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Analytical Solution to the Pneumatic Transient Rod System at ACRR

This report is intended as an update for my thesis advisors at Georgia Tech. It is not a document that is intended to be published in any manner. Portions of this document will certainly be used later in my Master's Thesis, but that document will go through R&A when it is finished.
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Problem Statement

The ACRR pulse is pneumatically driven by nitrogen in a system of pipes, valves and hoses up to the connection of the pneumatic system and mechanical linkages of the transient rod (TR). The main components of the TR pneumatic system are the regulator, accumulator, solenoid valve and piston-cylinder assembly.

The purpose of this analysis is to analyze the flow of nitrogen through the TR pneumatic system in order to develop a motion profile of the piston during the pulse and be able to predict the pressure distributions inside both the cylinder and accumulators. The predicted pressure distributions will be validated against pressure transducer data, while the motion profile will be compared to proximity switch data. By predicting the motion of the piston, pulse timing will be determined and provided to the engineers/operators for verification. The motion profile will provide an acceleration distribution to be used in Razorback to more accurately predict reactivity insertion into the system.

Problem Definition

The pneumatic TR system can be split into two sections: accumulator to solenoid valve and solenoid valve to cylinder.

Accumulator to Valve

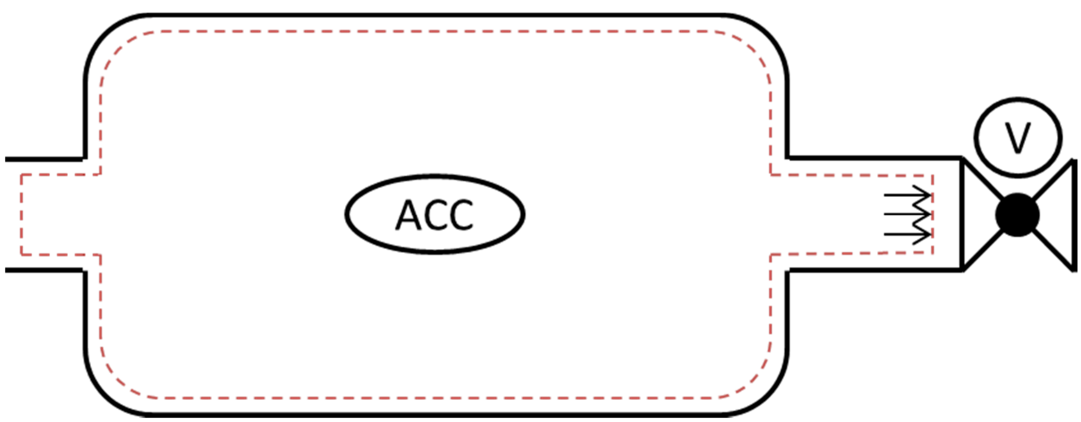


Figure 1. Geometric Representation of the Accumulator to Valve System. The red dotted lines represent the control volume denoted ACC

Given Conditions

Components: Accumulator, 2 1.25” 90 degree elbows, ~ 1’ of 1.25” piping, solenoid valve

Volumes: Accumulator- 675 in^3, Elbows & Piping – 25 in^3, Solenoid Valve – negligible

Constant system volume

Boundary Conditions

Temperature: Ambient = 70 F = 294 K

Pressure: 77.2 psia (65 psig) = 532275 Pa

Assumptions

The ~ 100 cubic inches of piping from the regulator to the accumulator is small enough to be ignored, as compared to the accumulator volume, on the short time scale of a pulse (see explanation in Assumption Justifications Section).

The volume due to the piping from the accumulator to the solenoid valve is small (~ 25 cubic inches); the accumulator-piping-solenoid valve system can be modeled as a single control volume with no inlets and one outlet.

The solenoid valve is modeled such that it is considered to have miniscule volume. It acts as an outlet of the accumulator and an inlet to the cylinder. Losses and geometry factors in the valve are inclusive in the equations used to calculate the mass flow rate, meaning that what leaves the accumulator-valve system is what enters the valve-cylinder system with no additional losses.

Valve to Cylinder

Given Conditions

Components: Solenoid Valve, ~ 1' of 1.25" flexible hose, 1 1.25" 90 degree elbow, manifold, pedestal, piston, cylinder

Volumes: Valve – negligible, Valve to pedestal – 51.3 in^3, Pedestal to Cylinder (piston on pedestal) – 0 in^3, Pedestal to Cylinder (piston full-up) – 90 in^3

Boundary Conditions

Temperature: ambient = 294 K = 70 F

Pressure: ambient = 12.2 psi = 84116 Pa

Assumptions

Again, the valve is considered to have negligible volume meaning that the fluid leaving the valve is the same as what is entering the valve-cylinder system.

The volume of this system is non-constant and varies depending on the location of the piston inside the cylinder. The pedestal to cylinder volume listed above is a range rather than a set value. If the piston starts by sitting on the pedestal, the initial volume in the pedestal-cylinder system is 0 and increases to 90 in^3 due to pressure increases forcing the piston upwards. This varying volume causes a headache in the calculations since both pressure and temperature are related to the volume, and during piston movement, all three are changing during each time step.

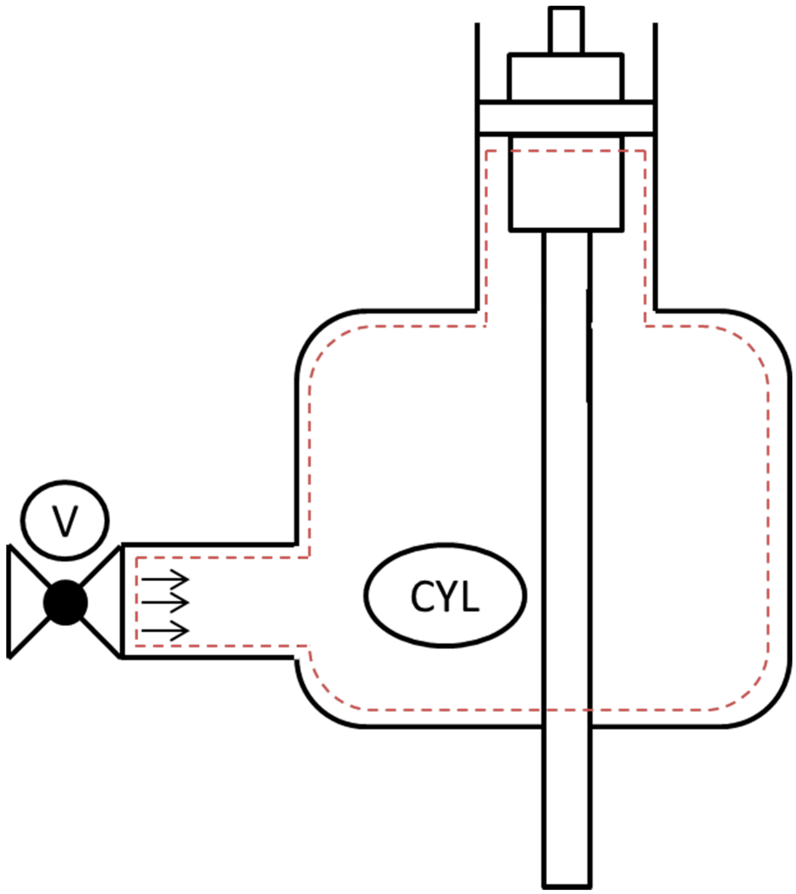


Figure 2. Geometric Representation of the Valve to Cylinder System, denoted CYL

Solenoid Valve

The solenoid valve in use by each TR is a Parker Hannifin Model H2001NC12501 1.25" NPTF Three-Way "Hustler" valve. The specification sheet for the H2000 series valve is located in Appendix A. This valve is a globe-type valve that operates with an internal piston moving vertically to allow for flow from port to cylinder or cylinder to exhaust. The manufacturer drawings of the solenoid valve are in Figure 3. Nitrogen enters Port P from the accumulator and exits at Port A as it moves up towards the cylinder. Port E is the exhaust port. The valve is normally closed meaning that in the deenergized state Port A is open to Port E, while Port P is blocked by the internal piston. When the valve energizes, the piston unseats from the Port P position allowing flow from Port P to Port A. Note that the valve doesn't have to be fully open for flow to exit the valve. When the valve closes, nitrogen flows back through the valve through Port A and exhausts through Port E.

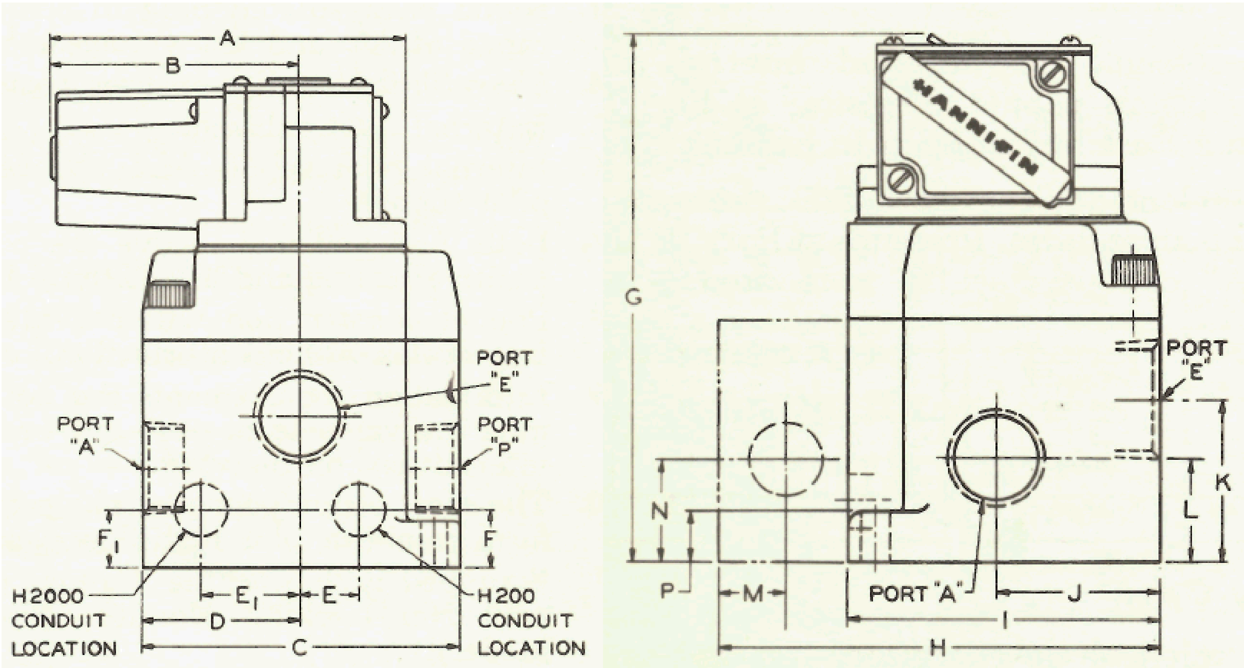


Figure 3. Front and Side Views of Solenoid Valve Respectively

Piston-Cylinder Geometry

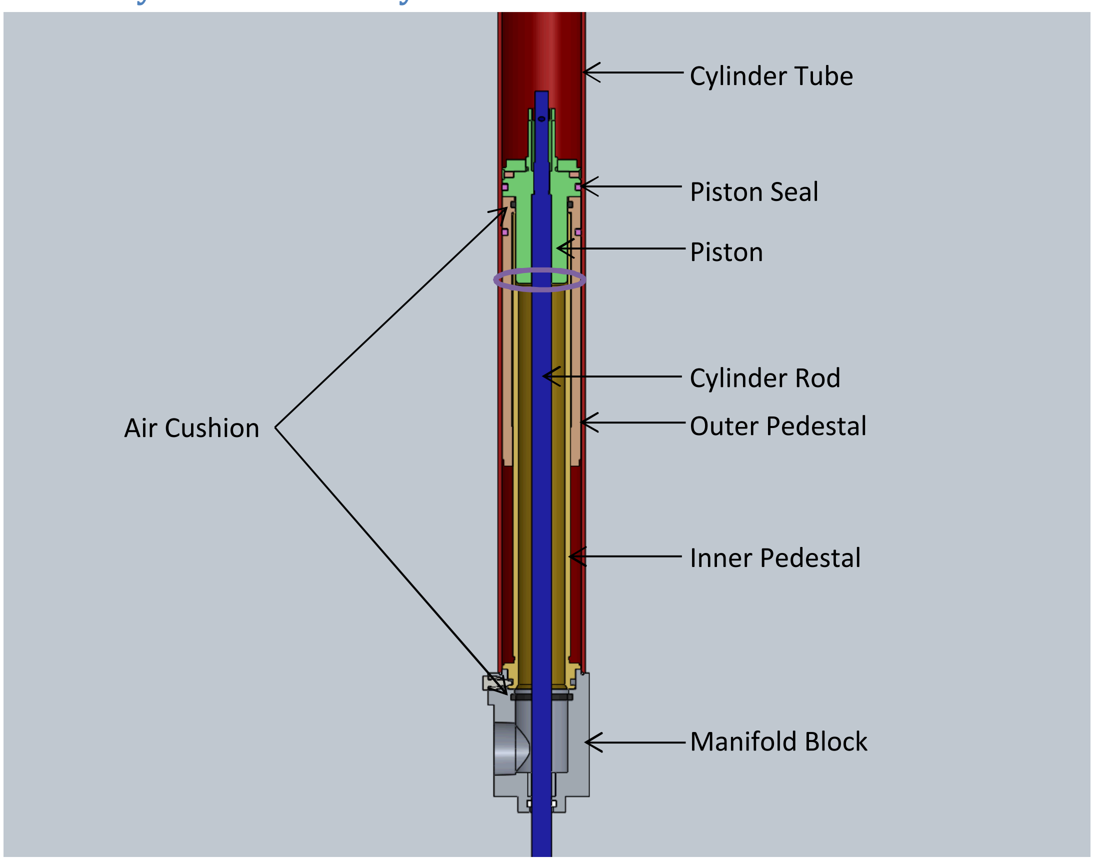


Figure 4. Cut Plot of Piston-Cylinder Geometry

The geometry of the piston-cylinder assembly is shown in Figure 3. Nitrogen flows into the manifold block from the left and travels to the bottom of the piston by way of the inner pedestal. The cylinder tube contains the whole assembly in an airtight manner, such that no nitrogen can escape out of the top. During the rod ejection, the piston seal plays an important role in keeping the volume under the piston airtight as the piston moves up the cylinder tube. An important factor during the rod ejection is that initially, the piston area is limited to the annulus inside the inner pedestal (denoted by the purple oval in Figure 3). After the piston clears the air cushion, the piston area expands to include the annulus on the upper portion of the piston (denoted by the orange ovals in Figure 4). Most ACRR pulses, however, do not pulse with the piston inside the pedestal (as seen in Figure 3, considered a max pulse), but rather pulse with the piston above the pedestal (denoted in Figure 4) by some predetermined height, measured in Rod Units (1 Rod Unit = 0.1mm).

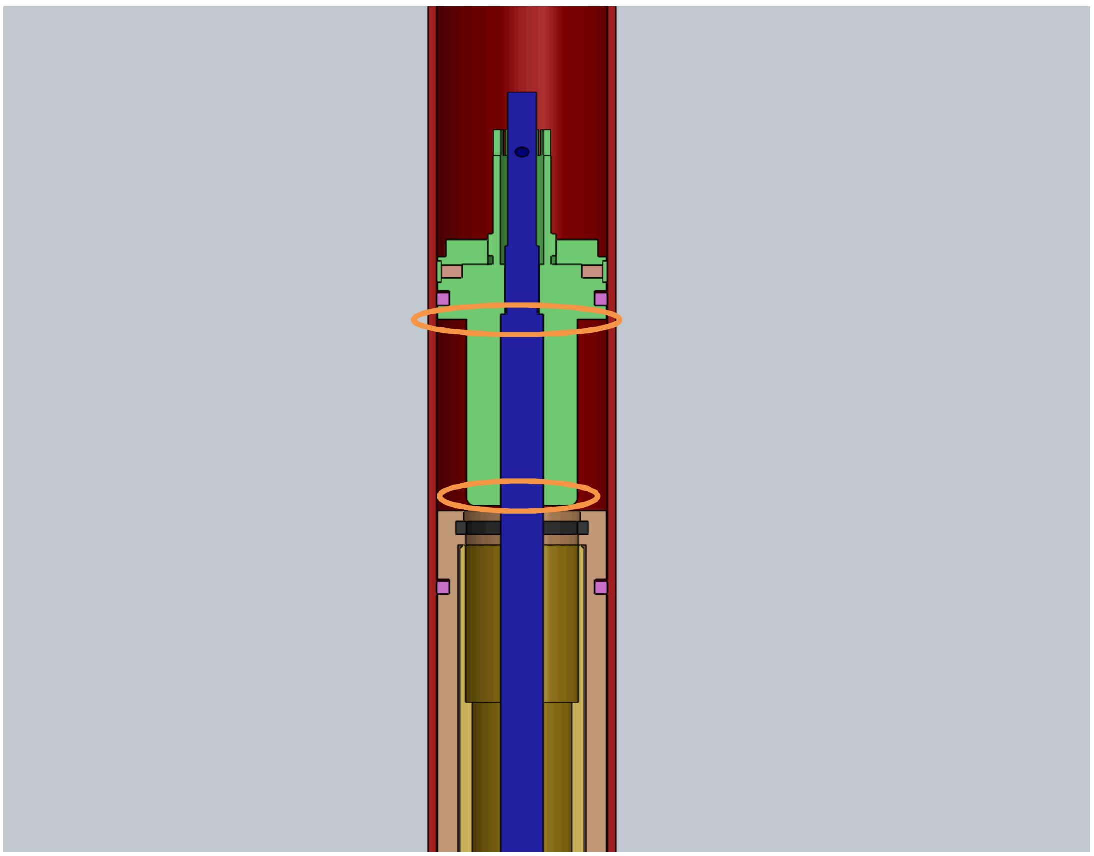


Figure 5. Piston Above Pedestal

During the rod drop, the air cushion plays a vital role in slowing the piston down once it reaches the pedestal. As the piston approaches the air cushion, a pocket of air is formed between the upper and lower faces of the piston, which is exhausted by a small needle valve in the pedestal. The impact forces are significantly decreased by using this method rather than having the piston directly impact the pedestal.

For the analytical solution, the piston-cylinder geometry will be referred to as simply the cylinder.

Methodology

The methodology contains three major sections: analytical derivations, TR timing, and mass flow rate calculation.

Analytical Derivations

The analytical derivations are the application of thermodynamic energy balances to the specific problem definitions as listed above. The two derivations vary greatly due to differing conditions and assumptions; however, they both use the foundational approach that is described in [1]. Here they are in full detail.

Nomenclature

$\mathrm{CV} = \text{control volume}$

$m_{cv} = m = \text{control volume mass}$

$\dot{m}_v = \text{mass flow rate through valve}$

$\dot{m}_k = \text{mass flow rates in and out of the control volume}$

$m_i = \text{mass in CV at time } t_i$

$m_{i-1} = \text{mass in CV at time } t_{i-1}$

$\Delta t = t_i - t_{i-1}$

$E_{cv} = \text{control volume energy}$

$\dot{W}_s = \text{rate of shaft work}$

$P_0 = \text{atmospheric pressure}$

$V = \text{volume of control volume}$

$\dot{Q}_0 = \text{heat transfer rate with atmosphere}$

$\dot{Q}_k = \text{heat transfer rate with other components}$

$h_k = \text{enthalpy of fluid at each orifice}$

$v_k = \text{velocity of fluid at each orifice}$

$g = \text{gravitational constant}$

z_k = height of each orifice

h_v = enthalpy of fluid in the valve

V_v = velocity of fluid through the valve

$u_{cv} = u$ = internal energy of control volume

T_i = Temperature in CV at time t_i

T_{i-1} = Temperature of fluid in CV at time t_{i-1}

C_v = constant volume specific heat constant of the fluid

P_i = pressure in CV at time t_i

ρ_i = density of fluid in CV at time t_i

R = specific gas constant

F_{net} = net force on piston

P_g = gauge pressure in CV

A_{piston} = surface area of bottom face of piston

F_{fr} = friction force on piston seal

F_M = weight of TR components

P_{pn} = pneumatic pressure on A_{piston}; same as P_i

D_{cyl} = diameter of cylinder

dV = incremental control volume in cylinder

a_i = acceleration of piston at time t_i

v_i = velocity of piston at time t_i

d_i = displacement of piston at time t_i

Accumulator to Valve

Assumptions

- Adiabatic
- Ideal gas law applies

- No shaft work
- No inlet, one outlet
- Negligible potential and kinetic energy effects
- Constant volume

Mass Balance

$$\frac{dm_{cv}}{dt} = \sum_k \dot{m}_k \tag{1}$$

After applying the third assumption, (1) simplifies to

$$\frac{dm_{cv}}{dt} = -\dot{m}_v \tag{2}$$

Where $\dot{m}_v$ is the mass flow rate through the valve.

Using finite difference in time, m_i can be represented as follows

$$m_i = m_{i-1} - \dot{m}_v \Delta t \tag{3}$$

Energy Balance

$$\frac{dE_{cv}}{dt} = -\dot{W}_s - P_0 \frac{dV}{dt} + \dot{Q}_0 + \sum_k \dot{Q}_k + \sum_k \dot{m}_k \left(h_k + \frac{v_k^2}{2} + g z_k \right) \tag{4}$$

After applying the assumptions, (4) simplifies to

$$\frac{dE_{cv}}{dt} = -\dot{m}_v h_v \tag{5}$$

Where

$$E_{cv} \equiv U_{cv} = m_{cv} u_{cv} = mu \tag{6}$$

Simplifying the left hand side,

$$\frac{dE_{cv}}{dt} = \frac{d(mu)}{dt} = m\frac{du}{dt} + u\frac{dm}{dt} = m\frac{du}{dt} - u\dot{m}_v \tag{7}$$

Recombining (5) and (7) yields

$$m\frac{du}{dt} - u\dot{m}_v = -\dot{m}_v h_v \tag{8}$$

Apply the finite difference method to (8) which defines the following variables as

$$m = m_{avg} = \frac{m_{i-1} + m_i}{2} \tag{9}$$

$$u = u_{avg} = \frac{u_{i-1} + u_i}{2} \tag{10}$$

$$\frac{du}{dt} = \frac{\Delta u}{\Delta t} = \frac{u_i - u_{i-1}}{\Delta t} \tag{11}$$

Multiply (8) by Δt along with some algebraic simplification yields the following equation for u_i

$$u_i = \frac{u_{i-1}(m_{i-1} + m_i + \dot{m}_v \Delta t) - 2\dot{m}_v h_v \Delta t}{m_{i-1} + m_i + \dot{m}_v \Delta t} \tag{12}$$

Utilizing the First Law of Thermodynamics to relate internal energy and temperature, T_i is represented by the following equation

$$T_i = \frac{u_i + u_{i-1}}{C_v} + T_{i-1} \tag{13}$$

The pressure is calculated using the Ideal Gas Law below

$$P_i = \rho_i R T_i \tag{14}$$

Where R is the specific gas constant and

$$\rho_i = \frac{m_i}{V} \tag{15}$$

Equations (3) and (12-15) are the basis to calculating the parameters for code implementation of the accumulator to valve system.

Valve to Cylinder

Assumptions

- Adiabatic
- No shaft work, significant piston work
- One inlet, no outlet during rod withdrawal
- One outlet, no inlet during rod drop
- Negligible potential and kinetic energy effects
- Non-constant volume
- Ideal gas law applies

Mass Balance

$$\frac{dm_{cv}}{dt} = \sum_k \dot{m}_k \tag{16}$$

After applying assumption three, (16) simplifies to

$$\frac{dm_{cv}}{dt} = \dot{m}_v \tag{17}$$

Using finite difference, m_i can be represented as follows

$$m_i = m_{i-1} + \dot{m}_v \Delta t \tag{18}$$

Energy Balance

After simplifying (4) with the assumptions for the Valve-Cylinder system and applying the simplification in (6), the energy balance reduces to

$$m\frac{du}{dt} + \dot{m}_v u + P_0\frac{dV}{dt} = \dot{m}_v h_v \tag{19}$$

Where P_0 refers to atmospheric pressure.

Applying the finite difference equations in (9-11) to (19) and solving for u_i

$$u_i = \frac{2\dot{m}_v \Delta t h_v - 2P_0 dV + u_{i-1}(m_{i-1} + m_i - \dot{m}_v \Delta t)}{m_{i-1} + m_i + \dot{m}_v \Delta t} \tag{20}$$

Note from (20) that when the piston is not moving, $dV = 0$; however, during piston motion dV is non-zero and significant.

Force Balance

In order to solve for dV, a force balance must be completed on the piston. Figure 6 shows all of the forces acting on the piston during the pulse. Note that F_{fr} changes directions when the piston is falling.

$$F_{net} = P_g A_{piston} - F_{fr} - F_M \tag{21}$$

Equation (21) represents the sum of the forces on the piston where

$$F_M = 14.234\ lbf \tag{22}$$

$$Gauge\ Pressure = P_g = P_{pn} - P_0 \tag{23}$$

$$F_{fr} = \pi D_{cyl}\left(1.5\frac{lbf}{in.}\right); D_{cyl} = 2.5\ in. \tag{24}$$

$$A_{piston} = 4.604\ in^2 \tag{25}$$

Equation (24) comes from the rule of thumb that Parker uses to calculate friction force on the specific piston seal being used in the grooves of the piston. The specific model of the piston seal is an 8400 series U-cup piston seal of 2.5" diameter made out of carboxlated nitrile. Since there is very little published friction data on carboxylated nitrile, the rule of thumb calculation from Parker will suffice. The energy balance in [1] assumes that the piston is simply connected to the control volume via the rigid structure; however, in order to correlate the change in volume to the force balance, the following relation was used

$$dV = A_{piston}dx \tag{26}$$

Equation (26) applies since the piston area stays constant throughout the duration of the pulse.

In terms of applying the finite difference method to calculating dV, the net force is calculated as follows;

$$F_{net,i} = (P_{cyl,i-1} - P_0)A_{piston} - F_{fr} - F_M \tag{27}$$

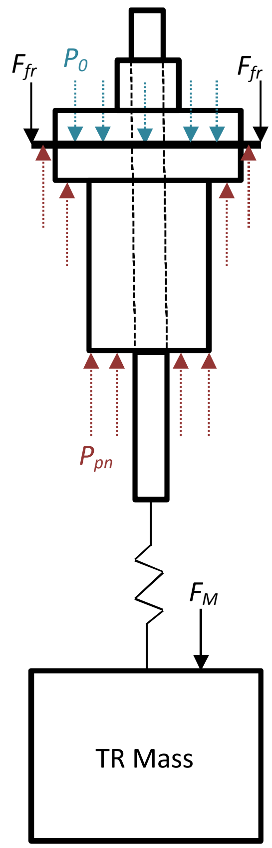


Figure 6. Free Body Diagram on Piston

Piston Kinematics

$F_{net,i}$ is used to calculate the new acceleration as shown in equation (28).

$$a_i = \frac{F_{net,i}}{m_{tr}} + a_{i-1} \tag{28}$$

$$v_i = a_i \Delta t + v_{i-1} \tag{29}$$

$$d_i = v_i \Delta t + d_{i-1} \tag{30}$$

$$dV_i = (d_i - d_{i-1}) A_{piston} \tag{31}$$

Velocity and displacement of the piston are calculated from equations (29-30) respectively. The dV calculated in (31) is used to determine the internal energy from equation (20). Temperature, pressure, and density are then calculated in the cylinder using (13-15) respectively.

Assumption Justifications

Nitrogen Supply into Accumulator

There are two sources of nitrogen that will flow into the accumulator during the pulse: nitrogen that has built up in the ~97 feet of .375 in tubing between the accumulator and the pressure regulator and any nitrogen that will flow through the regulator from the nitrogen bottles into the ~97 feet of tubing.

Due to the significant length of tubing that leads into the accumulator, there will be a lengthy delay for the flow passing through the regulator to reach the accumulator. Assuming that the max speed of nitrogen is the same as what is going through the valve (not likely to be this high since the pressure drop across the valve is at least 4 times greater than what would be seen between the piping and accumulator), the flow delay can be calculated as follows:

$$flow\ delay = \frac{flow\ length}{flow\ speed} \tag{32}$$

Where $flow\ length = 97\ feet = 1164\ in$, and $flow\ speed = 73\frac{m}{s} = 2874\frac{in}{s}$.

$$flow\ delay = \frac{1164\ in}{2874\frac{in}{s}} = 0.405\ s \tag{33}$$

Since the rod ejection portion of the pulse occurs on the order of 0.100s, any nitrogen that flows through the regulator after the valve has been opened would not make it to the accumulator before the piston reaches the full-up position.

Before the pulse starts, the 97 feet of tubing between the accumulators and regulator is pressurized to 65 psig like the accumulators. Upon firing, the nitrogen rushes to fill the three accumulators as the accumulators lose pressure through the valve. During the duration of the rod ejection (0.100s), the accumulators drop ~15 psig, implying that the piping leading up to the accumulators also drops about 15 psig. In order to find the amount of mass that enters the accumulators from the piping, the mass of nitrogen in the piping before and after the rod ejection must be compared.

$$mass\ before = Volume * \rho[65\ psig] \tag{34}$$

$$mass\ before = \frac{\pi(.305in)^2}{4} * 1164in * 2.213x10^{-4}\ \frac{lbs}{in^3} = 0.0188\ lbs \tag{35}$$

Where the internal diameter of .375 in. tubing is .305 in. (.035" wall). The densities of nitrogen at 65 psig and 50 psig are $2.213x10^{-4}\frac{lbs}{in^3}$ and $1.783x10^{-4}\frac{lbs}{in^3}$ respectively.

$$mass\ after = \frac{\pi(.305in)^2}{4} * (1164\ in) * 1.783x10^{-4}\frac{lbs}{in^3} = 0.0152\ lbs \tag{36}$$

The difference in mass between the two states is 0.0036 lbs pressurized at 50 psig. The difference in mass corresponds to a volume of 20.6 in^3. This volume is split between three accumulators, leaving an addition of 6.87 in^3per accumulator.

$$Volume\ Ratio = \frac{6.87\ in^3}{674\ in^3} = 1.02\% \tag{37}$$

The total volume added over the duration of the rod ejection is ~1% of the total volume of the accumulator. Note that this calculation is assumed to be free of losses, which is very unlikely and will drop the total volume addition to <1% per accumulator. Since the total addition from the piping is <1%, it has little effect on the total pressure in the accumulator or cylinder and can be ignored.

For longer RHU times (0.40s and longer), the valve stays open longer, allowing for more flow through the valve and into the accumulators. However, the rod ejection time stays fairly similar, meaning that the only difference is that the valve is open longer after the piston has reached the full-up position. While more nitrogen will flow into the accumulator during this time, it can be neglected due to the fact that the pressure between the cylinder and accumulators is close to equilibrium. The equilibration of the pressure causes the mass flow rate through the valve to drop towards zero, meaning that the flow into the accumulator will stay in the accumulator and not make it into the cylinder. Even for the longer duration of valve open time, the total volume of nitrogen added to the accumulator might reach up to 10% of the total accumulator volume, which considering very little makes it into the cylinder, means that it has little effect on the total system. Thereby, the assumption that the flow into the accumulator can be neglected is valid.

Flow Timing Delay from Valve to Cylinder

Once the nitrogen exits the valve on its way to the piston, there is a small delay in time for the nitrogen flow to reach the bottom of the piston. The nitrogen has to flow roughly 20 inches through one 1.25 in. diameter hose connected by a 1.25 in. 90 degree elbow before it reaches the cylinder manifold block (see Figure 7). The max flow rate of the nitrogen is near the beginning of the pulse when the pressure drop is the greatest. Based on the calculations with reference [6], the max flow speed is roughly 73 m/s or 2874 in/s. The flow delay is calculated as follows:

$$flow\ delay = \frac{flow\ length}{flow\ speed} = \frac{20\ in}{2874\frac{in}{s}} = .00696s \ \cong 7ms \tag{38}$$

This has a direct influence on the TR timing such that the pressure transducers don't start picking up the increased pressure until roughly 7 ms after the flow starts in the valve.

During the rod drop, the flow delay is greater due to a decreased flow speed which is caused by a smaller drop in pressure. The pressure drop between the cylinder and atmosphere during rod drop starts at a max value of roughly 40 psig and drops to 0 psig (atmosphere). This translates to an increased flow delay of ~3 ms, totaling ~10ms total.

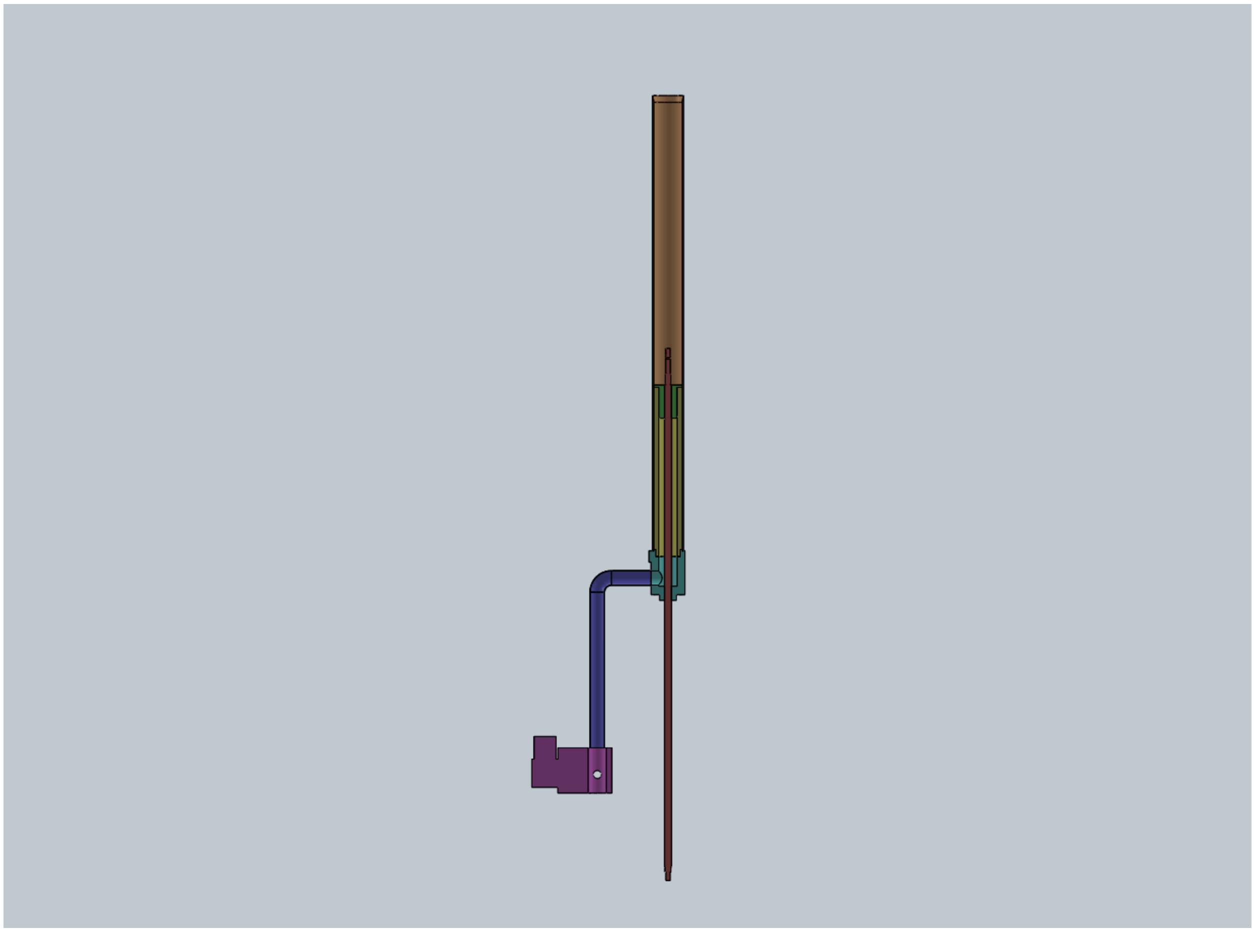


Figure 7. Visual Representation of Valve to Cylinder. The flow starts in the valve (pink) and flows through the piping to the manifold block and ending at the bottom of the piston. The valve and piping geometries are simplified for simulation.

Transient Rod Timing

In order to accurately model a pulse, the timing of how the TR fires and drops must be understood. However, while comparing the timing sequence provided by ACRR operators to experimental measurements (more on this in the results section), there happened to be significant discrepancies between the two. Here both the operator perspective and data perspectives will be explained.

Operator Perspective

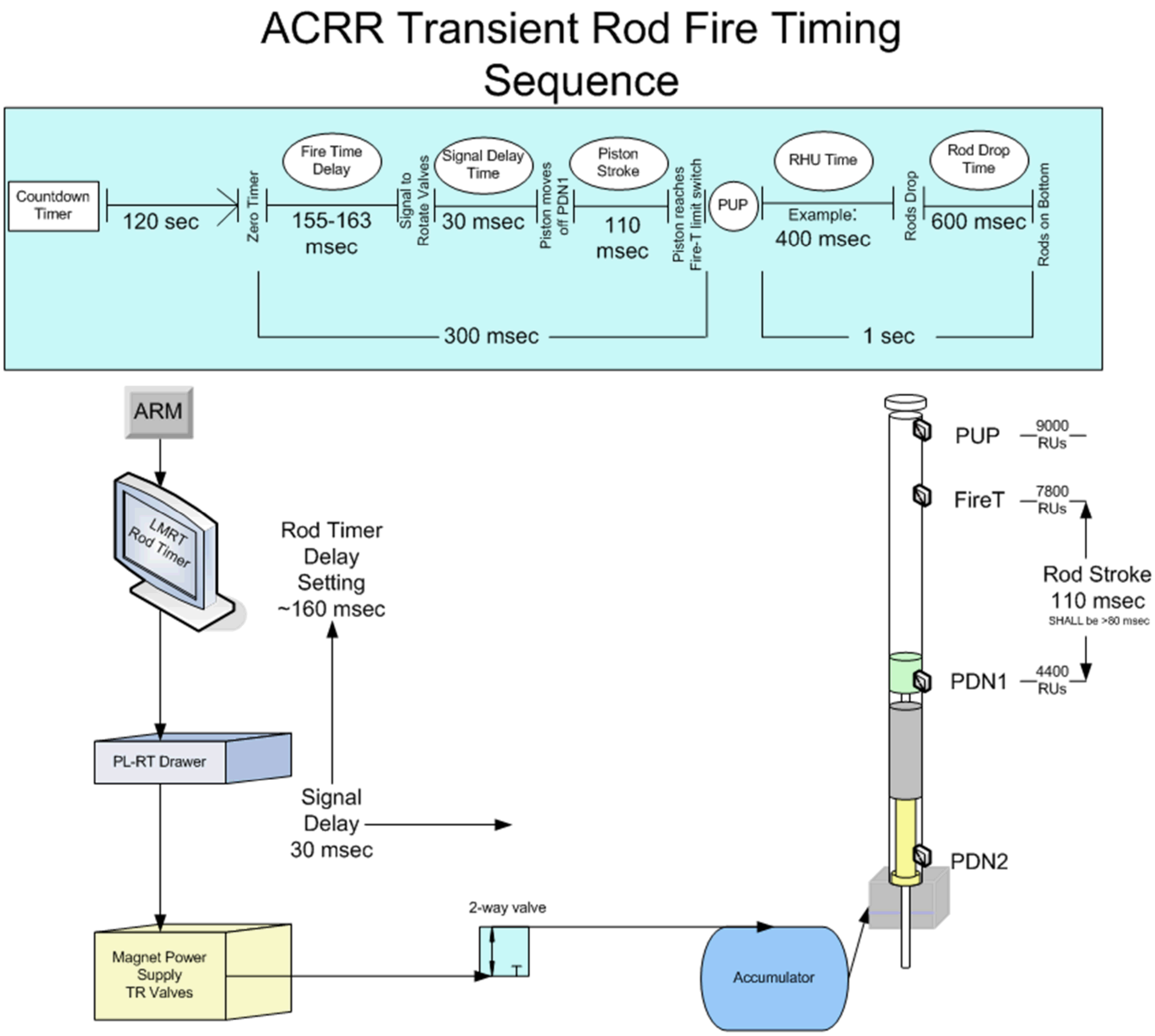


Figure 8. Diagram of the Transient Rod Timing Sequence

Figure 8 shows the reactor operator version of the transient rod timing sequence. The five important time ranges during a pulse are

- Fire Delay Time
- Signal Delay Time
- Piston Ejection/Stroke Time
- Rod Hold-Up (RHU) time
- Rod Drop Time

The countdown timer, which is set by a logic controller to two minutes in order for people in and around the reactor to know when the reactor is about to pulse, starts the pulse timing sequence. Once the countdown timer reaches 0, the zero timer starts and then cycles through the fire delay time for each transient rod. Fire delay time is set by the operators at certain times for each TR. The purpose of the fire delay time is to set each TR's timing sequence such that all three reach the FireT limit switch simultaneously. Once the fire delay time is reached, the logic controller for each TR sends an electronic signal to the solenoid valve. The time it takes for the signal to reach the valve from the controller is estimated at 30 ms, but that number has not been confirmed with any verifiable accuracy. Once the signal reaches the valve, the valve, which operates as a globe valve, actuates a solenoid inside causing a piston inside the flow volume to un-seat allowing nitrogen from the accumulator to start flowing into the cylinder. However, the process of actuating the piston inside the valve also takes time, which is called the valve opening time. The value for valve opening time is widely unknown, but a similar valve from the same manufacturer is said to open in roughly 40 ms [2]. During this 40 ms, the nitrogen starts to flow according to a linear opening profile (since the valve exhibits characteristics of a globe valve), meaning that the flow coefficient of the valve is linearly inhibited during the time it takes the valve to open [3]. From the operator's perspective, the valve opening time is built into the signal delay time and the beginning of the piston stroke.

Once the valve is fully open, the nitrogen flows uninhibited from the accumulator to the valve (mass flow rate calculations are described in the next section). The flow causes the piston to move rapidly up the cylinder. It is required by the Technical Safety Requirements (TSRs) for the ACRRF [8] that the full stroke piston time is greater than 80 ms, which gives a baseline for the piston ejection time. Piston ejection time is not a set value and varies depending on the size of the pulse. After the piston hits the full up position (measured by a magnet passing by a magnetically operated limit switch on the outside of the cylinder), the RHU time starts. The rod then stays in the full-up position for the duration of the RHU time. Once the RHU time is exhausted, the valve is signaled to close and allow the cylinder to exhaust to atmosphere. Another cycle of signal delay time and valve closing time (assumed to be the same as valve opening time) starts off the rod drop time portion. The TSR requires the rod drop time to be no greater than 2 seconds. After the piston has dropped back to the pedestal, the pulse is finished.

Data Perspective*

The operator and data perspective on TR timing differ primarily in RHU time. After thoroughly analyzing the data and pinpointing the locations along the pressure distributions where certain characteristics occur, the RHU time can be backed out as starting at the zero timer rather than after the piston hits the piston up limit switch. This phenomenon is especially noticeable with a 0.250s RHU time, in which the valve is signaled to close before the piston reaches the full up position. Figure 9 shows a visual representation of RHU discrepancy, along with a couple other additions, such as dislodging the valve opening time from the signal delay time and piston stroke, and further separation of piston motion. The methods to discovering the discrepancies and separations is explained later on in the results section.

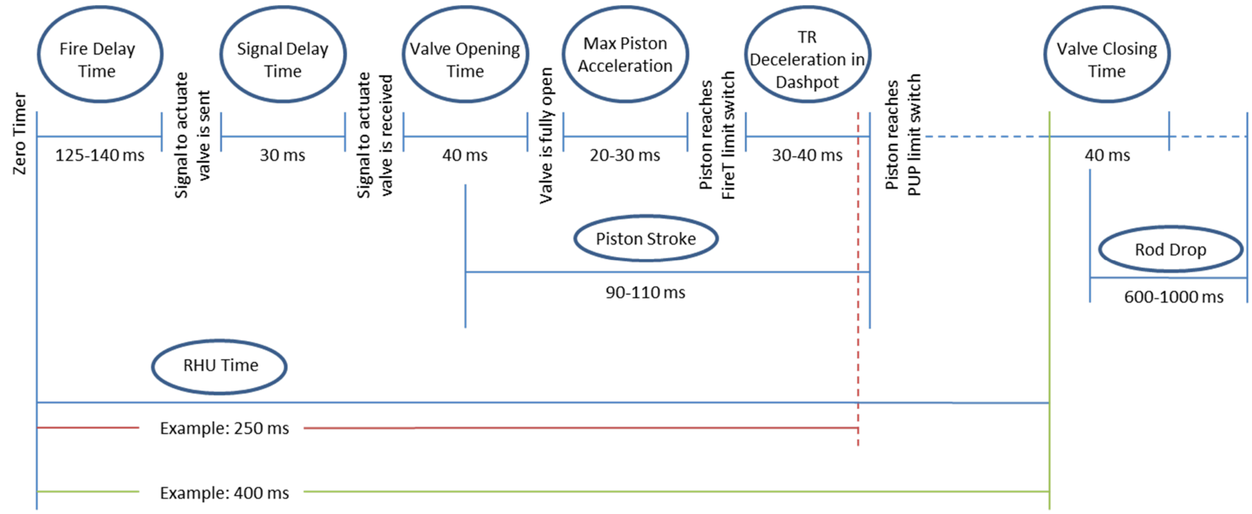


Figure 9. TR Timing Diagram Concluded from Experimental Data.

*Note that the data perspective may only apply to the .25 sec RHU time. It has not been verified with longer RHU times as of yet.

Mass Flow Rate

Arguably the most important parameter in determining pulse characteristics is the mass flow rate through the solenoid valve. While the mass flow rate is not too difficult to calculate from system parameters, there are many parameters, especially geometrical details that are either unknown or uncertain. This is where the published information on the specific solenoid valve comes into question.

Solenoid Valve Explanation

The solenoid valve being used in the ACRR is the Parker-Hannifin H2001NC12501 1.25 inch port, 3-Way valve. The NC stands for normally closed, meaning that during idle operation, the nitrogen will not flow from the accumulator to cylinder; instead the cylinder is open to the exhaust port, which is at room temperature and atmospheric pressure. One important note is that even though the valve is labeled as 3-way, there are only two flow directions that matter: inlet -> outlet and outlet -> exhaust. This piece of information has prompted most of the engineers working on the ACRR to call it the two way valve.

The Parker Hannifin valve being used in the system today is no longer in production and hasn't been for a couple decades. This makes finding information regarding its operation much more difficult. The specification sheet provides a flow coefficient, C_v , of 17.5, which is used the mass flow calculations [4]. When it comes to geometric parameters, the catalog entry and specification sheet provide plentiful measurements of the body and orifices of the valve, but none of the inner dimensions are listed [5]. Turns out, the inner dimensions are proprietary; however, the flow calculations can still be completed without them.

Mass Flow Rate Calculation

Mass flow rate is calculated using ISA-75.01.01-2007, *Flow Equations for Sizing Control Valves*, published by the International Society of Automation (ISA) [6]. Since the flow through the TR pneumatic system is nitrogen, it is subject to compressible flow effects. The flow through the specific Parker Valve on the TR system can be classified into two categories: choked and non-choked flow. Choked flow occurs when the pressure difference between the two sides of the valve is so great that the mass flow rate is limited by the geometry of the valve. The result of choked flow is that the mass flow rate is only a function of the inlet pressure rather than the pressure difference; thereby, causing the mass flow rate to decrease in a more linear fashion for transient flow. The guidelines for choked vs. non-choked flow, as published by Swagelok (the company that manufactures the majority of parts that make up the TR pneumatic system), states that flow will be choked when the downstream pressure of the valve is less than half of the upstream pressure [7]. With this in mind, here are the equations used by the analysis to calculate the mass flow rate.

Choked Flow

If $X > F_\gamma X_{TP}$,

$$\dot{m} = \frac{0.667 C_v N_6 F_P \sqrt{F_\gamma X_{TP} P_1 \rho_1}}{3600} \qquad \left[\frac{lbm}{s}\right] \tag{39}$$

Where

$$C_v = Flow\ Coefficient = \mathbf{17.5} \tag{40}$$

$$N_6 = constant = \mathbf{63.3} \tag{41}$$

$$F_p = Piping\ Geometry\ Factor = \frac{1}{\sqrt{1 + \frac{\sum \xi}{N_2}\left(\frac{C_i}{d^2}\right)^2}} \tag{42}$$

$$F_\gamma = Specific\ Heat\ Ratio\ Factor = \frac{\gamma}{1.4} = \mathbf{1}\ for\ N_2 \tag{43}$$

$$X_{TP} = \frac{\frac{X_T}{F_p^2}}{1 + \frac{X_T \xi_i}{N_5}\left(\frac{C_i}{d^2}\right)^2} \tag{44}$$

$$X = \frac{P_1 - P_2}{P_1} \tag{45}$$

$$N_2 = 890; N_5 = 1000 \tag{46}$$

$$C_i = 1.3\ C_v \tag{47}$$

And P_1 and ρ_1 are the upstream pressure and density in units of psia and $\frac{lbm}{ft^3}$ respectively. X_{TP} is the pressure differential ratio factor of a control valve with attached fittings at choked flow, while X_T is the same factor without attached fittings. X_T for globe valves with a contoured plug is .72 [6]. ξ is the velocity head loss of a reducer, expander, or other fitting attached to a control valve or trim. The velocity head loss occurs twice at each inlet and exit, as shown below

$$\sum \xi = \xi_1 + \xi_2 + \xi_{B1} + \xi_{B2} \tag{48}$$

Where

$$\xi_1 = 0.5\left[1-\left(\frac{d}{D_1}\right)^2\right] \rightarrow inlet\ expander \tag{49}$$

$$\xi_2 = 1.0\left[1-\left(\frac{d}{D_2}\right)^2\right] \rightarrow outlet\ expander \tag{50}$$

$$\xi_{Bi} = 1-\left(\frac{d}{D_i}\right)^4 \tag{51}$$

d is the nominal pipe diameter of 1.25 inches, and the inner diameter of the piping leading up to and away from the valve is 1.36 inches (D=D_1=D_2=1.36). Plugging in the values for the nominal and inner diameters of the pipes, $\sum\xi = .2304$. C_i is an assumed flow coefficient used for iterative purposes, and if both ends of the valve are the same size (they are), C_i can be used in place of C_v if $C_i \geq \frac{C_v}{F_P}$. Solving for F_p using C_i yields

$$F_P = \frac{1}{\sqrt{1+\frac{.2304}{890}\left(\frac{1.3*17.5}{1.25^2}\right)^2}} = .974 \tag{52}$$

Since $1.3*17.5 = 22.75 \geq \frac{17.5}{.974} = 18.0$, F_P stands.

ξ_i from (44) equals the sum of (49) and (51) where $i = 1$. When the values are plugged in, $\xi_i = \xi_1 = .3604$. Solving for X_{TP}

$$X_{TP} = \frac{\frac{0.72}{0.974^2}}{1+\frac{0.72*.3604}{1000}\left(\frac{1.3*17.5}{1.25^2}\right)^2} = .7202 \tag{53}$$

Simplifying (53) with all of the factors and constants, yields the following equation for mass flow rate

$$\dot{m} = .1696\sqrt{P_1\rho_1} \qquad \left[\frac{lbm}{s}\right] \tag{54}$$

When

$$P_1 - P_2 \geq F_\gamma X_{TP} = .7202 \tag{55}$$

Non-Choked Flow

If $X < F_\gamma X_{TP} = .7202$,

$$\dot{m} = \frac{C_v N_6 F_p Y\sqrt{XP_1\rho_1}}{3600} \qquad \left[\frac{lbm}{s}\right] \tag{56}$$

Where

$$Y = 1-\frac{P_1-P_2}{3P_1F_\gamma X_T} \tag{57}$$

Results and Findings

Two sets of flow results will be presented in this section: experimental pressure transducer results and analytical coding results. The two will be compared to explain discrepancies and nuances.

.25 sec RHU

Pressure Distributions

Experimental Data

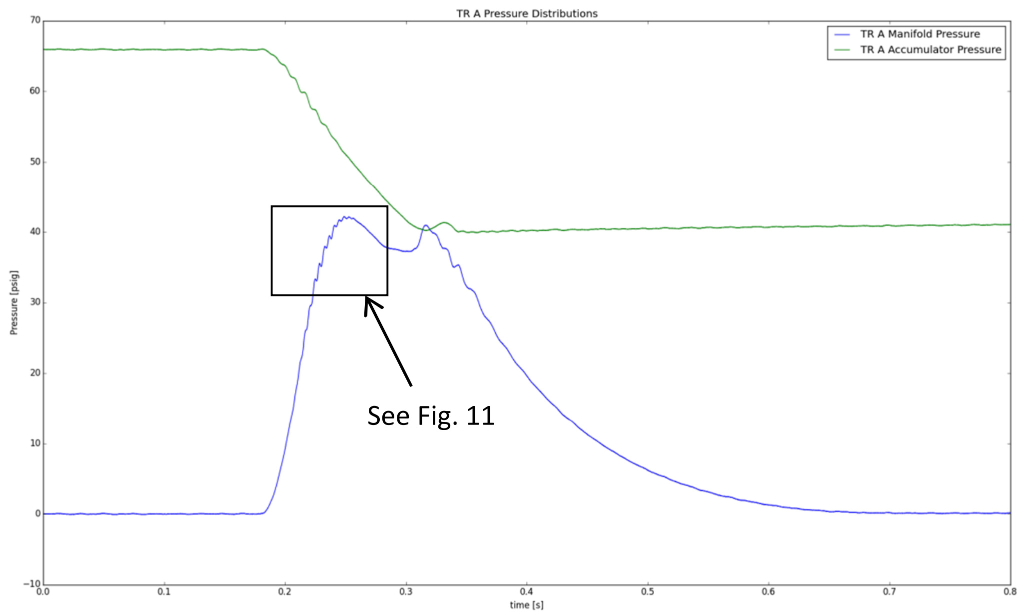


Figure 10. Pressure (Gauge) Distributions of TR A for both the Accumulator and Manifold during Pulse 11616. 11616 is a $1.657 pulse operated in Pulse-Reduced Tail mode with a .25 sec RHU time and piston starting at 5299 Rod Units

The data in Fig. 10 was taken from Setra 209 pressure transducers mounted on the accumulator and cylinder manifold blocks for all three TRs. The specific example above is only for TR A, but the other two rods exhibit similar results for the pulse. Notable timing of certain events is summarized below in Table 1.

Table 1. TR Timing for Experimental Pressure Data

Event	Duration	Elapsed Time	Significance
Zero Time	0.000s	0.000s	Transient rods are primed for ejection
Fire Time Delay	0.133s (for A, differs for B and C)	0.133s	Signal is generated to actuate valve
Signal Delay Time	0.010s	0.143s	Time for the signal to reach the valve
Valve Opening Time	0.040s	0.183s	Time for valve to

			actuate (Note that fluid starts flowing through valve before it is fully open)
Flow Delay Time	0.007s	0.190s	Time for nitrogen to travel from valve to cylinder
Jagged Edges in Rise	~ 0.040s	~0.210s-0.260s	Piston enters the dashpot and slows as it moves through the dashpot
Rod Hold Up (RHU) time	0.250s (from Zero Time)	0.250s	Signal to close valve is generated
Piston Full Up	~0.70-0.90s (from first piston motion)	~0.260s	Piston hits full up position
Signal Delay Time	0.010s	0.260s	Signal reaches valve to close it
Valve Fully Closed	0.040s	0.300s	Nitrogen starts venting out of the cylinder through the valve
Flow Delay Time	~0.010s	0.310s	Manifold pressure starts dropping

A few notable events are derived from Table 1. First of all, there are many delays that occur before the piston even starts to move (roughly around .180s). Other than the set delay (Fire Time), there is a roughly 10 ms delay from sending the signal from the Log Master Rod Timer (LMRT) through the Pulse Logic Rod Timer to the Magnetic Power Supply (MPS) ending at the solenoid valve. The MPS, which contains three magnetic power relays, has a maximum delay of roughly 10 ms due to the mechanical actuation of the three magnetic relays.

The flow delay time, explained in greater detail in Assumption Justifications Section, accounts for a roughly 7 ms delay on rod ejection; however, during the rod drop, the flow delay time is increased to roughly 10 ms due to a decreased pressure drop (which results in a smaller mass flow rate and flow velocity).

An important discovery can be found in the jagged edges between .21s and .26s. Looking more closely at the geometry of the dashpot, a correlation is found between the local peaks and holes in the dashpot. The dashpot, shown in Figure 11, has three sets of four .625 in. diameter holes separated by .75 in. and 6 sets of three .1875 in. holes of uneven spacing. In total, there are 9 sets of holes, which correlate to the 9 local peaks starting with the one at .225s. This implies that as the piston passes by the last peak at .257s, it is within ~.25 in. of full up position. This allows for pinpointing the full up position at roughly .260s.

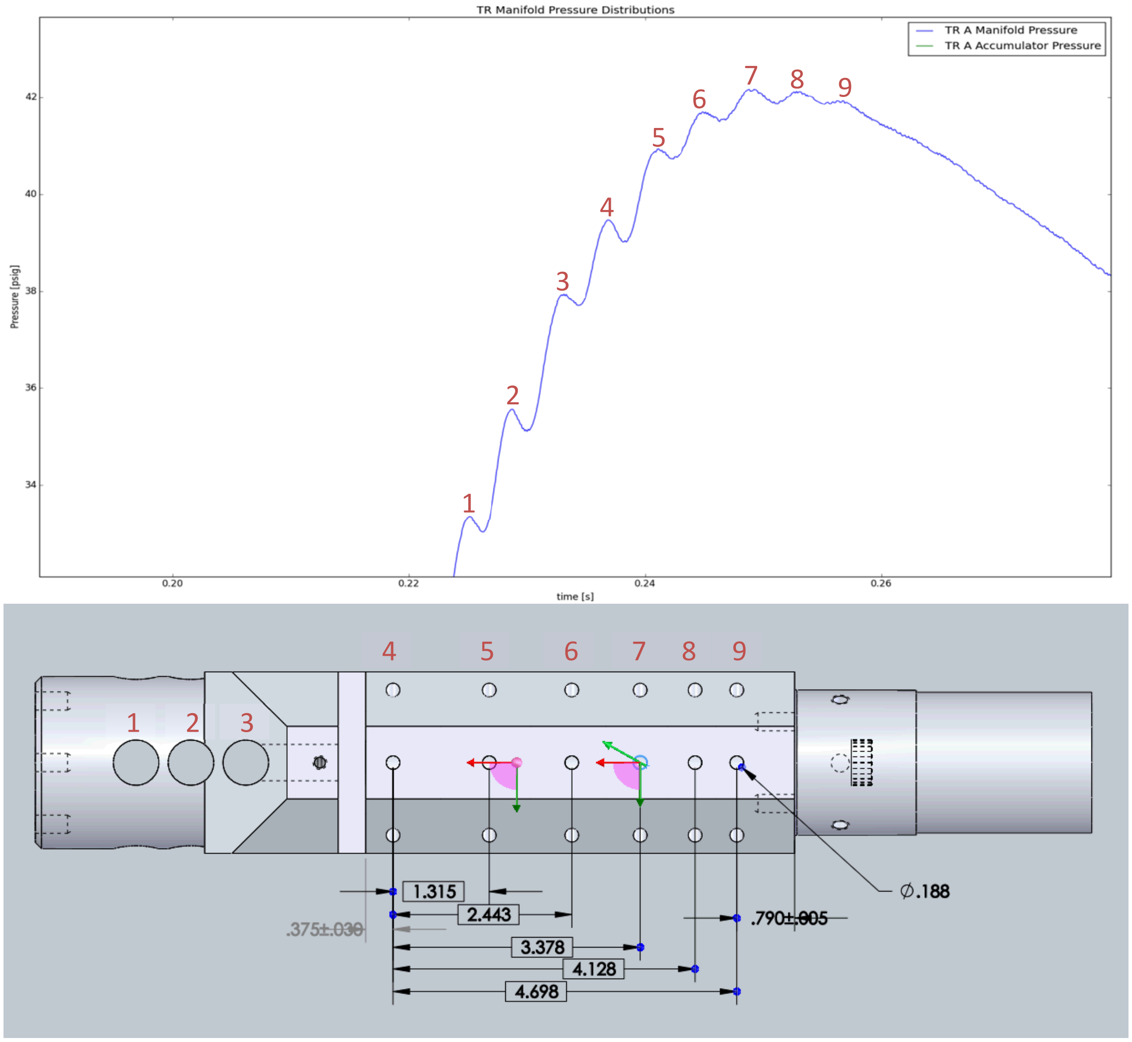


Figure 11. Overlay of Dashpot Geometry with Local Peaks on Pressure Transducer Data. The numbered peaks correspond to the row of holes on the dashpot.

Analytical Data

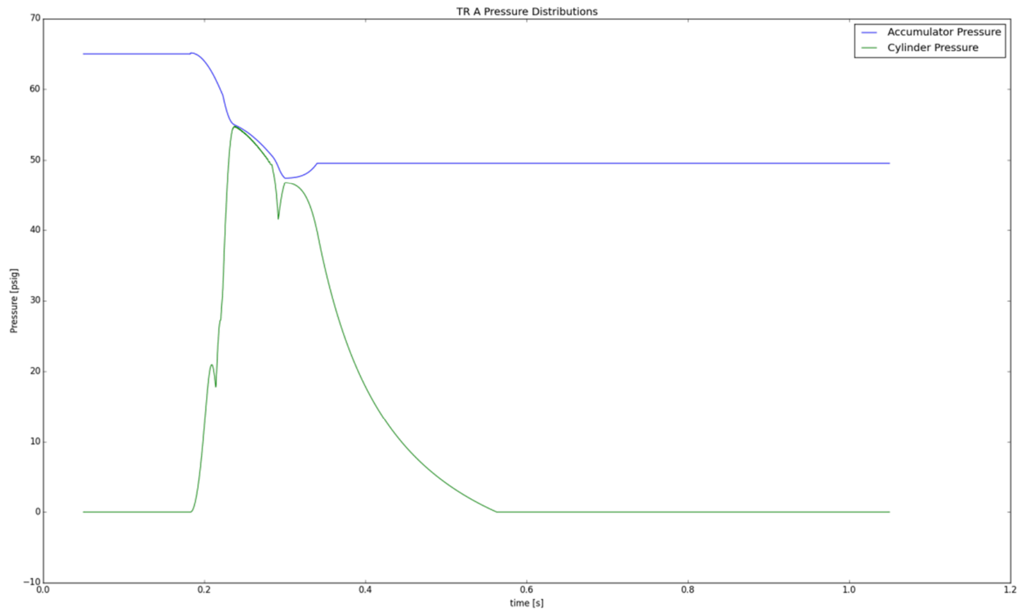


Figure 12. Analytical Pressure (Gauge) Distributions of TR A for both the Accumulator and Cylinder. Data is based off of the same 11616 pulse with .25 sec RHU time and starting piston position of 5299 Rod Units.

Figure 12 shows the pressure distributions as generated from the coding of the analytical solution. The most distinct differences can be seen between the peaks in the cylinder data, and also the higher pressure readings as compared to the experimental data. The major and minor differences are stated and explained in the list below:

- The small hump in the cylinder distribution at about .21 sec is due to the volume expansion from the movement of the piston in the cylinder being greater than the increase in mass of the system.
- The mushroom cap enters the dashpot around .22 sec, leading to a more jagged pressure distribution. This is largely caused by the assumption of how the deceleration in the dashpot works.
- The deceleration is assumed to work as follows: as the mushroom cap enters, the velocity of the piston decreases by ~ 10 m/s per time step. What happens from this is that the piston velocity (which is ~ 40 m/s as it enters the dashpot) decreases very significantly in a short amount of time and then fluctuates between positive and negative for the next .100s or so. The result from this is a jagged, but relatively flat pressure distribution (from .23 sec to .28s) and an oscillating displacement profile during the same period (explained in more detail in the next section).
- The justification for the choice in deceleration profiles is due to the fit of the pressure graph as compared to the experimental data.
- The full up position of the piston (at .29 sec) is much later than in the experimental data, due to oscillatory motion in the dashpot.
- As can be seen from Fig. 13, the accumulator pressures do not align too well. This is due to the algorithm used to calculate the mass flow rate. As the pressures are closer in magnitude, the mass

flow rate drops significantly causing less mass to flow from the accumulator. This keeps the accumulator pressure artificially high.

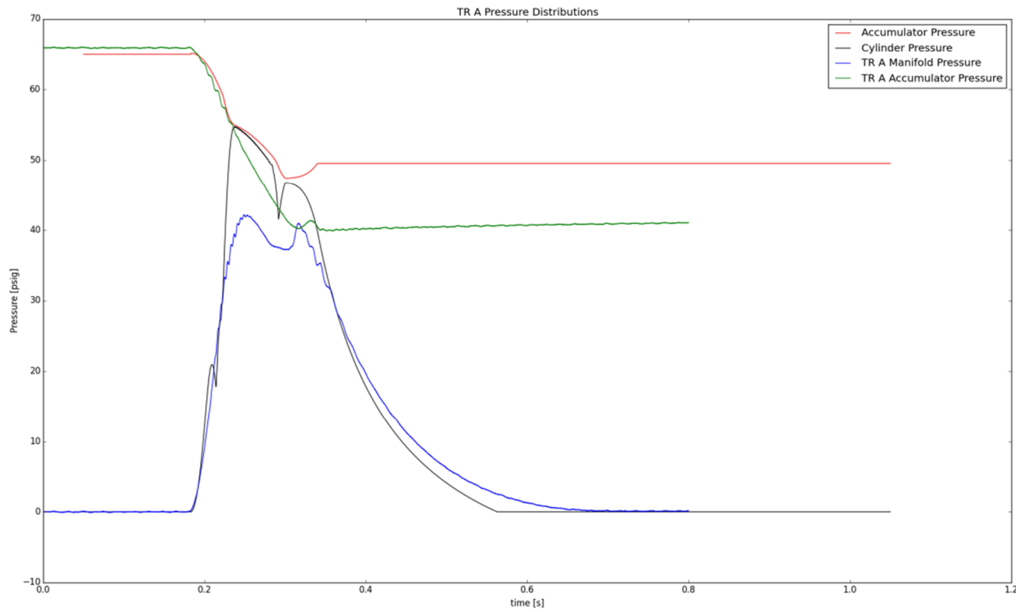


Figure 13. Overlay of Analytical and Experimental Pressure Distributions for Pulse 11616.

Table 2 explains the TR timing sequence implemented in the coding of the analytical solution; as well as, notable events during the pulse.

Table 2. TR Timing for Analytical Pressure Data

Event	Duration	Time	Significance
Zero Time	0.000s	0.000s	Transient rods are primed for ejection
Fire Time Delay	0.133s (for A, differs for B and C)	0.133s	Signal is generated to actuate valve
Signal Delay Time	0.010s	0.143s	Time for the signal to reach the valve
Valve Opening Time	0.040s	0.183s	Time for valve to actuate (Note that fluid starts flowing through valve before it is fully open)
Jagged Edge in Rise (Volume Expansion)	~ 0.010s	~0.210s	Pressure drops due to expansion of volume in the cylinder
Rod Hold Up (RHU) time	0.250s (from Zero Time)	0.250s	Signal to close valve is generated
Signal Delay Time	0.010s	0.260s	Signal reaches valve to

			close it
Piston Full Up	~0.90-0.110s from first piston motion	~0.290s	Piston hits full up position
Valve Fully Closed	0.040s	0.300s	Nitrogen starts venting out of the cylinder

Displacement Profiles

Note that the following displacement profiles only include the rise of the piston. The rod drop is still in the development phase.

Experimental Data

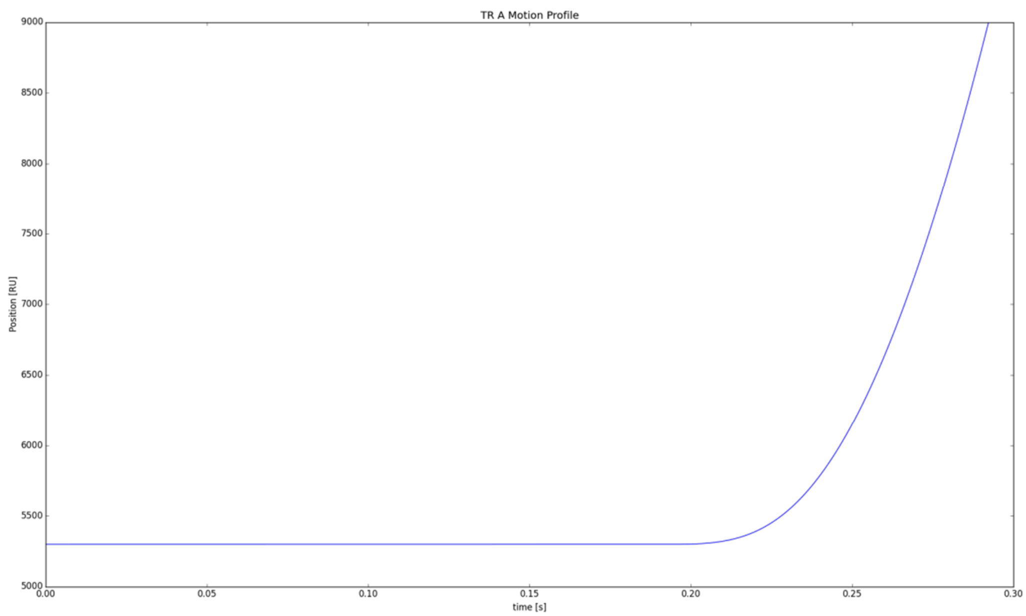


Figure 14. Motion of the Piston in TR A during Pulse 11616. Time Starts at Fire Time.

The displacement of the piston in the experimental data (shown in Figure 14) assumes one major piece of data: the piston moves smoothly based on the pressure distribution. This, in reality, is not true due to massive deceleration in the dashpot. Dashpot deceleration has currently not been estimated for this data set, but will be soon. The time for full piston motion is ~ 90 ms, but this will likely change with dashpot deceleration.

Analytical Data

There are massive discrepancies between the experimental and analytical data set for motion profiles. This is due solely to the assumption of dashpot deceleration. Since the deceleration assumed in the code model is based on decreasing velocity, there are sections of negative and flat motion. From what is thought about the piston motion, this case is not true; however, some insight can be gleaned from this data. The deceleration in the dashpot is based solely on the concept of a hydro-lock brake, which works by compressing a volume of liquid until it can no longer be compressed, bringing a piston (mushroom cap in this case) to an abrupt stop. Since this system uses the hydro-lock brake in a unique way as compared to what is seen with an automotive engine (which is highly undesirable), the motion in the dashpot is highly uncertain. There will certainly be some oscillations in the motion, and it is highly likely that the piston will rise and drop due to the massive deceleration. The rise and drop lead to mechanical vibration and stress in the TR components, and the drop could lead to the TR components transitioning from tension to compression. In terms of the motion distribution in Figure 15, it is unlikely that the flat portion is completely accurate; nevertheless, this is what the code outputs at the moment.

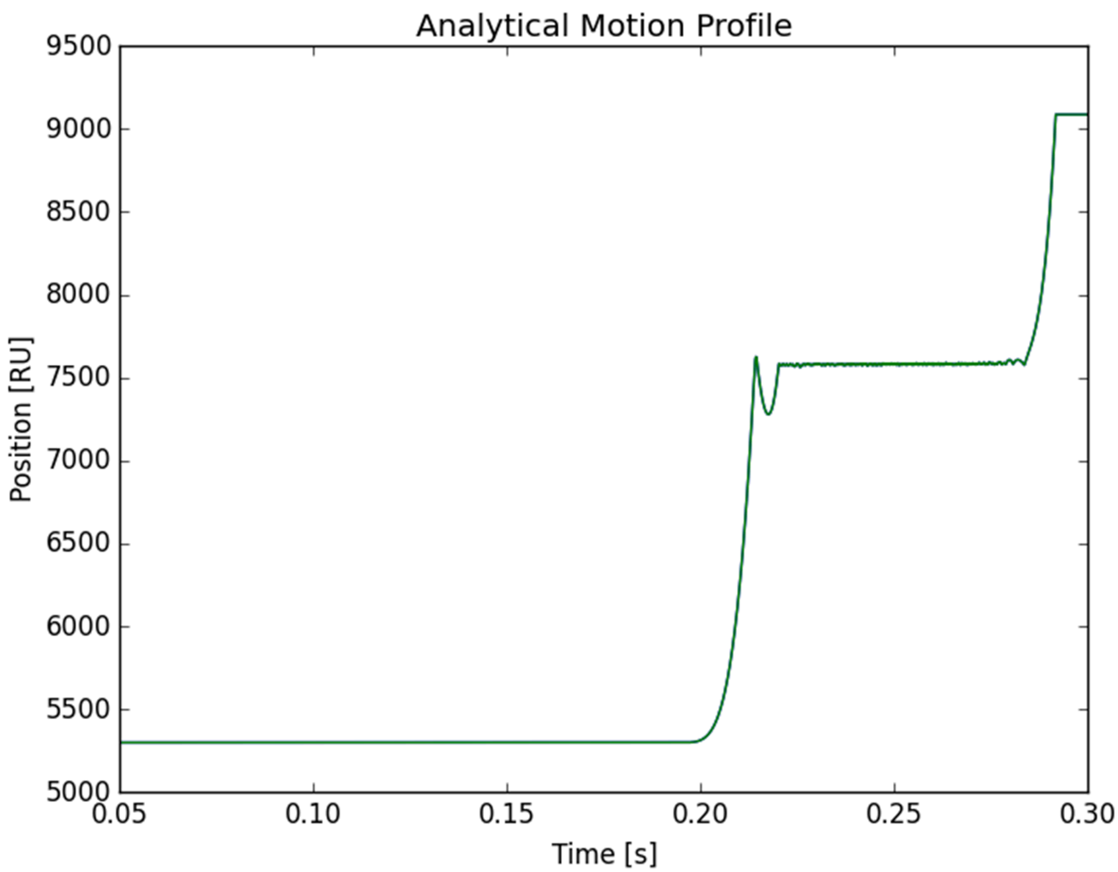


Figure 15. Motion of the Piston during Pulse 11616 according to the Analytical Method.

When comparing the data from Figure 16, one notices that the rise time for both is nearly identical. This timing should be expected; however, is not completely accurate due to the inclusion (and assumption) of dashpot acceleration in the analytical model, and the exclusion of dashpot acceleration in the experimental data.

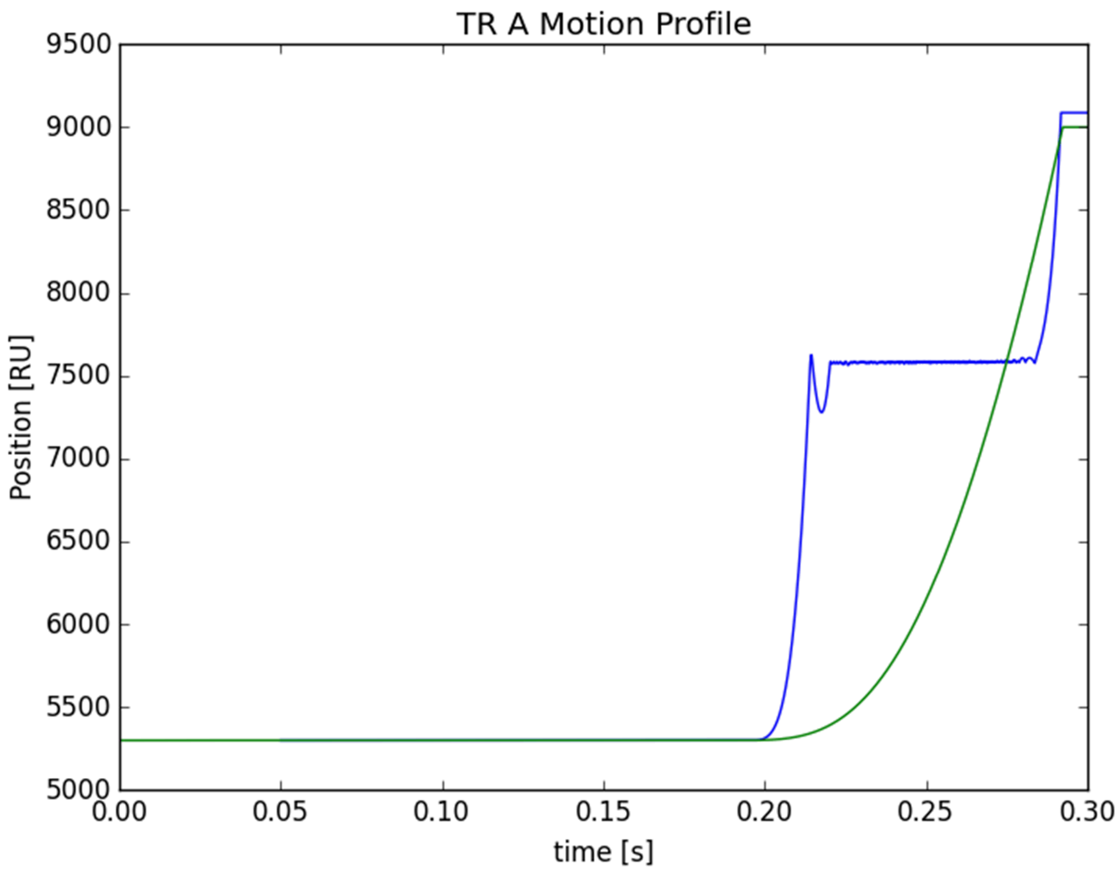


Figure 16. Comparison of Analytical and Experimental Piston Motion. Notice that the Timing to Full-Up is Nearly Exact.

.40 sec RHU and longer

The code analysis currently does not have a solution for pulses with a RHU longer than .25 sec. This feature is under development and should be included in the near future.

Future Work

Some aspects of future work include:

- Solving for all three TRs
- Extend analytical solution to solve for longer RHU times
- Verify that TR timing chart in Figure 5 is correct
- Develop acceleration profile for Razorback

References

[1] Gyftopoulos, E. P., and Gian Paolo. Beretta. "Bulk Flow." *Thermodynamics: Foundations and Applications*. Mineola, NY: Dover Publications, 2005. 360-69. Print.

[2] Parker Pneumatic Division Applications Engineering (Personal Communication, August 27, 2015)

[3] Knight, Erin, Matthew Russell, Dipti Sawalka, and Spencer Yendell. "ValveModeling." *Control Valve Wiki*. University of Michigan, 26 Sept. 2006. Web. 28 Nov. 2015.

[4] Specification Sheet, Parker H2000 Valves. Accumulator Relocation Mod Sharepoint Site

[5] Catalog Entry, Parker H2000 Valves (Personal Communication with Paker PDN Dvision, October 12, 2015)

[6] ISA-75.01.01-2007 (60534-2-1 Mod), "Flow Equations for Sizing Control Valves". 2007

[7] "Valve Sizing", Swagelok Flow Catalog.

[8] Sandia Report SAND2008-5637

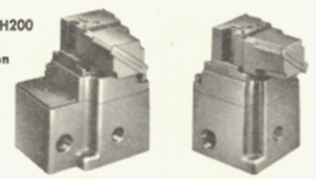
Appendix A: Solenoid Valve Spec Sheet

PARKER HANNIFIN
AIR CONTROL VALVES

CATALOG 0635
FILE A84.2

series
H200 and H2000
3-way "HUSTLER"

Series H200 With Junction Box



Series H200 Without Junction Box

SPECIFICATIONS

1. A base mounted, plug-in valve that can be serviced or converted from normally closed to normally open without disturbing the piping or wiring.
2. Valve can be converted from normally closed to normally open by simply changing the cartridge assembly.
3. Nylon encapsulated plug-in coil available in a wide range of AC and DC voltages.
4. *Shur-Shift* chamber ensures positive shift.
5. External pilot exhaust.
6. Internal pilot supply—may be converted to external pilot supply.
7. Both pilot and main valve are poppet type, ensuring positive sealing and insensitivity to foreign material.
8. Poppet construction will provide millions of trouble free cycles without lubrication.
9. Lubrication requirements are not critical, which permits the valve to operate dry or with heavy, light or intermittent oil delivery.
10. The short piston poppet stroke provides speed and uniform response, plus high flow capacity.
11. Equipped with Locking Manual Override–(Available with Non-Locking Manual Override–Add M1A to Model).
12. Built to J.I.C. Specifications.
13. Compact—takes less space than other three-way valves.
14. Response Time and Flow Capacities (Cv) are related in the table below.

USES

1. Brake and clutch actuation.
2. Single-acting cylinders.
3. In pairs to operate double-acting cylinders for high speeds, positioning or inching.
4. Selection of two different pressures or pressure sources.

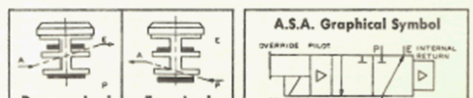
- 3-WAY (AND 2-WAY) 2-POSITION
- SOLENOID ACTUATED — MAINTAINED CONTACT
- PILOT OPERATED — INTERNAL PRESSURE RETURNED
- PISTON POPPET
- 1/2", 3/4", 1", 1-1/4" NPTF TAPPED SIDE PORTS
  Alternate Porting Available (See "How to Order")
- PRESSURES — 15 TO 150 PSI AIR
- 115/120 AND 230 VOLT, 50/60 CYCLE CONTINUOUSLY RATED SOLENOIDS STANDARD — 7.7 Watts (0.24 Amps Inrush, 0.14 Amp Holding on 115-120/50-60 Cycle). Other AC and DC Solenoids Available

HOW IT WORKS

Normally Closed

With the solenoid de-energized, internal pressure raises the piston poppet. Pressure at Port "P" is blocked and Port "A" is open to exhaust Port "E".

When the solenoid is energized, pilot pressure drives the piston poppet down. Pressure at Port "P" is open to Port "A", exhaust Port "E" is blocked.

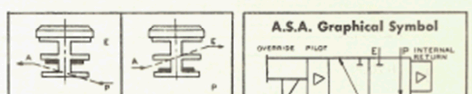


De-energized | Energized | A.S.A. Graphical Symbol

Normally Open

With the solenoid de-energized, internal pressure raises the piston poppet. Pressure at Port "P" is open to Port "A", exhaust Port "E" is blocked.

When the solenoid is energized, pilot pressure drives the piston poppet down. Pressure at Port "P" is blocked and Port "A" is open to exhaust Port "E".



De-energized | Energized | A.S.A. Graphical Symbol

Response Times and Capacity Coefficients

Valve Series	Valve Size	Response Time (Seconds)				Capacity Coefficients(Cv)†	
		12 Cubic In. Chamber		100 Cubic In. Chamber		Flow Directions	
		Fill	Exh.	Fill	Exh.	P to A	A to E
H200	½" NC	.025	.036	.066	.090	6.28	9.21
	½" NO	.031	.052	.073	.120	6.46	8.32
	¾" NC	.025	.029	.060	.088	7.89	9.39
	¾" NO	.031	.036	.063	.100	8.25	8.48
Valve Series	Valve Size	100 Cubic In. Chamber		500 Cubic In. Chamber		Flow Directions	
		Fill	Exh.	Fill	Exh.	P to A	A to E
H2000	¾" NC	.069	.087	.200	.221	10.66	17.06
	¾" NO	.080	.093	.203	.256	10.99	14.71
	1" NC	.067	.078	.173	.190	14.92	19.74
	1" NO	.071	.093	.185	.235	14.92	16.02
	1¼" NC	.062	.072	.162	.176	17.24	19.74
	1¼" NO	.068	.090	.156	.204	17.24	16.39

†See reverse side for Cv formula.

HOW TO ORDER

Valve Model Numbers

PIPE SIZE -50= 1/2" NPTF -75= 3/4" NPTF -100=1" NPTF -125=1-1/4" NPTF		SERIES H200			SERIES H2000		
		Side Ports Tapped*	Bottom O-Ring Ports	Side & Bottom Ports Tapped*	Side Ports Tapped ⧫	Bottom O-Ring Ports	Side & Bottom Ports Tapped
3-Way Normally Closed	Without Junction Box	H201NC-50 -75	H203NC-75	H205NC-50	H2001NC-100 -125	H2003NC-125	H2005NC-75 -100
	With Junction Box	H202NC-50 -75	H204NC-75	H206NC-50	H2002NC-100 -125	H2004NC-125	H2006NC-75 -100
3-Way Normally Open	Without Junction Box	H201NO-50 -75	H203NO-75	H205NO-50	H2001NO-100 -125	H2003NO-125	H2005NO-75 -100
	With Junction Box	H202NO-50 -75	H204NO-75	H206NO-50	H2002NO-100 -125	H2004NO-125	H2006NO-75 -100

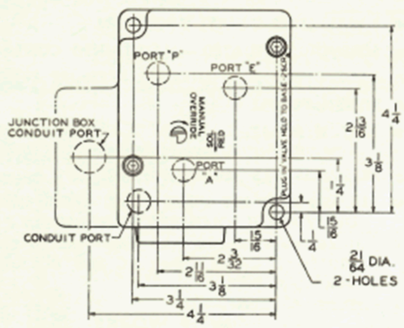
*Available with 3/8" porting. ⧫Available with 3/4" porting. Specify solenoid voltage and cycles.

Installation, Operation and Maintenance Instruction Form V-249 packed with every valve shipped.

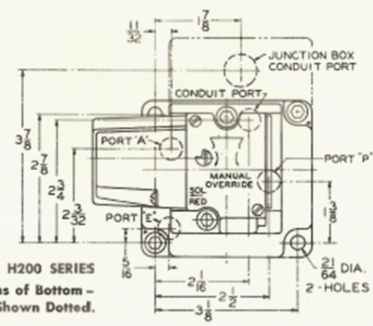
PARKER HANNIFIN

PNEUMATIC DIVISION
PARKER HANNIFIN CORPORATION
OTSEGO, MICHIGAN

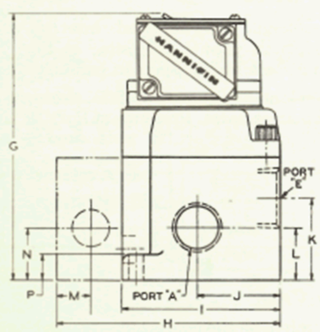
17



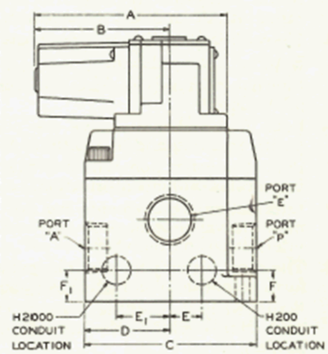
TOP VIEW H2000 SERIES
Port Locations of Bottom - Ported Models Shown Dotted.



TOP VIEW H200 SERIES
Port Locations of Bottom - Ported Models Shown Dotted.



- These valves will operate mounted in any position.
- Red electrical wires in junction box lead to Solenoid Red.
- Valves without junction boxes are supplied with 6′ leads. Valves with junction boxes have 9″ leads with attached terminals.
- Filtered and lubricated air recommended for maximum valve life and minimum maintenance.



SIDE VIEW H200 AND H2000 SERIES

FRONT VIEW H200 AND H2000 SERIES

VALVE DIMENSIONS

VALVE SERIES	PIPE SIZE	A	B	C	D	E	E_1	F	F_1	G	H	I	J	K	L	M	N	P
H200	½″ ¾″	4¼	2²¹⁄₃₂	3¾	1⅞	¹¹⁄₁₆	—	¹³⁄₁₆	—	5⅞	4¹⁵⁄₁₆	3½	1¹³⁄₁₆	1¹³⁄₁₆	1⁵⁄₁₆	¾	1⅛	⁹⁄₁₆
H2000	¾″ 1″ 1¼″	4¼	2²¹⁄₃₂	4⅞	2⁹⁄₁₆	—	1¾	—	¹³⁄₁₆	7⁵⁄₁₆	5⁵⁄₁₆	3⅞	2¹⁄₁₆	1¾	1¾	¾	1⅛	⁹⁄₁₆

PORT DIMENSIONS

	SIDE PORTS				SIDE & BOTTOM PORTED			BOTTOM O-RING PORTS			
SERIES	H200		H2000		H200	H2000		H200—¾″		H2000—1¼″	
VALVE SIZE	½″	¾″	1″	1¼″	½″	¾″	1″	COUNTER-BORED HOLE	O-RING GROOVE	COUNTER-BORED HOLE	O-RING GROOVE
PORT P*	½″	¾″	1″	1¼″	½″	¾″	1″	⁴⁵⁄₆₄″	1¹⁄₁₆″	1⁵⁄₃₂″	1⁹⁄₁₆″
PORT A*	½″	¾″	1″	1¼″	½″	¾″	1″	⁴⁵⁄₆₄″	1¹⁄₁₆″	1⁵⁄₃₂″	1⁹⁄₁₆″
PORT E*	1″	1″	1½″	1½″	½″	1″	1″	⁴⁵⁄₆₄″	1¹⁄₁₆″	1⁵⁄₃₂″	1⁹⁄₁₆″
CONDUIT PORT†	½″	½″	½″	½″	½″	½″	½″	½″	¾″	½″	¾″
JUNCTION BOX† CONDUIT PORT	½″	½″	½″	½″	½″	½″	½″	⁴⁷⁄₆₄″	1¹⁄₁₆″	⁴⁷⁄₆₄″	1¹⁄₁₆″

* Pipe Threads are NPTF † Conduit Threads are NPSM

C_v—Capacity Coefficients—(sometimes called Flow Factors). Each flow path through the valve has its own C_v value. All C_v ratings for each valve cataloged on this page are listed on the front side of this sheet.

$$C_v = \frac{Q}{22.67}\sqrt{\frac{GT}{(P_1-P_2)P_2}}$$

Q = Flow in Standard Cubic Feet per minute (14.7 PSIA at 60°F.)
P_1 = Inlet Absolute Pressure (gauge pressure +14.7)
P_2 = Outlet Absolute Pressure (gauge pressure +14.7)
Note: P_2 must be greater than .53 x P_1
G = Specific Gravity of flowing medium (Air, G=1)
T = Absolute Temperature of Air (460+°F.)