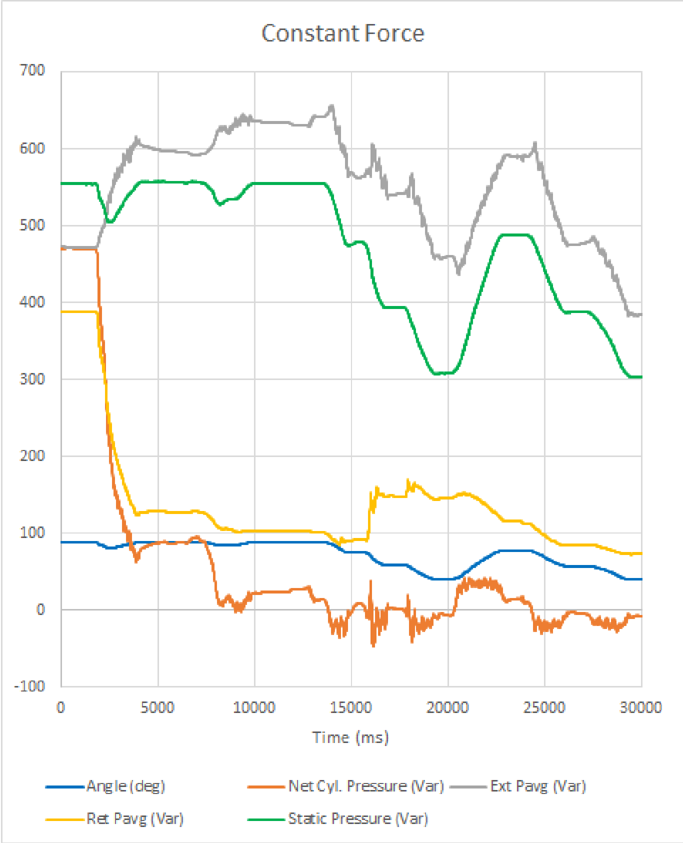


Figure 15: Force Compensation with User Controlled Motion, 40lbm end load

5. DISCUSSION

As the results have shown, pneumatics are a viable candidate for precise positioning control. With a properly tuned control system, accuracy within 0.1 degree (0.05" linear) is possible, with a repeatable precision of 0.25 degree (0.08" linear). However, at such accuracy levels, the controller transition speed will need to be sufficiently small so as to prevent oscillation about the target point, as the pneumatic cylinder will tend to act as a very close approximation to a mass-spring oscillator.

The utilization of pneumatic control systems in applications requiring force limitations, such as human interface devices, is also shown to be viable, but with limitations. Repeatability of the constant force aspect of this experiment is excellent, with little to no deviation noted between trials and a current force compensation of over 85% (Users need apply the remaining 15% to induce motion). The latter revision shows the most promise, with exceptional response and force attenuation. It does, however, possess a tendency to experience destructive oscillation during fast, high magnitude, positioning operations.

In addition, the results of this project have demonstrated that operating in a negative configuration (system initiates with a pressurized system and medium is released to induce motion) offers exceptional position control relative to a positive configuration (system movements initiate at atmospheric pressure, medium is added to induce motion). However, while the negative configuration offers increased position control and response, a positive configuration will always possess a higher transit speed at low applied forces due to the properties of the gaseous medium as it passes through the restriction points (solenoid values).

The limitations imposed on a pneumatic control system are extensive and will be divided into two classes, general limitations and limitations specific to a certain aspect of this project. General limitations that impact pneumatic control systems are primarily related to the compressible nature of the medium. Pneumatic systems should, first and foremost, possess a supply source of sufficient flow rate to prevent large pressure variations during operation, as such variations can easily cause a formerly stable control system to become unstable as the pressures (and flow rates) drop below expected values.

Next, pneumatic systems that utilize PWM modulation to vary the volumetric flow rate through a standard pneumatic solenoid valve must take into account the required actuation speed and size their solenoids appropriately. While smaller solenoids will offer higher response due to the reduced mass of the electromechanical components, they will possess an extremely low system actuation rate due to flow restrictions through the valves, themselves. In addition, care must be taken to position pressure transducers as far as possible from sources of noise, such as solenoids acting under PWM modulation. The inclusion of vibration canceling design methodologies, such as restriction orifices or interference feedback loops may help alleviate noise at pressure transducer ports.

Any pneumatic system must take into account the compressibility of the medium when the method of operation is being designed. As was noted at the beginning of this report, a positive configuration will allow the highest activation rate and speed, but only when a large pressure differential exists between the medium (air) supply and the pneumatic cylinder. Whereas operating in a negative configuration causes a significant increase in the density of the medium leading to a more controllable flow rate for large pressure differentials. Lastly, a pneumatic control system must be designed such that the maximum pressure required must never exceed one half of the maximum supply pressure. This is due to the system requiring a positive pressure differential to adequately dampen the system. Operating a pneumatic control system in such a way as the required pressure equals the supply pressure, would be identical to operating a car without brakes: The system will move but be unable to stop before it hits a wall (the limit of pneumatic cylinder travel).

Limitations specific to the “defined target angle” configuration of this project include limitations on actuation speed and frequency, as well as limitations due to the operation of the PID control scheme. The limitations on actuation speed are due both to the previously mentioned negative actuation scheme and the relatively high inertia associated with the simulated leg. Methods to alleviate these limitations include increasing the pressure differential between the supply & destination and/or changing the geometry of the simulated leg so as to decrease the torque reduction ratio about the axis of rotation.

The use of a traditional PID controller demonstrated a severe inability of the controller to compensate for changing pressure differentials (medium compressibility) at the different stages of its operation. This effectively caused a PID controller, which had been tuned at one pressure, to malfunction at a different pressure. To maximize position accuracy, it is recommended that the proportional constant be set to a lower than normal level, and the integral term of the controller utilized for the bulk of movement control, as this will allow the controller to compensate in situations where the pressure differential is minimal near the end of the movement (minimal pressure differential necessitates a higher PWM duty cycle to induce the same volumetric flow rate as at higher pressure differential).

The only configuration specific limitations applicable to the constant force configuration, other than those already mentioned, are the inability for the controller to function without a proportional-only control scheme and the difficulty in accurately determining the static pressure required to hold the leg in equilibrium during motion. A possible solution is to incorporate a more comprehensive user interface device so as to isolate user imparted force from the positional force detection scheme. In addition, the implementation of a proportional controller that utilizes a predictive algorithm to filter pressure transducer noise and proportionally control the volumetric flow rate through the solenoids in response to expected pressure variations as the angle changes would alleviate the control limitation.

6. CONCLUSION AND FUTURE WORK

In conclusion, the results have shown that a pneumatic control system is a viable candidate for precise position and force control, provided the innate limitations of the system have been taken into account. The major limitations of a pneumatic control system are an inability for the control system to compensate for impulse loading without deflection and a requirement that any control system be able to quickly compensate for changes in pressure differentials without excessive overshoot.

Future modifications to the project apparatus will include a redesign of the pressure transducer subsystem, so as to reposition the transducers in order to minimize noise due to PWM solenoid actuation. It addition, the implementation of a predictive control system, tuned to compensate for the changes in medium compressibility throughout the control cycle, may be implemented. Modifications to the testing apparatus will include the incorporation of volumetric flow transducers and an accurate force application device so as to allow precise measurement of the volumetric flow rates into and out of the pneumatic cylinder under various loads. This data should allow for further characterization of the response of the system.

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