DELTA
COMPUTER SYSTEMS, INC.

IFPE 2005 Technical Conference

Title: Trends in Closed Loop Motion Control and Dynamic Response of Fluid Power Systems

Abstract:

Machine Design requires sophisticated automation of hydraulic and pneumatic motion control for improved speed, accuracy, and extended machine life. New and retrofit projects increasingly use motion controllers, servo or servo grade proportional valves and higher resolution transducer technology.

Dynamic hydraulic or pneumatic system response is a key aspect of this requirement and can be a control challenge since dynamic responses are inherently nonlinear. Today's motion controller manufacturers are developing methods to address these nonlinear dynamics; however, it is still vital that hydraulic system be designed to be as linear as possible.

Motion controllers along with associated servo quality valves and feedback devices deliver more complete and consistent closed loop control. Motion controllers using advanced algorithm control techniques now account for parameters beyond typical PID including accelerations and velocities. This allows for faster settling times and smaller following errors while delivering precise, sophisticated position and pressure profiles for servo-hydraulic or pneumatic applications. This is accomplished through easy to use software programs, set up procedures, tuning and graphing tools yielding positive results sought by engineers, designers, technicians and operations personnel.

Since communication to and from motion controllers is of growing importance it is necessary to have flexible connection capabilities. Controllers using open fieldbus communications techniques such as EtherNet/IP or PROFIbus for HMI or PLC communications deliver that connectivity. For feedback devices, connecting seamlessly to multiple types of mix and matched devices such as SSI (synchronous serial interfaces), MDT (magnetostrictive displacement transducers), encoders, analog

transducers and digital I/O allows flexibility for multiple configurations.

Combining good machine design with these innovations in motion control can result in lasting value at lower lifecycle cost.

Outline of the Paper -

Demand for smoother motion is driving innovation in motion control

Dynamic response profiling can diagnose system performance issues and improve motion controller operation.

Choosing well-connected motion controllers improves system flexibility and performance

Smoother and faster moves result when using control parameters beyond simple PID. Pressure/force control in addition to position/velocity control can improve machine productivity.

Graphical tuning tools shorten development times and provide more optimal results.

Designers and machine owners should expect quality support from their motion controller manufacturers.

Author's Background:

The author is Peter Nachtwey, President of Delta Computer Systems, Inc.

Nachtwey has more than 20 years of experience developing motion control systems for hydraulic, pneumatic and electric systems in industrial applications. He graduated from Oregon State University in 1975 with a BSEE and served in the U.S. Navy until 1980. He was a systems engineer for I.E.C.C. and Applied Theory Inc prior to joining Delta Computer Systems. Nachtwey became President of Delta in 1992.

Delta Computer Systems, Inc. founded in 1977 and incorporated in 1982 develops, manufactures and markets industrial motion control products. Delta's customers are in the forest products, metals, automotive, entertainment, plastics, glass, and food processing industries.

Delta manufactures the RMC Family of motion controllers for multi-axis servo-hydraulic and servo-motor applications featuring fieldbus communications, Ethernet, PROFIBUS-DP, Modbus Plus, Serial and Digital I/O (25+ protocols). Users can mix and match I/O transducer modules with Delta products to allow more than 500 configurations. Powerful RMC software is used to easily setup, program, tune and diagnose applications.

IFPE 2005 Technical Conference Theme: Lasts Longer, Costs Less – Innovations in Machine Design

Paper Title: Trends in Closed Loop Motion Control and Dynamic Response of Fluid Power Systems

Author: Peter Nachtwey, President, Delta Computer Systems, Inc.

The fundamental purpose of a motion controller is to provide fast, consistent closed-loop control. Over the last two decades, motion controller manufacturers have gained significant knowledge from control systems theory and real world applications experience in enhancing their offerings to improve machine productivity and decrease lifecycle costs.

No matter how much attention is given to the selection and programming the motion controller, however, system control performance can never be fully optimized if the fluid power system design has physical limitations. Therefore, one of the trends that we see taking on increasing importance in system design is profiling the machine's dynamic response with simulations before the motion controller programming and tuning begins, preferably prior to fluid power system build.

System Dynamic Response Simulations

Good fluid power system design needs to start with proper sizing and location of the hydraulic or pneumatic system components. Mathematical simulation techniques that analyze dynamic response can predict and validate correct system design choices.

Factors that a hydraulic servo simulation program must take into account include the following:

1. To predict whether an actuator is able to move quickly enough, one must calculate maximum velocities, accelerations and decelerations in each direction.
2. One must know all the forces acting on the actuator including the force it takes to push the oil out the exhausting side of the cylinder.
3. Special care needs to be given to decelerating while extending. The rod surface area is smaller than the cap surface area that accelerated the load. The pressure on the rod side can intensify to pressures above the supply pressure.
4. One must be aware that as the pressures in the system changes the system gain is also changing.

Net Applied Force Issues

If one thinks about controlling a dynamic fluid power system as a series of design steps, the first key step is to know what forces are involved in the system and determining how to control the net applied force. This includes the force on the working side of the piston, the force on the exhausting side, and the inertia and friction that must be overcome, to accelerate and decelerate the load at the desired rates. Many higher performance systems spend much of their time accelerating and decelerating as they quickly move from point to point. In some cases the theoretical top speed is never reached because the system must decelerate before top speed is reached.

Cylinder Sizing

A common design oversight is to use cylinders whose diameters are too small. In these cases the actuator may not deliver enough force for the system to reach maximum speed during a typical move. Increasing the diameter of the cylinder increases the natural frequency of the system. The higher natural frequency allows the motion controller to support higher system bandwidth when properly tuned. This allows for fast accelerations and decelerations. After this step, pressures and gains should next be considered.

Supply Pressure and System Gains

The third step in designing a precisely-controllable fluid power system is picking the supply pressure. Increasing the supply pressure increases the system gain allowing the cylinder to will go faster with the same control signal. Increasing the system gain has an added beneficial side effect. It helps to keep the system linear and therefore the keeps system gains as constant as possible. This is important because motion controllers must compensate for the rapidly changing system gains.

Keep in mind that system gain falls as the applied force on the load increases.

$$\sqrt{K \cdot Ps \cdot Ape}$$

System gain with no load where
K is flow coefficient
Ps is supply pressure
Ape is area on the powered side of the piston

$$\sqrt{K \cdot (Ps \cdot Ape - Fl)}$$

System gain with load where Fl is load

If the system applies so much force that there is little pressure drop across the valve, the system gain can approach zero. This happens when the pressure in the cylinder reaches the system pressure. At this point opening the valve more will not cause more oil to flow. The system gain gets smaller as the system goes between no load point and the point where the gain is zero. For best performance, it is necessary that the pressure in the cylinders be kept to about the middle third of the range between zero and the supply pressure.

Because the supply pressure plays such an important part it is also important to keep the supply pressure as constant as possible. Pumps alone are not fast enough to respond to dynamic fluid pressure demands in a high performance system (The pressure could drop significantly before the pump started to respond). Therefore an accumulator is required. As a rule of thumb, make the accumulator large enough so the pressure does not drop more than ten percent during system operation, and charge it with just enough fluid to insure that it does not run out of oil. Keeping the gas bubble in the accumulator as big as possible is the key. For example, the pressure drop that occurs as the gas bubble expands from 5 to 5.5 gallons is much bigger than the pressure drop that occurs when the gas bubble expands from 9 to 9.5 gallons, because the change is a smaller percentage of the original size.

Next, select a servo or servo-quality proportional valve with a zero-overlap spool, sized such that it will supply the necessary flow for the diameter and desired speed of the actuator. Plus, one should consider the response of the valve: the frequency response and phase lag at the frequency at which the valve is going to be used. This requires care in reviewing the valve specifications to ensure the valve is adequate for intended use. *Caution*: Most high-speed applications are running with the valve opening to the 80% range, but some valves' frequency and phase lag ratings will drop by half when the specification for from 10% control signals to 80% control signals.

A Real World Example

We were recently called in to diagnose problems that a metals industry customer was having with getting their system to operate correctly. After obtaining customer specific data we performed a hydraulic servo system evaluation. Our simulation modeling of the current system suggested that cylinder sizing was the issue. We could see graphically that more force was necessary, and the simulation was changed to evaluate bigger diameter cylinders. The result was that the customer had to change out the 2-inch diameter cylinders they had originally specified for 3.25-inch diameter cylinders and the valves needed to be replaced with ones that were appropriately large. Figure 1. is a typical Control and Spool position vs. time graph for the three different cylinder sizes evaluated. A comparison between the graphs in figures 2a and 2b shows the dramatic improvements in system control that were predicted by increasing the cylinder diameter in the model. See the appendix of this paper for more details on tables of specific data, graphs for 2.5-inch cylinder simulations and support graphs of force vs. time and flow vs. time.

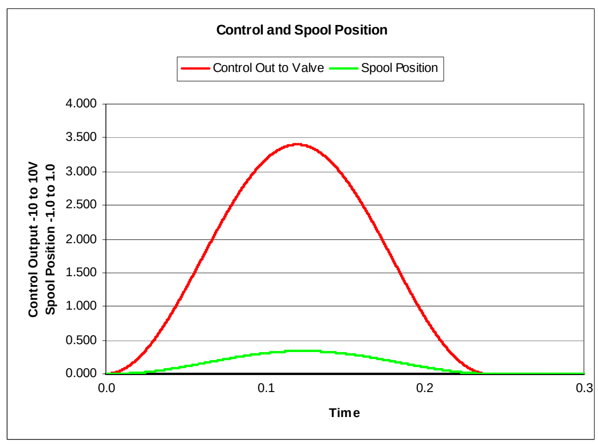


Figure 1. Typical Control Output vs. time for Control and Spool Position

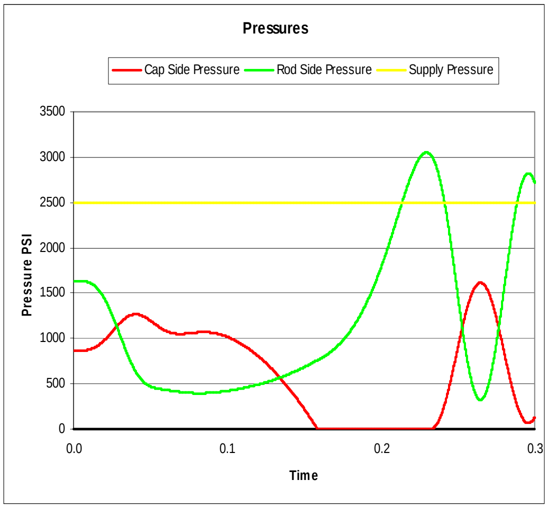


Figure 2a. 2-inch cylinder

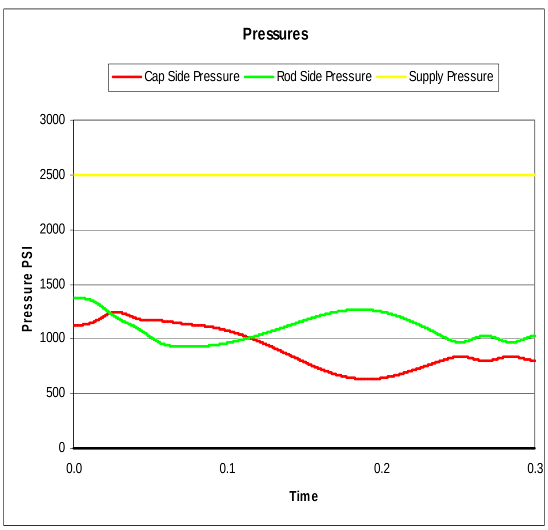


Figure 2b. 3.25-inch cylinder

Cylinder bore size simulation showing Cylinder Pressures vs. time. Note Figure 2b. demonstrates relatively stable pressures.

Notice in Figure 2a how the pressure intensifies on the rod side and cavitates on the cap side of the 2-inch cylinder. In this case when the servo controller tried to apply more force on the rod side the force actually decreased because the oil escaped the rod side into the oil supply. Obviously, this is not a controllable condition. The larger 3.25-inch cylinder (in Figure 2b) accelerates the load at the same rate but the pressures don't to change as much because of the greater surface area. The pressures are in the middle of the pressure range and there is plenty of pressure drop across the spool lands to maintain control. Referring back to equations above one can see the ratio of the load to no-load gains is also more constant.

For good background educational information on hydraulic servo systems, we suggest you consult the book *Basic Electronics for Hydraulic Motion Control* by Jack L. Johnson. To go even further, refer to the author of this paper's published articles and papers about getting optimal results from today's digital hydraulic motion controllers.

Once the elements in the hydraulic or pneumatic circuit are selected and sized, the choice and application of the motion controller and transducers can enable the further optimization of the system's function. Motion controller developments are improving the productivity and decreasing long-term costs in three key areas:

Enhanced Connectivity

The use of flexible connections to other system elements has increased the usefulness of motion controllers, as on-the-fly data communications to and from the devices has enabled more precise and adaptive control. Standardized interfaces enable mixing and matching best-in-class components, freeing machine owners and designers from being locked-in to the products of a single manufacturer.

Controllers using fieldbuses such as EtherNet/IP and PROFIbus can have motion programs and parameters downloaded at high speed from PLCs, while the same interfaces can be used to upload motion status information to HMIs and quality control systems. Delta was one of the first companies to be ODVA and PTO certified with this communications capability having recognized this as a practical trend in the late 1990s.

EtherNet/IP deserves special consideration as a method for time-deterministic data and information transfer. By 2001, Ethernet became the most popular communications option for our RMC100 Series motion controllers. And market acceptance has continued to increase, resulting in a doubling of motion controllers with Ethernet over the past two years (2002 to 2004). Most of these are using Ethernet/IP communication protocol.

For feedback devices, connecting seamlessly to devices such as SSI (synchronous serial interfaces), MDTs (magnetostrictive displacement transducers), absolute and incremental encoders, analog transducers and digital I/O interfaces, enables the motion controller to execute high-speed control loops without interfacing delays.

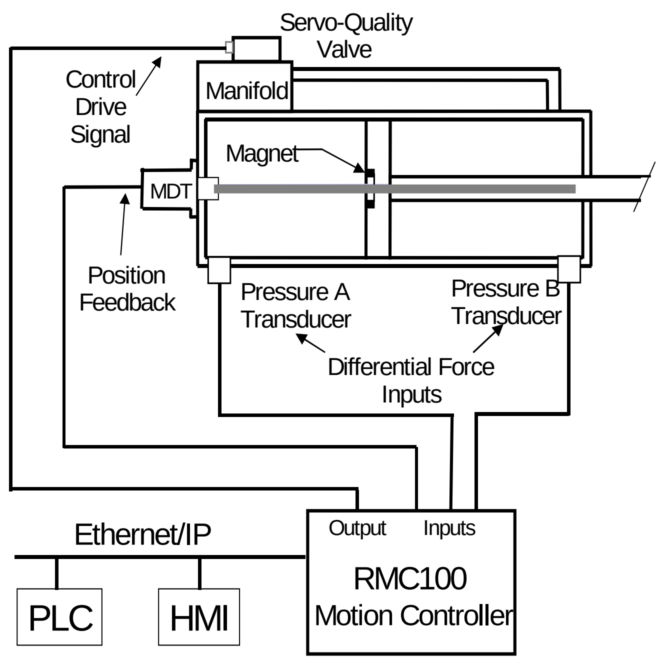


Figure 3. A well-connected motion controller

Advanced Control Techniques

Better motion control often means moving from open loop to closed loop control. “On-and-off” or “bang-bang” controls in hydraulic and pneumatic applications are increasingly being replaced by closed-loop control of proportional and servo valves to yield smoother operation.

Motion Controllers using advanced algorithms now support the use of control parameters in fluid power applications beyond typical PID components to include feed-forwards (i.e., predictive terms) for directional velocities, accelerations and decelerations. Using predictive parameters allows for faster settling times and smaller following errors while delivering precise and sophisticated motion profiles in fluid power applications.

In pneumatics for example, there is growing interest in closed loop motion control with active damping allowing smoother, faster and more precise motion compared to previously accepted standards.

Beyond simply controlling position or velocity, more motion control applications are adding control of pressure or force to the mix. This is particularly useful in press applications where execution of tightly controlled pressure profiles can enhance the quality and consistency of production output.

Another trend is for motion control companies to provide a family of controller products that have multi-axis control capabilities. Figures 4a and 4b show two motion controllers that have advanced multi-axis capabilities including “camming” and gearing of the operation of slave axes in relation to masters.

Figure 4a. Multi-Axis Motion Controller

Figure 4b. 1 & 2 axis Motion Controller

Optimization Support

After setting up the fluid power system, including connecting the motion controller to transducers, system controllers and other devices, and establishing basic motion functionality, the tuning and optimization process begins. While early motion controllers provided little support for optimizing systems, the newest controllers come with powerful development and tuning tools.

In order to tune an application easily and quickly during factory testing and field startup, one needs easy access to tuning parameters and step-by-step documented procedures to optimize control of each axis and also for different axes working together. Graphics capability is essential. Graphical tools not only assist in the tuning process during design and startup but also can be used for troubleshooting should a production hiccup occur.

Figure 5 shows the computer screen that Delta's RMCWin programming software provides to allow parameter tuning, and figure 6 shows a graph that is generated by RMCWin to allow the comparison of target and actual motion profiles.

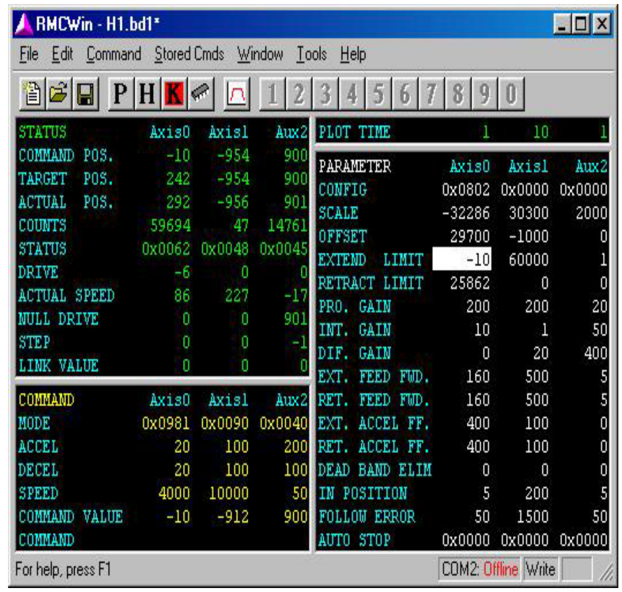


Figure 5.
Shows a display screen indicating status, command, plot times and parameters for an easy overview of axis information

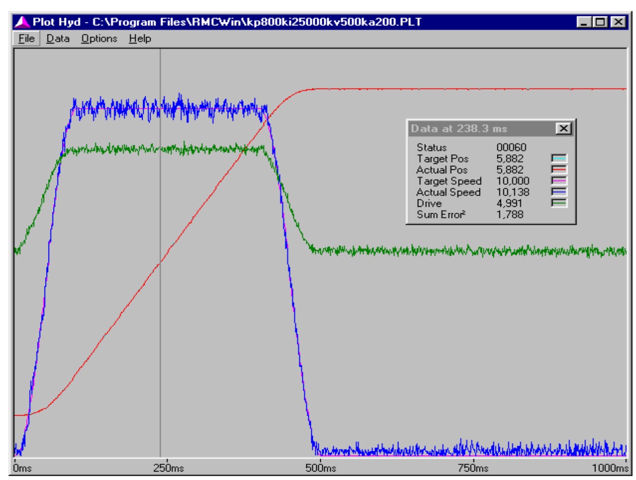


Figure 6. shows a plot of position speed and drive for a position axis.

Looking ahead

In the near future, even more powerful tuning tools will become available. Automated tools, such as improved and robust "tuning wizards" can exercise machines' dynamic response ranges to allow OEMs, systems integrators and/or end users to easily move beyond trial and error tuning methods. By establishing a process that focuses on system results instead of motion control loop gains, this type of tool saves time while yielding long haul benefits of machine life extension and minimizing downtime. *For more details, refer to the press release on Delta's Tuning Wizard product and a related technical paper available at* www.deltamotion.com.

Customer Support

Using motion controllers becomes easier when one realizes that motion control manufacturers are approachable and interested in discussing applications related to a machine's dynamic response during the development process. Machine designers should not be afraid to pick up the phone and discuss their application with a technically adept and responsive motion control manufacturer. Together they can determine what controller configuration and control algorithms best serve the needs of his or her application.

Conclusions

Fluid Power applications are utilizing closed loop motion control for hydraulics and pneumatics in more and better ways. The last 20 years has led to significant gains in applications experience, with better understanding customer requirements and through more scientific efforts to determine the system dynamic response affects (or personalities) of machines. With good fluid power system design and simulation and by selecting the best motion control approach, one can connect, control and optimize each system for best results.

APPENDIX

Hydraulic Servo Simulation Graphs
Metals Industry Case Study

Three Cylinders

Three different cylinders were simulated with diameters of 2.0 to 2.5 to 3.25 inches based upon specific data provided by a metals industry customer.

Tables

Tables 1, 2 and 3 below show the specific data that was used for the results shown in figures. Appendix Figures 1 – 6 show the dynamic response effects of three different simulations based upon values described for three different size cylinders using Delta's Hydraulic Servo Simulator.

Figures

Figure 1 is typical to all three sizes and plots control output and spool position over time. Figures 2a and 2b plot the different cylinder size affects on pressure versus time. Figures 4 and 5 plot different cylinder size flows versus time.

Three Different Responses

From these figures one can see this application simulates three very different responses to a sinusoidal input. Appendix Figure 2 shows how a 2-inch cylinder will reach cavitation as the cap side pressure goes to zero. Notice also that the rod side pressure is oscillating. Appendix Figure 2b shows a 2.5-inch cylinder that also experiences cavitation but has less oscillation. Figure 3a shows a 3.25-inch cylinder that has no cavitation and reasonable practical cap side and rod side pressure variations.

This is better than figure 3a for the 2.5 inch cylinder and much more acceptable than the 2.0 inch cylinder graph. Figures 4 and 5 plot flows over time and demonstrate that flow becomes more sinusoidal like the control signal as the cylinder size increases making the system more linear thus making more precise control possible. Figure 6 shows force vs. time and can be compared to the pressure vs. time graph and is similar in response excepting the magnitudes are quite different.

Hydraulic Servo Simulator
Table 3 – 3.25 inch cylinder

Time Increment	0.00001	Cylinder Diameter	3.250
Load Weight pounds	3000	Rod Diameter	1.375
Load Type	0	Cylinder Length	22.000
Damping Force lbf/ips	10	Initial Position	10.000
Valve GPM	14.53	Ratio	1.000
Rated PSID	145		
Valve Response Hz	42		
Supply Pressure	2500		

Data Tables for three different cylinder sizes

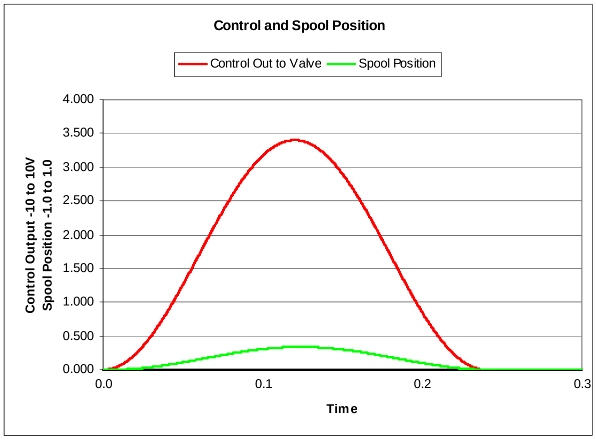
Hydraulic Servo Simulator
.
Table 1 – 2.0 inch Cylinder

Time Increment	0.00001	Cylinder Diameter	2.000
Load Weight pounds	3000	Rod Diameter	1.375
Load Type	0	Cylinder Length	22.000
Damping Force lbf/ips	10	Initial Position	10.000
Valve GPM	14.53	Ratio	1.000
Rated PSID	145		
Valve Response Hz	42		
Supply Pressure	2500		

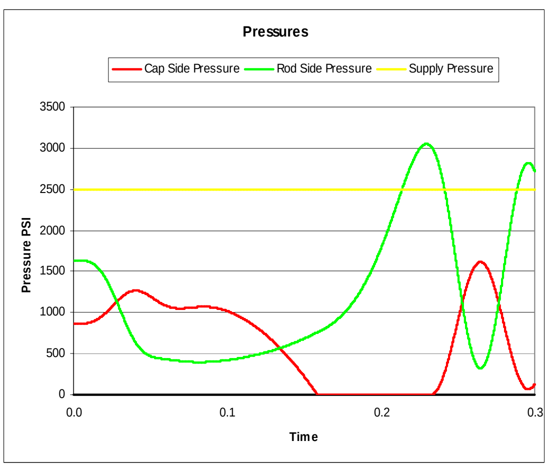
Hydraulic Servo Simulator

Table2. 2.5 inch cylinder

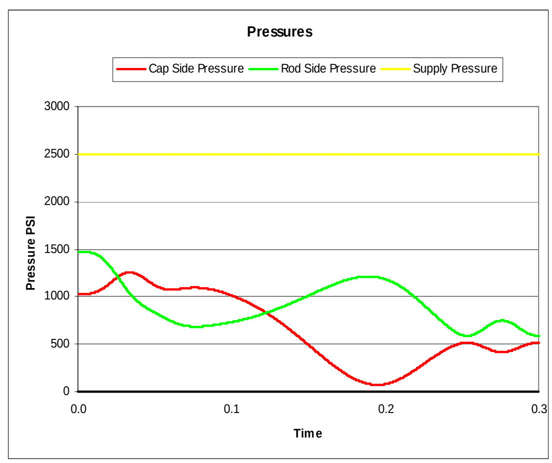
Time Increment	0.00001	Cylinder Diameter	2.500
Load Weight pounds	3000	Rod Diameter	1.375
Load Type	0	Cylinder Length	22.000
Damping Force lbf/ips	10	Initial Position	10.000
Valve GPM	14.53	Ratio	1.000
Rated PSID	145		
Valve Response Hz	42		
Supply Pressure	2500		



Appendix Figure 1. Typical Control Output vs. time for Control and Spool Position



Appendix Figure 2a.
Cylinder Pressures vs. time for a 2.0 inch compared to a 2.5 inch bore simulation



Appendix Figure 2b.

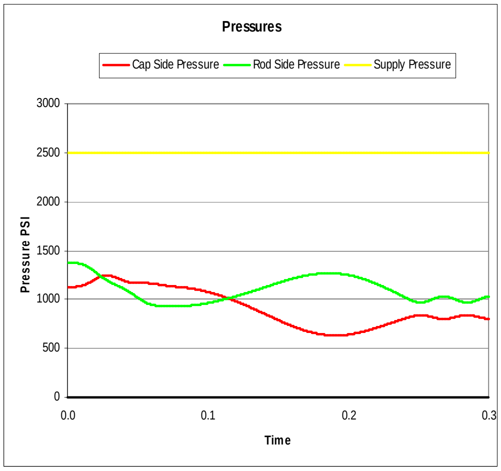


Figure 3a. 3.25 inch

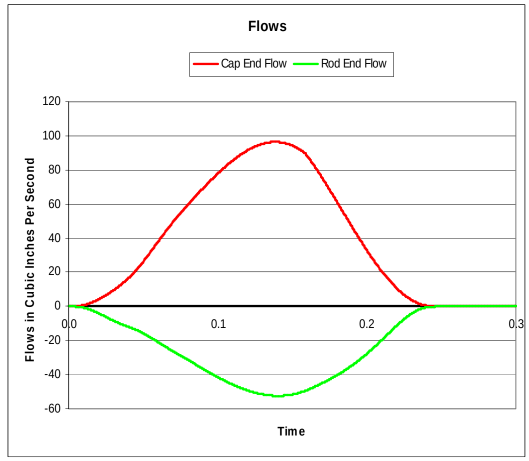


Figure 4a.

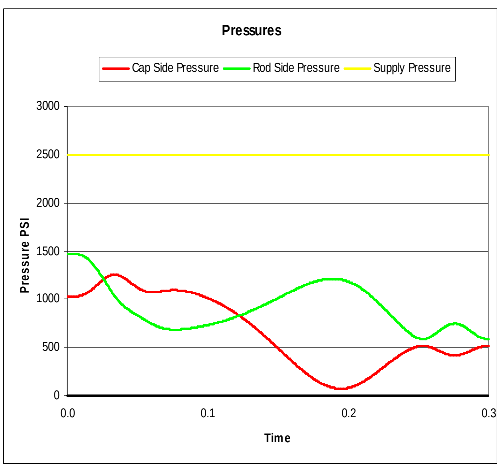


Figure 3b. 2.5 inch

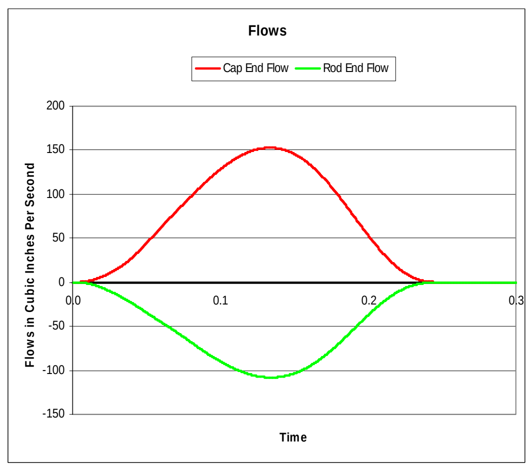


Figure 4b.

Cylinder Pressure vs. time for a 3.25 inch cylinder bore simulation shows relatively stable pressures compared to a 2.5 inch cylinder

Appendix Figure 4. Flow vs. time for a 2.0 inch cylinder compared to a 2.5 inch cylinder

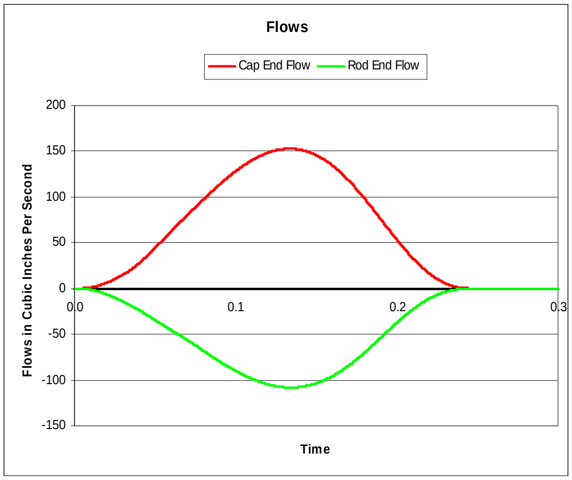
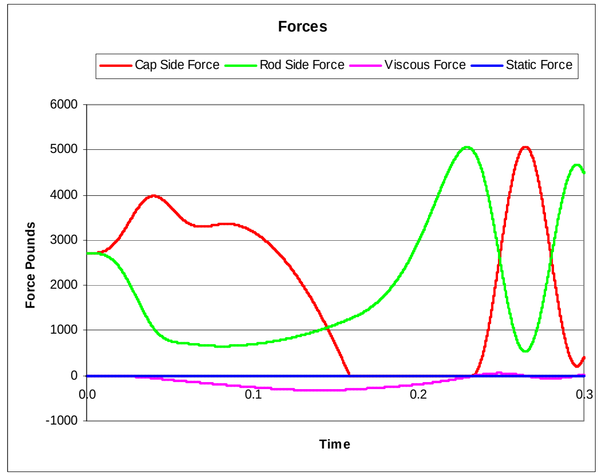


Figure 5a.



Appendix Figure 6.

Example of Different Cylinder Forces in Pounds vs. time for a 2.0 inch cylinder

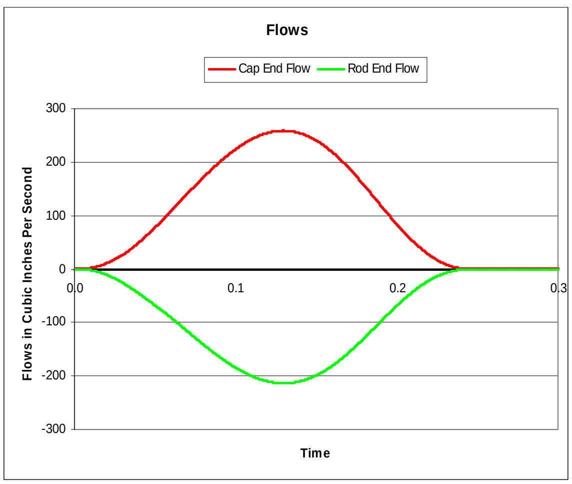


Figure 5b.

Appendix Figure 5. Flow vs. time comparing 2.5 inch cylinder to a 3.5 inch cylinder