Pneumatic Valve Documentation

CSI Rocketry: Fluids Team 2023-2024
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1. <u>High-Level Introduction</u>

Objective: Design, manufacture, and test a novel single-acting pneumatic valve for controlling nitrous oxide flow to Valkyrie Mk2's injector.

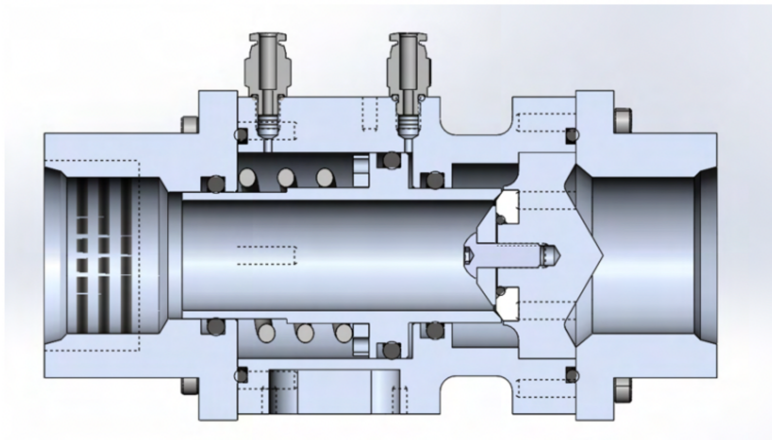
Previous Valve Designs: We have previously experimented with several valve designs. The earliest was a servo-actuated ball valve using a 3-D printed gear mechanism. While this was easily repeatable, the 3-D printed gears failed and led to a launch abort.

The following year, the team designed a pyro valve. This valve worked by using a piston to impede flow. When actuated, a gunpowder charge would be fired and the piston would slide forward, permitting flow. There were several challenges with this design, however. The first was the radial offset, which required additional plumbing with either a flex hose or bent pipe to mate to the injector body. The second was that, when testing, it would require the use of gunpowder, which was both hard to obtain and unsafe, a maintenance period after each static fire and cold flow attempt, and head loss due to turbulence induced in the flow.

For the 2022-2023 rocket PleaseGoUp, we designed and tested a ¾" ball valve with servo actuation. To prevent the binding issues of previous years, we changed the torque transfer mechanism from gears to a two-bar linkage. This proved to be robust even at low temperatures and the valve successfully flew and was recovered.

Two legacy ball valves. Left is the 2019 valve; right is the 2022-23 valve pre-flight.

Design Inspiration: We were initially inspired to build a pneumatic valve from the work of the University of Waterloo on their 2022-23 IREC entrant, Leviathan of the Sky. They were extremely successful using this valve on their vehicle and it enabled them to fly an extremely powerful O motor in a 6" diameter airframe and hit 31,000', which is a significant feat due to the constraints of designing nitrous oxide blowdown systems.



Leviathan of the Sky pneumatic valve, courtesy of Waterloo Rocketry

Pneumatic Valve Rationale: While the PleaseGoUp servo-actuated ball valve was successful, it has several drawbacks that make it infeasible for the latest rocket, tentatively named PleaseGoHigher. The first was the volumetric flow rate. For Valkyrie Mk2, the combustion chamber requires a higher oxidizer mass flow rate, lower density of nitrous oxide due to high saturated vapor pressures, and a shorter burn time compared to previous engine parameters.

The previous ¾" ball valve would not have been suitable for several reasons. Compared to last year, our volumetric flow rate requirements increased in several areas: nominal burn time *decreased* (8s to 6s), nominal liquid nitrous oxide mass *increased* (23 lbs to 30 lbs), and nominal head *increased* (~750 psi to ~900 psi). This is due to several changed parameters in the thrust profile, including accommodating a higher vehicle dry mass by increasing initial thrust on the rail, improved models of paraffin-nitrous oxide combustion, and higher combustion chamber performance. This means we not only have more liquid nitrous oxide mass, but that it will be less dense (due to the higher equilibrium saturated pressure) and must be discharged in a shorter period.

We can approximate the performance of the ¾ ball valve based on the updated parameters using the coefficient of velocity C_v. The ball valve used in PleaseGoUp had a manufacturer-provided $C_v\ of$ 6.4. The formula for the flow coefficient C_v is given below:

$$C_v = Q\sqrt{\frac{SG}{\Delta P}}$$

Where *SG* is the specific gravity of the fluid and ΔP is the pressure drop we are solving for. Since nitrous oxide is compressible, this is a slight overestimation. Additionally, the expected nitrous oxide volume is solved below. It uses a density $\rho = 5.53\ lbs/gal$, which is the nitrous oxide density at a saturation pressure of 950 psi, and liquid mass $m = 30.65\ lbs$. While the density will increase during the burn as the nitrous oxide cools due to vapor boil-off, any choked flow is unacceptable, especially at the beginning of the burn when thrust is critical for vehicle aerodynamic stability, and so calculating for the initial state is valid.

$$V = \frac{m}{\rho} = \frac{30.65}{5.53} = 5.54\ gal$$

Then, using a burn time $t = 6.06\ s$ to solve for the volumetric flow rate, Q (this value is measured in gallons per minute in the US):

$$Q = \frac{5.54 \cdot 60}{6.06} = 54.85\ gal/min$$

We can find the specific gravity of nitrous oxide at our density value $\rho = 5.53\ \ lbs/gal$, using an accepted value for water of $\rho_{water} = 8.33\ \ lbs/gal$:

$$SG = \frac{5.53}{8.33} = 0.67$$

Now, solving for the pressure drop across the ball valve:

$$6.4 = 54.85\sqrt{\frac{0.67}{\Delta P}}$$

$$\sqrt{\frac{0.67}{\Delta P}} = 0.117$$

$$\frac{\Delta P}{0.67} = 73.45$$

$$\therefore \Delta P = 49.21 \text{ psi}$$

This is a pretty significant pressure drop and will negatively impact engine performance. Thus, a larger ball valve would be required, and it would be a significant engineering challenge to install both a 1" ball valve and lightweight, high-torque actuation mechanism within a 6" airframe.

The second reason for building a pneumatic valve is mass. High-pressure, commercial off-the-shelf ball valves are typically machined from 316 Stainless Steel, which is very heavy and requires electronic or pneumatic actuation systems which further increase system mass. By comparison, the pneumatic valve can be machined of a lightweight alloy, like 7075-T6 Aluminum, and the working gas can be provided from the ground support infrastructure via a quick disconnect.

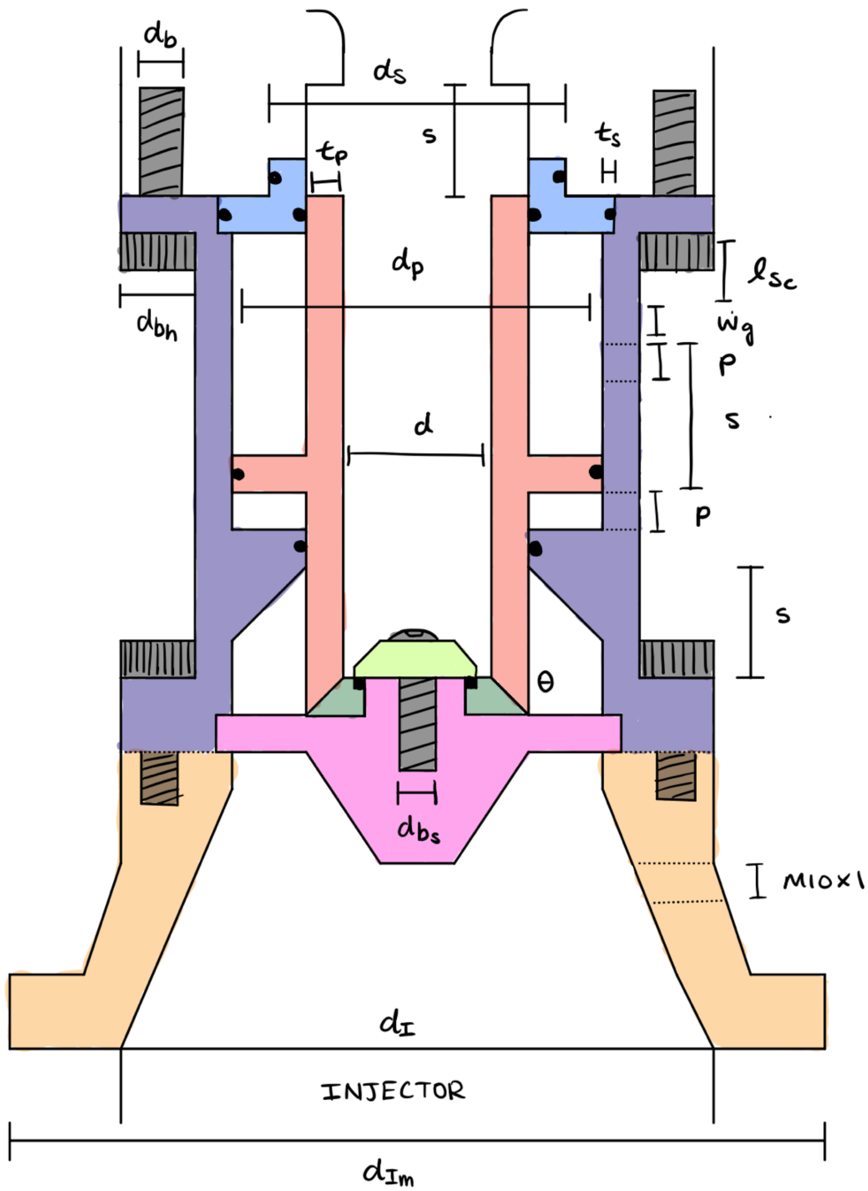
2. Constraining Parameters

Constraints: There are several constraints around the design of the pneumatic valve:
A. Fit within a 6" diameter cylindrical airframe.
B. Interface compactly with both the oxidizer tank and injector manifold.
C. Permit liquid nitrous oxide flow without choking or cavitation.
D. Actuate in a short time frame (< 0.1 seconds) to maximize I_{sp} of Valkyrie Mk 2.

Constraining Dimension: Based on empirical data compiled on nitrous oxide hybrid rocket motors, it was determined that to prevent choking in the plumbing, all valves must have an area at least 3x that of the injector. Thus, we select a 1" ID valve configuration based on injector sizing from Burnell's equations for our target mass flow rate, including some margin for future increases in vehicle performance. All other dimensions are derived from this diameter.

Pneumatic Valve Preliminary Sketch

Key: Piston (Red), Seal (Green), Seal Retainer (Bright Green), Seal Seat (Magenta), Main Body (Purple), Injector Mate (Orange), Integration Ring (Blue), Bolts and Hardware (Black). Spring **not** shown.



Manufacturing Constraints: An additional objective for this valve is designing it such that all components could be machined on the Columbia Makerspace Haas ST-20Y 4-axis lathe. While not insurmountable, this requires some foresight to ensure all designs are easily machinable with in-house tooling.

Regarding the material choice, we plan to use 7075-T6 Aluminum alloy for its superior strength properties compared to 6061-T6, which is the material we use for the oxidizer tank.

Properties	**Aluminum 6061**	**Aluminum 7075**
Density(g/cm^3)	2.7	3.0
Melting Onset (°C)	580	480
Melting Completion (°C)	650	640
Thermal Conductivity (W/m-K)	170	130
Ultimate Tensile Strength (MPa)	310	560
Yield Tensile Strength (Mpa)	270	480
Brinell Hardness	93	150
Elongation at Break (%)	10	7.9
Fatigue Strength (Mpa)	96	160
Shear Strength (Mpa)	210	330

Courtesy of DDPrototype.com

Working Gas: For the pneumatic valve to function correctly, it needs a working gas that is used to open the valve. To minimize costs and ensure repeatable testing, a high-pressure carbon dioxide tank with a pressure regulator can be used. The benefits of using a pressure regulator are that we can fine-tune the actuation pressure depending on results for the static and kinetic O-ring friction. For testing, we can use a high-pressure regulated compressed air supply in the shop.

Closing Procedure: For initial testing, we can simply use a small 1/8 NPT ball valve to purge the ball valve. For cold flows, hot fire testing, and flight we can install a small solenoid that can be purged remotely by our fantastic avionics team.

Stroke Length: One of the most important parameters for the pneumatic valve is the stroke length. The higher the stroke length, the less perturbed the nitrous oxide flow is and the less likely it is that cavitation can occur. However, the stroke length also has a multiplier for the valve length. If the stroke length were twice as long, it would triple the valves length. Thus, finding a median between performance and size is critical.

The stroke length is a direct function of the angle of the PTFE seat. The larger the angle, the longer the stroke length. Since this was a purely experimental valve, it may be best to emulate our friends at Waterloo and use a 45° inclination for the PTFE seat, as they were successful in testing and flying with that value. Future simulations with Ansys Fluent can refine this value. Performing some trigonometry to find the stroke length s:

$$tan\ \frac{\pi}{4} = \frac{s}{d/2} = \frac{s}{0.5}$$

$$s = 0.5\text{ in}$$

Sealing Parameters: In the closed state, a spring is used to provide clamping force on the PTFE seat. We determined the clamping force by referencing a design study from the Marquardt Corporation in 1965, who found in gaseous helium testing that a clamping force of **4 lbf per linear inch of circumference** was sufficient. Thus, for a 1" valve:

$$F_s = 4\pi \approx 12.6\ lbf$$

However, because the team has no experience working with PTFE seats the spring should be able to provide up to 30 lbf and still actuate successfully. During the development cycle, if the clamping force F_s is insufficient, we can modify the housing to provide more force. Using the formula $F_s = -kx$ we can then identify a suitable spring with the stroke length *s*. Because the stroke length is at least 0.5", the free length of the spring must be at least $s + s_0$, plus some tolerance.

Seal Material: The standard for preserving a seal in coaxial pneumatic valves is using PTFE. Waterloo used 25% glass-filled PTFE in their valve, however, they also reported extrusion issues under pressure. I think we should attempt to use 25% glass-filled PTFE and carbon-doped PTFE, which is significantly more rigid, to determine which is more appropriate for our applications.

Piston Dimensions: The piston needs to be capable of containing 1.5x MEOP for the duration of the filling cycle, which is approximately 1,500 PSI. Using a yield strength value of 73,000 PSI for 7075-T6 aluminum, we can use the hoop stress to calculate the wall thickness, *t*.

$$\sigma_h = \frac{P \cdot d/2}{t} = \frac{1500 \cdot 0.5}{t}$$

$$t = 0.01\text{ in}$$

However, this is an extremely small value and is likely not machinable on our equipment. It also doesn't account for the axial stresses the piston experiences. We are constraining the static sealing force, F_s, to be at most 30 lbf, so the piston should be able to withstand these loads without buckling. The buckling of a simply supported cylindrical shell with axial pressures can be partially expressed with Donnell's Shell Theory (pg. 41):

$$N_x = k_x \frac{\pi^2 D}{L^2}$$

Where k_x is an empirical coefficient, *L* is the cylinder length, *t* is the wall thickness, and *r* is the radius. *D* is defined as the wall flexural stiffness per unit width:

$$D = \frac{Et^3}{12(1-\upsilon^2)}$$

Where *E* is the Young's Modulus and υ is Poisson's Ratio. An additional parameter, γZ, is also required to define k_x:

$$\gamma = 1 - 0.901\left(1 - e^{-\frac{1}{16}\sqrt{\frac{r}{t}}}\right)$$

$$Z = \frac{L^2}{rt}\sqrt{1-\upsilon^2}$$

A quick calculation shows for any value $\gamma Z > 2.85$, thus we can use an approximation for k_x:

$$k_x = \frac{4\sqrt{3}}{\pi^2}\gamma Z$$

Now, we can determine the critical stress σ_x under which the piston will exhibit shell buckling:

$$\sigma_x = \frac{\gamma E t}{r\sqrt{3(1-\upsilon^2)}}$$

Because isolating the thickness from the gamma term seems challenging, I solved it instead using code in Python. The calculation yielded a minimum thickness, derived using a ceiling of $F_s = 90$ lbf (FOS of 3):

$$t = 0.05 \text{ in}$$

If you're interested, I uploaded the code to the team Google Drive folder. For additional safety, I think it's best we double this value to account for any manufacturing imperfections or empirical underestimations, then adjust so it aligns with rod seal requirements in SAE AS4716 (pg. 132 of Parker ORD).

$$t = 0.123 \text{ in}$$

Therefore, the piston OD must be 1.246".

The second critical design feature of the pneumatic piston is the spring flange. This means we need to make a brief segue to spring design before returning here.

Spring Selection: The spring we select is one of the critical features of the pneumatic valve. As discussed previously, the spring needs to provide a static load of up to 30 lbf and have a free length of 0.5", including some tolerance, while also being compact.

I semi-arbitrarily selected this spring. It has sufficient load and free length to provide up to 30 lbf at the PTFE seat while compressing a further 0.5 inches for actuation, while being cheap and commercially available. In the future, once we are confident about the sealing force required to prevent leakage, we can size the spring exactly for that value. However, since this is our first attempt making a custom valve I think it's best to leave some margin for extra sealing force if its proven to be necessary.

Spring Parameters:
- OD: 1.640"
- ID: 1.370"
- Length: 2"
- Rate (k): 46.66 lbf/in
- Max Deflection: 1.325"

[Spring ordering link]

Spring Forces: Calculating the spring forces requires evaluating it at several initial conditions. The first initial condition is the closed state, which is always the minimum compression of the spring during its use. We can calculate this using our minimum closing force, 12.6 lbf, however we should also evaluate it at the maximum closing force, 30 lbf.

$$12.6 = 46.66 \cdot x_m$$

$$x_m = 0.27"$$

At the maximum designed closing force:

$$30 = 46.66 \cdot x_m$$

$$x_m = 0.64"$$

This means that we still have 0.18" of extra spring length post-actuation, which is excellent. Calculating the spring force for the designed case:

$$F_s = 46.66 \cdot (x_m + s) = 46.66 \cdot (0.27 + 0.5)$$

$$F_s = 35.93 \text{ lbs}$$

Major Bore Diameter: The last defining geometry is the major bore diameter, labeled d_p in the drawing. This must be at least 1.640" to accommodate the spring we chose. Since we're using pressure regulated carbon dioxide, we can choose an arbitrarily low PSI for the actuation pressure. The regulator can then increase the pressure to up to 140 PSI. Let's check to get an idea of what a reasonable diameter is by using 50 psi as a baseline. The working area is:

$$A = \pi(d_p/2)^2 - \pi(0.623)^2$$

Then, solving for d_p using the equation $P = F/A$:

$$\frac{35.93}{\pi(d_p/2)^2 - \pi(0.623)^2} = 50$$

$$d_p = 1.57 \text{ in}$$

Since this is less than the OD of our spring, we're in the clear. Let's use the spring to constrain it then and choose the pressure from there. To add tolerance, say we use a 1.678" bore diameter (this is selected from Parker ORD). Then the minimum pressure required to actuate the valve would be:

$$A = \pi(1.678/2)^2 - \pi(0.623)^2 = 0.99\ in^2$$

$$P = \frac{35.93}{0.99}$$

$$P = 36.2\text{ psi}$$

So we only need 36.2 psi of pressure to actuate the valve, **neglecting O-ring friction**, if the bore diameter is 1.678". Now we have all the parameters to design the valve! After the valve is designed, we can calculate a new actuation pressure with O-ring friction and consider the contingency case of 30 lbf of sealing pressure on the PTFE seat.

3. Valve Design

CAD Software: The valve is designed using Fusion 360. Screenshots of the file are displayed below, but the .f3d files are in the Columbia Space Initiative folder

Complete O-Ring Table:

O-Ring Num.	Type	Component	Quantity
130	Static	Integration Ring	1
132	Static	Integration Ring	1
139	Static	Main Body	1
218	Dynamic	IR, MB	2
221	Dynamic	Piston	1

O-Ring Friction: In addition to the spring, the other factor that must be considered when determining the actuation gas pressure is O-ring friction. O-rings experience break-out and running friction, where O-ring breakout friction is the force required to overcome static friction between the O-ring and surface and running friction is the "drag" imposed by O-rings on the actuator. It is empirically established that O-ring breakout friction is typically 3x O-ring running friction, according to the Parker ORD. Therefore, if we calculate the running friction using the Parker ORD handbook it will also be possible to calculate the break-out friction.

An important note about these calculations is that they widely vary as a function of several controllable characteristics: O-ring compound (elastomer), lubrication of the O-rings, and surface roughness. In our case, it will be challenging to have extremely low surface roughness due to limitations in machine precision, but we can experiment with different O-ring compounds and lubricants to see if that reduces breakout and running friction during pneumatic valve testing. Temperature (colder correlates to more friction) and resting time (how long the O-ring had been waiting in that position) also affect the O-ring friction, so as a part of the pre-fire SOP we should cycle the valve several times.

Returning to the initial calculations for the actuation pressure, this is calculated using *max* compression of the spring, so when the valve is fully open. However, the O-rings are only in breakout when the valve is transitioning from closed to open. So, in theory, the actuation gas pressure should only increase by the amount required to overcome *running* friction, assuming the force required to overcome breakout is less than that provided by 36.2 psi of gas.

In this case, there are three dynamic O-ring seals: 2 rod seals (218) and 1 piston seal (221). The rod seals will have a significant pressure gradient and will be different depending on the location. The Parker ORD handbook provides some information to calculate the running friction of an O-ring seal. This friction model has two components: friction due to the rubbing surface, i.e. O-ring compression, and dynamic friction due to hydraulic pressure in the valve. Assuming a nominal firing pressure of 850 psi and actuation pressure of 36 psi for all 3 cases:

Piston Groove	**Rod Groove**
$F_C = f_c \times L_p$	$F_C = f_c \times L_r$
$F_H = f_h \times A_p$	$F_h = f_h \times A_r$
$F = F_C + F_H$	$F = F_c + F_H$

A_p = Projected area of seal for piston groove applications.
A_r = Projected area of seal for rod groove applications.
F = Total seal friction in pounds.
F_C = Total friction due to seal compression.
F_H = Total friction due to hydraulic pressure on the seal.
fc = Friction due to O-ring compression obtained from Figure 5-9.
f_h = Friction due to fluid pressure obtained from Figure 5-10.
L_p = Length of seal rubbing surface in inches for piston groove applications.
L_r = Length of seal rubbing surface in inches for rod groove applications.

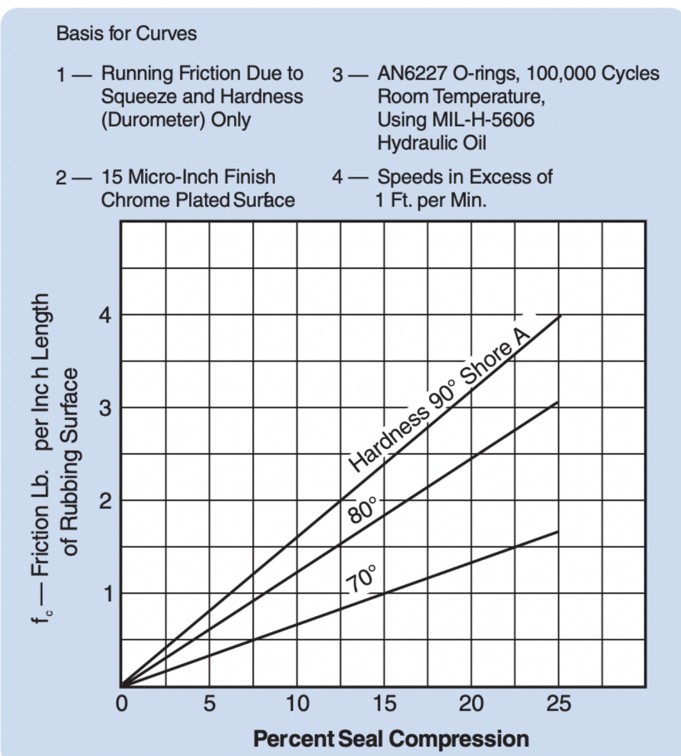


Figure 5-9: Friction Due to O-ring Compression

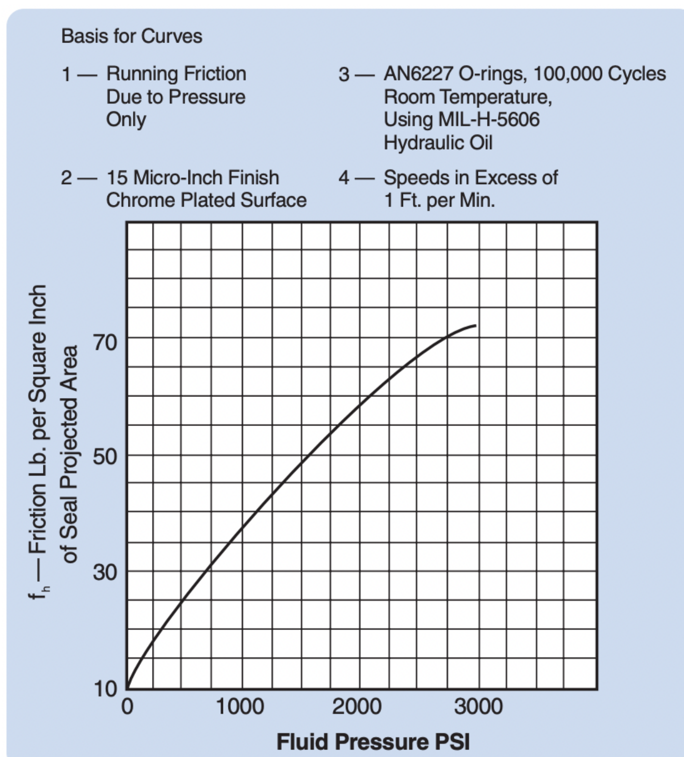


Figure 5-10: Friction Due to Fluid Pressure

Parker-Hannifin ORD Handbook, Section 5-7 pg. 113-114

Dynamic O-Ring 1: Integration Ring Rod Seal (218)

This is the gauge pressure of the tank, roughly:

$$\Delta P = 900\ psi$$

The below values are from the Parker ORD handbook tables 5-1, 5-4, 5-9, and 5-10. I found the seal compression by taking half the difference between the groove diameter and rod and dividing by the O-ring width (0.139”) to give a value of 11.5%. The values are reported for hardness 70 Shore A:

$$f_c = 0.65\ lbf/in$$

$$f_h = 35\ lbf/in^2$$

$$A_r = 0.53\ in^2$$

$$L_r = 3.92\ in$$

Solving for the total friction contributed by this O-ring seal:

$$F_r = F_c + F_h = f_c \cdot L_r + f_h \cdot A_r$$

$$F_r = 21.10\ lbf$$

Dynamic O-Ring 2: Main Body Rod Seal (218)

Same procedure as above:

$$\Delta P = 900 - 36 = 864\ psi$$

$$f_c = 0.65\ lbf/in$$

$$f_h = 33\ lbf/in^2$$

$$A_r = 0.53\ in^2$$

$$L_r = 3.92\ in$$

$$F_r = F_c + F_h = f_c \cdot L_r + f_h \cdot A_r$$

$$F_r = 20.04\ lbf$$

Dynamic O-Ring 3: Piston Seal (221)

Similar procedure, just using adjusted values for piston seals. In this case, the compression is 13%. Based on the results from the other two calculations, we can conclude that the piston pressure will have to be substantially larger. Since these are just approximations, we can calculate the additional pressure only from the two rod seals, who have first-order contributions due to the hydraulic pressure friction, and use that to approximate the piston seal, then finally solve for a target actuation pressure:

$$\Delta P = 36.2 + \frac{21.10 + 20.04}{0.99}$$

$$\Delta P = 77.76\ psi$$

$$f_c = 0.85\ lbf/in$$

$$f_h = 14\ lbf/in^2$$

$$A_p = 0.61\ in^2$$

$$L_p = 5.31\ in$$

$$F_p = F_c + F_h = f_c \cdot L_p + f_h \cdot A_p$$

$$F_p = 13.05\ lbf$$

Therefore, the running friction the valve experiences is:

$$\Sigma F_f = 54.16\ lbf$$

However, the breakout friction is substantially higher. Parker gives a general rule that breakout friction is 3x running friction. This is by no means a hard and fast rule, and we can experiment with different O-ring compounds, like Viton and Buna-N, as well as lubrication to lower this value in the field. We just want to use these calculations to verify that the dimensions of the pneumatic valve will be sufficient for liberal friction models.

$$F_B = 3 \cdot 54.16 + 12.6 = 175.08\ lbf$$

$$P = 176.85\ psi$$

This means we need at least 176.85 psi of gas to actuate the pneumatic valve, which is significantly more than is required to compress the spring.

Pivoting to the closing case, the spring force provided at the maximum spring compression for the designed seal compression of 12.6 lbf is:

$$F_s = 35.93\ \text{lbf}$$

The spring force will not be enough to overcome the friction of the O-rings. However, when the spring is closing the valve, i.e. at the end of a static fire, there will not be a pressure difference across any of the O-rings since the pressure will be purged. So, the only contributions will be from the friction due to seal compression, F_c. Replicating the steps above for only the F_c values:

$$\Sigma F_c = 9.61\ lbf$$

This means that the breakout friction, which is approximately 3 times the running friction of 9.61 lbf, will be 29 lbf, which should be within range. So, aggregating these results for every friction case for the pneumatic valve:

1. Closed to open (firing under pressure):

   The pneumatic valve is sealed, and the spring is providing at least 12.6 lbf of sealing force. The O-rings are at rest. After the igniter lights and has been burning for at least several seconds, we pressurize the pneumatic valve with at least 177 psi of carbon dioxide.

   The carbon dioxide pressure overcomes to the O-ring breakout friction and the piston begins moving. The piston is experiencing 54.16 lbf of running friction, plus additional force from the spring. I wrote a Python simulation file (in our team drive) that solves for the actuation time and velocity at impact with the bottom end cap. It produced the following values:

$$t = 0.0346\ s$$

$$v = 27.81\ in/s = 0.71\ m/s$$

2. Closed to open (testing, no pressure)

   The pneumatic valve is sealed, and the spring is providing at least 12.6 lbf of sealing force. The O-rings are at rest. This time, the running friction is roughly 18.15 lbf and the breakout friction is 54.44 lbf. This requires 55 psi of actuation gas, which is above that required for full spring compression. Solving for the actuation time and velocity:

$$t = 0.0805\ s$$

$$v = 9.81\ in/s = 0.25\ m/s$$

3. Open to closed (post-fire, testing):

   The pneumatic valve is open and pressurized with 177 psi or 55 psi of carbon dioxide. The solenoid is actuated and pressure is purged from the valve. The total running friction is 9.61 lbf and the total breakout friction is 29 lbf. The piston is compressed 0.77”, so the spring is providing 35.93 lbf of return which induces break-out. The piston returns to the seat because the force provided by the spring (> 12.6 lbf) always exceeds the running friction and the seal is restored.