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MODELING AND SIMULATION OF THE DYNAMIC EFFECTS OF PRESSURE VARIATIONS ON HYDRAULIC BLADDER AND PISTON STYLE ACCUMULATORS

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To the Graduate Council:

I am submitting herewith a thesis written by Colin Loudermilk entitled "MODELING AND SIMULATION OF THE DYNAMIC EFFECTS OF PRESSURE VARIATIONS ON HYDRAULIC BLADDER AND PISTON STYLE ACCUMULATORS." I have examined the final electronic copy of this thesis for form and content and recommend that it be accepted in partial fulfillment of the requirements for the degree of Master of Science, with a major in Mechanical Engineering.

Ahmad Vakili, Major Professor

We have read this thesis and recommend its acceptance:

Steve Brooks, Gregory Power

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(Original signatures are on file with official student records.)

MODELING AND SIMULATION OF THE DYNAMIC EFFECTS OF PRESSURE VARIATIONS ON HYDRAULIC BLADDER AND PISTON STYLE ACCUMULATORS

A Thesis Presented for the
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ABSTRACT

Hydraulic accumulators, being critical for system control, must meet performance parameters depending on system requirements. Multiple types of accumulators exist which provide varying levels of performance. These levels are not well-defined in most technical literature. A mathematical model was developed and computer simulation was used to fill some of the gap. Multiple accumulator systems were mathematically modeled in the Simulink environment and their performance characteristics were determined. Gas-charged bladder and double-acting piston accumulators were simulated with varying degrees of damping due to friction, the main factor that separates the two types.

It is shown that a bladder accumulator will in fact provide a faster response to the pressure fluctuations of a hydraulic system. However, the faster response is commonly under-damped. While a piston accumulator produces a slower response, the vast amount of damping provided by the accumulator piston produces a critically-damped to an over-damped response and would be advantageous for the designer looking for a more precise control.

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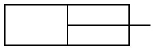
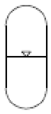
NOMENCLATURE

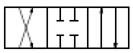
A	Area
C	Coefficient (subject to subscript)
c	damping coefficient
d	diameter
F	force
f	friction factor
G	Gain (numerical value)
K	spring constant
P	pressure
P_{pre}	pre-charge accumulator pressure
P_{sys}	system pressure
q	fluid flow rate
R	ideal Gas Constant
V	volume
T	temperature
t	time
v	fluid velocity
ρ	fluid density
ξ	damping ratio
ϵ	absolute surface roughness
ω_n	natural frequency

SYMBOLOGY

Fluid Power Symbols (International Organization for Standardization, 2012)

 Double-Acting Cylinder

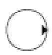
 Gas-Charged Accumulator
(Any Type)

 Four-Way Directional Control Valve

 Servo Valve Actuator

 Check Valve

 Fluid Reservoir

 Positive Displacement Pump

Fluid Conductor (Pipe or Tube)

CHAPTER 1: INTRODUCTION

Advanced hydraulic equipment used in manufacturing, construction, and scientific applications is often required to operate with high levels of precision, repeatability, and responsiveness. The emphasis on performance increases yearly and, with varying levels of pump quality, it is important that accumulators be viewed as a vital control component.

This study will serve to formulate, simulate, and observe the variations in the response time, the time required for an accumulator to come to equilibrium following a pressure change in the overall system, of two separate styles of hydraulic accumulators which can be subtly different. There are benefits to having this knowledge as it would allow system designers and maintainers to save time and space by better understanding which accumulators fit within their requirements and how it may impact overall system function. Basic hydraulic systems, like the one in Fig. 1, are the most common linear positioning devices in industry for applications requiring a large amount of force.

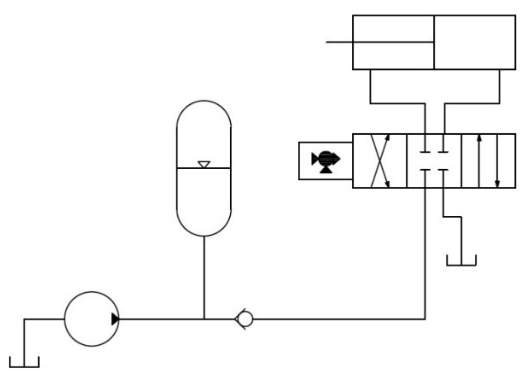


Figure 1 Basic hydraulic system

Consisting of few components, these systems can be deceptively complex when performance requirements are high. A high-performance unit will have a control system which can easily outperform its mechanical counterpart if the overall system is designed incorrectly. (Vickers, Inc, 1998) A controller can, for example, require more flow from a pump than is physically possible. For economical purposes, an accumulator is typically placed within the system to provide makeup flow for a controller placing excess demand on the pump but does not require constant motion, placing the accumulator's ability to respond to a pressure variation at the center of the system's ability to meet the control system's demands.

Two types of accumulator are most commonly used in position control systems with the intent of supplying flow to the control valve: bladder and piston types, which are explained further, later in the text. In broad terms, a bladder accumulator will have a shorter response time than a piston accumulator and is thus the end-users' preference, when speed is a major concern. But bladder accumulators also do require bulky spare parts, costing the maintainer valuable time and storeroom space whereas a piston accumulator repair kit requires little storage room and has a longer shelf life than a bladder (Parker Hannifin Corporation, 2003).

Where does the system manager draw the line on performance? Can he or she save the time and space by using an accumulator that can still meets demand despite going against a generalized rule of thumb? And where is that threshold located? These are some of the questions that will be explored in this study.

The primary objective of this study is to develop a better understanding of the various accumulators' operations, especially focused on their response time based on damping and design fundamentals. Additionally, it would help develop a better model that can accurately account for the minutiae in accumulator systems that make the difference in agility between the types. The model consists of a simple hydraulic circuit with the pump removed to isolate the performance of the accumulator from the extraneous perturbations of the pump.

1.1 Hydraulics

This section is intended to serve as an introductory review of hydraulic system operations as pertained to different accumulators' response time trends and behavior. For a full course on hydraulic systems, the author recommends Industrial Hydraulics Manual, 5th Ed. by Eaton Hydraulics Training Services

1.2 Accumulators

An accumulator consists of two critical components: Oil containment and some means of storing potential energy. The former is typically in the form of a tank or vessel while the latter can employ any number of methods. A weight, height of the vessel, ambient air (for water systems) or springs can be used; but the most common method in industry, however, employs a compressed gas separated from the hydraulic oil by some barrier, which can consist of pistons, rubber bladders, or rubber diaphragms. (Hedges & Womack, 1985)

The pre-charge is preferably an inert gas, commonly diatomic nitrogen, to prevent degradation of the construction materials. Using compressed air as a pre-charge can promote corrosion as well as combustion of petroleum based oils. (Gupta, Westcott, & Riescher, 2017)

This study will focus on the piston and bladder type accumulators as they are typically used to supply large volumes of oil flow to a system (rather than simply dampening pressure spikes) (Parker 2003).

A bladder accumulator (see Fig. 2) consists of a pressure vessel containing a rubber bladder (sometimes referred to as a bag) that contains pressurized nitrogen. Bladder accumulators are often used for the full range of accumulators function previously mentioned and can withstand a fair amount of contamination (Parker Hannifin Corporation, 2003). Additionally, because of their ease of operation, bladder accumulators are the type most widely used. (Yeaple, 1995)

Piston accumulators (see Fig. 2) also use a nitrogen pre-charge, but they separate gas from air using a metal piston. The piston has two seals around the OD. Any more than two seals can create undue frictional forces that the accumulator must overcome to operate. (Hedges & Womack, 1985) The piston accumulator cannot tolerate as much contamination as its bladder counterpart due to the rubber seals on a sliding surface; however, it can supply far more flow when demand is high. (Parker Hannifin Corporation, 2003)

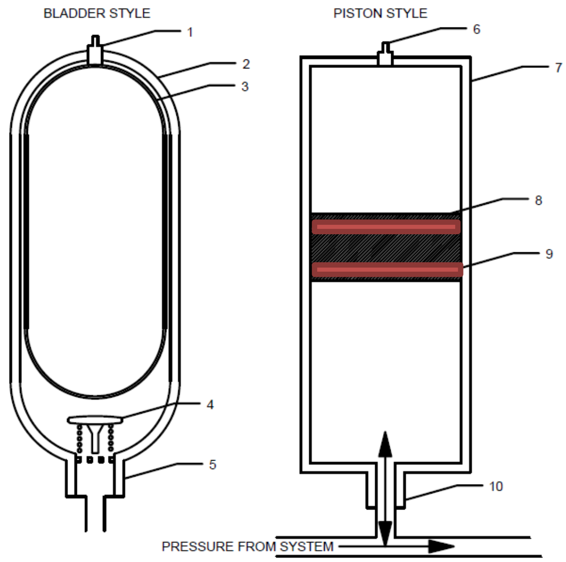


Figure 2 Typical layout and components of a bladder (left) and piston (right) accumulator

Components

1. Nitrogen Charge Valve
2. Shell
3. Bladder
4. Poppet Valve
5. Oil Port
6. Nitrogen Charge Valve
7. Shell
8. Piston
9. Piston Seals
10. Oil Port

Regarding regular maintenance of piston and bladder accumulators: 1) Rubber bladders, while acting as a fluid barrier, are still permeable to some extent and can allow the pre-charge to escape into the oil system over time while piston accumulators have far less rubber area (only the piston seals) in contact with both fluids. 2) Piston accumulators can be completely torn down and serviced, including any honing of the inside diameter as required while the shell of a bladder accumulator does not lend itself well to entry. 3) Spare bladders take up a lot more space in a maintenance inventory than a set of piston seals.

Regarding response time, as a general rule, bladder accumulators are encouraged in applications requiring fast response times; however, according to Parker, a piston accumulator can still be used in applications requiring as little as 25ms response time (Parker Hannifin Corporation, 2003).

The process of operation of a bladder accumulator is as follows:

1) Before the hydraulic system is operated, the bladder or gas charge chamber (of a piston accumulator) is pre-charged with diatomic nitrogen to a pre-determined pressure dependent on the application. For flow capacity, the subject of this study, the pre-charge pressure, P_0, is typically 90% of the minimum working pressure (Hydac, 2013) which will vary as a percentage of the system operating pressure depending on the equipment. The force of the bladder's expansion closes an anti-extrusion valve while a piston will deadhead at the bottom of the shell.
2) The system is started and brought up to pressure. When the system pressure exceeds the pre-charge pressure, either (for a bladder style) the anti-extrusion valve opens allowing oil to fill the space around the bladder and compress the nitrogen pre-charge or (for a piston style) the system pressure causes the piston to compress the nitrogen pre-charge and oil fills the resulting space inside the shell.
3) After the system reaches operating pressure and the nitrogen charge has reached equilibrium at P_1, a demand is placed on the system. The system's control valve

opens causing a pressure drop which in turn causes the nitrogen pre-charge to expand, pushing oil from the accumulator into the system at a rate of q_{demand} - $q_{pump,}$ lowering the pressure of the pre-charge to P_2.

4) When the demand on the system is satisfied, the control valve closes and the accumulator is refilled by the pump, the charge pressure returning to P_1. Steps 1 through 4 are repeated as required.

5) When the system is turned off, the volume of oil in the accumulator can either be used for emergency operation of equipment (i.e. in the event of a power failure) or, over time, leakage in the system will allow the accumulator to dispel its oil volume back to the reservoir and the charge pressure will return to P_0.

1.3 Directional Control Valves

What describes a hydraulic system as “high-performance” often depends on the actuation method of its directional control valves. Directional control valves come in several types for different applications. Table 1 lists each type along with its corresponding response time. For the purposes of this study, only High Performance Proportional Solenoid Valves and Servo Valves will be considered due to their response times being in the sub-25ms range and subject to a larger degree on the response time of an accumulator supplying flow.

Using Fig. 3 as a reference, the operation of a servovalve is as follows:

1) Pressurized hydraulic fluid enters the valve at ports P (in reality this is typically only one port but some valves have two), equally pressurizing either side of the

Table 1 Typical response times of hydraulic directional control valves (Vickers, Inc, 1998)

DCV Actuation Type	**Typical Response Time**
Solenoid	20-100 ms
Proportional Solenoid	50-150 ms
Feedback Proportional Solenoid	12-37 ms
High Performance Proportional Solenoid	10 ms
Servo	<10 ms

valve spool and the flapper. As long as the pressure is equal, the spool will remain centered.

2) A current is imposed on the torque motor causing the flapper to deflect to one side creating a pressure differential across the upper orifices. The pressure differential causes the valve spool to shift, connecting C1 and C2 (the output ports) to the pressure side, P, and the return side (low pressure), R depending on the direction of the deflection.
3) The feedback probe follows the spool and pulls the flapper to its initial null position when the spool has reached its desired location causing flow from P to R to stop. Steps 1-3 are repeated based on demand.

1.4 Hydraulic System Simulation

Texts such as Merritt 1967 and Stringer 1976 were written with the technology of their times in mind: Computers took up entire rooms and you had to ask for permission

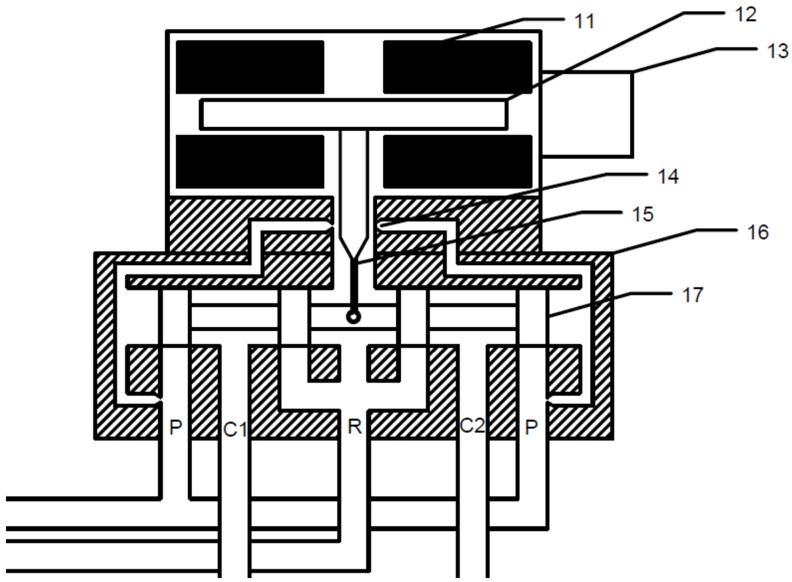


Figure 3 Typical layout and components of an electrohydraulic servovalve

Components

11. Torque Motor
12. Armature
13. Electrical Connector
14. Nozzle
15. Flapper/Feedback
16. Housing
17. Spool

to use one. They were not intended to give someone exact answers to every variable but rather how to find the answers to only a handful. Today, when everyone has a computer within arm's reach (or in their hands in the form of a cell phone) at any given moment, text such as these really come into their own, when the entire breadth of the knowledge bases they present can be calculated in short order and with little expense relative to when they were first published (Merritt, 1967).

With the availability of personal computers, it is common practice to simulate a hydraulic system design before sending it to fabrication as minor issues with the design can balloon in cost as the unit is assembled. Common programs used in industry are Mathworks® Simscape Fluids[1], FESTO FluidSIM®[2], and Famic Technologies, Inc. Automation Studio[3] amongst several others. Simscape Fluids will be used for the bulk of this study.

Simscape is a toolset within Matlab's Simulink environment that aids in the modeling and simulation of physical systems. Unlike Simulink where data is transferred forward through a flow chart, Simscape facilitates bi-directional flows of information between blocks so that the system can be modeled just as if it were a physical system. Beginning in 2006, a new set of Simscape blocks was published specifically for the simulation of hydraulic equipment called SimHydraulics. Before many components within the Simscape environment would need to be fashioned together into rudimentary hydraulic model. (Mathworks, 2018)

[1] https://www.mathworks.com/products/simhydraulics.html
[2] https://www.festo-didactic.com/int-en/learning-systems/software-e-learning/fluidsim/fluidsim-5.htm
[3] https://www.famictech.com/pro/index.html

CHAPTER 2: MODELING

2.1 Algebraic Solution

Assuming Nitrogen is an ideal gas, the position of the barrier for either type accumulator can be represented as

$$x = \frac{P_{pre}L}{P_{sys}}$$

2.2 Second-Order ODE Solution

A general second-order motion of a damped spring-mass system is composed of the following elements:

$$\frac{1}{\omega_n} D^2 x_2 + \frac{2\zeta}{\omega_n} D x_2 + x_2 = x_1 \text{ (Stringer, 1976)}$$

Accumulators can be modeled in the form of a second-order spring-mass-damper system, with a few assumptions, like Fig. 4:

1. A bladder does not physically stretch (i.e. with no pressure it is the size of the internal volume of the shell) and the surface area remains constant.
2. Diatomic nitrogen is used as a pre-charge and acts as an ideal gas.
3. Hydraulic fluid compresses so little compared to the compression of the nitrogen that it can be treated as incompressible

Starting with the equation for a spring mass damper system like that in Appendix VI:

$$M \frac{d^2x}{dt^2} = -K(x_2) - c\frac{dx}{dt}$$

$$M \frac{d^2x}{dt^2} + c\frac{dx}{dt} + Kx = f_{sys}$$

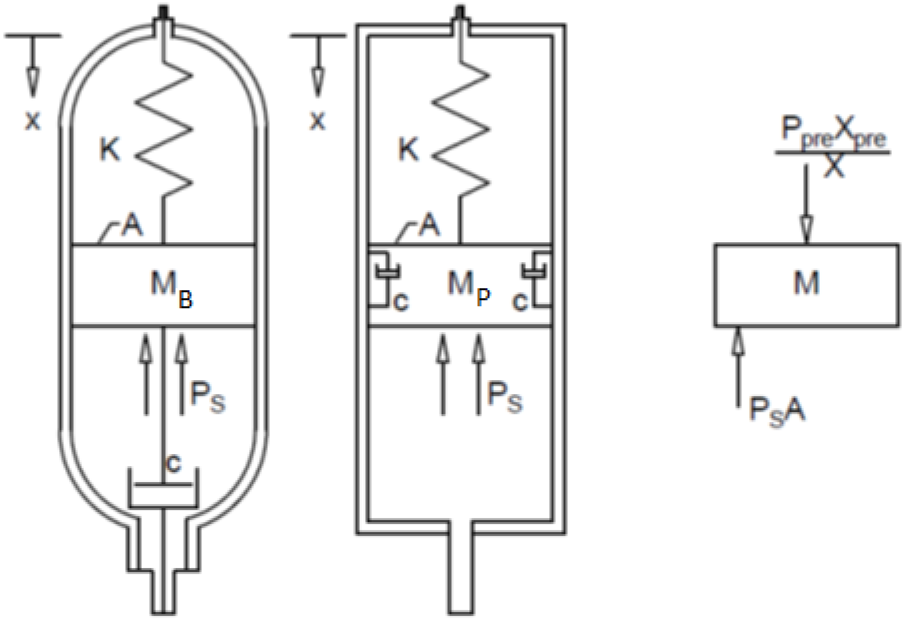


Figure 4 Accumulators modeled as damped spring-mass systems

2.3 Spring Force

Assuming the spring force comes solely from the N2 compression and that it acts as an isothermal ideal gas:

$$P_1V_1 = P_2V_2$$

Given that:

- $A_1 = A_2 \rightarrow V_2 = xA$, where A is the area of the barrier and x is the distance *away from* the charge valve
- $V_1 = LA$, where L is the length of the accumulator
- $P_1 = P_{pre} = \frac{F_{pre}}{A}$, Nitrogen pre-charge pressure and force
- $P_2 = \frac{F_{comp}}{A}$, Hydraulic system pressure and force

Therefore:

$$F_{comp} = \frac{F_{pre}L}{x}$$

2.4 Damping Force

Damping in an accumulator system comes primarily in the form of frictional forces acting on either the gas/fluid barrier or the hydraulic fluid.

A common form of damping for both styles is viscous friction which acts on the fluid due to surface roughness of the interior of the vessel. The resultant pressure drop can be determined by the Darcy-Weisbach equation:

$$\Delta P = f_{Darcy}\frac{\Delta L}{D}\rho\frac{v^2}{2}$$

The Darcy friction factor can be found by solving the Colebrook equation:

$$\frac{1}{\sqrt{f}} = -0.86ln\left(\frac{\epsilon}{3.7D} + \frac{2.51}{\sqrt{f}}\right)$$

The Colebrook equation can be solved numerically or using the Moody chart. An ISO 32 oil was used in the model and oil properties can be found in Appendix I.

Piston seals are commonly constructed of rubber and other polymers and are a source of friction within these types of accumulators; however, the seals are lubricated with hydraulic oil (or sometimes vacuum grease upon installation) and the coefficient of friction of the sealing material provides little in the way of friction (Parker-Hannifin, 2018). Friction is rather caused by the compression of the seal against the vessel and the hydraulic pressure acting on the seal (Parker-Hannifin, 2018):

$$F_s = F_c + F_H$$

Where:

F_s = total friction force acting on the piston and:

$$F_c = f_c L_P$$

$$F_H = f_h A_p$$

Where:

F_c = force due to seal compression friction

f_c = friction factor due to compression and can be found in manufacturer’s literature

L_p = length of the sealing surface

F_H = force due to hydraulic force frictional

f_h= friction factor due to hydraulic force and can be found in manufacturer’s literature

A_p = area of the seal which extends past the OD of the piston

In the model, all friction forces are permuted slightly as a function of velocity and given an individual gain value.

Therefore, the total damping coefficient of a piston accumulator is:

$$f_{Damping} = (G_V f_{Darcy} \frac{\Delta L}{D} \rho \frac{v}{2} A + G_{F_c} F_c + G_{F_H} F_H) v$$

The rubber barrier of a bladder accumulator tends to compress in on itself when subjected to a net positive pressure (Parker Hannifin Corporation, 2003) and is therefore not subjected to sliding friction. The anti-extrusion valve however has a poppet that causes damping through hydrodynamic drag.

The force of the drag can be estimated by:

$$F_D = \frac{1}{2}\rho v^2 C_D A$$

Where:

F_D = total drag force

C_D = drag coefficient

The drag coefficient can be found through calculation or through a table. At Re 10^5, for a flat plate with flow moving perpendicular to it, which is essentially what the poppet valve is, a standard C_D is 2.0 (Bertin & Cummings, 2008).

Another source of damping in a bladder accumulator can be found in the boundary between the bladder and the shell and is modeled in the same manner as the seal friction of the piston accumulator with a gain value turned down since the bladder compresses in on itself when the system is pressurized and will not see the kind of friction that the piston accumulator seals see.

Total Damping Coefficient of a Bladder Accumulators:

$$f_{Damping} = \left(G_V\left(f_{Darcy}\frac{\Delta L}{D}\rho\frac{A}{2} + \frac{1}{2}\rho C_D A\right) + G_{F_c}F_c + G_{F_H}F_H\right)v^2$$

2.5 Simulink Model

Several models of the above system were generated using Matlab and Simulink. The models and code can be found in Appendix IIa-f. The models were run using a step function block to simulate a pressure change in a hydraulic system with each compared with the model of the algebraic solution (Appendix IIa). The results of these simulations can be found in the next section.

CHAPTER 3: RESULTS AND DISCUSSION

3.1 The Effect of Barrier Mass on the System Response

One of the most intriguing effects of the system was the magnitude of the system response when the mass value of the barrier was changed.

3.1.1 Bladder accumulator: As can be seen in Appendix IIIa the fastest (relatively stable) settling time of a 10 gallon bladder accumulator given various bladder weights is 0.17 seconds and, as the weight of the bladder decreases, the damping of the system goes down until the system becomes unstable. It also appears through further analysis that 10-12 lbs is the point of diminishing returns when designing a 10 gallon bladder. The same simulation was performed on a 5 gallon (Appendix IIIb) and a 1 gallon (Appendix IIIc) bladder accumulator with the point of diminishing returns being 8-9 lbs for a 5 gallon bladder accumulator however it doesn't appear that any reasonable weight (for the size) will result in an approximate critically damped system. Note: the bladder accumulator simulations were run with a seal friction gain of 0.25, which should be accurate given how much the bladder actually contacts the shell of the accumulator.

3.1.2 Piston accumulator: Appendices IIId and IIIe show the "sweet spot" of a 10 gallon piston accumulator to be somewhere around 20 lbs. For a 1 gallon accumulator, Appendix IIIf gives the same result as the 1 gallon bladder accumulator in that any barrier large enough to result in a critically damped system would also be too large for the accumulator, in this case around 40 lbs.

3.2 Variations in Response between Bladder and Piston Accumulators

See Appendix IIIg and IIIh. The bladder accumulator response tended to move faster but overshoot and oscillate for some time after reaching the desired position, which in reality would lead to pressure fluctuations in the hydraulic system. Meanwhile, the piston accumulator tended to be more damped and have far less overshoot, leading to fewer hydraulic pressure variance in the system as a whole. The settling (equilibrium) time for both styles of accumulators, however, was roughly equal.

These effects were also dependent on the amount of pressure change as can be seen in Appendix IIIj. As a general rule, the effect of pressure change on the time delay between piston and bladder accumulators was 50 ms per 10% of pressure drop and 20 ms per 10% pressure gain.

3.3 The Effects of Rubber Friction on a Bladder Accumulator

The bladder model includes a factor of seal friction (the same as that used in the piston accumulator) with a drop-down gain because the effects should not be as pronounced in bladder accumulator due to the fact that the rubber bladder does not tend to slide against the wall. Appendix IIIi shows various gain values for the friction factor on a bladder. Given prior experience with the internals of bladder accumulators, the author tends to believe that 25% is a reasonable number and this percentage was used on all of the models. With more (proprietary) design information, a more accurate estimate for friction can be made.

3.4 Overall Magnitude of Damping

Appendices IIIk and IIIl show the magnitude of damping for the two types of accumulators. Both types had sources that were by far the most evident. A bladder accumulator experiences the vast majority of its damping from the drag force acting on the poppet valve while the piston accumulator experiences most of its damping from the force of the seal compression against the wall of the shell. In a 10 gallon accumulator the force of the drag is nearly 800 lbf more than the drag caused by the piston seals but, taken as a whole, the piston accumulator has more sources of damping than the bladder accumulator and thus produces a smoother response.

CHAPTER 4: CONCLUSIONS AND RECOMMENDATIONS

While for the vast majority of the hydraulic design and application population, the difference in response between a piston and a bladder accumulator is negligible, there are some strong differences between the two types when a short response time and/or precision is required. Based on the analysis and results developed in this study it has been shown that a bladder accumulator will in fact provide a faster response to the pressure fluctuations of a hydraulic system; however this fast response is commonly under-damped; whereas, while a piston accumulator produces a slower response, the vast amount of damping provided by the accumulator produces a critically-to-over-damped response and would be advantageous for the designer looking for a more precise control.

It should also be noted that various aspects of a hydraulic accumulator can change the response time, specifically the mass of the barrier (piston). By creating a piston accumulator with very little mass, one could improve the response time by several millisecond which, incidentally, is the difference between a bladder and a piston accumulator as has been found. Carbon fiber composite pistons have the potential to provide the structural integrity and low weight for improved time response.

More research should be performed on this topic using a live hydraulic system and accumulators with internal feedbacks tracking the position of the barrier.

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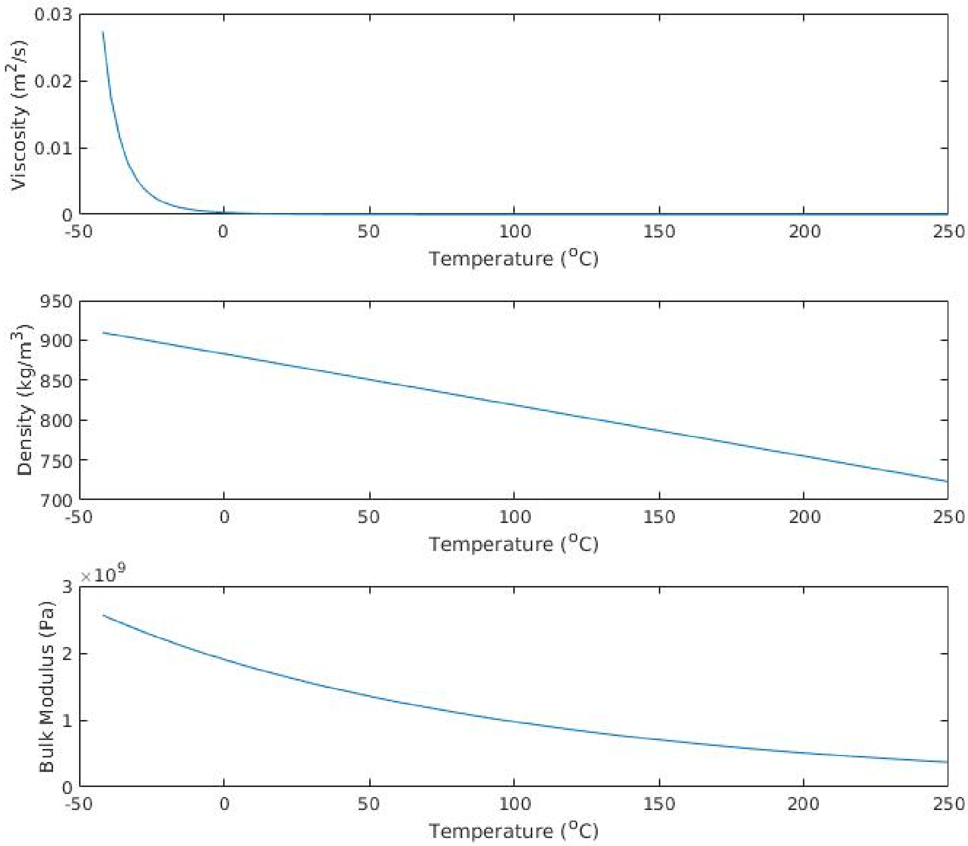
APPENDICES

APPENDIX I

Hydraulic oil properties.

ISO Grade	Kinematic Viscosity centiStokes		Density kg/m^3
	$40\,^oC$	$100\,^oC$	
32	32	5.4	857

ISO Grade 32 Oil Properties (The Engineering Toolbox, 2008)

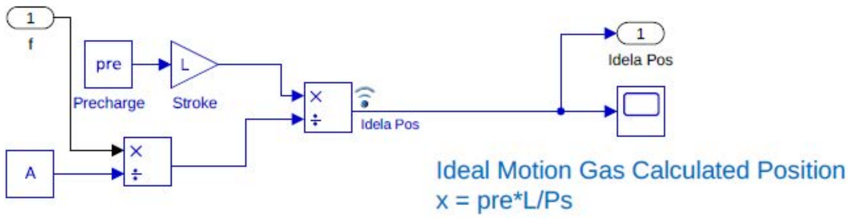


Properties of ISO 32 Hydraulic Fluids (Mathworks, 2019)

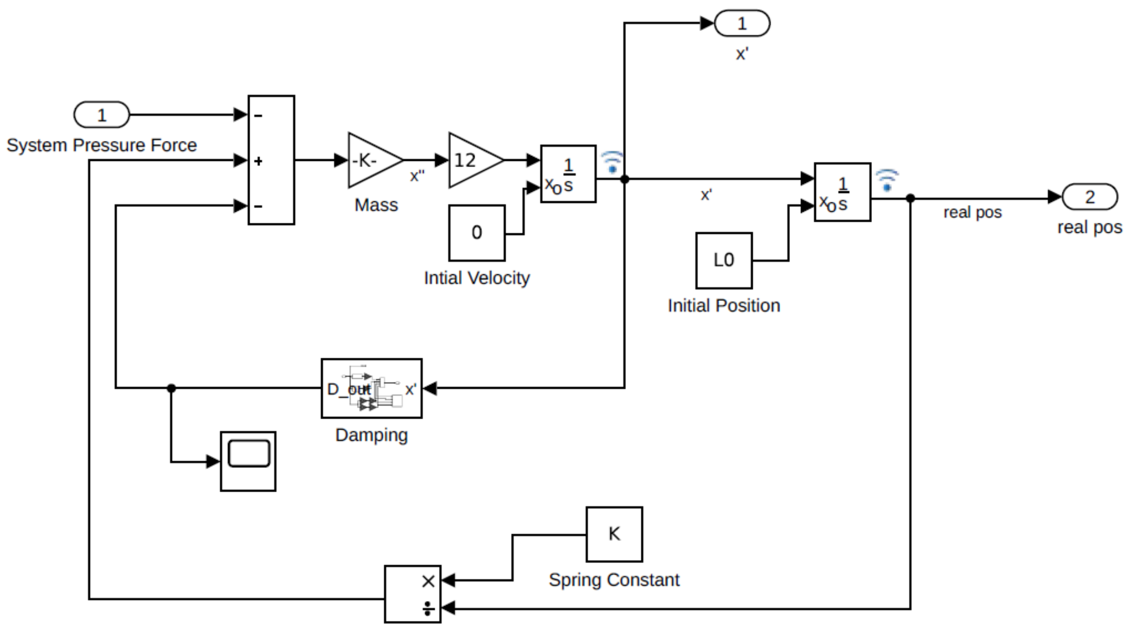
APPENDIX II

Matlab/Simulink/Simscape System Model and Code

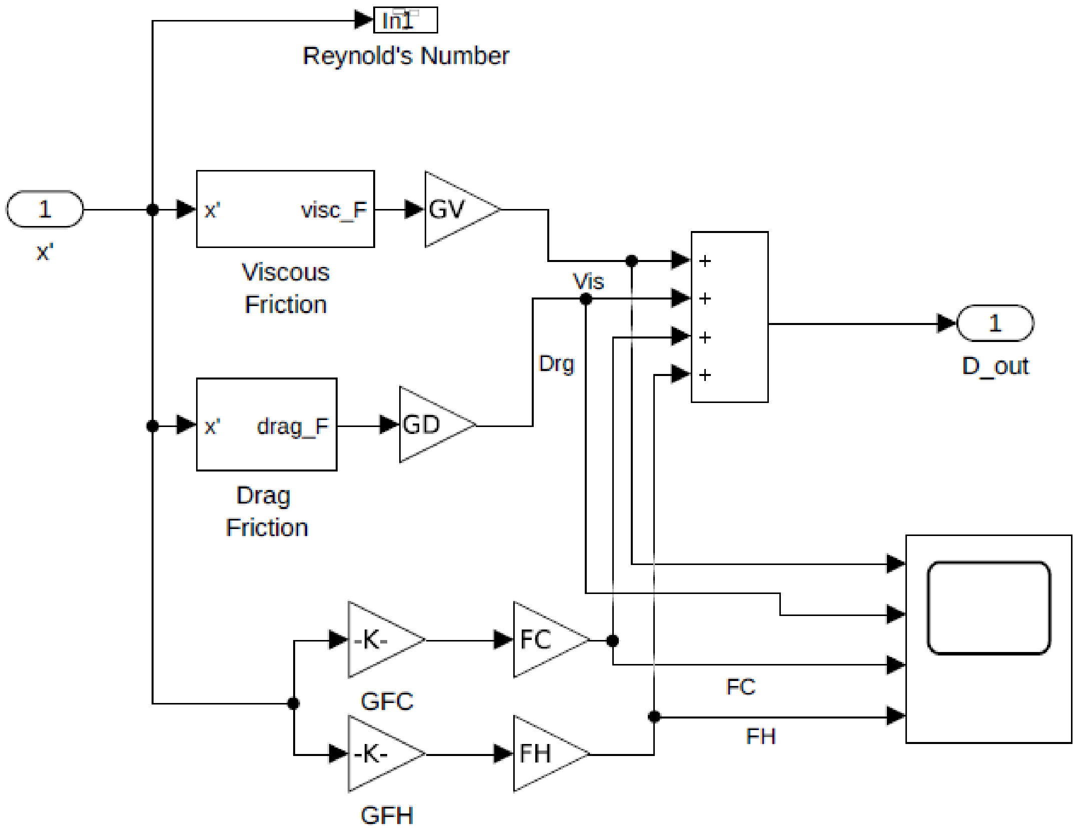
(a) Algebraic Model



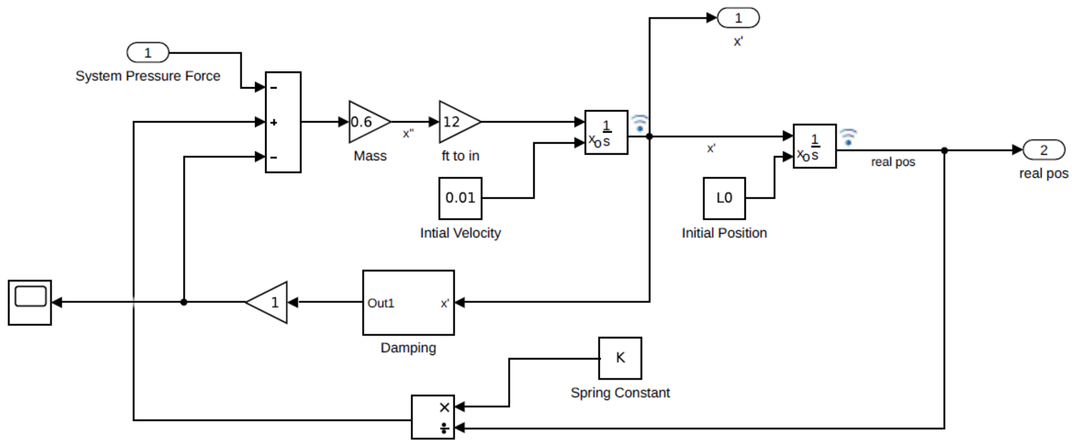
(b) Bladder Accumulator Model



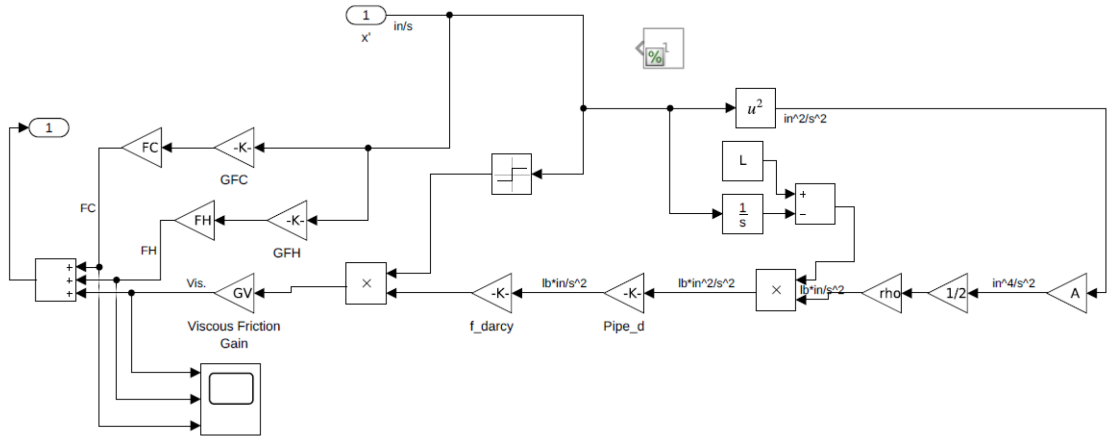
(c) Bladder Accumulator Damping Model



(d) Piston Accumulator Model



(e) Piston Accumulator Damping Model



(f) Input Data M-File

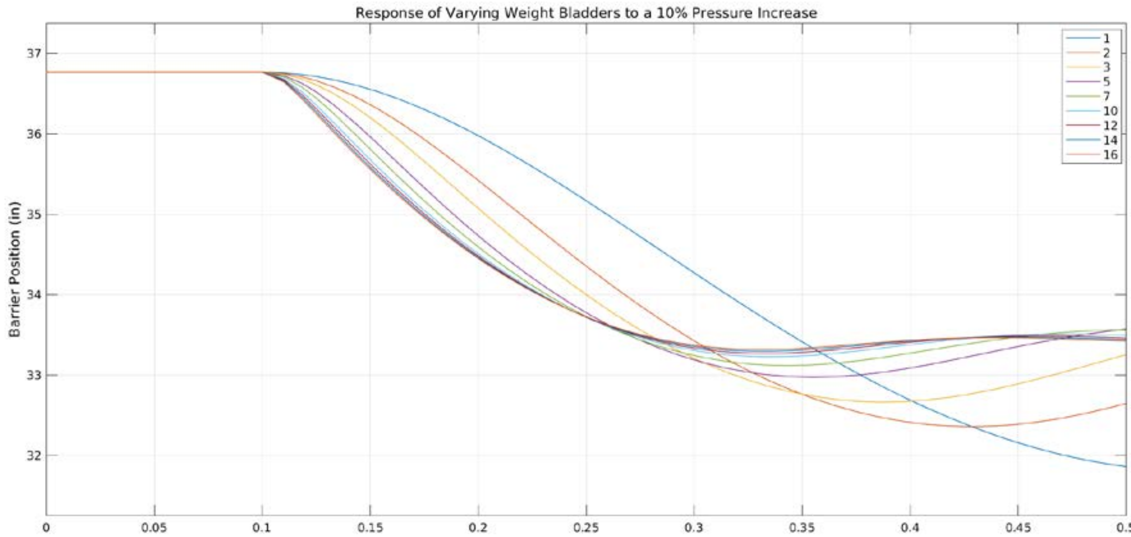
```
%This m-file is to be run before running the Simulink Accumulator model.
%It sets up most of the variables in the model%%%%%%%%%%%%%%%%%%%


%                          INPUTS

clc

Temp_F = 70; %Avg. Temperature in F

D = 8; %ID of the accumulator body in inches

V = 10; %Accumulator volume in gallons

pre = 750; %Nitrogen precharge, psi

m = 10; %mass in lbm

% E = 1; %damping ratio

Ps = 1500; %system pressure in psi

s_rough = 0.0012; %Surface roughness of stainless steel in inches (0.02mm).

%See https://neutrium.net/fluid_flow/absolute-roughness/

pipe_d = 1.5; %Inlet pipe diameter in inches

pipe_L = 4; %Inlet pipe length in inches

duro = 90; %Piston seal durometer

s_comp = 17; %Percent seal compression

mu = 32; % Fluid viscosity in cSt at 104 degrees F

rho =  0.0310; %Fluid density in lb/in^3

f_darcy = 0.025; %Darcy friction factor: median value acquired from Moody
                 %diagram

C_D = 2; %Drag Coefficient of the poppet valve

%                        CALCS
% Miscellany

Temp_C = (Temp_F-32)*5/9; %Converts tempt to C
```

```
pipe_L_m = pipe_L/39.37; %Converts inlet pipe length from inches to Meters

i_2_m = 1/39.37; %converts inches to meters

% Accumulator sizing

Vin = V*231; %Accumulator volume in cubic inches

A = pi*(D^2)/4; %Cross sectional area of the accumulator

L = Vin/A; %Length of the accumulator body (stroke length)

Z = s_rough/pipe_d;  %Surface roughness divided by hydraulic diameter



%Forces

f= Ps*A;  %Force exherted on the mass by the hydraulic pressure

fpre = pre*A; %Force exherted on the mass by the precharge pressure



% Spring Constant

K = fpre*L; %Spring Constant

% K = A/(L0-x0); %Spring Constant



% Initial Position

L0 = L*pre/Ps; %Length of the gas charge side at equilibrium

% x0 = L-L0; %Length of the oil side at equilibrium


% Seal Friction

if duro == 70
   f_c = (1/15)*s_comp;
elseif duro == 80
   f_c = (1.5/12.5)*s_comp;
elseif duro == 90
   f_c = (2/12.5)*s_comp;
```

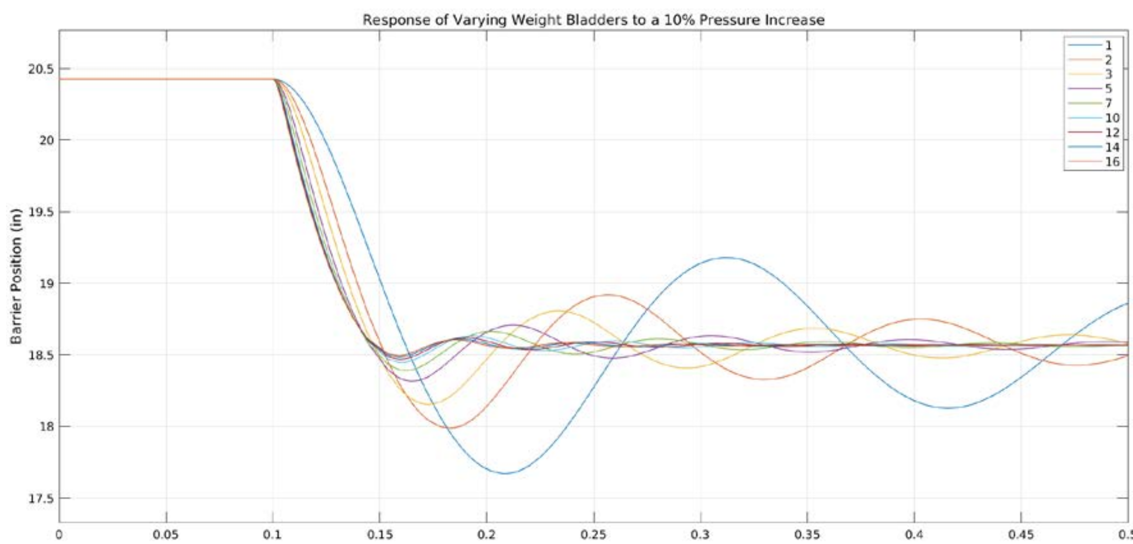
```
else
    f_c = 1;
end

f_h = -0.0000032213*Ps^2 + 0.030482*Ps + 10.15814; %found from a trendline
%in Parker's seal handbook

L_p = D*pi; %projected length of seal
A_p = pi*(D^2/4)*0.005; %projected area of seal

FC = f_c*L_p; %Force of seal compression

FH = f_h*A_p; %Hydraulic force on the seal

F_seal = FC+FH; %total sealing force resisting motion

%%%%%%%%%%%%%%%%%%  GAINS

GV = 1; %Viscous Friction Gain
GFC = 1; %Seal Compression Gain
GFH = 1; %Hydraulic Friction Gain
GD = 1; %Drag Friction Gain
```

APPENDIX III

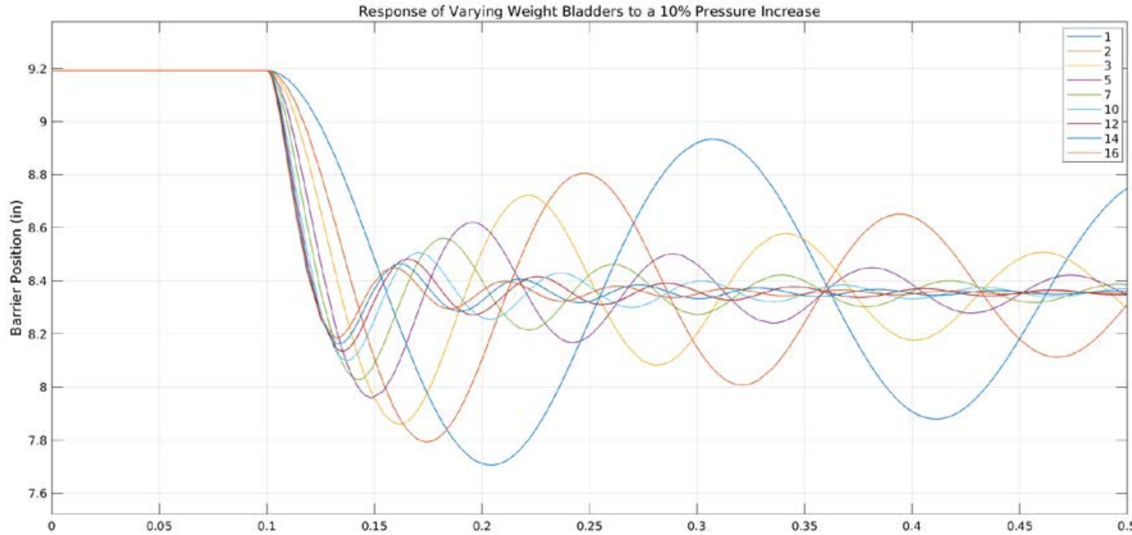
Output
(Note: All x-axes are in seconds)



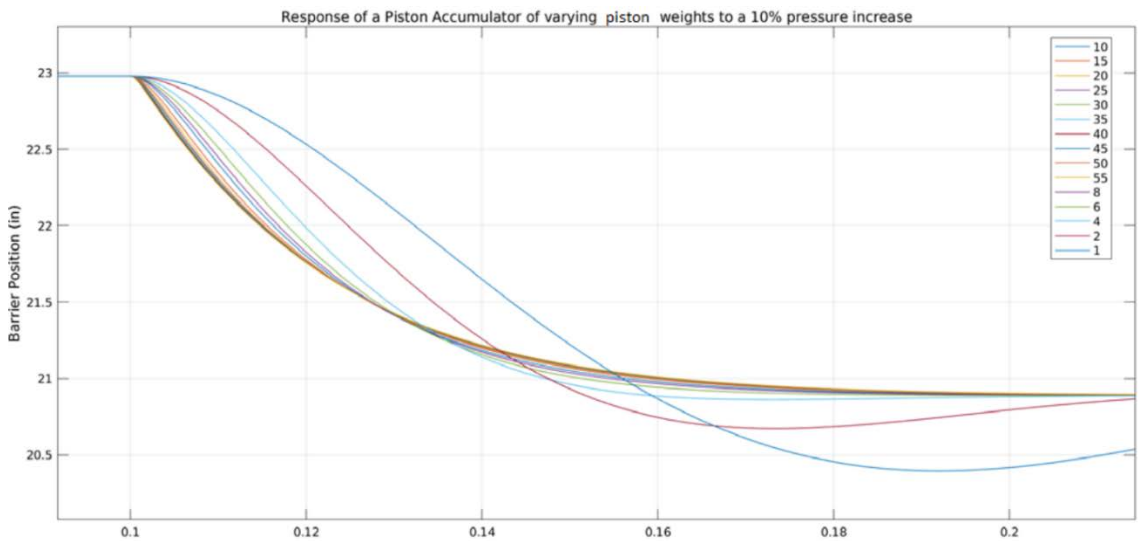
(a) Response of a 10 Gallon Bladder Accumulator of varying bladder weights to a 10% pressure increase



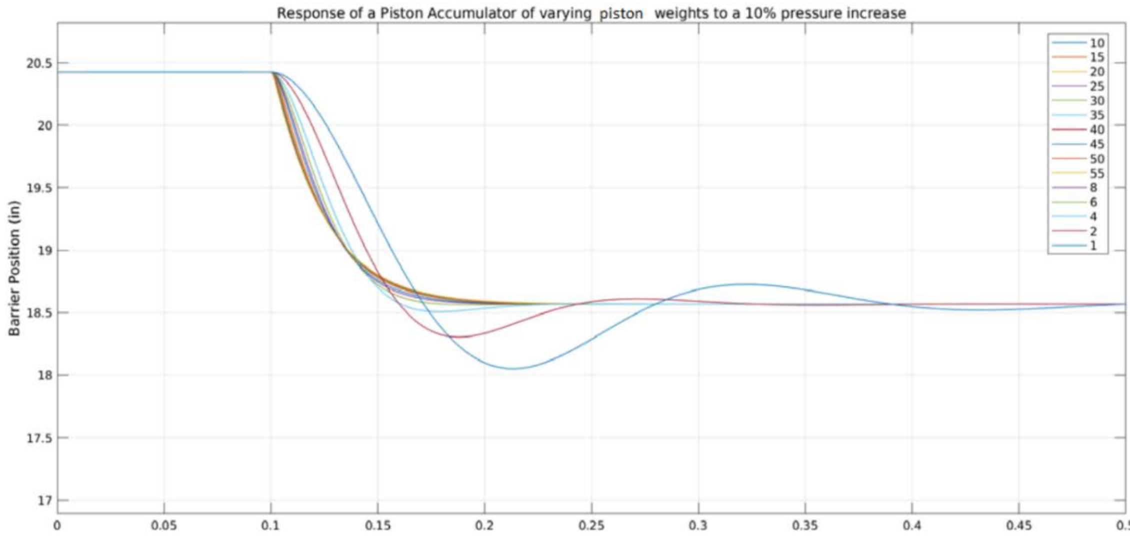
(b) Response of a 5 Gallon Bladder Accumulator of varying bladder weights to a 10% pressure increase



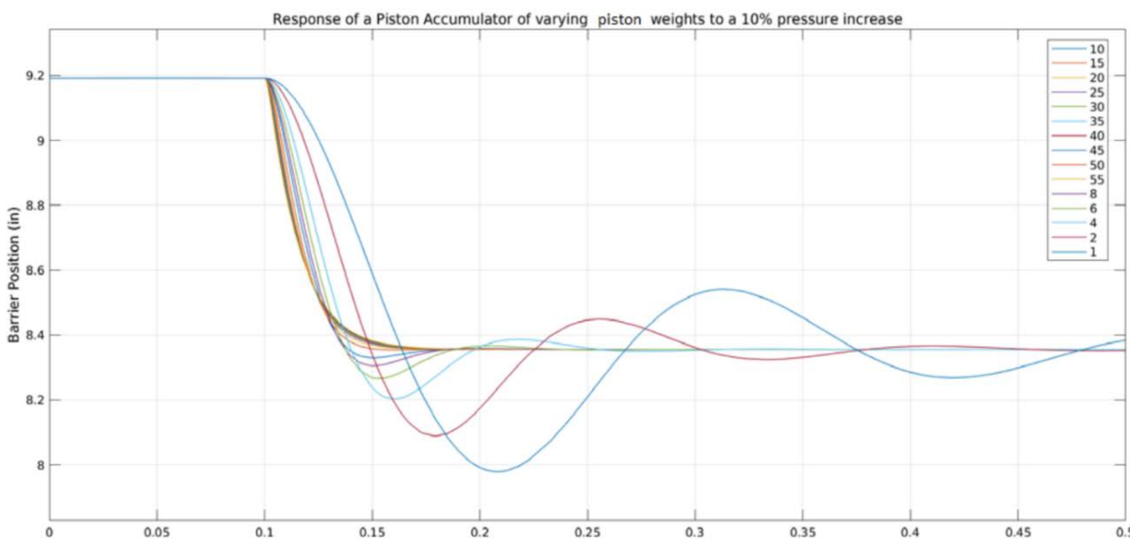
(c) Response of a 1 Gallon Bladder Accumulator of varying bladder weights to a 10% pressure increase



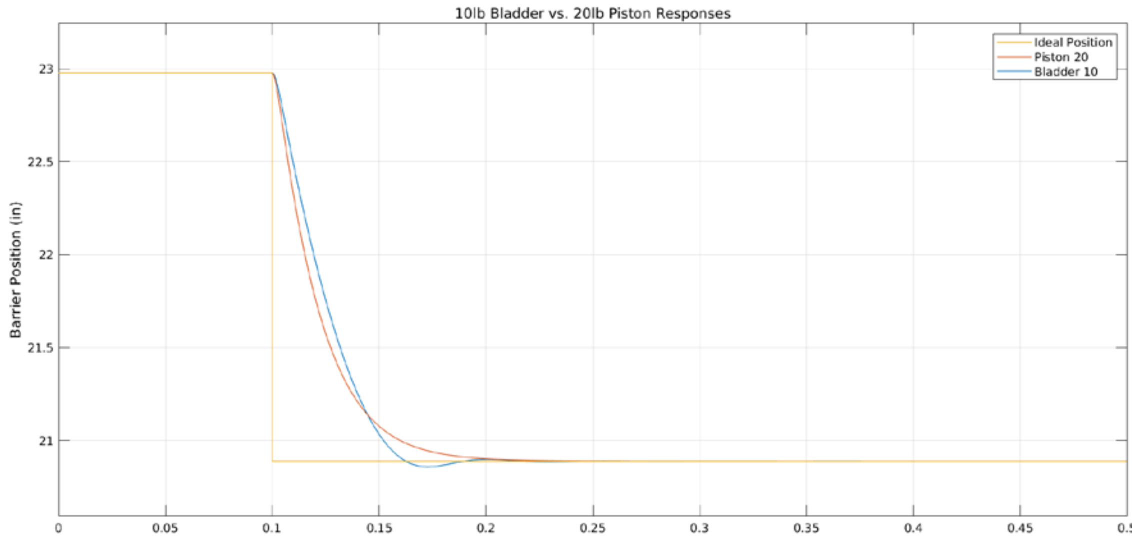
(d) Response of a 10 Gallon Piston Accumulator of varying piston weights to a 10% pressure increase



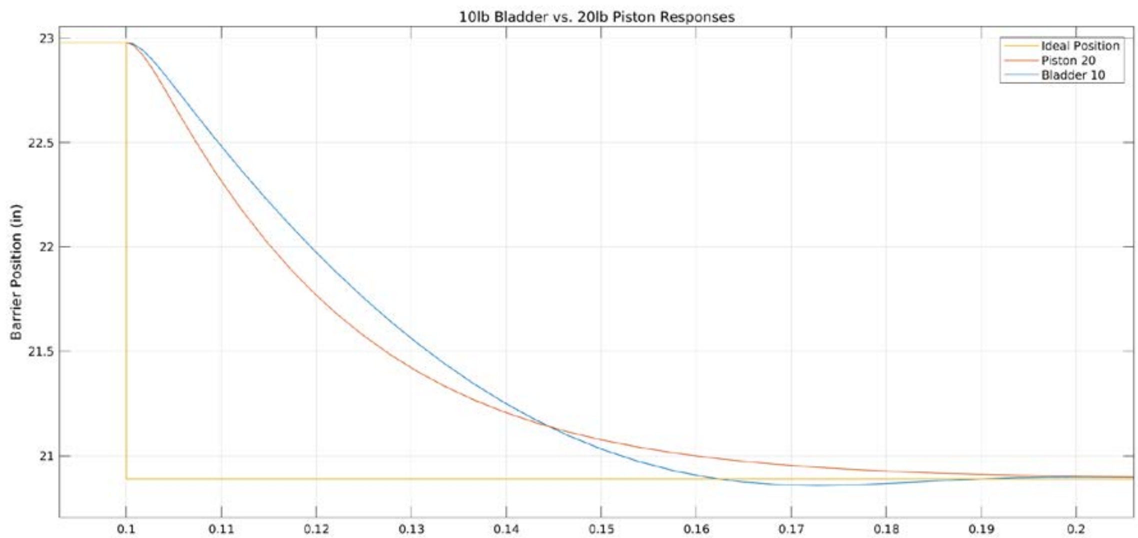
(e) Response of a 5 Gallon Piston Accumulator of varying piston weights to a 10% pressure increase



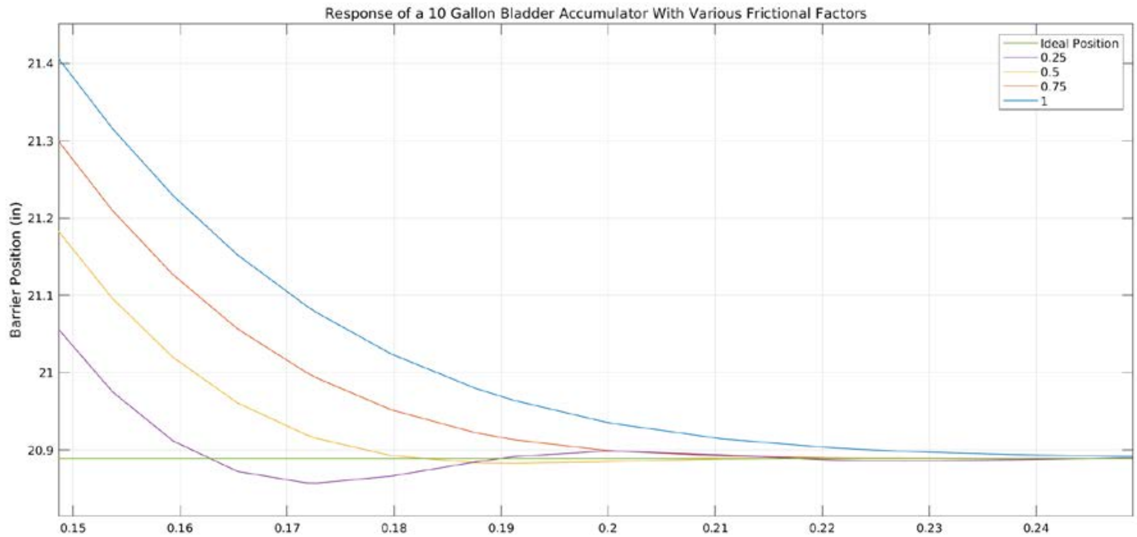
(f) Response of a 1 Gallon Piston Accumulator of varying piston weights to a 10% pressure increase



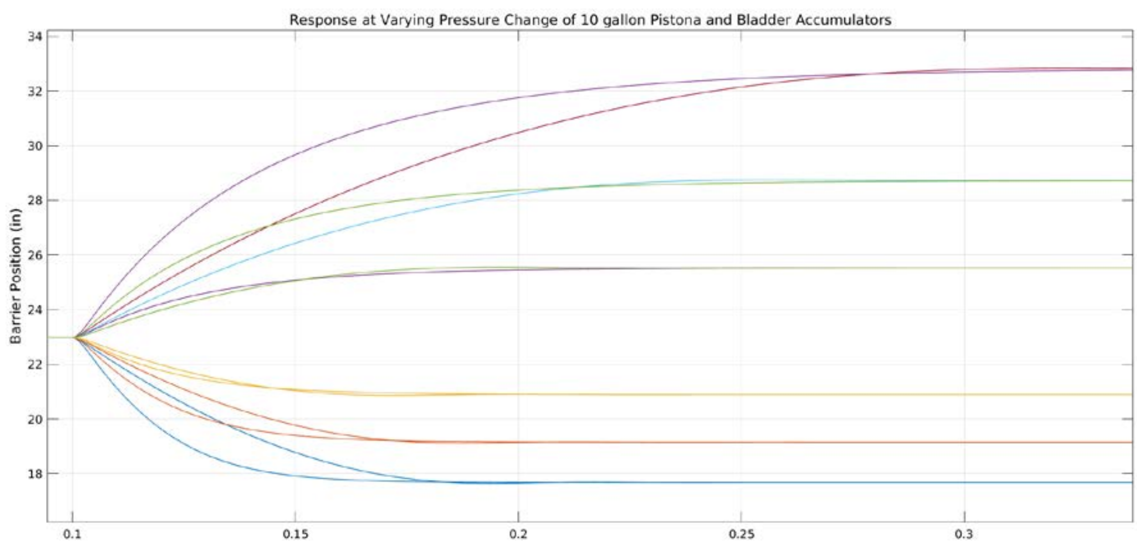
(g) Response of a 10 gallon Piston and Bladder accumulator to a 10% pressure increase with a 10 lb bladder and a 20 lb piston



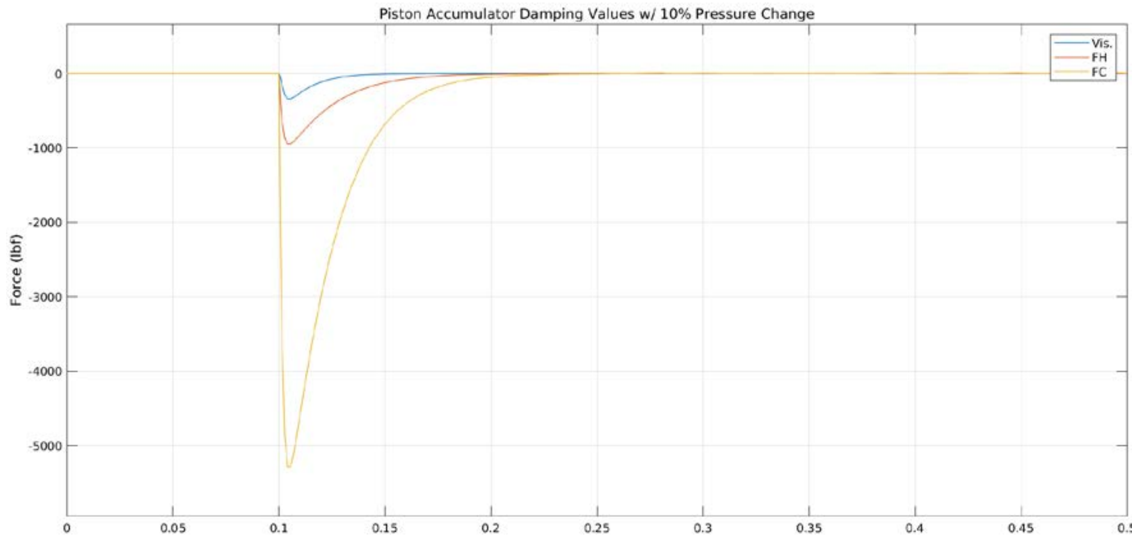
(h) Detail of the response of a 10 gallon Piston and Bladder accumulator to a 10% pressure increase with a 10 lb bladder and a 20 lb piston. Notice the 40 ms lag between bladder and piston crossing the ideal position (position with no losses or temperature effects).



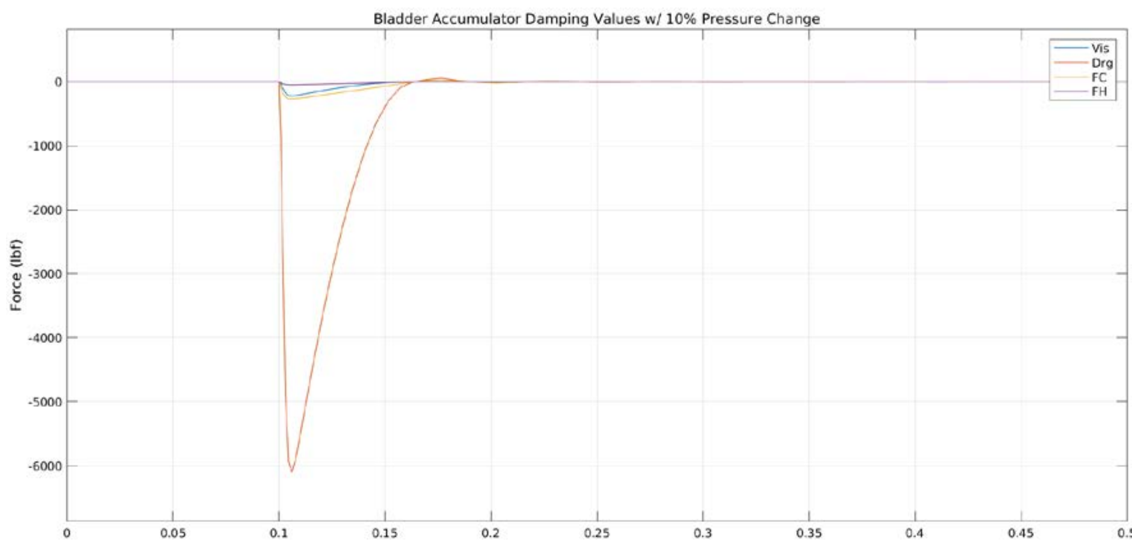
(i) Response of a bladder accumulator with various coefficients of seal friction proportional to that of a piston accumulator.



(j) Response of pressure variations of 10 gallon piston and bladder accumulators. The piston response is more damped an the bladder response is less damped. From top to bottom: -30%, -20%, -10%, +10%, +20%, +30%



(k) Damping values of a 10 gallon piston accumulator with a 10% pressure change



(l) Damping values of a 10 gallon piston accumulator with a 10% pressure change

VITA

Colin Loudermilk was born in Estill Springs, Tennessee. He earned his Bachelor of Science Degree in Mechanical Engineering from Tennessee Technological University in 2011. He currently works as a mechanical design engineer for National Aerospace Solutions at Arnold Engineering Development Complex at Arnold Air Force Base in Tennessee.