Final Design Report

Team: CP Check Valve – Team 53

Team Members: Jessica Dent, Skylar Tusting, Alec Der Matoian

Team Contact: cpcheckvalve@gmail.com

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Abstract

This report is to serve as a final design review and update to the project sponsors at Zurn Wilkins for the Check Valve Design Senior Project. The senior project team was tasked with improving upon the design of Zurn Wilkins' backflow prevention assembly for small diameter pipes by reducing the pressure loss created by the check valves within the double check backflow assembly. The information contained in this report builds off the information contained in the Critical Design Report (CDR), as well as feedback and further investigation suggested during an Intermediary Design Review. Based on conclusions from the CDR, our team settled on developing one main design, referred to as the double-disk check valve. The goal of this design is to use the mechanical advantage of two actuating half-disk poppets connected to a central hinge to allow for greater cross-sectional flow area during open flow conditions, reducing pressure loss for each check valve.

This report explains the decision behind the double-disk design and includes discussion on design alternatives that were considered. Our team provides analysis of data defending the final design direction, manufacturing plans, material selections, anticipated costs, and the results of iterative testing of various prototypes. Prior to March 18th, 2020, our team anticipated using the remaining months of the project to iterate upon the double-disk design through prototyping and testing at Zurn Wilkins' wet lab facility. Due to the COVID-19 pandemic, the team has needed to modify the scope of the project. The double-disk design was instead finalized using analytical methods. However, due to health restrictions the team was not able to manufacture a model of this proposed design. We were able to work with our sponsor at Zurn to have one final prototype tested to help validate our models. The final design proposed by the team is a culmination of the testing, research, and analysis performed over the course of this project and is intended to serve as a stepping stone for future work in the reduction of pressure loss in the double check backflow assembly for ¼"- 2" diameter systems.

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1.0 Introduction

The purpose of this report is to present the final design for sponsor approval by the senior project team assigned to the Check Valve Design challenge for Zurn Wilkins in Paso Robles, CA. An overview will be presented on topics discussed in the preliminary design review, including an explanation of the objectives, conceptual prototype process, preliminary design, testing plan, and design paths considered. The report includes the results of testing and analysis used in the development of the final design. This document also includes an overview of how the team managed its progress and accountability for the project.

Zurn Wilkins is a manufacturer of water and plumbing solutions, targeting commercial, industrial, and municipal markets. One of their many product groups includes water safety devices, more specifically, backflow prevention assemblies. These devices are placed in series with an existing water supply line that feed anything from single rooms in a house, to entire commercial complexes. When the flow of water is stopped or even reversed, the backflow preventer closes an internal valve that prevents any downstream contamination from traveling upstream. **Figure 1** represents a cross-section view of the internals of a double-check valve.

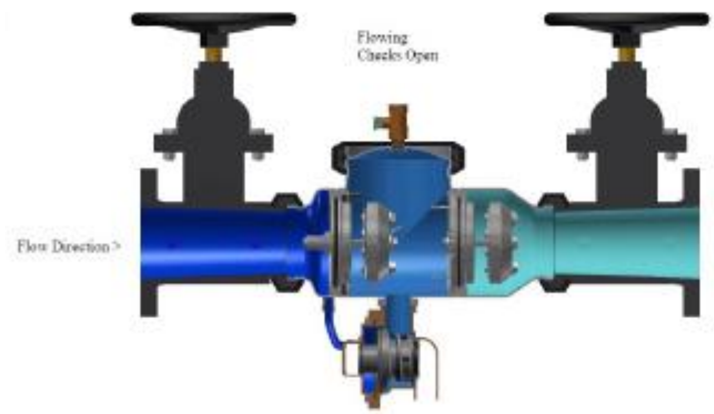

Figure 1. Cross Section of a Double-Check Backflow Preventer

Zurn has proposed a project that aims to design a new type of mechanically actuated backflow prevention system, focusing on the check valve. The new design should improve upon existing Zurn check valve designs by minimizing pressure loss and be scalable to different diameter valve assemblies, ranging from ¼” to 2”.

In addition to offering engineering oversight, Zurn provides the team access to their “wet” testing facility. Qualities such as pressure loss, differential pressure holding, and material integrity (burst/leak pressures) can be tested in the lab. Zurn also provides the opportunity for in-house rapid prototyping via two liquid/laser 3D printers.

The project team assigned to Zurn consists of three mechanical engineering undergraduates. The team assigned to this project participated in three consecutive quarters of senior project mentorship and lab instruction. By June 2020, the team is expected to present all project deliverables to the sponsor and the project.

Near the end of Cal Poly's 2020 winter quarter term, the unfortunate consequences of the COVID-19 outbreak began to affect critical operations of the campus, including learning activities and projects; off-campus travel, on-campus resources, and overall access to labs and facilities were eventually suspended for the remainder of the academic year. Since this project is heavily reliant upon the use experimental test results driving the critical design changes of the next prototype iteration, having no direct access to experimental testing or rapid prototyping severely undercuts this iterative testing cycle.

These impacts have prevented our team from completing this project with the deliverables anticipated at the time of presenting the Critical Design Review (CDR), which includes a fully-functioning prototype that could operate similar to a production model. The team was not able to manufacture a model that includes proper sealing and backflow prevention, which was to be tested in Zurn's facilities.

As a workaround, our team and sponsors at Zurn agreed on modifying the scope of the project. Our team was to use the remaining project time to provide a sound foundation for the use of the double-disk check valve design in backflow prevention assemblies via computational fluid dynamics (CFD) modeling and analytical methods. After working with our sponsor to complete one final prototype, the test data proved that the double-disk is capable of reducing pressure loss. Our team is confident that the project can act as a foundation for Zurn Wilkins engineers to integrate the double-disk check valve into a fully-functioning product intended for water supply lines.

2.0 Background

Any project requires sufficient knowledge of the system/environment that is involves the product or service to be provided. Multiple sources were used to research background information, patent history, and current industry device designs.

2.1 Existing Designs and Patent Research

A check valve is a valve that allows fluid flow in only one direction. There are a wide variety of check valves used in various applications, each with unique performance characteristics. Listed below are a few of the existing types of check valves and a brief description of their operation, applications, performance, and limitations.

2.1.1 Swing Check Valve

A swing check valve contains a disc that swings on a hinge or shaft. A cross-section of two styles of swing checks is shown in **Figure 2**. The disc swings off the valve seat to allow forward flow. When the flow is stopped, the disc swings back onto the seat to block reverse flow. Often, a lever/weight or a lever/spring combination is mounted to the disk to achieve improved performance. Although swing check valves come in various sizes, they are typically used in larger diameter lines. A common issue for swing check valves is water hammer. It can occur when the disk closes rapidly and abruptly stops the flow. This causes a surge in pressure that can result in high velocity shock waves and place a large stress on the piping in the system. [Perry's]

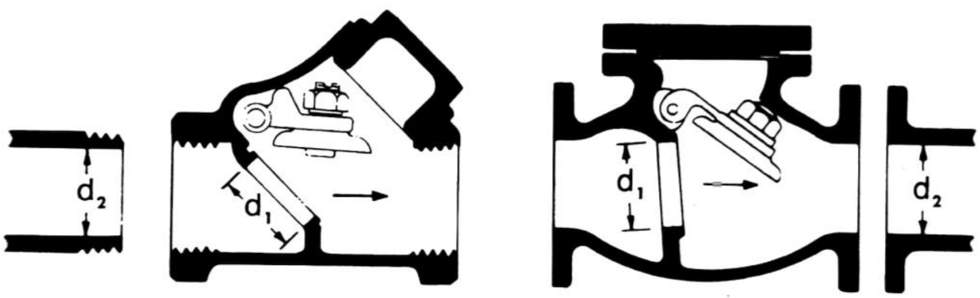

Figure *2*. Swing Check Valves (Crane)

2.1.2 Tilting Disk Check Valve

A tilting disk check valve is very similar to a swing check valve but differs in that the pivot is in the middle of the gate. A cross-section view is shown in **Figure 3**. These valves may be installed in a horizontal line or in lines in which the flow is vertically upward. Compared with swing check valves of the same size the pressure drop in tilting disc valves is less at low velocities but greater at high velocities. These valves can close quickly at the instant the flow reverses. [Perry's]

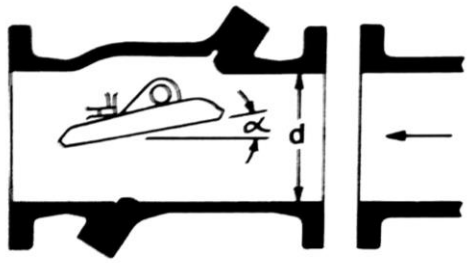

Figure 3. Tilting Disc Check Valve (Crane)

2.1.3 Dual Disk Check Valve

A dual plate or dual disk check valve has two halves of a disk that fold at the center around a common pivot or shaft. The two half plates rest on the valve seat when in the closed position. A torsion spring at the pivot point helps maintain closure when upstream pressure is lacking. The pressure loss is greatly reduced because the disc folds into a more streamlined profile thus reducing the drag, as can be seen in **Figure 4**. If the pressure is not high enough the valve may not fully open and have a larger pressure loss as compared to other valves. In addition to a rise in energy consumption, insufficient flow velocity can wear the valve prematurely. This can lead to issues with proper sealing, especially when used in vertical orientations were additional spring force is necessary to seal the valve against gravity. [Sotoodeh] They are also sometimes known as butterfly or wafer check valves. [2016 ASHRAE]

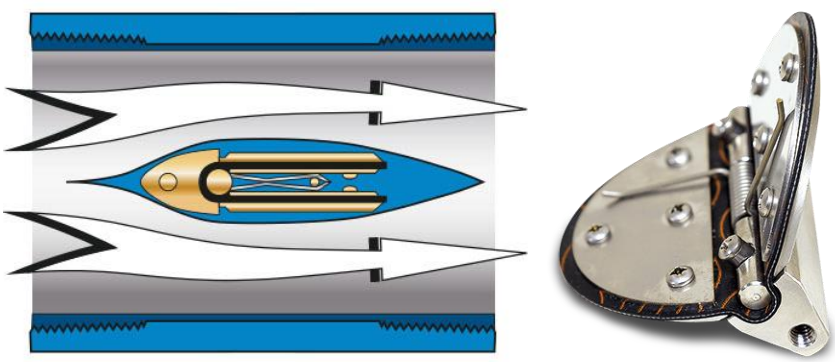

Figure 4. Dual Disk Check Valve (US Valve)

2.1.4 Lift Check Valve

Lift check valves have a body design like a globe or angle valve body with a similar disk seating. A cross-section view of two styles of lift valves can be seen in **Figure 5**. The guided valve disk is forced open by the flow and closes when flow reverses. Because of the body design, the pressure drop is higher than that of a swing check valve. Lift check valves are recommended for gas, compressed air, or in fluid systems not having critical pressure drops. [2016 ASHRAE]

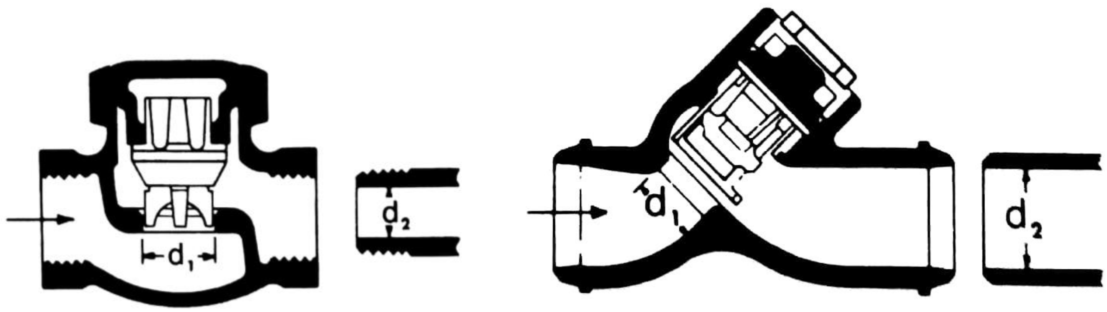

Figure 5. Lift Check Valves

2.1.5 Inline Spring Check Valve

Inline spring-loaded check valves are common and have a fairly simple design. When flow enters the inlet port of the valve, it must have enough pressure to overcome the cracking pressure and the force of the spring. Once overcome, it pushes the disk open and allows fluid to flow through the valve, as shown by **Figure 6**. When the pressure is no longer high enough, or there is a backpressure, the spring compresses the disc against the seal and shuts the valve. The spring and the short travel distance allow for quick reclosing time when the pressure is not sufficient. This design also helps avoid pressure surges in the line and thus prevents water hammer. They can be installed in the vertical or horizontal positions. They typically have poor pressure loss performance since the flow must overcome the force of the spring and the disk remains in the flow path.

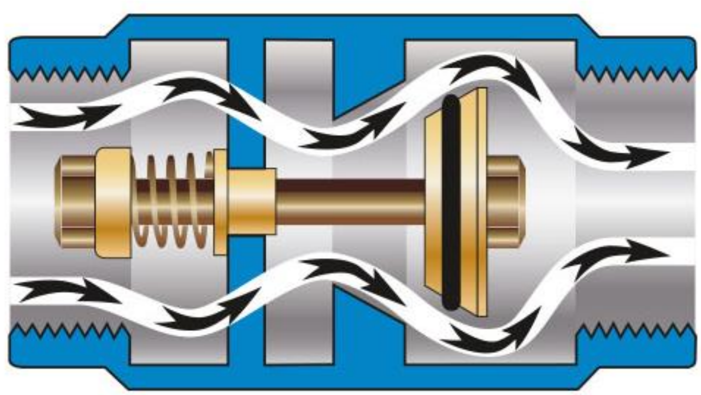

Figure 6. Inline Spring Check Valve (US Valve)

2.1.6 Backflow Direct Dual-Action Check™

The Dual-Action Check™ is a compound movement check valve produced by Backflow Direct. The stroke of the valve can be separated into two phases: First, for the initial 25% of the valve's stroke, linear separation occurs between the valve and the valve seat. The initial movement of the valve is very beneficial for consistent sealing and the holding of pressure differentials. Then, for the remainder of the stroke, the valve continues to rotate until its effective aspect ratio relative to the flow of water is reduced. This compound movement allows for the valve to create less pressure drop as full-flow is achieved, and for the valve to effectively distance itself away more than a simple in-line check valve

could do. **Figure 7** illustrates a side view of the check valve assembly, with the valve entering the rotation phase of the stroke. (US 8,875,733)

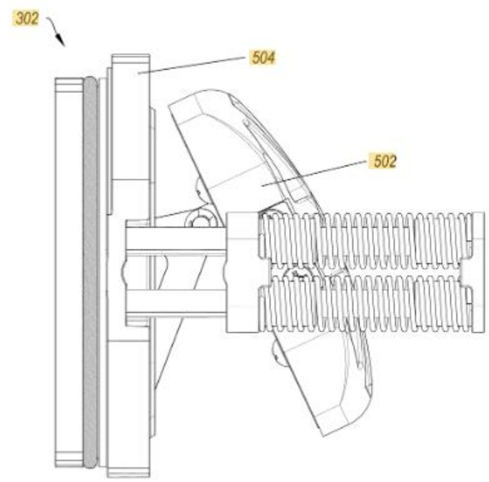

FIG. 6C

Figure 7. Dual Action Check Valve, Adapted from US 8,875,733, Fig. 6C

2.2 Patent Research

A list of relevant patents is listed in **Appendix A – Patent Table**; however, there are two of these designs that we find most interesting with regards to the design challenge.

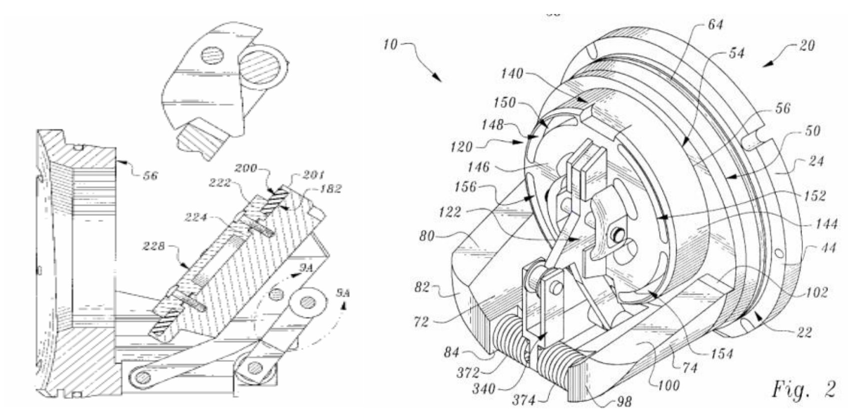

Figure 8. Variable Opening Force Check Valve.

The check valve having a variable opening-force threshold (US 6648013 B1), shown in **Figure 8**, is noteworthy because the design reduces the amount of force necessary to open the valve as the fluid flow rate increases. This reduction in pressure loss is of interest to our team. [Ray, Ernest B.]

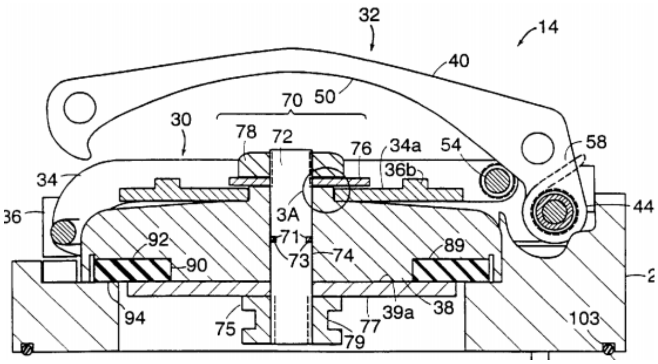

Figure 9. Flapper Check Valve patent design.

The flapper check valve (6050293), shown in **Figure 9**, is of interest to the team because it uses the mechanical advantage of lever arms to hold the valve in a closed position. Utilizing a lever arm is a potential solution to our design problem. [Lin, Ping, and Rand Ackroyd]

2.3 List of Applicable Industry Codes, Standards, and Regulations

There are many standards, industry codes and regulations surrounding valves, backflow prevention assemblies, and check valves. We have listed some of the most relevant below. See **Appendix B – Applicable Industry Codes, Standards, & Regulations** for a more detailed explanation of each standard. **Appendix C – Glossary** also provides a glossary for common technical terms used in the field of backflow prevention and pipe flow.

- ASME B16.34
- ASSE 1015
- CSA B64.5
- AWWA C510-17
- Cal OSHA Title 8, Subchapter 7, Group 2, Article 9, §3363(h)
- USC Foundation for Cross-Connection Control and Hydraulic Research Manual of Cross Connection Control, Tenth Edition

3.0 Objectives

Due to the conditions and changes of plans caused by COVID-19, the objectives of this project have changed substantially. These changed include how much improvement to the current design was forecasted from CDR to the end of the project, and which engineering specifications were quantitatively met with the latest-developed prototype.

3.1 Problem Statement

Zurn Wilkins, a plumbing parts manufacturing company, is requesting a new check valve design that uses mechanical advantage and fluid dynamic principles to reduce pressure loss comparative to their existing products. This design must meet industry standards for water supply backflow prevention.

3.2 Boundary Diagram

The boundary diagram for this project can be seen in **Figure 10**. Our team's work for this project will remain within the boundary of the valve housing in Zurn's current product lines. We will need to consider the design's interface with the check valve enclosure (housing), the test plugs, and the sealing surfaces connecting the check valve to the isolation ball valves on either end.

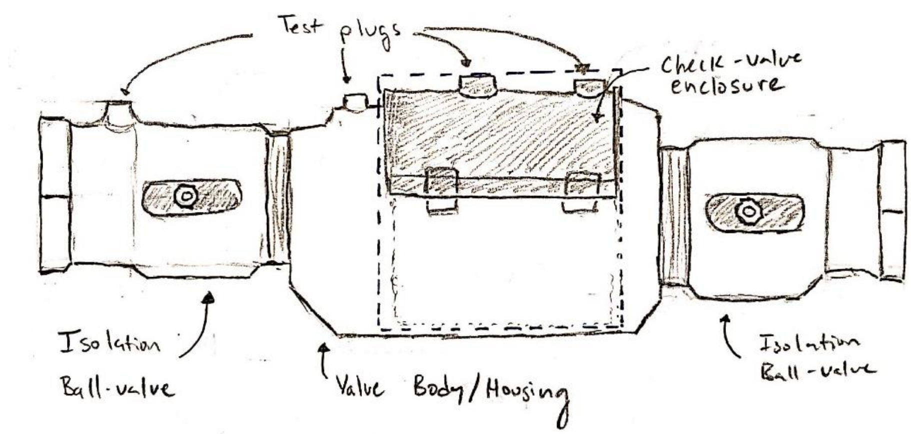

Figure 10. Boundary Diagram Sketch

3.3 Customer Needs

The main customer for this project is Zurn Wilkins. The major customer needs fall within the category of improving performance. The customer needs are as follows:

- **Reduce the pressure loss:** The current product has pressure loss due to the inline disc disrupting the flow path
- **Maintain a static pressure differential:** A minimum pressure differential between the inflow and outflow of the check valve. This requirement is crucial to the functionality of the check valve to prevent backflow.
- **Mechanically driven:** The check valve must open and close using only mechanical means.
- **Meets industry requirements:** This includes flow, pressure, and safety regulations for backflow prevention devices.

    - **Water compliant materials:** The design must be made of materials that will not rust or corrode.

3.4 Customer Wants

Customer wants are design criteria that are important to take into consideration and would improve the quality of the design. Customer wants are important for the function of the design and are considered lower priority than customer needs. The customer wants are listed below:

- **Adaptable design:** The design should be able to fit within backflow systems between 3/4" and 2" in diameter. The design should also be able to scale to the various standard pipe diameters within the ¾" to 2" range while maintaining proper functionality.
- **Manufacturability:** The ease of manufacturing the design should be taken into consideration throughout the development process. Designs that are simple and utilize conventional manufacturing techniques, such as injection molding, are preferred.
    - **Standard tooling:** The design dimensions should follow US Customary unit standard sizes and be manufactured using standing tooling.
- **Horizontal or vertical position:** The valve must function properly if placed either horizontally or vertically. The closing mechanism should not be significantly affected by the direction of gravity.

3.5 Design Considerations

The following are factors to be considered during the design process, but not required for a successful product.

- **Compatible with Zurn's current design:** The user should be able to swap out the new design for the existing one without making alterations to the existing valve housing or connecting surfaces.
- **Reduced complexity:** The new design should aim for simplicity. A mechanism with less parts has less potential to break and is easier to maintain.
- **Cost comparable:** Our team's design should aim to be comparable in costs to Zurn's current design. This means no exotic materials or uncommon manufacturing processes should be used.

3.6 Customer Needs & Wants Summary

The customer needs, wants, and design considerations are summarized in **Table 1**. Many of the listed items have some interdependence. The customer needs will be prioritized for the final design.

Table 1. Summary of Customer Wants, Needs, and Design Considerations

Customer Needs	Customer Wants	Design Considerations
Reduced pressure loss	Adaptable design	Compatible with Zurn's current design
Maintains static pressure differential	Designed for manufacturability	Reduced complexity
Mechanically driven	Adapt design of DC to RP setup	Cost comparable
Meets industry standard (As listed in Section 2.3)		

To empirically test that the Customer Needs are achieved, the project team will be using various test bench setups and "wet" testing lines housed within Zurn's Paso Robles location.

3.7 Quality Function Deployment Process

To better define the problem being addressed in this project, our team used a Quality Function Deployment (QFD) process. The QFD method is used to translate the customer needs, wants, and thoughts into engineering specifications which can be measured and evaluated. Our team utilized a QFD tool called the House of Quality which can be seen in **Appendix D – House of Quality.**

From our QFD process, we have found engineering specifications to meet each customer need. The team was able to match the customer wants and needs either directly or indirectly through the specifications listed in **Table 2**.

Table 2. Engineering Specifications Table

Specification No.	Engineering Specification	Description	Requirement or Target	Risk	Compliance
1.	Pressure Loss	< Current Product	< 5 PSI. loss	High	Test, Analysis
2.	Size	Fits within current housing	+-0.005 in	Medium	Inspect
3.	Maximum Allowable Water Pressure (MAWP)	175 psi	Min.	Low	Test
4.	Maximum Allowable Working Temperature (MAWT)	180°F	Min.	Low	Test
5.	Static Pressure Differential	7 psi /valve (RP) 2 psi/valve (DC)	Min.	High	Test
6.	Assembly Time	≤ current product	Max.	Medium	Inspect
7.	Cracking Pressure	= to Current Products	Min.	Medium	Test

The engineering specifications have associated risks, from low to high. High risk specifications are considered the most challenging for the team to complete. Reducing the pressure loss and holding the static differential requirement is critical for the design to be successful. In the compliance category, the method of verifying each specification is listed. “Test” means formal testing will be done, likely at Zurn’s facility. “Inspect” will be a go/no-go compliance check. “Analysis” means the specification will be investigated through computer programs or studies such as FEA and CFD.

The engineering specification for pressure loss refers to the change in pressure across the valve due to friction losses, measured in pounds of pressure difference between the inlet and outlet of the valve. The current Zurn design has a pressure loss of 5 PSI per valve, so to have a successful design our valve must have less than this value. The size specification of the valve is intended to keep the new design within reasonable size constraints to allow for an easy transition to the new valve. The maximum allowable water pressure and water temperatures (MAWP and MAWT) are the values of the extreme maximum conditions the valve is expected to fully operate before failure. The valve our team designs should be able to meet these specifications to be competitive with the current design and to meet industry standards. The assembly time specification is intended to keep the valve economically competitive and

of lower priority than many of the other specifications. The cracking pressure specification refers to the amount of pressure differential needed to change the valve from its normally closed state to the open state. This is an important value for piping system designers and needs to be equal to the cracking pressure required for Zurn's current in-line spring check valve.

The results of the QFD predict that the most important characteristics of the project are as follows:

- Maintain Desired Cracking Pressure (@ 18% Relative Weight [RW] of QFD)
- Minimize # of components (@ 14% RW)
- Reduce Pressure Loss (@ 13% RW)
- Reduce Assembly Time (@ 14% RW)

It should be noted that after further discussion with Mr. Yale, the objectives of reduced assembly time and component count/complexity, that appear to fall under the category of "Design for Manufacturability (DFM)," are considered a want more than a need. Therefore, the team will still target the maintenance of desired cracking pressures for the Double Check configuration as a priority need, as well as reducing the pressure loss of the new design.

4.0 Concept Development

This section serves as an overview of the ideation, concept prototyping, and design selection progress. In this section, the project team provides a list of the ideation methods used, along with examples of some of the concept sketches and prototypes developed. Idea refinement and selection was performed using a design matrix, and preliminary rough CAD models were produced to begin exploring some of the selected design elements. After the Preliminary Design Review the team decided to pursue a two different concept designs to test and compare the effectiveness of each model. These two design paths and their performance will be discussed.

4.1 Ideation Processes

The following section describes the several ideation methods the team used to generate large amounts of simple, isolated concepts that could later be developed or combined. By the end of the ideation period of the design cycle, decision-making tools such as a weighted decision matrix were used to select the most viable designs for further refinement.

4.1.1 Functional Decomposition, Brainstorming, & Brainwriting

To make use of the goals developed during the Scope of Work in terms of check valve operation, our team used functional decomposition sessions to break down the goals of the project into manageable aspects that were targeted individually. The broken-down characteristics included minimizing activation force, changing the entry/exit shape, and reducing flow resistance.

After functional decomposition was completed, a series of brainstorming and brainwriting sessions were conducted to further develop the design challenges identified. These exercises included the production of sketches that attempt to solve the design challenges listed.

Since the team consists of 3 members, the commonly-used "Brainwriting 6-3-5" method, meaning 6 members, producing 3 ideas each, in 5 minutes, was adapted to 3-3-5. Each Brainwriting table took a main function from the functional decomposition results as the problem statement. Then, each member provided ideas on how to solve or improve such issues. After several Brainwriting sessions were conducted, some of the recurring ideas or concepts were used in formulating the design options listed within our design matrix, detailed in **Table 6.**

In **Table 3**, team member brainwriting results are provided for attempting to "Reduce Flow Resistance." Pressure loss reduction is important to the overall success of the valve design since pressure loss is considered by the industry to be a critical performance indicator.

One recurring idea involved using an aperture-style, or origami-folding valve that would be able to completely clear the flow path of the water passing through the assembly. Since most conventional designs involve a large object or assembly of parts that obstruct water flow, having a valve that removes itself completely from the flow path, much like a gate valve, would produce low pressure loss.

Table 3. Flow Resistance Brainwriting Results

Problem: How to Reduce Flow Resistance?			
Member:	Idea 1:	Idea 2:	Idea 3:
Jess	Conical Valve	Gate Valving	Aperture Valve
Skylar	Smooth contouring	Origami Valve	Laminar flow production
Alec	Smooth material	Channeling flow	Use of internal airfoils

In **Table 4**, team member brainwriting results are provided for attempting to "Minimize the force required for open-valve flow." All energy used to maintain the open state of the valve after cracking pressure has been exceeded is considered wasted energy. Thus, allowing the water to maintain as much energy as possible during open flow conditions is an important goal to meet. Here, a compound or multi-link spring system proved popular. The concept would involve using multiple springs to allow for varying spring forces at different times during the valve stroke. That is to say, the valve can experience a multitude of spring force constants as distance traveled changes. This concept was further refined to be operable in a translational, or rotational nature. Further visualization of these two design paths is presented in **Figure 19** and **Figure 20**.

Table 4. Flow Force Brainwriting Results

Problem: How to Minimize Open Flow Force?			
Member:	Idea 1:	Idea 2:	Idea 3:
Jess	Self-closing orifices	Rail-slider	Multi-link pivot
Skylar	Ratcheting system	Use gravity	Mechanical Advantage
Alec	Locking pins	Compound spring	Torsional latch

In **Table 5**, team member brainwriting results are provided for attempting to "Change the Entry/Exit Shape of the Valve Body." This functional decomposition result was thought to be important since reducing the cross-sectional area that the water flow "sees" in its flow path translates to greater flow capacity and lower pressure drop. The brainwriting results pointed to designs that involved either "flipping" the valve via a system of rails & guide channels, or a multi-face valve, such as a double-disk valve or butterfly valve.

Table 5. Entry/Exit Shape Brainwriting Results

Problem: How to Change Entry/Exit Shape?			
Member:	Idea 1:	Idea 2:	Idea 3:
Jess	Aperture design	Rotating Valve seat	Origami Valve
Skylar	double-flap	Origami Valve	Overlapping valve flaps
Alec	Non-constant orifice sizes	Compound Movement Valve	Flexible housing

4.1.2 Concept Prototyping Session

The figures presented in this section are the result of a concept design session using craft and low-cost materials. The purpose of this exercise was to take the large number of design ideas and generations from the ideation sessions described in **Section 4.1.1** and produce low-resolution concept models that could allow for better visualization and description of a certain idea. These models serve as representations of isolated functions of what the final design might entail.

4.2 Initial Concept Sketches

From our initial ideation sessions, we made more detailed sketches of our top ideas. The top ideas the team selected to detail out were the Folding Aperture, Modified Double-Disk, Sliding Rail, and the Counter Weighted Lever Swing. These designs were selected because they either will reduce the drag of the closing mechanism or they will take less force to hold open than the current Zurn design. These designs all have significantly different forms and their effectiveness of their ability to reduce pressure loss is best determined in real test conditions.

4.2.1 Aperture Check Valve

The inspiration for this valve, shown in **Figure 11**, was a camera aperture mechanism. The primary benefit of this design is that when it is in the full-open position there is no obstruction of flow. The valve would have losses close to that of an equivalent length of pipe, assuming the entrance and exit regions of the valve line up with the internal diameter of the adjoined piping. The difficulty of this design is determining how to have the valve actuate from a closed to an open position based on pressure and flow direction using mechanical elements. Another main concern is that sealing with multiple elements could be difficult. Through prototyping we found that finding a material that properly seals while allowing the sliding of the aperture elements is difficult to find. Similar ideas we considered involved folding mechanisms akin to origami. Upon further investigation, we decided most of these were variations of the aperture or double-disk designs.

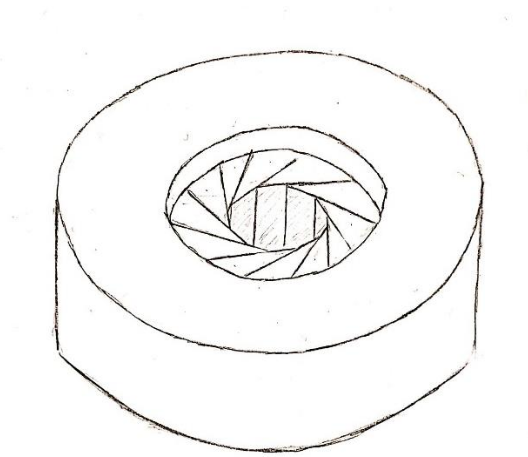

Figure 11. Sketch of aperture-inspired check valve design

4.2.2 Modified Double-disk Check Valve

This design is based on the existing double-disk check valve design. The details of operation and functionality of this valve were previously described in **Section 2.3.3** and are shown in **Figure 12**. The primary difference in this design as compared to a standard double-disk wafer check valve is that it could utilize a compound spring mechanism. The compound spring mechanism would allow the holding pressure of the valve to be reduced as the valve opens more. Zurn has an inline check valve that utilizes a set of rollers and spring bar to create a compound spring element. An adaption of this existing design could be used on a double-disk check valve combining the positive aspects of each design

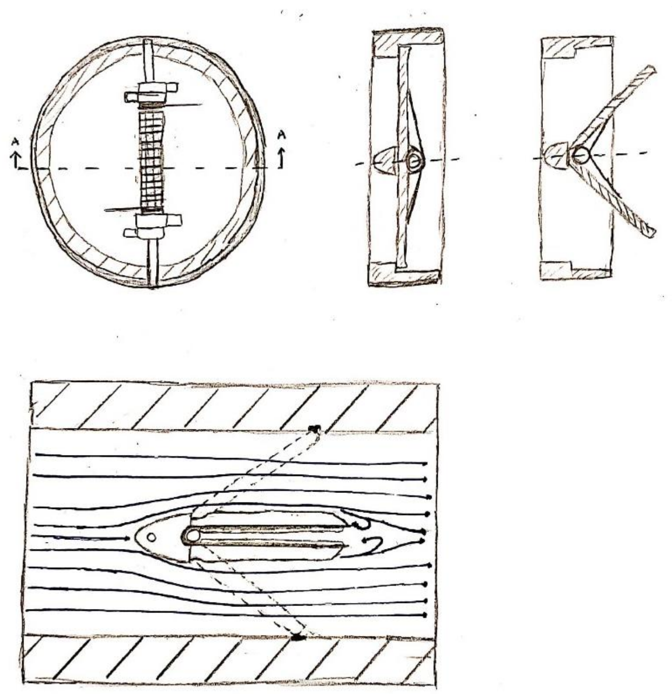

Figure 12. Concept sketch of double-disk design.

4.2.3 Sliding Rail Check Valve

This design involves having the poppet mounted on rails that would slide and rotate in and out of the closed and open position. When the valve is in the fully open position the poppet is completely parallel to the flow direction thus minimizing the obstruction of the flow and reducing pressure loss. **Figure 13** shows the valve as it would rotate from the closed position (vertical to flow) to the open position (horizontal to flow).

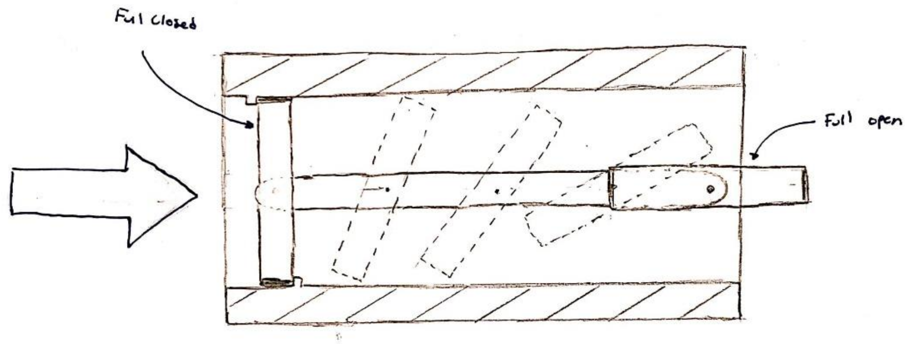

Figure 13. Sliding Rail Sketch

4.2.4 Counter Weighted Lever Swing Check Valve

This design is based on a typical swing check valve with the addition of a lever arm with a weight. The lever arm and weight help offset the center of gravity of the swing assembly. By pushing the center of gravity further away from the point of rotation we take advantage of the lever arm. As the valve opens the weight and thus the center of gravity shifts closer to the point of rotation reducing the amount of torque. This, in turn, reduces the amount of force required to hold the valve open and reducing pressure loss. **Figure 14** shows this aspect of the design.

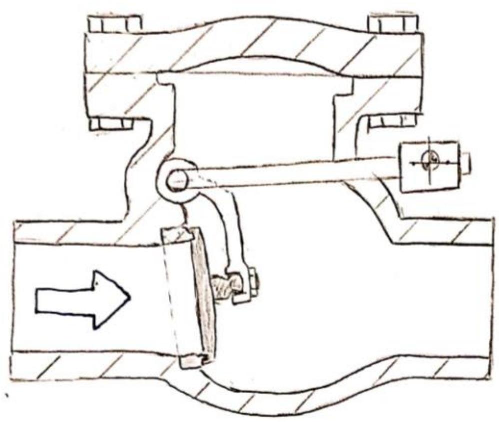

Figure 14. Counter-Weight Swing Check Sketch

4.3 Concept Selection & Weighted Decision Matrix

The top five selected concept designs were chosen because they each represented a major category of each of the ideated concepts. Each operates differently and can be refined through future iterations. The benefits and drawbacks of the selected designs are considered concerning to Zurn's current 1.5" diameter in-line spring check valve. These top designs were also chosen because we think they might be most effective at reducing pressure loss, either by introducing a non-linear opening mechanism or by reducing the cross-sectional area of the valve when in an open state. The top designs were then placed in a Decision Matrix seen in **Table 6.** The matrix rates the designs according to criteria set by customer needs and wants.

Table 6. Decision Matrix comparing top conceptual designs.

Criteria	Weight	Design Option									
		Aperture		Double-Disk Mod		Nonlinear Link		Rail Slider		Lever Swing	
		Score	Total	Score	Total	Score	Total	Score	Total	Score	Total
Minimize Open-State Force	4	3	12	3	12	5	20	3	12	5	20
Minimize Component #	2	1	2	4	8	2	4	3	6	3	6
Reduce Pressure Loss	5	5	25	4	20	3	15	3	15	4	20
Reduce Assembly Time	2	1	2	3	6	1	2	3	6	2	4
Ease of Design Scalability	3	3	9	4	12	3	9	3	9	2	6
TOTAL:		50		58		50		48		56	

The outcome of the matrix ranked the designs as follows: double-disk adaptations and counterweighted swing check as most likely to meet the customer needs. These were followed by the folding aperture, Non-linear linkage, and sliding rail. The double-disk and counterweighted check valve designs are ranked best is because they are expected to reduce pressure loss more than other designs since they are similar to patents and conventional designs. Another key criterion ranks how well each design might minimize the force required to hold the valve in its open state. This favored the designs that utilized mechanical advantage. One criterion that was not explicitly considered in **Table 6** is the consistency of sealing for each valve. This can be speculated however the team believes this will be best understood through reliability testing for each design. The designs that were chosen for modeling are the double-disk adaptation, the non-linear linkage, and the folding/aperture design. These designs were chosen for prototyping because we wanted to better understand how they would function.

4.4 Concept CAD & Preliminary Calculations

After deciding upon the top designs, the team made rough CAD models. Three of the designs are shown in **Figure 15.** The team also did some preliminary calculations to determine the static loading on the double-check to compare the design to Zurn's inline spring valve design.

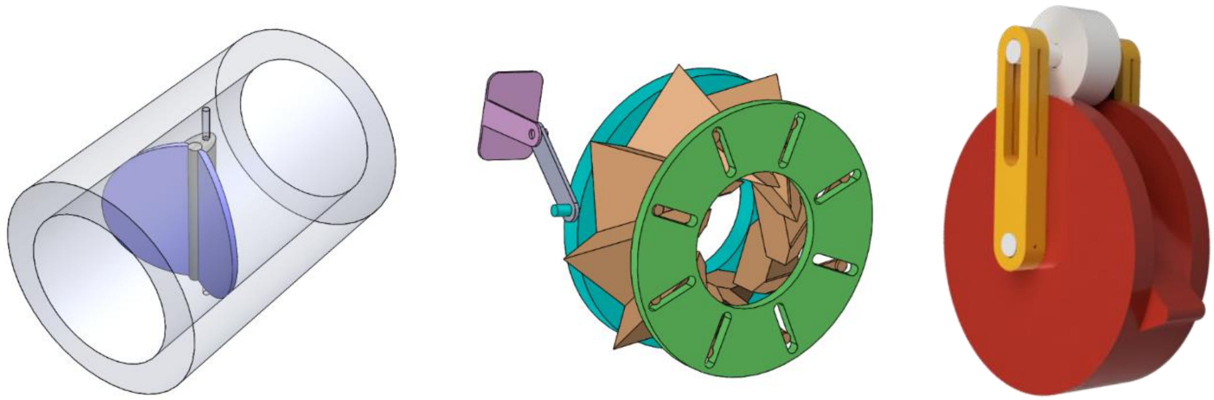

Figure 15. Isometric views of the top concept designs.

4.4.1 Modified Double-Disk

The concept CAD model for the double-disk adaptation is shown in detail in **Figure 16.** The benefit of this design is that it nearly eliminates all obstruction of flow when it is open. Another benefit is the valve's symmetric nature, allowing for streamlined manufacturing and component design over non-symmetrical design. The valve requires only one degree of freedom, rotation about the axis of the linkage pin, reducing the overall mechanical complexity and risk of premature mechanical failure. One of our adaptations is to start the disks at a steeper angle than is conventional to decrease the amount of travel necessary to reach a fully open state. The diameter of the disks, when spread out, is larger than the internal diameter of the pipe, ideally minimizing sealing issues. This was tested at a later phase of the design process. The purple disks and gray central mount would be made of conventional plastic and the pin through it all would be a corrosion-resistant metal. This design would meet all customer requirements if standard materials are used and the proper spring rates are selected. The double-disk design is known in industry to have lower pressure losses than an in-line spring design.

To meet the closing pressure for the double-disk the spring, each half-disk for this cross-sectional flow area would require a spring rate of *k =2.22lb/in.* In comparison, Zurn's current in-line check has a spring force of *k = 4.43 lb/in*. An advantage of the double-disk is that each spring would be smaller and relatively more flexible, allowing it to be placed in locations that minimize the blockage of the flow. As noted by Mr. Corral in a conversation, the main drawback of the double-disk is that it has a higher potential for incomplete seating, and which leads to issues with sealing during backflow conditions.

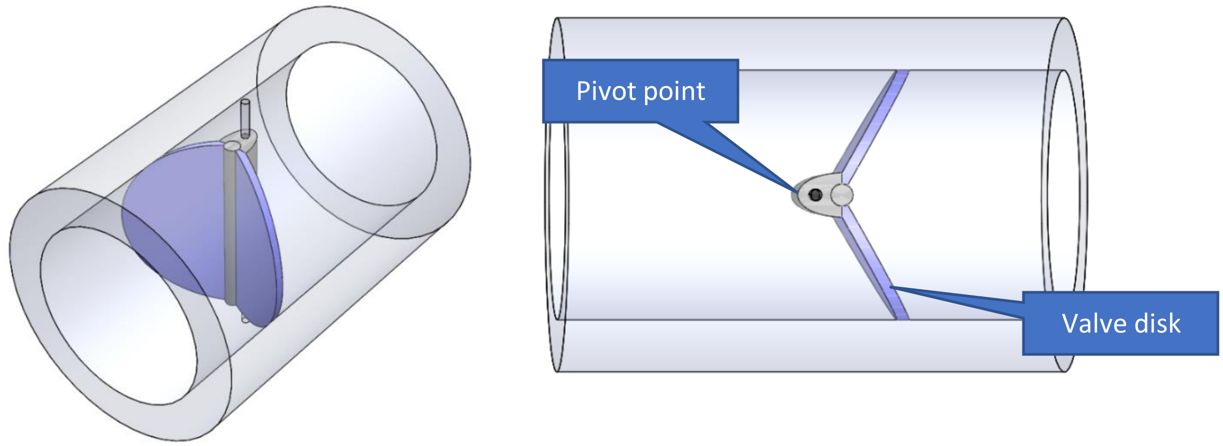

Figure 16. Adaptation of the double-disk design CAD.

4.4.2 Aperture Style Design

The next design was inspired by the way that rotation moves the pieces of the camera aperture away from the closed position. The aperture-style check valve can be seen in **Figure 17**. The benefit to this design is that it could fully remove any obstruction to the flow path. However, it is also complicated and there is more room for issues to arise with the high number of parts. The valve would meet all engineering specifications and customer requirements, especially those regarding pressure loss. The only concern with meeting specifications is that the assembly time would be high and the mechanism to regulate the cracking pressure is not fully defined yet.

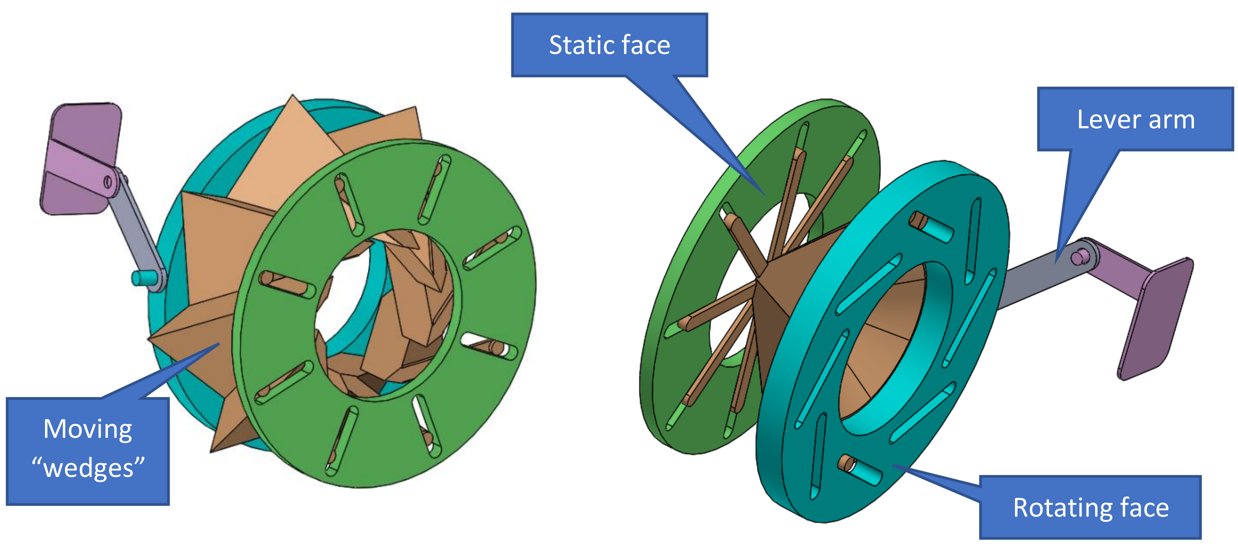

Figure 17. Camera aperture-inspired design CAD.

The valve rotates the orange "wedges" along the slots by turning the blue disk. The mechanism for closing the blue slotted disk is not yet fully defined but is expected to utilize a lever arm that is pushed open there is enough pressure differential to reach the desired cracking pressure. There would likely be a torsional spring to apply the closing force on the valve.

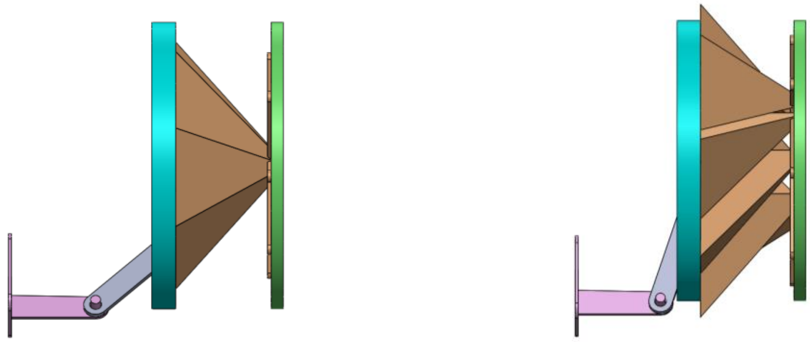

Figure 18. Visual of the aperture valve in closed and open positions.

The valve is shown in **Figure 18**. In the open (right) and closed (left) positions. The wedges would likely be made of a plastic with the contacting sides being coated or covered in a low-friction substance. The number of wedges and shape of the moving pieces will need to be refined through testing if this design is selected. Potential disadvantages of the moving "wedges" are custom manufacturing and the introduction of friction between the sliding pieces.

<u>4.4.3 Non-Linear Linkage Designs</u>

A third major design consideration is the use of a non-linear spring force linkage. These designs are specifcally made with the intent to reduce the amount of force required to hold the valve in the open position after the required cracking pressure has been met. This should reduce the amount of energy lost to holding the valve open. The non-linear design meets the required customer specifications and would specifically excel at the criterion of minimizing the force needed to hold the valve open. This is achieved using non-linear mechanisms such as the dual spring assembly concept shown in **Figure 19.**

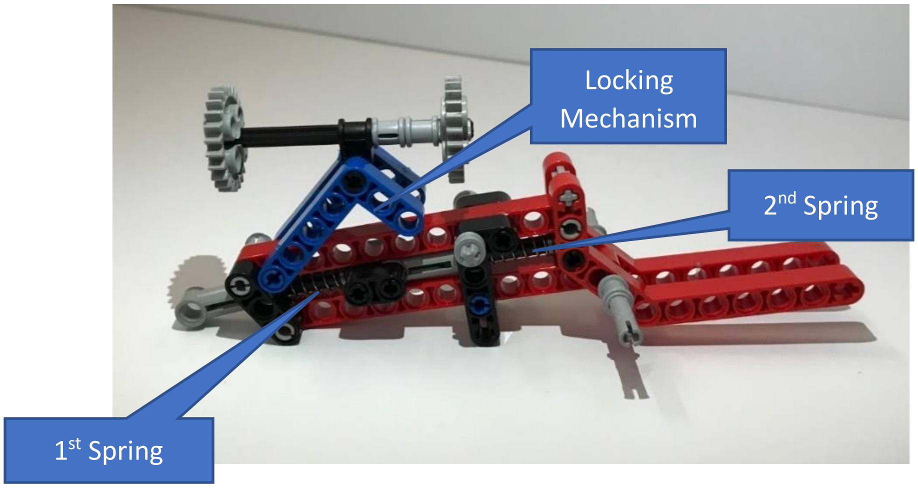

Figure 19. Non-linear linkage design.

Since the springs are in series, the spring rate can be changed depending on which springs are allowed to actuate. As the stroke of the first spring reaches the maximum, a locking mechanism (detailed blue in **Figure 19**) is pushed out of the way, allowing the second spring to compress in series with the first. One disadvantage of this concept is that the assembly containing the springs would most likely be positioned directly behind the check valve. Having this large volume obstruct the flow path can potentially cause poor pressure loss results. However, this increase in flow resistance could be remedied if coupled with a valve design that changes its aspect ratio as the valve stroke is completed (such as the double-disk valve collapsing the cross-sectional area of its valves in **Figure 16**).

Another conceptual design that falls within the non-linear category utilizes a cam and follower system. This is modeled in **Figure 20.** The bar shown in yellow would be a spring and the cam profile would be perpendicular to the flow.

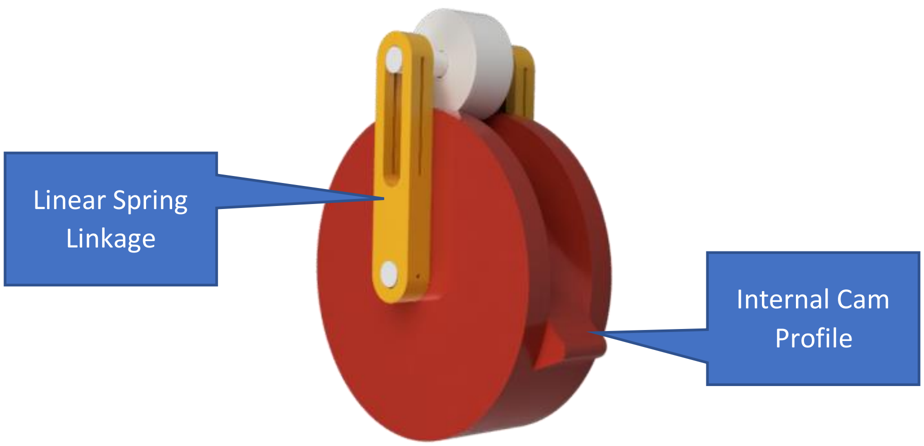

Figure 20. Conceptual CAD of the non-linear spring driven check valve.

Like the other designs, the materials used would be plastic for the extruded parts and linkages, with non-corrosive metals used at pins and for the springs. Much of the cam design is undefined such as the integration of it to the closing mechanism of the valve. The concept of a cam that is non-linear cam is the major interest in this design. Unlike the translational version of the non-linear spring, this rotational version can be mounted in such a way that it does not impede the flow of water. A usage example of the rotational spring includes the spring assembly in **Figure 20** hinged to a check disk, while affixed to the valve body in a recess in the valve walls, as opposed to being centered in the flow path. While this ability to mount the rotational spring assembly out of the water flow path allows for improved pressure loss operation, the small physical footprint of the design would introduce the added difficulty of having to manufacture smaller components and use of smaller springs.

<u>4.4.4 Preliminary Calculations</u>

The introduction of a secondary spring in series requires knowledge of each spring rate and the combined spring rates**.** To meet the desired cracking pressure of the valve of 2 psi, and given a stroke length of 2 inches which is typical of a 2.5” diameter swing check valve, a spring constant of 0.814 lbf/in would be necessary for each spring.

4.5 Risks, Challenges, and Unknowns

Each of these designs has its own challenges. For example, the double-disk adaptation may have issues with sealing during backflow, while the aperture one has challenges regarding the synchronous movement of the pieces. All designs for this project have minimal safety risks. The evaluation of safety hazards can be seen in **Appendix E – Design Hazard Checklist (Updated).** The major unknown variable for each of these designs is their performance regarding to pressure loss. This can be modeled using a simulation and will be tested after the top designs are prototyped. The next challenge for each of these designs is to develop rapid prototypes that represent each design well enough to be tested and compared to Zurn's current in-line spring check valve. The most important aspect of the first phase of this design is to select the best performing design to iterate upon later in the design process.

4.6 Preliminary Design Paths

Based on the conclusions of the preliminary design review, the team decided to test and further develop two designs, the modified in-line check and the double-disk check. Both these designs were 3D printed using Zurn Wilkins' Formlabs SLA printers. We then assembled and placed them into the backflow housing and tested each design for pressure loss in Zurn's wet lab facility. Based on testing results, the team made modifications to each design and repeated this process of printing and testing. Modifications were based on common fluid mechanics principles to reduce pressure loss, such as reducing the area of geometry in the flow path and rounding sharp edges. Specifically, for the double-disk, major changes were made to the seat angle and spring retention. For the in-line design path, changes were made to the poppet geometry and the spring retainer. Based on this testing, the double-disk was adopted as the final design path due to its superior performance over the in-line design.

5.0 Final Design as of CDR

The final design for our project progressed the double-disk design as far as possible, given the abrupt end to iterative testing and re-design capabilities from the impact of the COVID-19 outbreak.

This final design section will explore the final changes made to the current double-disk model, how our team expects these changes to meet project specifications, a detailed description of the most current double-disk assembly, and considerations of safety, cost, and manufacturing.

5.1 Progressing the Double-disk as Final Design

At the time of submitting the Critical Design Review, our final design path consisted of using the most current test data at the time to justify selecting the double disk as the final design.

Figure 21 is a compilation of our team's test data from February 27th, 2020 at the Zurn Wilkins Paso Robles test facility. Three pressure drop curves are plotted, showing performance results for the split inline, double disk design as of CDR (without a spring), and a baseline for the Zurn 350XL inline check (without a spring). The smaller-sloped trendlines show ultimately lower values of pressure drop, meaning a better performing valve.

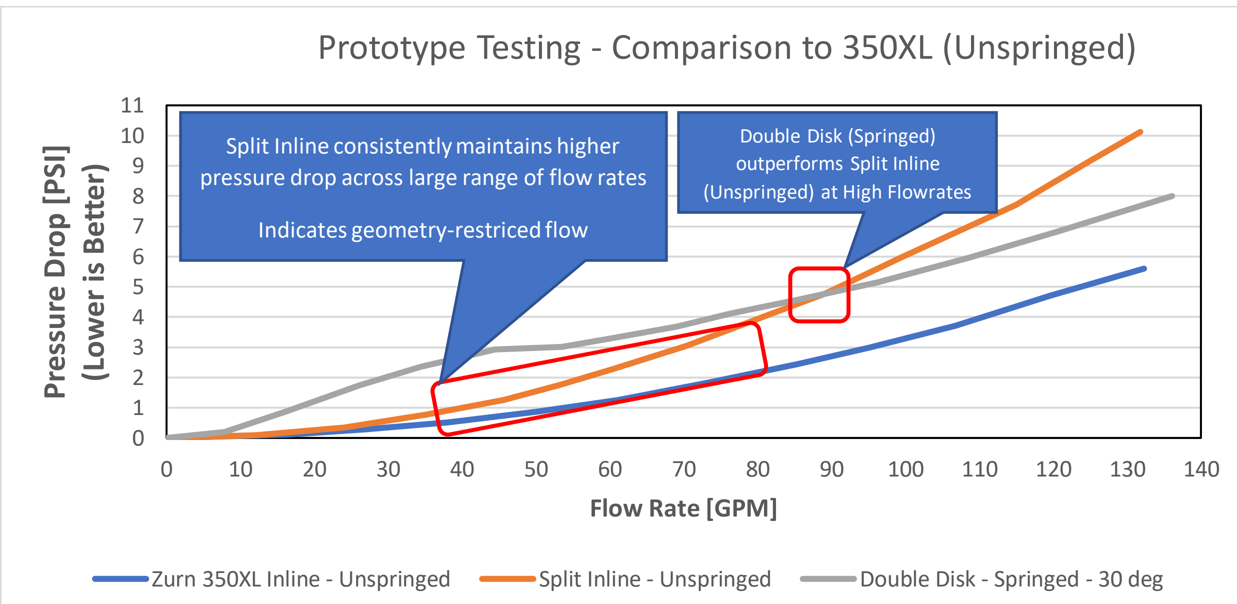

Figure 21. Comparison of Performance of (as of 2/29/2020) Latest Prototypes to Zurn 350XL Baseline

There are a couple key takeaways from the data presented in **Figure 21**:

- The best-performing iteration of the split inline (orange curve) does not at perform better than the baseline performance of the 350XL check valve.
- The difference in pressure drop between the split inline and 350XL appear to be linear for a large position of flow rate.
- The sprung double disk appears to perform better than the split in-line with no spring at high flow rates, starting at approximately 86 GPM.

The results from 2/27/2020 testing also included testing a variety of different poppet geometries. The resulting data indicated that with the current geometry the split inline cannot surpass the 350XL in performance, regardless of flow rate. By analyzing the fact that the difference in pressure drop between the split inline and the 350XL is fairly constant for a considerable range of flow rates, the conclusion has been made that this decrease in performance is inherently due to the constricting nature of the valve's geometry, where any benefit made by producing a central flow cavity in the poppet cannot overcome the consequence of reducing the inlet cross-sectional area. **Figure 22 and Figure 23** help illustrate the areas of concern regarding this issue with the split inline.

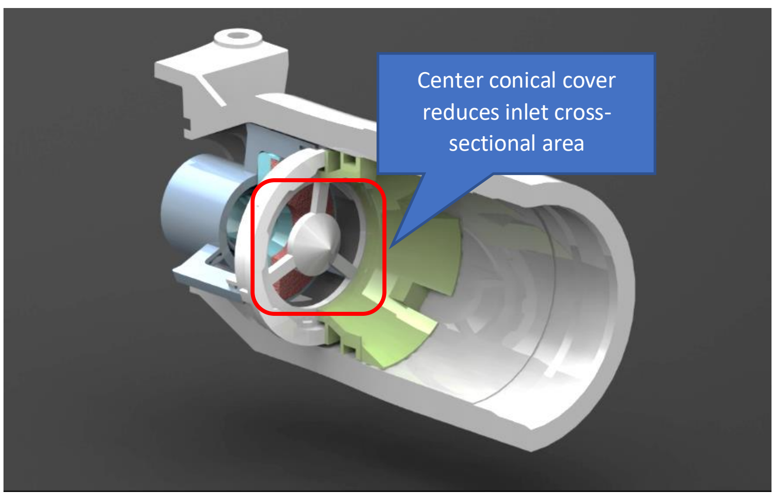

Figure 22. Sectioned View Showing Restricted Cross-Sectional Area of Split Inline

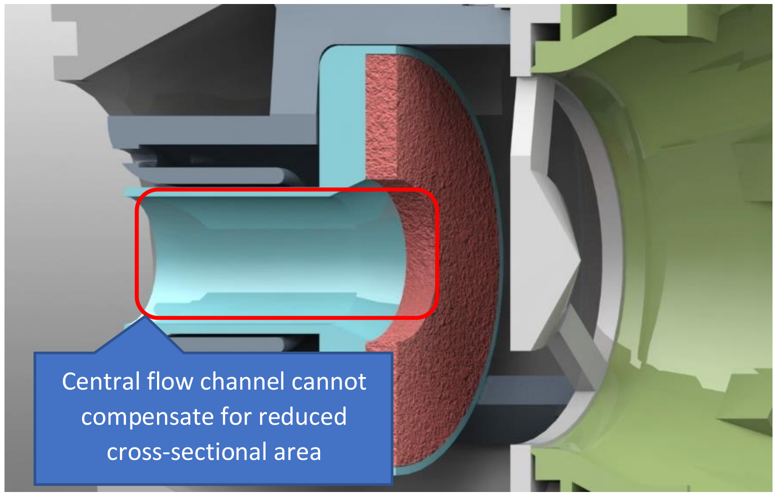

Figure 23. Close-Up Showing Central Flow Channel of Split Inline

The results from testing presented in **Figure 21** confirm that out of the various designs that our team developed throughout the project, the double disk proved to have the greatest potential of achieving the engineering specifications defined in the scope of this project, most importantly including pressure loss reduction.

5.1.3 Design Advantages of the Double Disk

By revisiting the established engineering specifications of this project and constructing a new Pugh Matrix (**Appendix F – Revisited Final Design Decision Matrix (Pugh Matrix))**, our team determined there was enough support to continue forth with the double disk design. The driving factor of this decision was that the valve must first and foremost be able to reduce pressure loss comparative to competing products.

The double disk has a few key advantages inherent to its design that will hopefully overcome the issues encountered by the split inline in terms of performance:

- The overall frame of the double disk allows for a larger inlet area to the valve, compared to the restriction in diameter that the valve seat caused in the split inline.
- The disks of the valve actuate backwards, away from the oncoming flow. This produces an effectively smaller aspect ratio of the valve as the flow increases.
- The disks can be seated at varying angles, which can allow for our team to determine what angle of valve seating is advantageous to pressure loss performance.

Figure 24 illustrates the current geometrical advantage of the double disk valve. Note that the valve does not restrict the inlet diameter unlike the split in-line with the valve seat (seen in green in **Figure 22**). However, as of CDR the team had not designed the double disk to meet all engineering specifications, such as sealing, spring subassemblies, and hinge design.

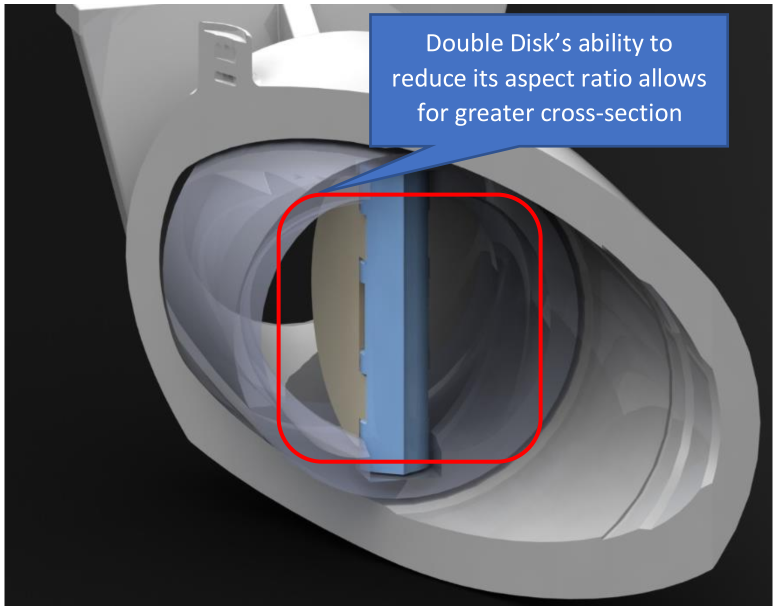

Figure 24. Sectioned View Showing Increased Cross-Sectional Area of Double Disk

5.2 Preparing the Double Disk to Meet Specifications

As noted, the double disk has so far proven to be capable of enhanced pressure drop performance over the baseline 350XL valve. After CDR, our team developed a timeline for the final 10 weeks of the project to ensure the double disk will satisfy other engineering specifications not yet verified.

5.2.1 Improving Central Hinge & Angle of Disks

The current design of the double disk hinge has so far sufficed in testing and dynamic operation. However, an improved design of the central supporting arm and connecting hinges will lead itself to improved sealing capacity and ability accept a wider range of springs for further testing iteration.

Furthermore, the angle of the disks when fully seated or closed, can play a role in how the valve responds to flow rate changes, and consequentially, performance in pressure drop. By conducting further iterations of modeling and CFD testing, our team anticipates selecting a disk angle that will benefit its pressure drop performance. Based on the results of CFD analysis, the final design uses a valve seat that is perpendicular to the flow direction. This is further discussed in Section

7.5 CFD Simulation Results**.**

5.2.2 Sealing, Spring Forces, & Dynamic Operation

Arguably, the most difficult aspect of adopting the double disk will be to ensure that the valve is able to seal fully to prevent backflow according to proper test standards and guidelines such as *USC FCCCHR*. Our team needed to progress a design that ensures proper sealing while not compromising improved performance of the valve. The sealing performance of the final valve was not able to be tested due to COVID-19 restrictions, however the team proposed sealing designs that followed standard O-ring compression practices.

The manner in which the double disk provides counterforce or returning-force to itself via springs is a major factor in how the valve performs under dynamic flow conditions. One approach our team is considered was the use of a buckling spring mechanism. The premise of this device is to provide a discontinuous spring force curve, while controlling the stroke distance at which this discontinuity in force occurs. **Figure 25** illustrates such a mechanism providing nonlinear resistive force dependent upon the stroke distance of the spring. In this case, the stroke is the compression distance of the spring.

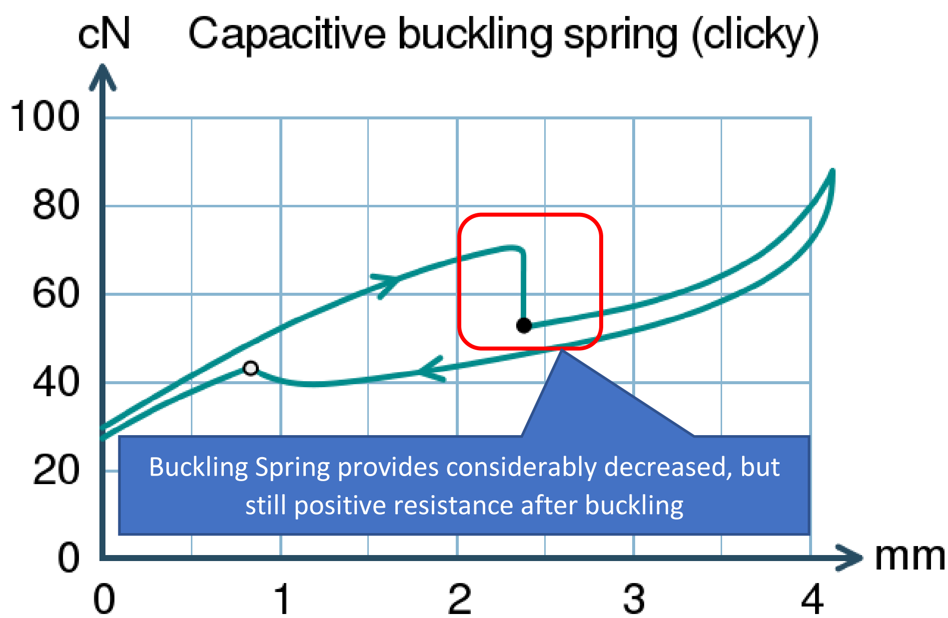

Figure 25. Force vs Stroke Graph Illustrating Force of Buckling Spring

The team was unable to test any buckling springs, however we did briefly look into the mechanism prior to CDR. The challenge regarding implementing buckling springs in a double-disk design is that current buckling springs used in industry are of linear nature, where the resistive elements used in the double disk have so far only been of torsional nature.

5.2 Configuration of the Double Disk – Pre CDR

Figure 26 and **Figure 27** shows the Pre-CDR configuration of the Double Disk in a double check assembly, meaning that two individual checks are placed within the same valve housing. Annotations are provided to mirror the names of the components as seen in the Indented Bill of Materials (**See Appendix G – Indented Bill of Materials**).

In the configuration as of CDR, the double disk valve assembly contained 6 different part groups/sub-assemblies:

- Check Frame: Provides a structural component for smaller components to be housed in
- Hinge: Main supporting structure that connects valve disks to the check frame
- 3/32" Pins: Metal pins that locate the valve disks concentrically to the check frame
- Torsional Spring: Resistive force element that provides a returning force to the check disks
- Poppet Disk: Two of these disks are the elements that prevent backflow through the valve during no-flow conditions.
- Sealing Components (Sub-group): Various components (O-Ring, Seat Seal, etc.) that allow for complete sealing of the valve during no-flow conditions.

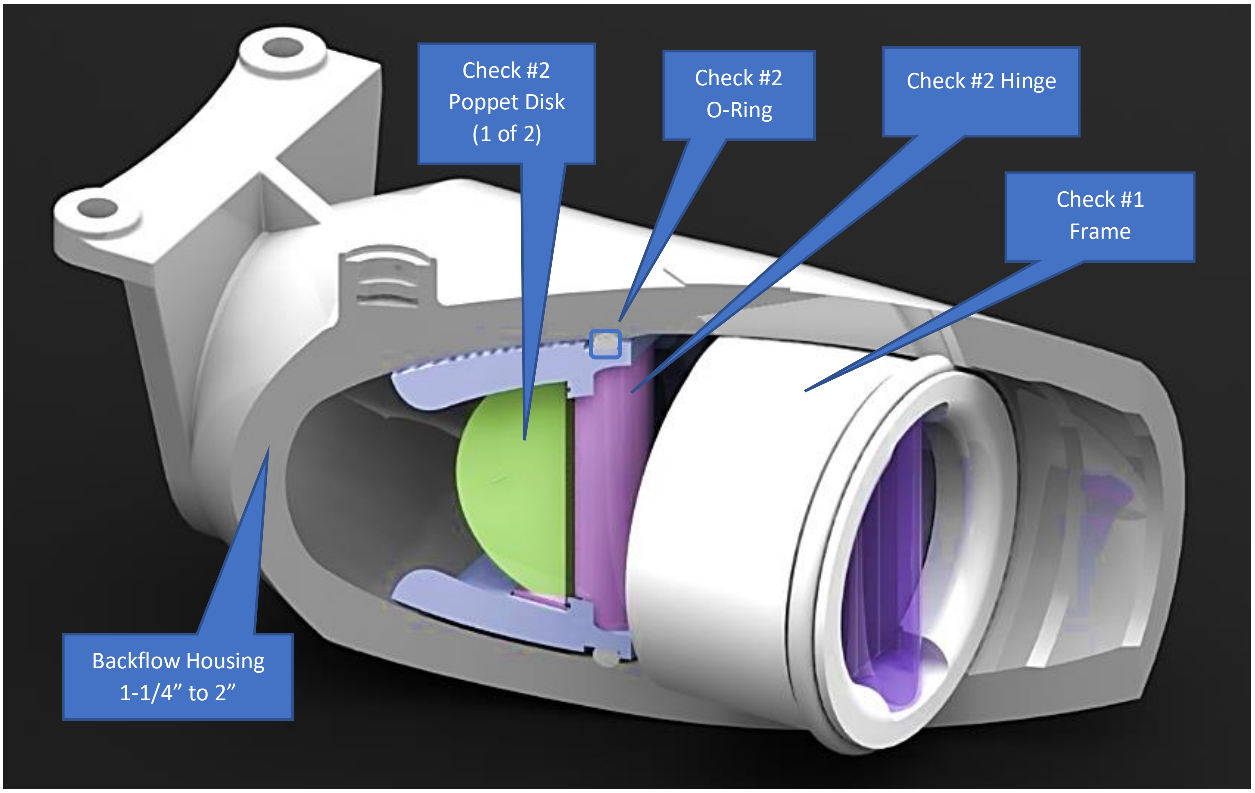

Figure 26. Isometric Sectional View of Double Disk Valve Assembly

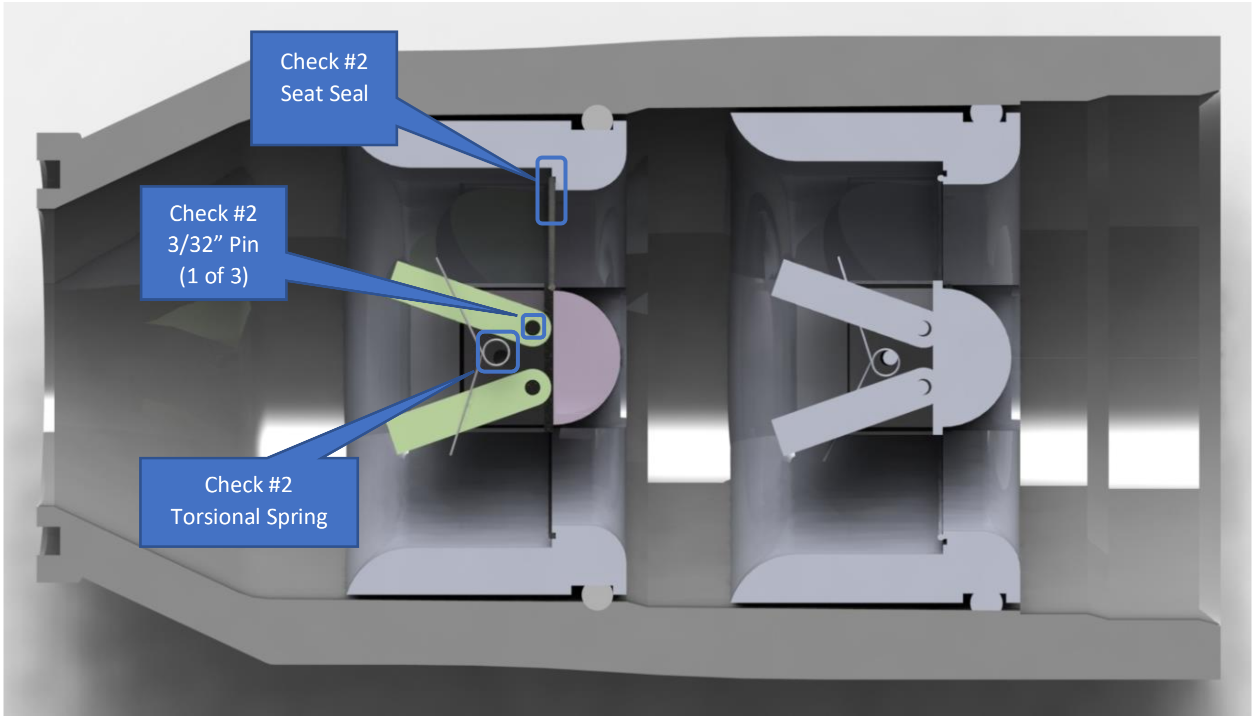

Figure 27. Cross-Sectional Overhead View of Double Disk

5.3 Safety, Manufacturing Considerations, & Cost

Since the cost of transitioning a check valve design from pre-production and rapid prototyping into small-volume runs with custom-manufactured components can be in the range of tens of thousands of dollars, which drastically over exceeds the provided budget for our project, our team plans to source materials and prototyped components through Zurn vendors and companies specializing in prototype manufacturing and small-batch production. **Section 6.0 Completed Manufacturing & Testing** for further details this anticipated manufacturing strategy, along with other information detailing the assembly of the current double disk valve.

As for safety and hazard considerations, a Failure Modes & Effects Analysis (FMEA) and Design Hazard Checklist (Presented in **Appendix H – Failure Modes & Effects Analysis (FMEA) & Appendix E – Design Hazard Checklist (Updated)**, respectively), detail the possible hazards and points of failure of our check valve design. The FMEA shows points of failure being similar to those of industry-current check valves, and there are no outstanding issues indicated within the hazard checklist. In other words, there are no novel hazards or failure points that our team or Zurn must consider when prototyping, manufacturing, or testing our designs.

Despite the fact that our current prototypes have been developed using rapid prototyping methods (SLA printing, our team still presents a package of technical drawings for the components involved in the current double disk design **(See** Error! Reference source not found.**)**.

The drawings are meant to take advantage of Geometric Dimensioning and Tolerancing (GD&T) to specify critical surfaces and curvatures of the parts. For example, any mating surfaces that involve use of the stainless steel rods must be located precisely to prevent unnecessary binding from through-hole misalignment. Additionally, any seal surfaces must be manufactured to a specified surface

finish/roughness so as to avoid premature valve/seal separation. Within the frame, the outer support structure must be concentric so as to fit within the existing Zurn 350 XL valve body. Cylindricity of the frame is also a concern, since any major tilt in the geometry in the check valve subassembly will cause a constant bias in shearing of the water when passing through the frame.

5.4 Final Double Disk Design Preview – Post CDR

After in-person testing and any project-related visits to Zurn's facilities ceased in mid-March 2020, the corrected scope and objective of the project pushed for one final design iteration of the double disk model. **Figure 28** is a rendered isometric view of the final design proposal for the double disk. A few of the major design changes/updates compared to the Pre-CDR design are highlighted:

- Modifications to the interior geometry of the valve housing
    - A slotted valve spacer (seen in purple) is used to locate the two valve assemblies in relation to each other, as opposed to relying on interior grooves in the valve housing to locate each of the 2 check valve assemblies individually. The valve spacer also serves as a sealing surface for the upstream check
    - Each valve assembly's valve seat/hinge is now mechanically fastened in place via a threaded pin that protrudes through the wall of the valve housing. This change enhances structural integrity and sealing capability of the valve
    - An O-ring is the sealing component for the interface of the valve hinge (seen in yellow) and the interior wall of the valve housing. This is an improvement in sealing capability over the previous design
- Dovetail glands are used in addition to a sealing medium (partial O-ring or gasket material) to seal the interface between the valve hinge (seen in yellow) and the poppets (seen in orange)

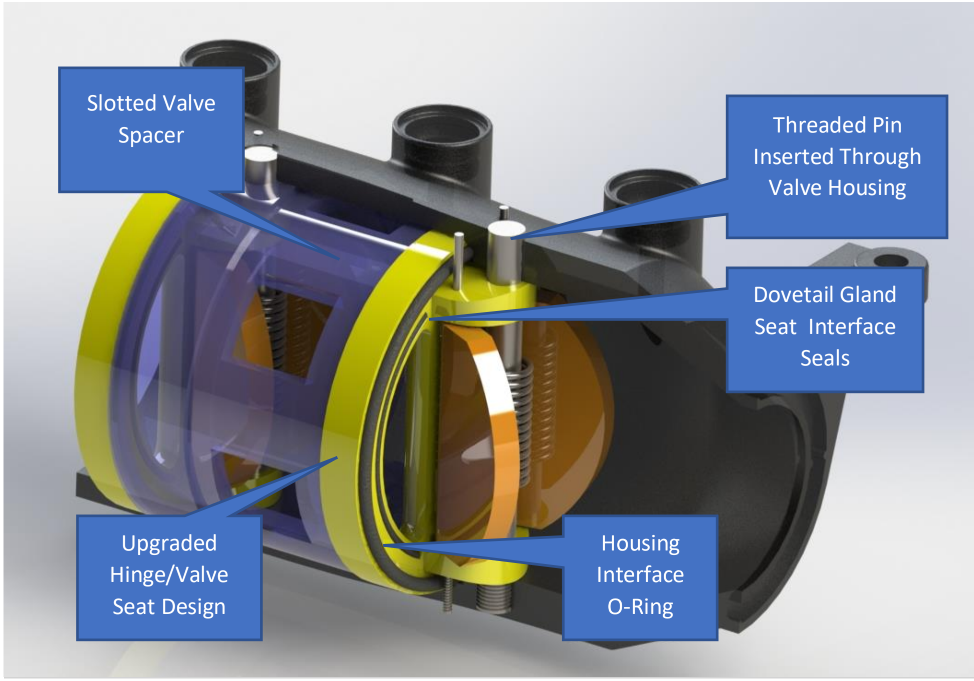

Figure 28. New Features of Proposed Final Design

Detailed design justification, description of operation, and assembly views involving the changes made in the final proposed design will be provided in **Section 7.6 Proposed Final Design**.

6.0 Completed Manufacturing & Testing

Due to the nature of this project, the iterations leading to the final design were initially done by using 3D printed models at Zurn's facility. These printed prototypes were tested until a design conclusion was reached. Since the team was not able to continue testing during the final months of the project, the valve design was determined using the conclusions from completed testing and further analysis. The team developed a manufacturing plan that could be used for the full-scale manufacturing of these valves since no physical prototype of the final design was permitted to be built.

6.1 Procurement

Most of the components for this design are intended to be custom-made from Delrin 150. This material was selected because of its resistance to warpage when submerged in water, and its capability to be both injection-molded and post-machined. Delrin is also a common material for valve components in industry. The check valve hinge, poppet disks, and spacer will be manufactured through a combination of manual and CNC machining.

For a new valve design, Zurn Wilkins typically works with their current vendors to develop all procurement and manufacturing plans. Since the team was not able to manufacture a final valve, all procurement costs summarized in **Table 7** are rough estimates for full-scale manufacturing. Many of the custom parts costs are estimates based on costs of similar parts currently produced by Zurn and by cost per weight of the material. The full budget is detailed in **Appendix J – Full Budget Breakdown.**

Table 7. Summary of Project Budget for Final Design

Name	Description	Vendor	Vendor Part #	Qty	Cost	Total Cost
Modified Backflow Housing	Nylon N72333 STHL, Black, UV Resistant. Modified from the 350XL, machined to include modifications	Zurn Wilkins	354-1A	1	$ 25.09	$ 25.09
Check Hinge	Delrin 150, custom injection molded & post-machined	Protolabs	----	2	$ 3.83	$ 7.66
Torsion Spring-Custom	Torsion Spring, 210 Degree Angle, Left-Hand Wound, 0.585" OD, 0.070" Wire Diameter, 0.5" Leg Length, 13.083 Coils	International Industrial Springs	----	1	$ 10.00	$ 10.00
Poppet Disk	Delrin 150, custom injection molded & post-machined	Protolabs	----	4	$ 0.47	$ 1.88
Poppet O-ring Arcs	Buna-N O-Ring 1/16 Fractional Width, Dash Number 040, [Zurn #040N]	McMaster-Carr	9452K128	1	$ 14.26	$ 14.26
Check O-ring Seals	Buna-N O-Ring 1/8 Fractional Width, Dash Number 237. [Zurn #273N]	McMaster-Carr	9452K166	1	$ 13.41	$ 13.41
3/16 Stainless Pin	3/16" 303 Stainless Steel Rod, 2 ft	McMaster-Carr	8984K13	1	$ 3.81	$ 3.81

3/8 Pin	3/8" 303 Stainless Steel Rod, 1 ft	McMaster-Carr	8984K99	1	$ 4.54	$ 4.54
Check Spacer	Delrin 150, custom injection molded & post-machined	Protolabs	---	1	$ 2.55	$ 2.55
					TOTAL COST	$ 83.20

The parts needed for sealing will be either be purchased from a vendor or custom manufactured. The team considered sending the custom seals to be manufactured by the company Protolabs, which our sponsor has worked with before, however due to the high initial cost of tooling for custom seals, the team decided to custom manufacture any seals that can't be easily purchased using commercially available O-rings. All sealing grooves were adjusted to fit standard available O-rings.

The only other hardware required for the final design are a torsion spring and stainless-steel pins used in the disk hinges. The pins (3/16" and 3/8" in diameter) are made of 304 stainless steel to prevent any corrosion. For prototyping, the team used stainless steel welding filler rod, however for the final product these would be sourced from an online vendor such as Grainger Industrial or McMaster-Carr. In the final design the pins need slight modification by cutting their respective UNF threads using a manual thread die.

6.2 Manufacturing Operations

Due to the high tooling costs of injection molding and CNC machining, the process used for prototyping and testing is different compared to the final products. The process proposed for manufacturing the final design is more complicated than that used for making the prototypes. In this section our team will outline both manufacturing methods, for the prototypes and the proposed design.

6.2.1 Prototype Manufacturing

The double-disk valves that the team has tested have been rapid prototyped by 3D printing the valve pieces with Zurn Wilkins' SLA printers. The support structure is removed, sharp edges are filed off, and the pins and springs inserted. The whole check valve is placed in Zurn's backflow housing, and the system is ready for testing.

For the in-line check valve prototypes, custom seals made of 1/16" thick neoprene rubber were cut using one of the mechanical engineering department's laser cutters. This method is shown in Figure 29. The seals used in the final prototype of the double disk valve were made of standard O-rings, some of which were cut to length to fit within dovetail grooves that retained them.

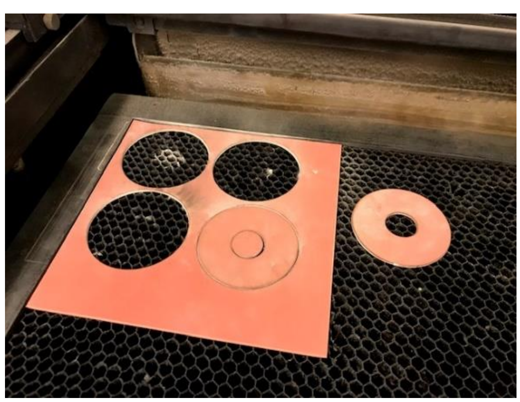

Figure 29: Example of poppet seals cut using the ME machine shop laser cutter

One thing to note is that the final prototype is assembled with 5 small torsion springs that match the torque requirement of the check valve. These springs do not meet the fatigue requirements of industry test standards but were adequate for the short pressure loss tests performed by our sponsor.

6.3 Assembly

One of the more frustrating aspects of the double-disk designs that have been tested so far is the length of time it takes to properly assemble the prototypes. The presence of thin features has led to parts of the valve hinge breaking during the assembly process. The assembly process for the most recently tested double-disk valve is outlined below. This prototype is referred to as the “final prototype” and is most similar to our proposed final design. Some of the breaking issues were removed by simplifying the hinge design and making the poppet disks thicker in the final design.

Since the team was not able to build the final double disk prototype in person, our senior project sponsor Mr. Yale graciously 3D printed and assembled the components for testing. The assembly procedure he used is as follows:

1. Cut the steel rod into 3 lengths of 2.75-inch segments. See Figure 30.

Figure 30. Cutting steel pins to length

2. Cut the smaller diameter O-rings (-015) into semi-circles. These should match the length of the arcs that the hinge has on each end. See Figure 31.

Figure 31. Hinge seals cut to length and placed on hinge

3. Cut the medium diameter O-rings (-142) into semi-circles, as shown in Figure 32. These should match the length of the dovetail grooves on the check frame. Press these O-rings pieces into the grooves.

Figure 32. Cut O-rings to length for dovetail grooves

4. Insert a pin through the check hinge and through each poppet disks.
5. Insert the third pin through the check hinge and 5 of the torsion springs. The spring legs should compress the poppets back as seen in Figure 33.

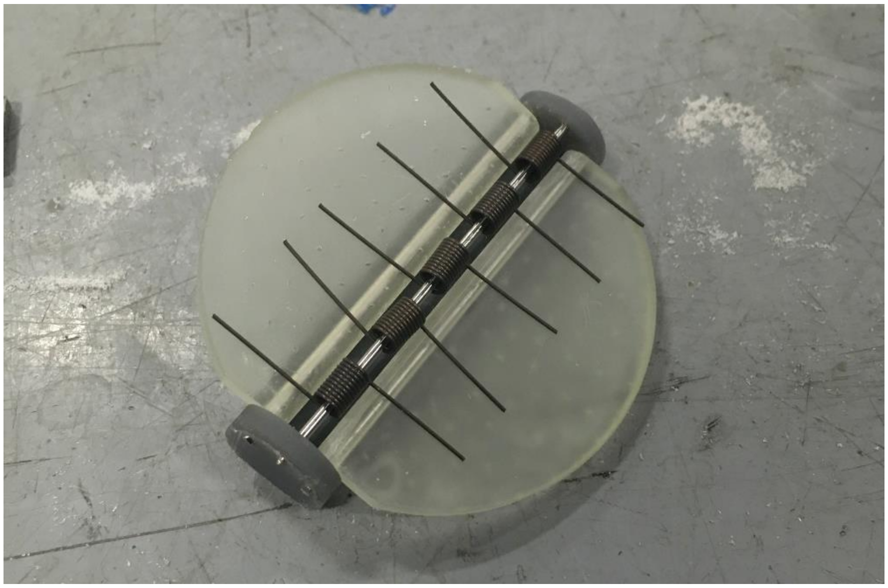

Figure 33. Final prototype with pins inserted through springs and poppets

6. Slide the assembled hinge with seals, disks, and springs into the check frame. This requires folding back the poppet discs so they are able to get "downstream" of the poppet seats.

Figure 34. Hing and poppets placed inside the check frame

7. Place the largest O-ring, (-336) in the groove around the assembled check frame.

Figure 35. Final prototype check valve, fully assembled

8. Insert the whole check valve into a 1.5" 350XL housing

Figure 36. Final check prototype inserted into backflow housing for testing

6.4 Outsourcing

If necessary, the components made from custom seals would be outsourced to Protolabs, the preferred company for this type of rapid prototyped seals. Originally, the team considered outsourcing the seals as a possibility for the final prototype, specifically for these sealing components. Since the final assembly was not manufactured, outsourcing is no longer the suggested production method. For the final design we recommend sealing components be manufactured by Zurn Wilkins industrial suppliers.

The team was not able to find a torsion spring that precisely meets all requirements and therefore suggests a custom spring be manufactured by a company such as International Industrial Springs. One major tradeoff that was found when designing the spring is that as the torsion spring coil diameter decreased It became easier to package, but more susceptible to fatigue. Thus, the springs used in the prototypes were design to meet static stresses and did not meet the fatigue safety factor.

6.5 Suspension of Manufacturing and Testing Operations

All manufacturing and testing operations were suspended as of March 18th 2020 due to the rapidly changing events surrounding COVID-19. As our design process was based on iterative design driven by test data and supported by rapid prototyping our design process was greatly impacted. Prior to the suspension of testing operations our team had completed 18 pressure loss tests at Zurn's wet lab facility. Tests were performed on multiple iterations of two different design trees. At the time of manufacturing operation suspension, we had manufactured 7 complete prototypes and additional components such as poppets with varied geometries to test other designs modifications via rapid manufacturing with Zurn Wilkins' Formlabs SLA printers.

Despite the suspension of manufacturing and testing operations for students we were able to work with our project sponsor to get our final prototype printed, assembled and tested.

7.0 New Project Scope & Results

Due to the COVID-19 pandemic, the scope and outcome of this project were altered. The team was no longer able to perform in-person testing and no longer had access to the Cal Poly machine shops or Zurn Wilkins' wet lab facility. Through discussion with the project sponsor, the scope of the project was shifted to a more analytical focus.

7.1 New Problem Statement & Objectives

In **Section** 3.1 Problem Statement, the original problem statement for our project is as follows:

> *"Zurn Wilkins, a plumbing parts manufacturing company, is requesting a new check valve design that uses mechanical advantage and fluid dynamic principles to reduce pressure loss comparative to their existing products. This design must meet industry standards for water supply backflow prevention."*

While the engineering specifications and problem statement that Zurn is attempting to resolve hasn't necessarily changed, what has been considerably modified is the scope and objectives that our team, along with Zurn, has agreed to satisfy before the end of the project timeline (June 4th, 2020). Below, our team summarizes the change in scope and reasonable objectives to be achieved for the remainder of this project:

Given the impacts made by the COVID-19 outbreak, Zurn Wilkins and the Check Valve design team has shifted their project objectives to focus on delivering a semi-functional final prototype, final test data, and additional support research such as CFD analysis and additional design research, as final deliverables for this project. These deliverables combined will serve as a foundation for Zurn to later build off for use of the use of a double disk check valve in backflow prevention products built to standards required for potable and treated water systems.

7.2 Original Design Verification Plan

In order to verify our check valve design and its ability to meet our design specifications, testing protocols were designed. Our group began testing some of the specifications of the current iterations of our designs in January at Zurn Wilkins wet lab. We have developed tests to ensure that each one of our specifications was been met properly, we tested individual functions to confirm that specific specifications are met.

7.2.1 Specification Discussion

As part of the Quality Function Deployment (QFD) process in which we translated our customer needs wants and thoughts into engineering specifications. Based on these initial specifications a Design Verification Plan & Report (DVP&R) was developed and can be found in **Appendix K – Design Verification Plan (DVP&R)**. These initial specifications included:

1. Reducing the pressure loss as compared to the current check valve assembly
2. Size compatible with Zurn's current design
3. Maximum Allowable Working Pressure (MAWP) per ASME B16.34
4. Maximum Allowable Working Temperature (MAWT) per ASME B16.34
5. Static pressure differential
6. Assembly time
7. Cracking pressure

Further in our design it had become apparent that some specifications may not be reasonable to test until later stages of development, are not as important as once considered, or do not need specific test plans. For this reason, the team modified our existing specifications, and will only explain tests that were feasible and necessary speciation's at the time of CDR. The modified specifications that we planned on testing directly include:

1. Reducing the pressure loss as compared to the current check valve assembly
2. Valve does not leak in reversal of flow condition
3. MAWP at coincident temperature rated to 175 PSI @ 180°F (Hydrotest to 350 PSI @ 180°F)

It should be noted that the second specification to be tested was not laid out initially and was added at a later design phase. This specification is crucial to the basic function of a check valve and was likely overlooked. Due to the circumstances of the final months of this project, the second and third specifications were not tested.

7.2.2 Test Specifications and Test Equipment

The previous specifications will be tested using the following testing methods laid out in this section. The pressure loss tests have already been running for about a month and the next tests are planned to begin in the future at the appropriate time in the design process. Outlined below are all the tests that have been conducted and were planned for before the change in scope.

Pressure Loss Test

This test involves placing the desired check valve or component to be tested inside the 350 XL 1 ½" Valve Body Housing (354-1) ensuring that all valves are in their appropriate closed or open position, all seals and O-rings are installed, and that the valve body is properly secured by the four flange bolts (352-11). The primary inlet shutoff valve is then verified to be in the full closed position before the appropriate pump is then turned on to circulate water from the water tank to test stand.

A rough process flow diagram is shown in **Figure 37** that includes only the essential components. Zurn Wilkins' wet lab acts as a closed loop system that circulates water at semi-constant pressure through the test stands.

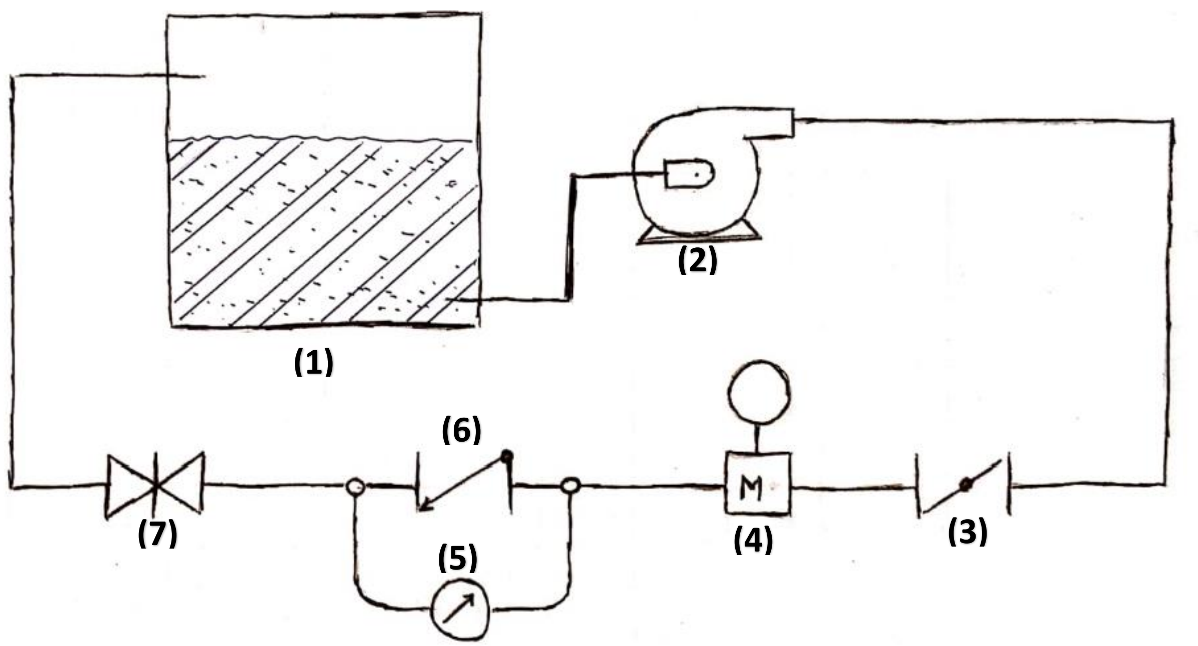

Figure 37. Pressure loss test stand PFD

The large reservoir of water is stored in a large underground water tank **(1).** This holds the water that circulates through all the various piping manifolds and test stands. The water is pressurized and circulated by several centrifugal pumps **(2)** capable of being set up in a parallel flow configuration to allow for larger flow capacity conditions. Under standard conditions the water that enters the test stand sits just under 100 psi_g and the flow is varied from 0 up to 140 GPM. Before the water circulates through the test stand its flow path is stopped by the butterfly isolation valve **(3)** detailed in **Figure 38**. This valve is used to prevent the test stand from receiving pressure until it is ready and has been properly lined up. This valve could be considered a safety device to some extent.

Figure 38. Inlet isolation butterfly valve (3)

Water then flows through the magnetic flow meter **(4),** or mag-meter. This device allows us to read the flow rate of water moving through the test stand in GPM. The operating principle of the magnetic flow meter is based on Faraday's Law of electromagnetic induction and it is designed to measure the volumetric flow rate of fluid. An insulated pipe is placed vertically to the direction of a magnetic field. When an electrically conductive fluid flows through the pipe, an electrode voltage is induced between a pair of electrodes placed at right angles to the direction of the magnetic field. The electrode voltage is directly proportional to the average fluid velocity and with a known cross-sectional area the volumetric flow rate can be determined. The specific mag-meter being used is seen in **Figure 39** and its data sheet can be found in **Appendix L – Instrumentation Documentation**.

Figure 39. Magnetic flow meter (4)

Pressure is then read from the pressure differential digital gauge **(5)** in psi_d. Pressure is measured across the test stand and the pressure taps are several pipe diameters upstream and downstream of the test stand. The pressure taps come off of large bulbous expansions in the piping, this is done to reduce the effects of the dynamic pressure and measure the static pressure at these regions as accurately as possible. The pressure gauge hoses have needle valves for purging any air that may remain in the system and to remove the effects of having compressible fluids in the instrumentation. The specific pressure gauge being used can be seen in **Figure 40** and its data sheet can be found in **Appendix L – Instrumentation Documentation**.

Figure 40. Digital pressure gauge (5)

Water flows through the test stand **(6)** and desired check valve configuration to be tested. It is important to ensure that all valves are in the appropriate positions and all seals are in place to prevent unwanted leakage of the test stand during testing. The test stand with a Zurn 350XL double check backflow preventer and the pressure taps can be seen in **Figure 41**.

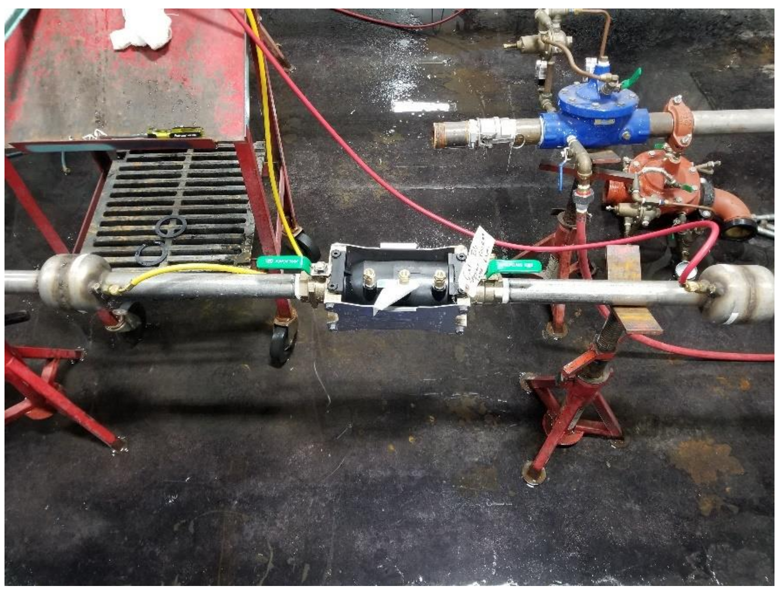

Figure 41. Test stand

The last critical component for testing is the hand actuated gate valve **(7)** being used for flow control. Once the system has been verified to be in the proper condition the gate valve is in the full closed position the isolation valve **(3)** can be opened to expose the system to full test pressure. Pressure drop at the no flow condition is then captured. Using the gate valve, the flow rate is stepped up in moderate increments of about 10 GPM and pressure loss is recorded at each flowrate. The specific gate valve can be seen in **Figure 42**. A full test operating procedure can be found in **Appendix M – Pressure Loss Testing Procedure.**

Figure 42. Gate valve for flow control

Reverse Flow and Valve Leakage Test

The purpose of this test is to verify if the check valve can seal and prevent reversal of flow in backflow situations. This test does not come directly from any of our initial specifications but is key to the success of a check valve design as it is the primary functional purpose of a check valve.

This can be tested at the early stages quite simply. First, the desired valve assembly to be tested is fitted into a housing. Then, the valve is connected in the reverse flow condition to an appropriate water pressure source (tap) with the necessary connections and adapters. Once the valve apparatus is connected to the water pressure source, the valve can be exposed to a reverse flow condition and verified if it properly seals and prevents the reversal of flow.

This test can be further improved at later stages by adding a pressure control valve and a pressure indicator to modulate the pressure from zero to the max pressure condition and determine how high of a pressure the valve can hold while preventing reversal of flow.

MAWP & MAWT Test

The purpose of this test is to verify that the design can meet withstand a minimum Maximum Working Allowable Pressure (MAWP) and Maximum Allowable Working Temperature (MAWT). This will be accomplished via a hydrostatic test of the final concept per the testing procedure laid out by ASME B16.34 and other relevant industry standards.

A typical hydrostatic test for a valve follows these basic steps:

1. The valve and adjacent piping to be hydrotested are filled with the testing fluid at the specified temperature
2. The specified pressure is applied for the specified length of time and the components can come up to test temperature if appropriate (Typically at least 2X MAWP at coincident temperature)
3. Leakage is measured across the valve element of interest using both measuring instruments and visual examination methods (most standards specify that no visually detectable leakage is allowed)

4. A visual inspection is performed to ensure the valve has not been damaged during the testing procedure
5. If necessary, additional inspection may occur via Nondestructive Testing (NDT) methods

The hydrostatic test is akin to the reversal of flow and valve leakage test and both could be run simultaneously.

7.3 Final Prototype Design

The team was only able to produce one prototype for testing, and this prototype needed to be assembled out of commercially available components due to long leads times for custom components because of the COVID-19 outbreak. The final prototype was slightly different from the final design proposed by the team in Section **7.6 Proposed Final Design** because of these limitations. The team did not build or test this prototype but instead worked with our project sponsor, Mr. Brian Yale, to print, assemble, and test the prototype double disk check valve.

7.3.1 Changes from Previous Prototypes

The final prototype builds off the results and experiences from our previous double disk prototypes. This includes a reduction in material in the flow path, alterations to the hinge geometry, and most significantly, the introduction of sealing components. Images of the CAD model for the final prototype are found in Figure 43. Note that the valve has three sealing surfaces and is housed within a check frame. The frame was removed in the final design, as is discussed in Section 7.6 Proposed Final Design.

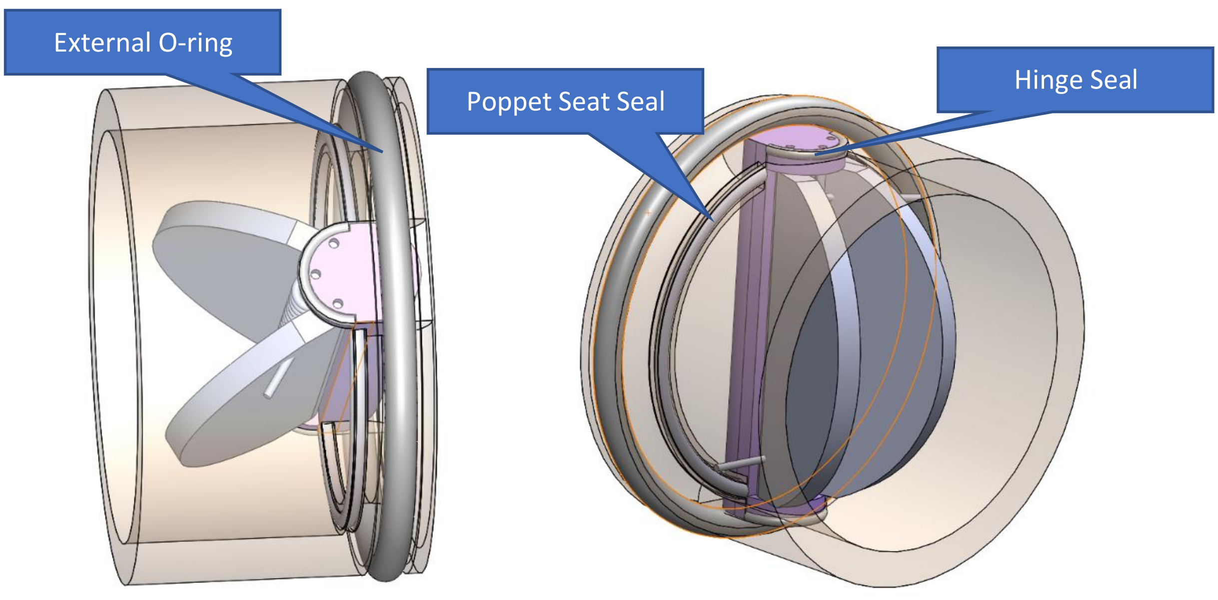

Figure 43. CAD Model of the Final Prototype

All three seals were made from standard O-rings, and both the poppet seat and hinge seal were made by cutting these O-rings to length. The external seal is not altered in any way. One goal of this prototype was to enlarge the area of flow by reducing the width of the hinge. This in turn requires that the poppet disks be thinner.

7.3.2 Manufacturing Challenges

The team was concerned that reducing the poppet disks' thickness would increase their susceptibility to breaking. To address this, we made the hole for the pins smaller in diameter, now 1/16". One challenge that the team ran into throughout the project is the tolerance on the Formlabs printers. We found that the external geometries were typically quite consistent, within 0.015-0.020" of the intended dimensions, however there was greater variability for internal geometries such as grooves and holes. Because of this, we were concerned that our sponsor might run into issues with the smaller pin size. This was a problem experienced by our sponsor when he was assembling the final prototype. However, the team was able to re-print the poppet disks with pine holes enlarged by 0.020" and our sponsor was able to successfully test this prototype.

7.4 Test Data and Numerical Analysis

As mentioned before, testing for pressure loss had already been started at Zurn Wilkins wet lab. Detailed data and trends can be found in **Appendix N – Test Data**.

Our first test runs were primarily for the purpose of learning how to properly conduct a test and operate the equipment. Once we were comfortable using the testing equipment and capturing data, we ran a few tests to verify that the data we were collecting was in line with our expectations.

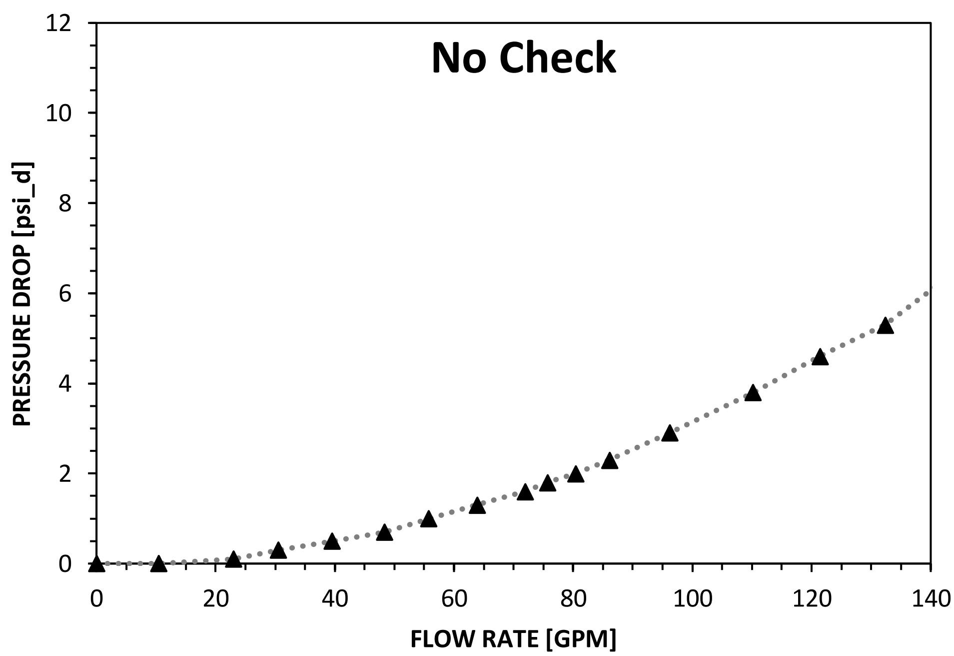

Figure 44. Test data to verify expectations

Figure 44 shows test data from the test stand with no check valves. As seen, the results of the test are in line with our expectation that pressure loss is proportional to the square of flowrate. With this confirmed, further tests of varying types were conducted to build a collection of test data.

From our early stages of testing, we were able to determine which design paths had more potential. **Figure 45** shows the pressure drop results for one of the split inline poppet designs. In this design, the

poppet had an internal flow channel as shown previously in **Figure 23**. The flow channel through the poppet had a smaller inlet as compared to its inlet, similar to a diffuser.

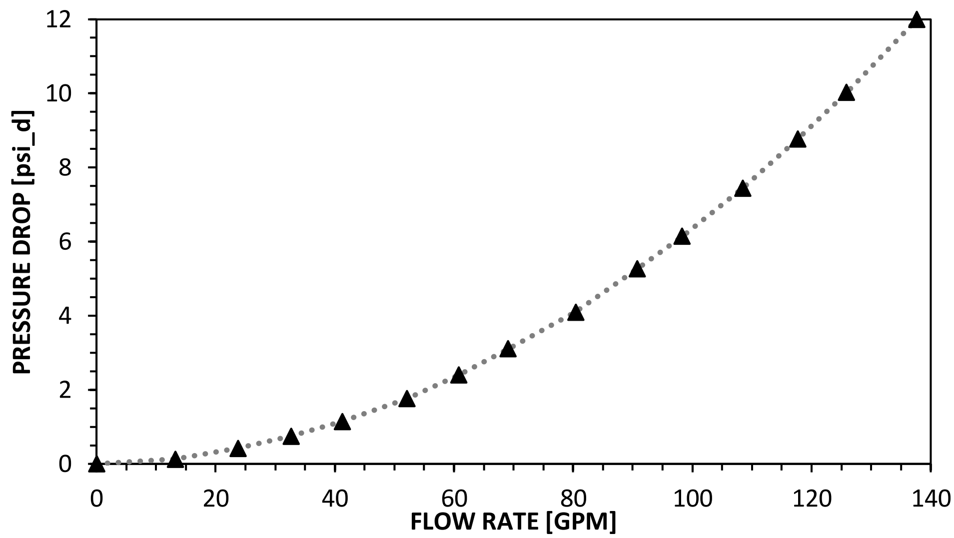

Figure 45. Split Inline Poppet: Diffuser test data

This pressure loss curve is comparatively steep relative to other tests performed. In testing the split inline variants no springs were installed. The pressure loss curve would be even steeper had we tested with springs installed. Other variations on the split inline followed a similar trend.

In **Figure 46** pressure drop results for one of the later iterations of the double disk can be seen. The disks would seat at a 30°offset as opposed to being completely perpendicular to the flow path.

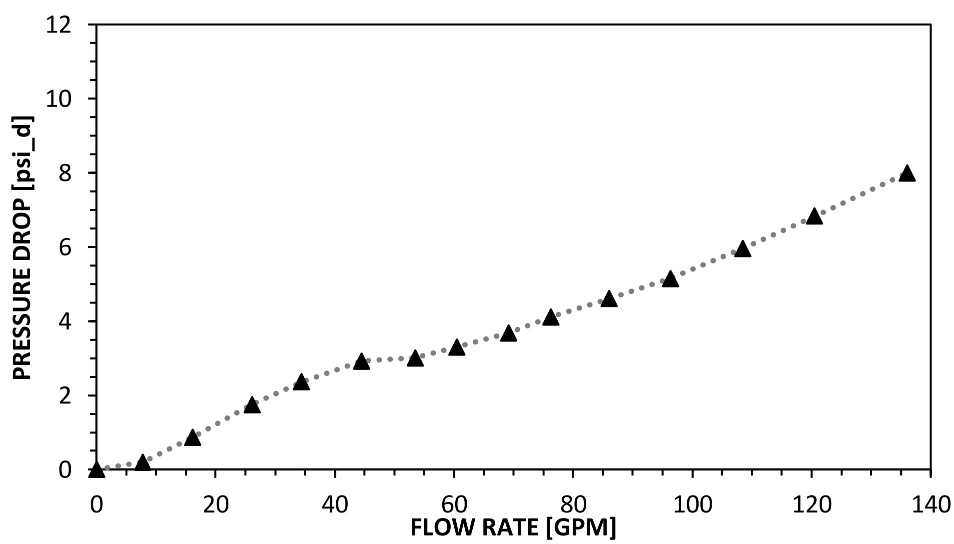

Figure 46. Double Disk V4 30° test data

This pressure loss curve was comparatively shallow relative to other tests performed. Springs were installed in all poppet seat angle variations that were tested, referred to collectively as the V4 double disks. With the 30° and 45° seat angle versions, we saw no significant improvement over the normal seat angle in terms of performance.

Based on the test data, we decided to progress the Double Disk design. This design decision is discussed in greater detail in **5.1 Progressing the Double-disk as Final Design**.

After our final prototype (V6) was printed and assembled by Mr. Yale, one final head loss test was conducted to quantitatively compare the performance of the most recent prototype version to the 350 XL inline check valve, which has been our team's baseline, or control, for the vast majority of our test runs. **Figure 47** shows the graphed test run data collected by Mr. Yale. Three data sets are graphed to indicate pressure drop across the entry and exit of the 350 XL valve housing with differing single check valves, or lack thereof, installed in the housing. The theoretical minimum amount of pressure drop possible is represented by data collected while the 350 XL housing was vacant (Light Blue). The maximum allowable pressure drops to still meet our project's objective of pressure loss reduction is represented by the data collected with the single inline check valve installed in the housing (Yellow). Nested in-between these two test runs lie the results of the V6 double-disk (Dark Blue). A few key points are listed regarding these test results:

- While ignoring backpressure and sealing capabilities, the V6 prototype does appear to produce a lower pressure drop nearly across the entire testing range of flow rate. However, it must be considered that while the V6 double disk is producing improved results over the inline, if the double-disk will require heavier spring loads for adequate sealing and backpressure testing, higher pressure loss may result in later prototype testing.
- During low flow rates (0-80 GPM), the general curve of the V6 doubles disk appears to represent more closely that of the "no-check" results, versus the inline result. This could be indicative that the nature of the double-disk design allows for less flow constriction than the inline.

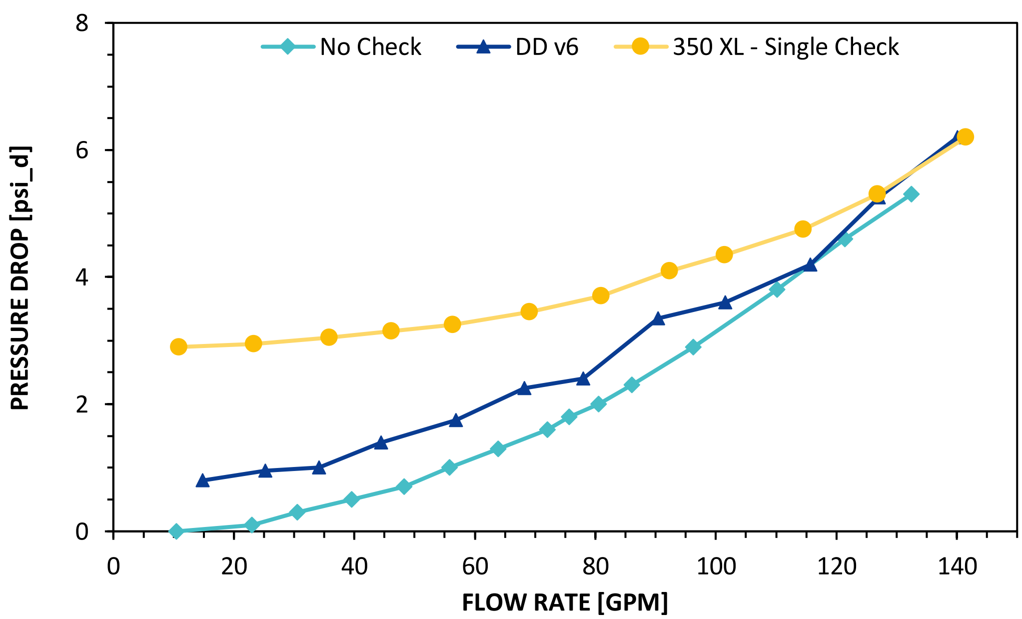

Figure 47. Test Results comparing v6 to Zurn Design

7.5 CFD Simulation Results

The primary engineering analysis we performed was Computational Fluid Dynamics (CFD). We utilized the Autodesk® CFD Ultimate package to run our flow simulations per Zurn Wilkin's recommendation. We had initial success in accurately simulating flow conditions within our test stand. In this first render, **Figure 48**, we can see a cut plane of the test stand with no check valve installed. The purpose of this simulation was to determine the validity of our model and flow parameters by comparing it to physical test data. Our simulation came within 4% of the test data. Colors here represent the pressure gradient developed across the valve at specified flow conditions.

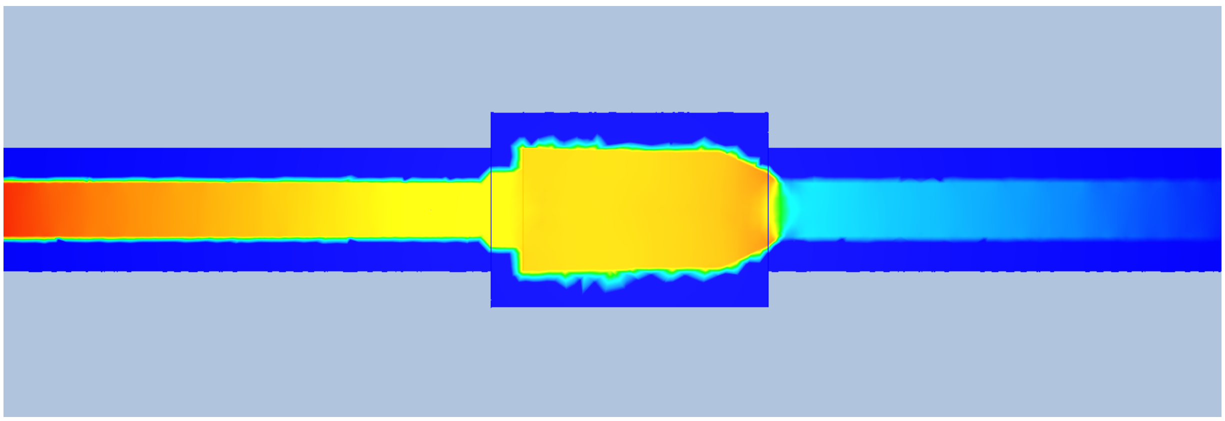

Figure 48. Results of initial CFD

We then experimented with simulations with transient boundary conditions such as varying the flow rate at a boundary. **Figure 49** displays the function used to define how the flow rate would change with time. We defined the flowrate as a ramp function that would step up from 0 to 140 GPM in increments of 10 GPM. The ramp function had a settling time of 30 seconds for each flowrate and would step up to next increment over 5 seconds.

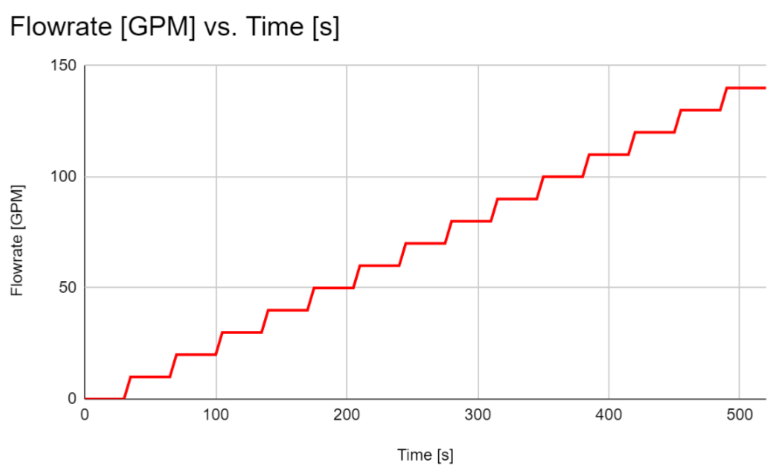

Figure 49. Flow rate ramp function

However, these models proved to be difficult to stabilize and would consistently produce highly inaccurate data. For this reason, reports from these simulations were omitted from this report.

As our testing operations were suspended at the later stages of design, we relied more heavily on simulation data to drive design choices and analyze our current designs. **Figure 50** and **Figure 51** show the pressure gradient developed across our final prototype model in static configurations. **Figure 50** has the poppet disks placed at 45° offset from the sealing surface. The pressure loss developed across the valve in this simulation was 10 psi at a flowrate of 100 GPM. For the simulations of our final prototype, 3D models with reduced geometric complexity were used to assist the simulation software. Changes include removing O-rings, seal grooves, pins and pin holes. The changes made are fairly minor and should not have a significant impact on the flow simulation results.

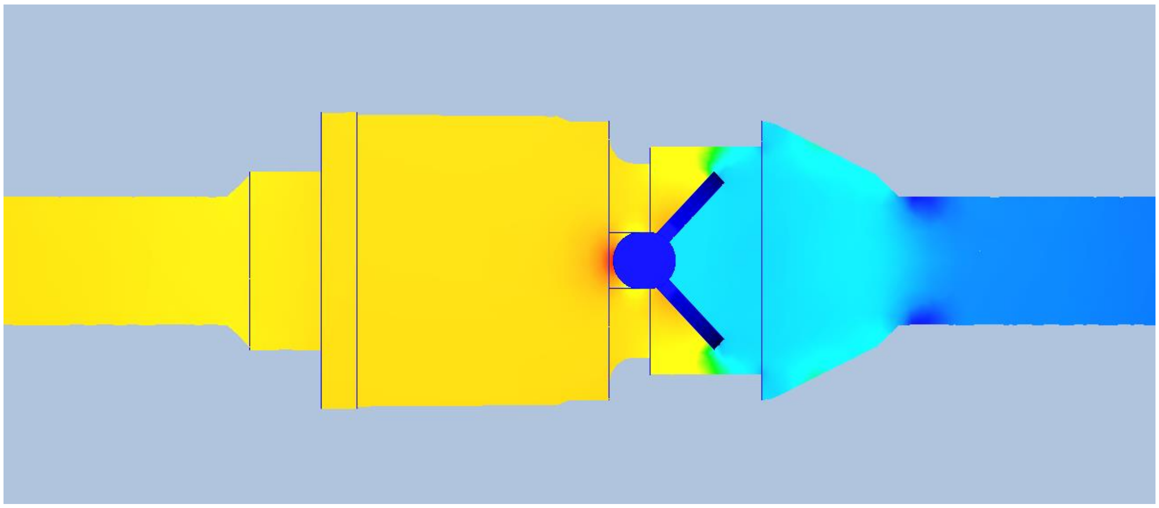

Figure 50. V6 45° static pressure gradient

Figure 51 shows the same model but with the poppet disks 90° offset from the sealing surface. The pressure loss developed across the valve in this simulation was 3.7 psi at a flowrate of 100GPM. This reduction in pressure loss was in line with our predictions, as in the first case the flow is more restricted. This case would also represent a lower bound for pressure loss of simulations without springs present and at specified flow conditions.

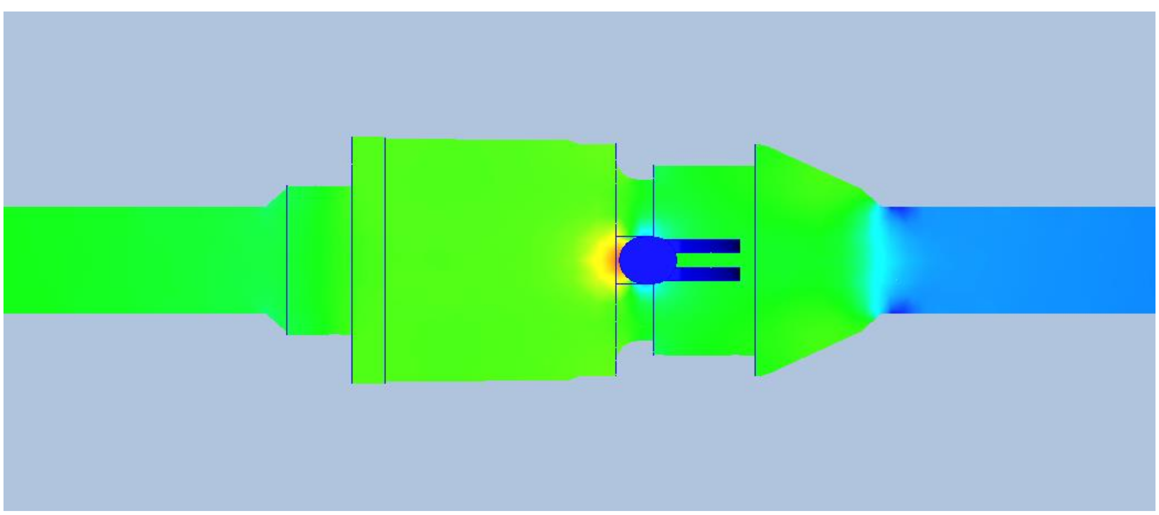

Figure 51. V6 90° static pressure gradient

The simulation shown in **Figure 52** is a transient model and has flow driven angular motion-defined for the poppets. The motion of flow-driven objects is influenced by the flow as well as user specified resistive forces to oppose the motion. A torsional spring was modeled as a resistive element with the appropriate parameters defined. The pressure loss developed across the valve in this transient simulation was 4.7 psi at a flowrate of 100GPM.

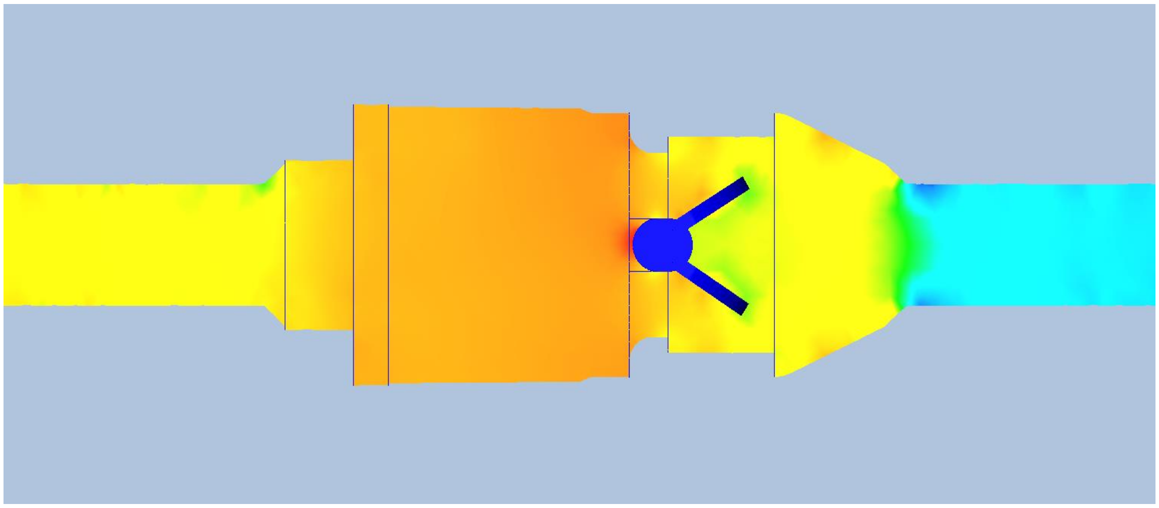

Figure 52. V6 motion elements pressure gradient

The pressure loss of the simulation was ~20% higher than test data collected later. This difference could be attributed to a variety of parameters. However, it was able to predict the performance of the prototype to an acceptable level. The complete report for all simulations discussed can be found in **Appendix O – CFD Results**.

7.6 Proposed Final Design

Unfortunately, the final design proposed by the team was not able to be manufactured or be tested due to the COVID-19 safety limitations in place at this time. The final prototype built and tested by our sponsor (as described in Section **7.3 Final Prototype Design**) was similar to our final design, however it lacked some of the features due to our team's inability to modify the backflow housing and procure the proper spring. The design described in this section is the result of our prior testing, CFD analysis, and iterations for the double-disk check valves. As mentioned before, the main advantage of this design is the reduction of cross-sectional area of the poppets as the valve opens.

7.6.1 Features of the Final Design

A render of the final design can be seen in **Figure 53**. The design includes two identical check assemblies separated by a spacer. The double check housing is modified to include a step which constrains the second check, check #2 (yellow, on the right), while the spacer (purple) acts a sealing surface and stop for the upstream check, check #1(yellow, on the left).

Figure 53. Final proposed design

Similar to the current Zurn design, our proposed design requires some assembly of the check valves outside of the housing. Since the check assemblies are identical, either one can be placed in the valve in any order. The torsion springs will compress the poppet disks flat, but unlike the prototype the disks do not need to be folded. This is because the poppet seats are integrated onto the check hinge. The check assembly is shown in **Figure 54.**

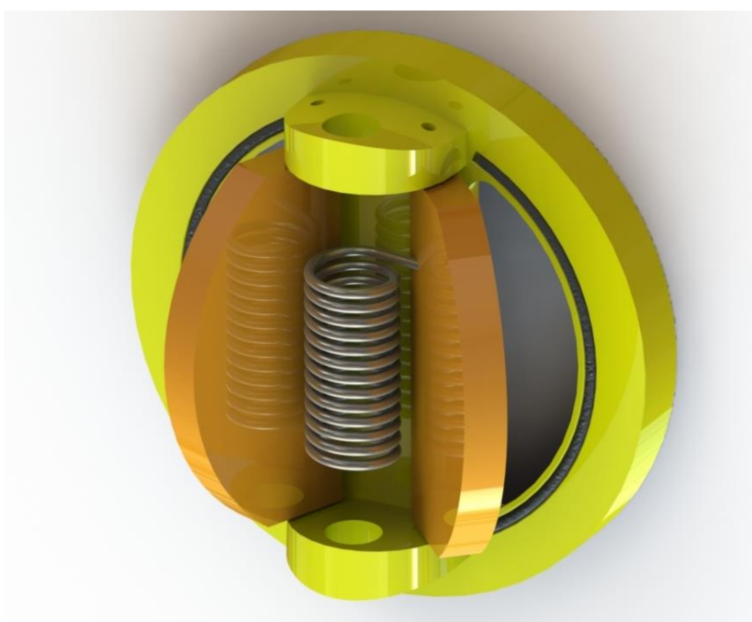

Figure 54. Proposed double-disk check final design

There are O-ring grooves in the step on the housing and on the end of the spacer for sealing. The seals are depicted in **Figure 55** and **Figure 56**. These utilize the forward pressure of the fluid and check hinges to compress the O-rings. Per standard procedures, grease should also be used.

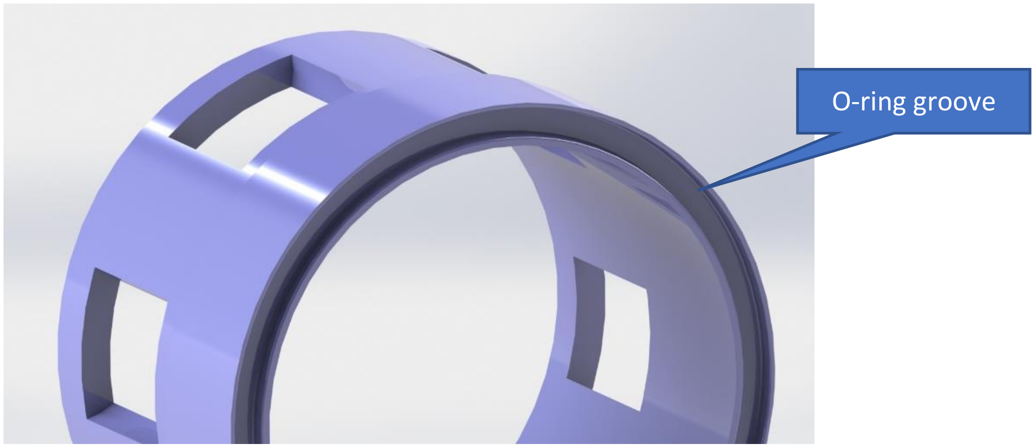

Figure 55. Groove detail on check spacer

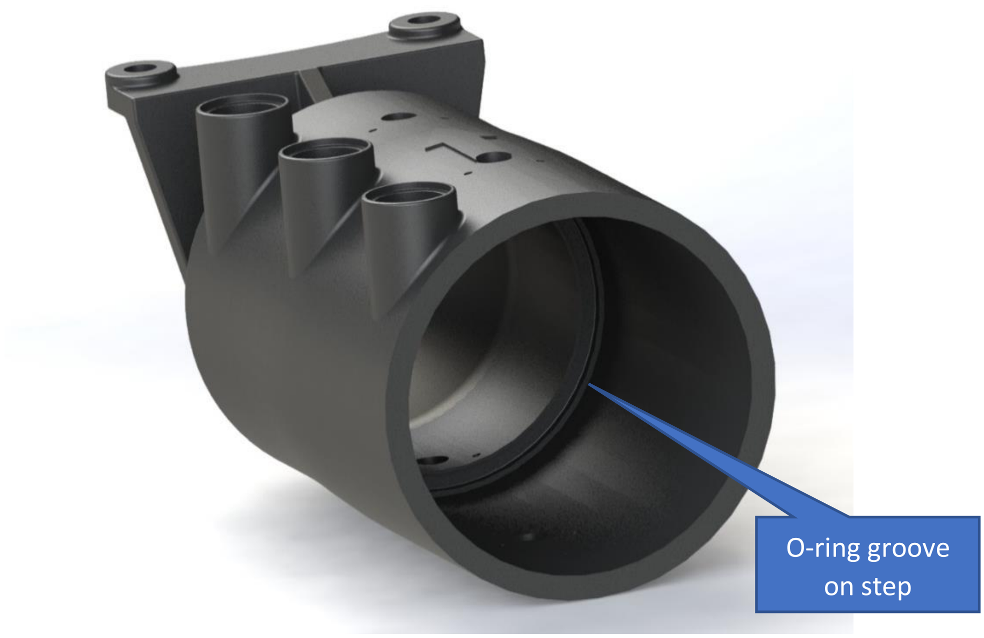

Figure 56. Groove detail on modified check housing

One of the major new features of this design is that the check housing used in the prototypes was eliminated. Each check valve is now held in place by three pins: a 3/8” stainless pin (1132) which also constrains the torsion spring, and two 3/16” diameter stainless pins which act as the pins constraining the poppet disks. This fixes the issue of aligning the checks so that the hinges are vertical, and more importantly, opens the cross-sectional area to allow for more flow through the double check assembly. The pins are threaded on the end and thread into the bottom of the check assembly. The pins and threads are shown in **Figure 57** and **Figure 58**, respectively.


Figure 57. Threaded pin

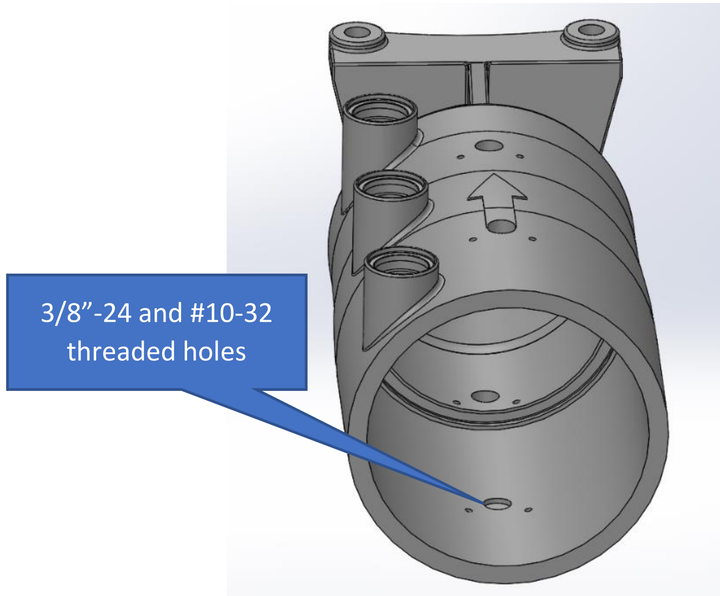

Figure 58. Housing with threaded holes

The reason for this change was largely driven by the design of the torsion spring. When designing the torsion spring, the team found that to meet industry standards the check valve must be able to withstand 100,000 cycles of open to close. This number comes from communication with an experienced engineer at Zurn Wilkins and is specific to the lifetime cycling that would be expected in relatively small residential water lines and schools. The 100,000-cycle number was used for torsion spring design and exceeds the number of cycles required by most testing standards. USC test standard 10.1.2.3.3.8 (10^{th} ed.) is outlines the requirements for evaluating performance for specified cycles. There are 4 test cases, each requiring 1,250 cycles for a total of 5,000 cycles required to meet this test requirement. Since Zurn Wilkins would like to maintain designs that far exceed the industry requirements, our team decided to go with the 100,000-cycle number. This resulted in a spring diameter that was unavoidably large, around 0.5-0.6” in diameter.

The larger spring diameter forced the check hinge to be wider, increasing the aspect ratio of the valve significantly. The advantage of the double-disk lies in its ability to reduce the aspect ratio of the valve as it opens, causing us to integrate the checks into a larger housing. The final torsion spring was custom designed to be a 210-degree angle, left-hand wound, 0.585" OD, 0.070" wire diameter, 0.5" leg length, with 13.083 coils.

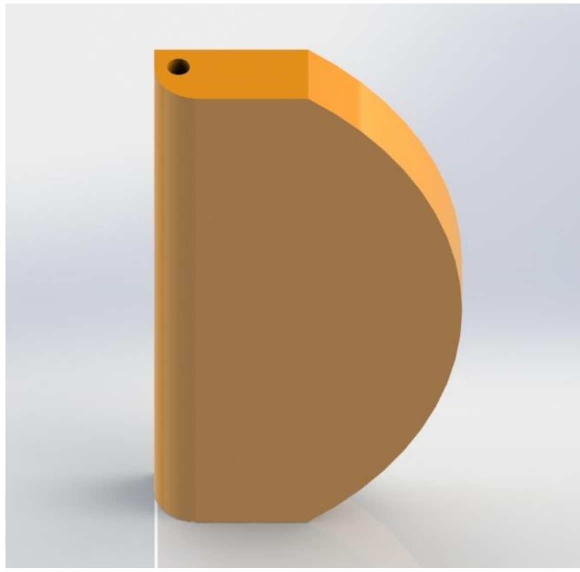

Figure 59. Asymmetrical Poppet

The poppets in this design utilize an asymmetrical shape, as seen in **Figure 59**. This solution was adopted by the team to thicken the disks without causing them to interfere with the larger spring. This also helps give them more structural integrity. **Figure 60** demonstrates how the thicker side of the disk seals onto the poppet seat, while the thinner side faces the torsion spring.

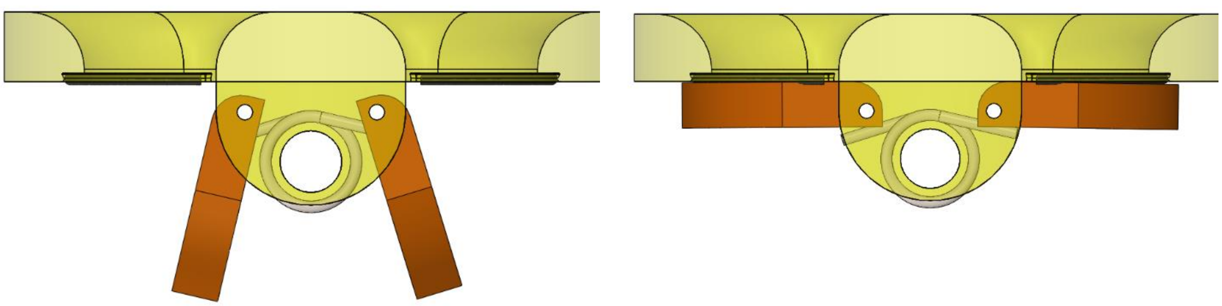

Figure 60. Top view of asymmetrical poppet disks

An additional feature of this design is the implementation of dovetail grooves to contain the poppet seat seals, referred to as the poppet seal arcs due to their shape. The dovetail shape of the groove can be seen in **Figure 61**, and is used in this location because the seals are on the downstream side of the valve. The dovetail helps capture the seals, and more about this sealing method can be found in **Section**

7.8 Additional Research – Sealing Methods**.**

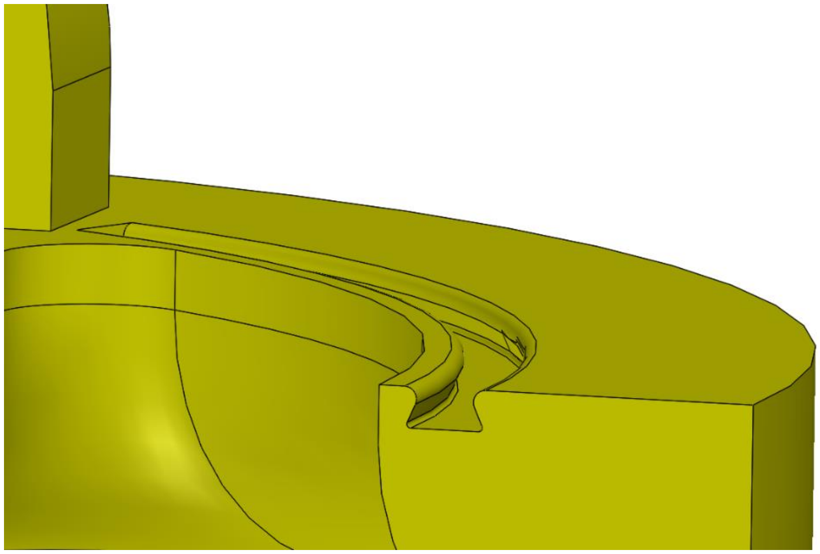

Figure 61. Detail view of dovetail groove

7.6.2 Further Design Refinement

There are a number of aspects of this design that the team would suggest further investigation and refinement on. The proposed final design addresses many of the challenges that the team experienced from our iterative testing, but since we were not able to extensively test the sealing valve with the designed torsion spring, there are still many areas we believe will need further development.

- The spacer utilizes cutouts to reduce material usage and to allow water to flow out of the test port. We do not know the effect of these windows on the pressure loss of the valve, and the material could be reduced further. Better yet, the spacer could be eliminated if modifications to the check housing allow for proper sealing of check #1.

- The sealing on the pins at the top of the valve could use further development. The team would suggest some sort of cap or gasket but has not modeled a sealing mechanism in the final design.

- The constraint of the torsion spring could be improved. The 3/8" pin keeps it in place but does not align it vertically in the valve. Early in the design process the team tried integrating the ends of the torsion springs into the poppet disks themselves, as seen in **Figure 62**, but this lead to issues such as increase friction between the spring and poppet disk, as well as weakening of the poppets themselves.

- Improved sealing between the poppet disks and hinge. The proposed design seals this region only with contact. A few ideas the team has come up with to address this include adding a sealing rubber to the back side (thicker) of the poppet, or to apply some sort of flexible seal around both the upstream side of the hinge and the poppets. Further development of this sealing surface is necessary.

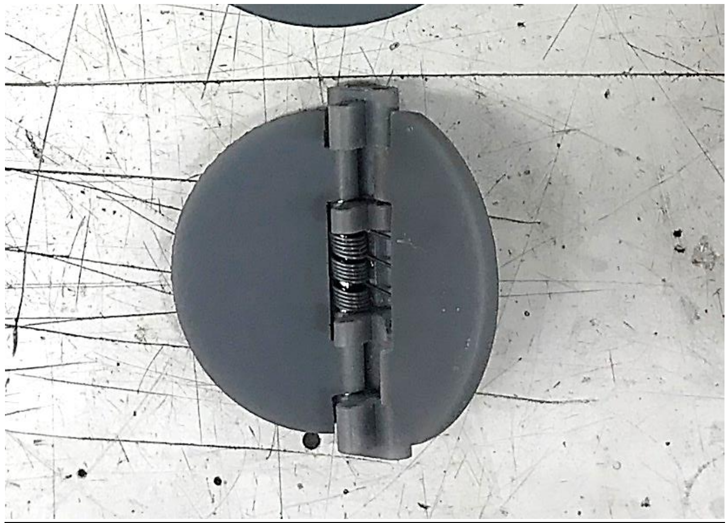

Figure 62. Old prototype with spring ends inserted within the disks

7.6.3 Final Design Manufacturing

The final design for this project was not manufactured, however the team has developed a manufacturing plan for commercial scale manufacturing of the proposed design. The pins and O-rings will be purchased and modified. Many components will still need to be custom made, these would be through plastic injection-molding and some post-machining if necessary. The springs will also need to be custom manufactured. See Section **6.4 Outsourcing** for more details.

All parts to be purchased or manufactured are assigned unique part numbers which are listed in **Appendix G – Indented Bill of Materials**. These may be referenced when discussing each part. The components that require custom manufacturing are the check hinge (1111), poppet arc seals (1114), backflow housing (1100), poppet disks (1113), and the stainless pins (1131 & 1132). These parts require varying degrees of manufacturing.

- The poppet arc seals (1114) will be manufactured by cutting a commercially available O-ring to arc lengths that fit within the poppet sealing grooves. These would be considered semi-custom since they take commercially available products and modify them.

- The backflow housing (1100) will be a modification of the current backflow housing used in the 350 XL backflow preventer assembly. This is Zurn part # 354-1A. A step will be bored out of the inflow end of the housing using a CNC mill, and an O-ring groove will be machined into this step.

- The stainless pins are made of stock 3/16" (1131) and 3/8" (1132) stainless steel rod. These will be cut down to length, and threads will be made on the end of each pin using a thread die. The fine threads (UNF) for each respective pin size should be used.

The remaining components are made of plastic and require injection molding and post-machining. These will all be made of Delrin 150. This material was selected because it is exhibits low moisture absorption, non-corrosive properties in water, and is easy to use for both casting and machining. Delrin is also used by Zurn Wilkins for many components in the current double check backflow assembly.

The manufacturing process used for the plastic components is heavily dependent on the final part geometries. The two methods that the team originally considered to manufacture these parts are as follows:

1. **Plastic Injection Molding.** This method required the manufacturing of dies and/or molds for each of the parts. In addition to the extra tooling, the process will also require some amount of post-machining for the grooves, holes, and tight tolerance surfaces. A pro of this method is that it is similar to that used in industry for large scale manufacturing runs.
2. **CNC Machining.** In this method the parts would be directly machined out of pieces of Delrin bar stock. Due to the unusual geometries of these pieces aluminum soft jaws would be required to fixture the parts in the machine. This would add to the overall cost of this manufacturing method; however, the parts would require little to no post-machining and would have repeatable tolerances.

After further discussion and consideration, the team decided that the best method of manufacturing a final prototype would be CNC machining the components out of stock Delrin. This method is only suggested for small scale runs and prototyping purposes, such as the final valve (were the team to manufacture it). This was determined by the reasoning that the complex geometries needed for plastic injection molds would likely need to be CNC machined. Additionally, injection molded parts often require post-processing that may include machining anyway.

Unfortunately, the team did not have the ability to manufacture a final prototype, and instead proposes making the remaining custom components (1111, 1113, 1140) with a plastic injection molding process, as would be common in commercial scale manufacturing. A complete list of the part numbers and their manufacturing methods are described in **Table 8**.

Table 8. Overview of Part Manufacturing Processes

Part Number	Part Name	Mfg. Process
1000	Final Assembly	-
1100	Modified Backflow Housing [354-1A]	Provided by Zurn, modified with CNC mill
1110	Double-disk Check Assembly	-
1111	Check Hinge	Injection molded, post-machined on CNC mill
1112	Torsion Spring	Outsourced
1113	Poppet Disk	Injection molded, post-machined on CNC mill
1114	Poppet O-ring Arcs Seals [040N]	Purchased, cut to length
1120	Check O-ring Seals [273N]	Purchased
1130	Stainless Pins	-
1131	3/16 Pin	Purchased, cut to length, cut with thread die
1132	3/8 Pin	Purchased, cut to length, cut with thread die
1140	Check Spacer	Injection molded

7.8 Additional Research – Sealing Methods

Due to the shutdown of all in-person testing capabilities as of mid-March 2020, our team was not able to successfully finalize the design parameter for functional valve sealing. With that said, however, our team has conducted additional research on interface sealing methods and seal design. As well, our team has even integrated some early applications of these concepts into our final prototype design (**See 7.6 Proposed Final Design)**.

The following section explores why sealing has proven difficult for the double disk design, the takeaways from the difficulties encountered during this project, and a summary of the seal design methods encountered during our research.

7.8.1 Difficulties of Double-disk Sealing

Figure 63 is a close-up of the valve seat and poppet sub-assembly in the closed position. Unlike the traditional inline check valve in the 350 XL, there is no direct fastener, such as a washer and bolt, that clamps the seat seal into position. This means that with the repeated contact force that the valve seat experiences from the poppets opening and closing during normal operation, the seal may degrade prematurely or become dislodge from its cavity.

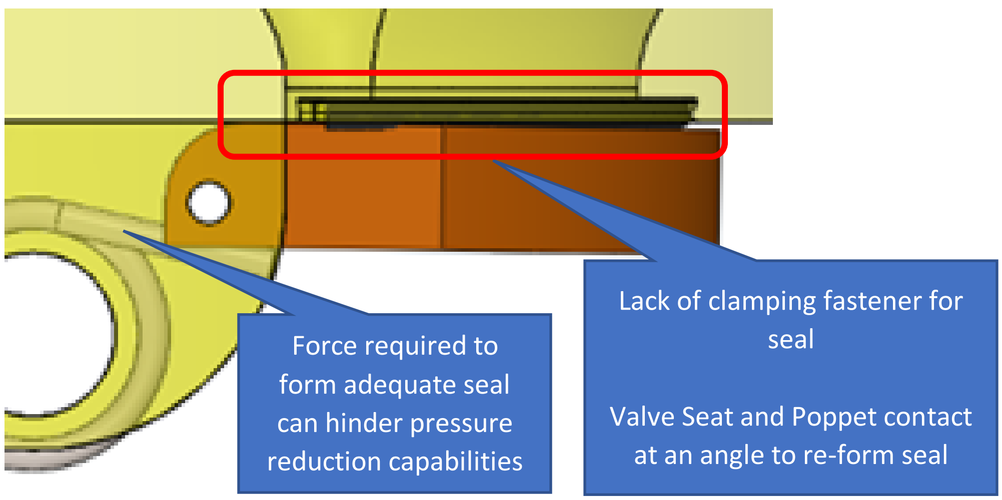

*Figure 63. Close-Up of Sealing interface of Poppets. Adapted from **Figure 60.***

As well, the double disk re-forms its seat seal in a “zippering” method, where one point of initial contact between the poppet and seat grows, eventually sealing the entire perimeter of the seat opening. The 350 XL inline instead re-forms its seal almost instantaneously, as there is no angled contact between the seat and poppet **(See Figure 64)**.

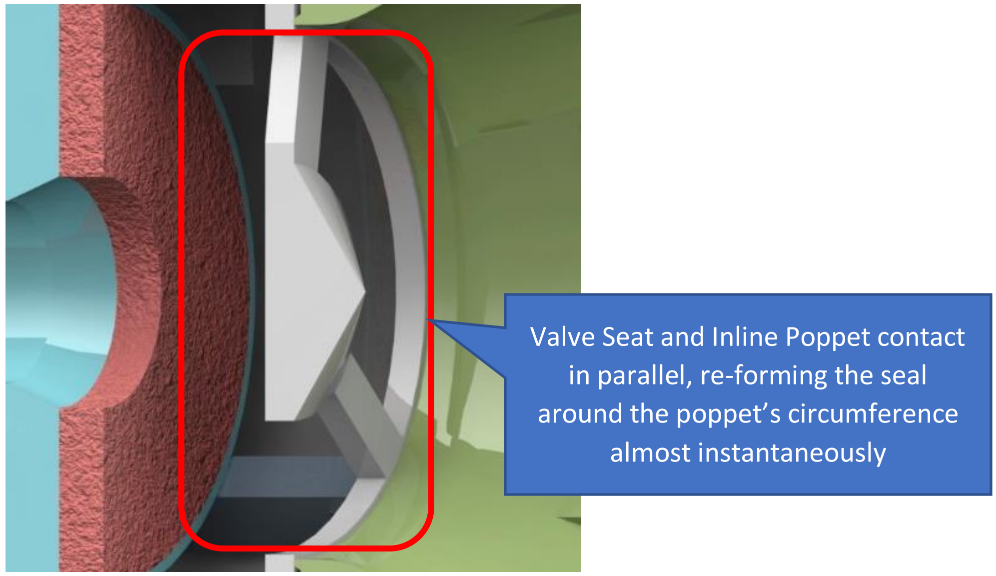

Figure 64. Close-Up Illustrating Inline Check's Flat Sealing Surface. Adapted from ***Figure 23.***

Below is a list of main concerns that a new seal design for the double-disk must address:

- Seal must have enough clamping force for the valve seat and poppet to form a complete seal while considering the wearing down of the seal due to repetitive open-close cycles.
- Seal must be located and secured in a manner that requires no direct mechanical or chemical fastener (Such as washers, pins, or glue/adhesive)
- The seal must be compliant enough for the poppet to seal against the valve seat completely, but rigid enough to not "slip" or be bumped out of its groove

The following subsections attempt to provide recommendations on how to correct these known issues of designing sealing for the final double-disk design.

7.8.2 Adjustable/Serviceable Seal Design

While researching methods to correct for the degradation of the valve seat seal due to operating conditions, the team found a design made by PBM Valve Solutions attempts to solve the issue of seal compression and degradation by taking advantage of regular servicing intervals already required of check valves. The solution, called the PBM Adjust-O-Seal®, uses bi-directional sealing in a ball check valve, with regular tightening of valve body/housing bolts to account for seal wear **(See Figure 65)**. [*PBM Three Piece Sanitary Ball Valve*]

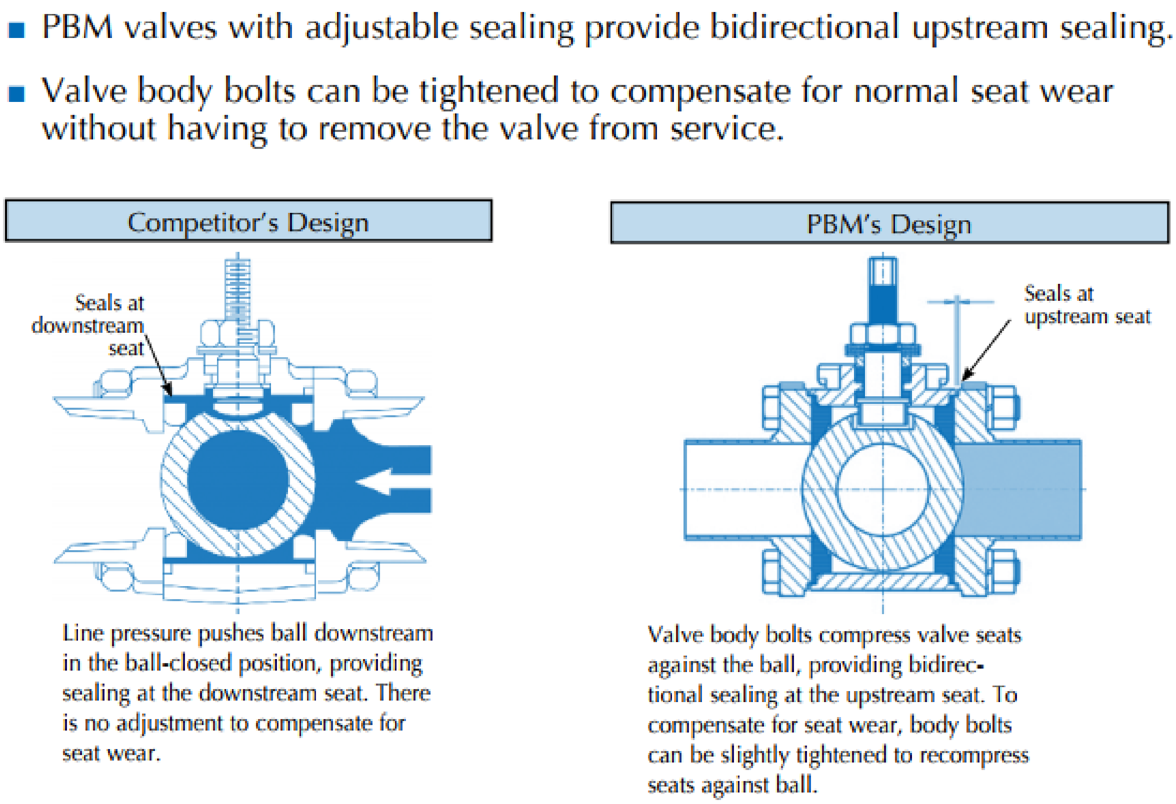

Figure 65. PBM Adjust-O-Seal Example. Adapted from PBM Valve Solutions

This method of sealing the check valve, despite being used with a ball check, can be translated over to the double-disk by using fasteners that can decrease the distance between the valve seat and axis of rotation of the poppets. While not currently integrated into the final prototype, use of a rack-and-pinion, or rotational-to-translational motion assembly can be used to re-tighten the seal interface. This may not be a large departure from the prototype's current configuration, as our team is already making modifications to the outer geometry of the valve housing, as evidenced in **Figure 66.**

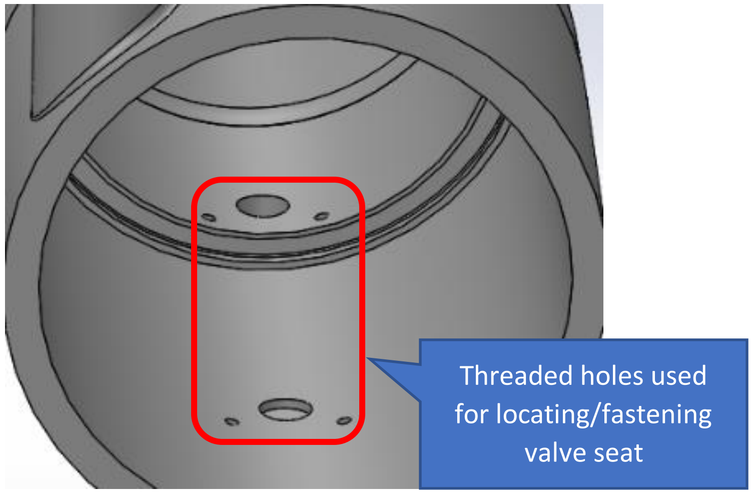

Figure 66. Close-Up Illustrating Threaded Pin Holes. Adapted from ***Figure 58.***

7.8.3 Types of Seals and Integration into Final Prototype

In order to address the complication of keeping the seal seated and avoid being knocked out of its groove due to poppet movement, a substitute for direct mechanical fastening (washers and pins) must be presented. Our team proposes that certain seals with self-locking or self-adhering properties be used. Two examples of these types of seals include a self-gripping seal, as seen is **Figure 67,** and the use of a dovetail gland to secure an O-ring or other gasket material, suggested in **Figure 68.**

The design advantage of the self-gripping seal is its wide commercial availability and minimal need for design modification. All that is generally required of the self-gripping seal is additional geometry, such as a raised lip or edge where the U-shaped gasket material can then interlock with ridges onto it. The cylindrical gasket material can then act as a sealing interface.

Some disadvantages of the self-gripping seal for our specific application are as follows:

- The air-filled gasket material may inhibit proper sealing if too little pressure is applied to the valve seat by the poppet. Reducing the volume of air or sourcing a seal with a solid sealing surface may alleviate the additional compliance caused by the air pocket.
- Unless the geometry which the self-gripping seal is attached around imparts extreme tension on the seal, or produces a large amount of friction for adhering the seal, there is a possibility that a surge of fluid within the valve can produce an updraft on the lower edge of the seal, effectively lifting it off and around the gripping surface.
- Dependent upon the cycling caused by the seal due to opening and closing, the connection of the cylindrical seal material and the U-shaped gasket may rupture. Selecting a gasket material that balances rigidity with tear resistance may alleviate this issue.

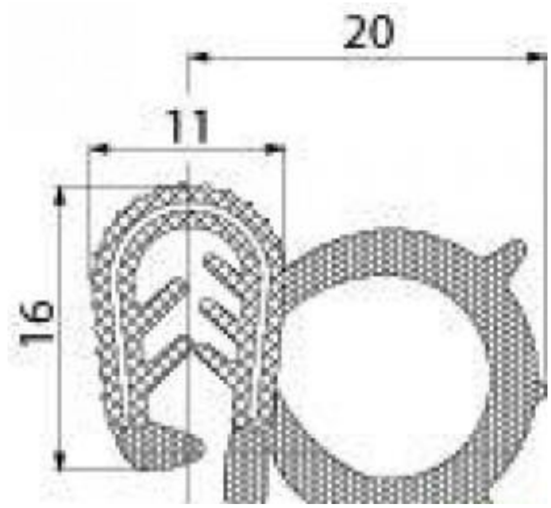

Figure 67. Cross-Section of Self-Gripping Seal /Gasket

The dovetail gland seal traditionally uses a circular cross-section O-ring as the media that produces the seal itself, rather, the way the O-ring is secured within the part's geometry is the unique design application of the dovetail seal. By relying on the frictional constraint that the dovetail gland produces, there is virtually no way (except for extreme pressure and force conditions) where the O-ring is extruded from the gland. In the case of check valve use, where there is no solid-on-solid sliding movement on the O-ring, the chances for seal extrusion (the event of the O-ring being forced and disformed out of its gland) is extremely low.

However, some disadvantages do present themselves with using a dovetail gland seal:

- The dimensioning of the gland and material selection of the O-ring is critical, as inadequate tolerancing or poor material selection may lead to sealing failure. (Note the small amount of O-ring material protruding above the gland in **Figure 68** to effectively seal the interface).
- Since the dovetail seal utilizes a traditional O-ring, the same points of failure for a standard O-ring seal apply to a dovetail seal. As a dynamic seal in a fluid, which is what would be used to seal the valve seat and poppets, excessive moisture absorption and ensuring the seal is not squeezed by the poppet are the highest-risk points of failure.
- Since the dovetail gland requires an undercut, there is a possibility that advanced manufacturing efforts are needed to produce the continuous geometry required for the gland. Using a parting line along the gland would introduce too high of a risk of water-tightness failure of the gland, so a casting or molding process must be used that allows for undercuts.

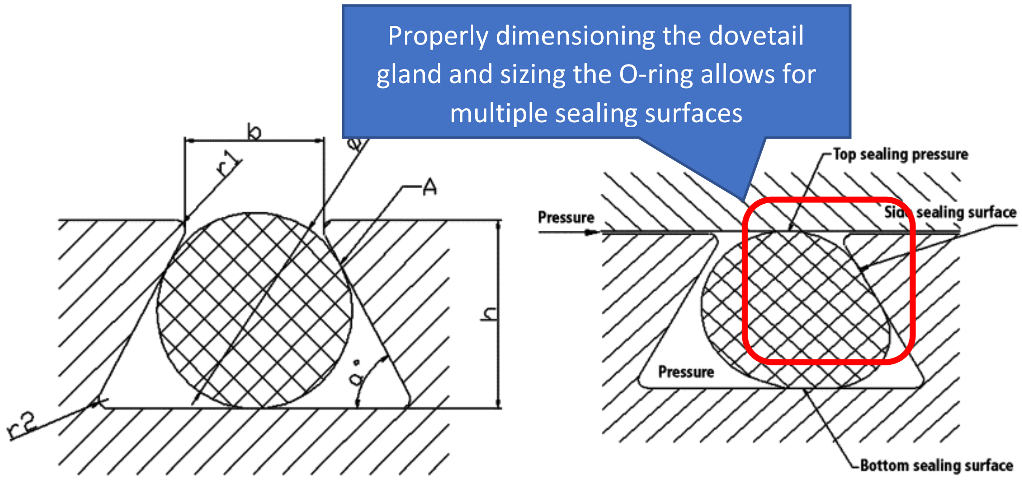

Figure 68. Cross-Section of Dovetail Gland with O-Ring Seal

Despite there being many more options for a seal that would serve as a potential contender for being used in the double-disk's final design, our team elected to use the dovetail gland seal in our final prototype, portrayed in **Figure 61.** We justified this design choice by emphasizing the dovetail seal's ability to withstand significant amounts of pressure and compressive force before seal failure. Additionally, the wide commercial availability of O-rings and circular cross-section gaskets is beneficial in selecting the appropriate material and size of the sealing material. While the current configuration of the dovetail seal in the final prototype is an early adoption of the design, further refinement of the seal must take place to endure the sealing is effective during backpressure testing. Some of the current limitations of the dovetail seal in the Final Prototype are listed:

- The sectioned nature of the dovetail gland on the valve seat of the final prototype means that a stock O-ring cannot be directly used in the design. Instead, a specific circumference of the O-ring must be cut during assembly or ordered as a custom component from a manufacturer.
- It has yet to be confirmed with Zurn if an undercut feature included in the dovetail grand can be made on the valve seat in mass-scale manufacturing. If this feature cannot be readily manufactured with current manufacturing and assembly techniques implemented by Zurn, the cost and time involved in modifying machinery and /or manufacture-provided components should be considered.

7.8.4 Seal Deformation and Compression

As evidenced by discussing the disadvantages of the dovetail design, even a well-implemented sealing design method can have its drawbacks. The highest-risk failure point of a dovetail seal used in a dynamic compressive force, fluid-filled environment, is seal deformation.

Seal deformation can be described as any effect that the environment may have on the sealing element so as to cause it to deform or move in a way where sealing failure can result. **Figure 69** is adapted from Apple Rubber Product's Seal Design Guide and illustrates that the most common mode of O-ring failure in terms of seal deformation is compression set [*Seal Design Guide, 2009*]. Causes of this type of failure can stem from, but not limited to, the following:

- Excessive swelling caused by fluid absorption in a fluid-filled working environment
- Excessive pressure applied to the seal in a dynamic force setting (repetitive opening and closing of the poppet in the double-disk check valve)

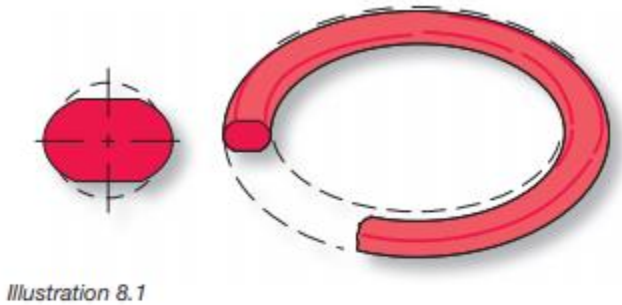

Figure 69. Representation of Compression Set. Adapted from Apple Rubber Products.

In order to remedy these causes of compression set, it is important to select a seal material that is suitable for permanent fluid submersion and can withstand dynamic forces while still remaining compliant to form an adequate seal at the interface of the valve seat and poppet face. These material attributes help reduce the effect of compression set in the seal. While every commercially-available seal material experiences some amount of compression set, as governed by the ASTM designation D395 test procedure [*Seal Design Guide, 2009*], compression set can at least be minimized using proper material selection for its particular application. Below are some example materials that would satisfy these requirements:

- Nitrile
    - As of Apple Product's 2009 Seal Design Guide writing, nitrile the mostly widely used and economical elastomer for seal manufacturing
    - Nitrile can be used in FDA-regulated applications and low-temperature environments
    - High tensile strength, low compression set, and high resistance to abrasion
- Cast Polyurethane
    - Extremely favorable for its high tensile strength, low compression set, and operating temperature range (with the exception of being subjected to extremely hot water or steam)
    - Used in applications where parts are subjected to continuous or periodic stress, such as wiper seals, shock absorbers, and bumpers.
- Silicone
    - Extremely wide operating temperature range
    - Low compression set and retention of flexibility
    - Compatible with FDA-regulated applications

While not directly applicable to the dovetail seal on the current final prototype of the double-disk, it should be noted briefly that with any situation where a sufficient force is applied to a surface of a seal, there is the potential for seal extrusion to occur. **Figure 70** provides a range of extrusion limits for different material hardness ratings. Even at a hardness of 40 Shore A, which is the hardness rating of some common gasket and seal materials for valves and manifolds, the pressure imparted on the valve seat by the water or force of the poppet does not pose a threat of extrusion.

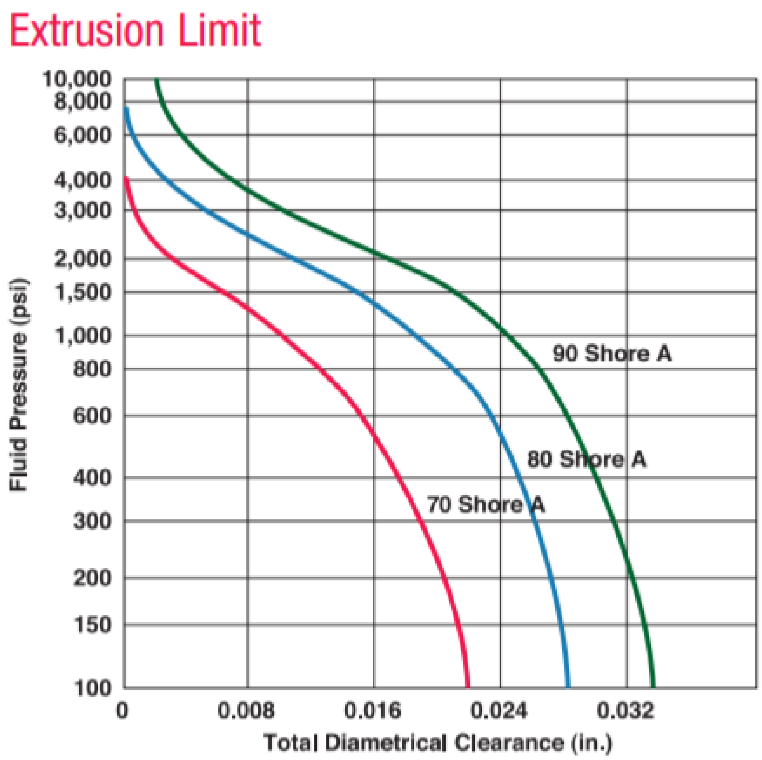

Figure 70. Extrusion Limits of Various Hardness O-Rings

Since the sealing capability of the dovetail seal is dependent upon a well-selected seal material and size, the careful design and tolerancing of the dovetail gland itself is tantamount. **Figure 71** is the table provided by Apple Rubber Products to effectively design a dovetail gland according to the size of the O-ring intended to be used. Note that a combination of symmetrical and unilateral tolerances are provided for the various dimensions of the dovetail gland.

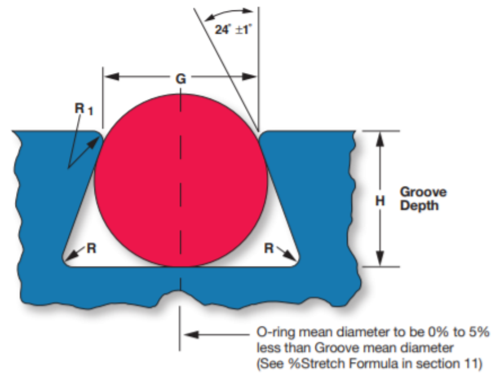

Table E Dovetail Gland Dimensions

AS-568* Number	O-Ring Cross Section				G Groove Width				H Groove Depth		R_1 Radius		R Radius	
					Sharp Edge		Rounded Edge							
	In.	±	mm	±	In. ±.002	mm ±.05	In. ±.002	mm ±.05	In. +.000 -.002	mm +.00 -.05	In.	mm	In.	mm
004 - 050	.070	.003	1.78	.08	.057	1.45	.063	1.60	.052	1.32	.005	.13	1/64	.40
102 - 178	.103	.003	2.62	.08	.085	2.16	.090	2.29	.083	2.11	.010	.25	1/64	.40
201 - 284	.139	.004	3.53	.10	.115	2.92	.120	3.05	.115	2.92	.010	.25	1/32	.79
309 - 395	.210	.005	5.33	.13	.160	4.06	.170	4.32	.180	4.57	.015	.38	1/32	.79
425 - 475	.275	.006	6.99	.15	.220	5.59	.235	5.97	.234	5.94	.015	.38	1/16	1.59

*Note: The current revision of the Standard is "C" but it changes periodically.

Figure 71. Dovetail Glan Dimensioning Guide. Adapted from Apple Rubber Products

In order for a dovetail seal to work effectively as a compression-type seal, the amount of material protruding above the grove depth (Dimension H in **Figure 71**) must be sufficient so that the seal is not compressed below the opening of the groove during the point of maximum pressure application that is expected in operating conditions. In order to determine this maximum amount of seal compression, Apple Rubber Products provides seal compression calculations using simple equations and a graph-lookup method **(See – Seal Design Charts: Compression Calculations & Rules of Thumb (Adapted from Apple Rubber Products)**.

In short, in order to overcome the inherent complications of using a dovetail seal as a sealing solution for the double-disk, the following considerations must be made:

- Ensure that an adequate material for the sealing media (O-ring or gasket) is used
    - Low compression set and resistance to fluid-induced swelling
    - Wide operating temperature range and rated for potable water use
- Dimension and tolerance the dovetail gland to ensure a proper fit of the sealing media, and allow for compression sealing during maximum pressure application
    - Perform seal compression calculations to determine maximum compression set
    - Ensure that the height of the seal material protruding from the seal gland is more than that of the material's maximum compression set

8.0 Project Management

To ensure that the team provides deliverables on-time and consistent of good quality, well-formed project management is a must. The project team has made several accommodations to help in achieve good project oversight, including the listing of key deliverables in a Gantt Chart (**Appendix Q – Gantt Chart – Updated as of 5/31/2020**) that details key actions up until June 4th, which is the scheduled submission date for final deliverables, including the Final Design Report.

Due to the significant changes of our project, this chapter will compare the originally intended plan for finalizing the project versus the adapted plan for project completion. As well, our team reflects on what aspects of the project management were helpful, and what aspects would prove beneficial in improving.

8.1 Post CDR Approach – Meeting Engineering Specifications

Since our team had selected a final design direction, our priority post-CDR was to shift our focus to meeting the engineering specifications listed in the Pugh Matrix (**Appendix F – Revisited Final Design Decision Matrix (Pugh Matrix))** as best as possible.

Originally, out team had planned to progress the design the double-disk and perform 3 to 4 iterative design-test cycle between the submission of CDR and the project deadline. This would allow us to take full advantage of Zurn's in-house rapid prototyping and testing facilities. By the end of the last design-test iteration, we had anticipated to have made significant progress on, or completed, the engineering specifications of pressure loss reduction, and pressure differential holding (backpressure testing), highlighted in red in **Table 9.**

Table 9. Expected Actions to Satisfy Engineering Specifications

Engineering Specification (Per Pugh Matrix – Appendix F)	Intended Modification(s) to Design	Date Expected
Reduce Pressure Loss	Spring/Buckling Spring Subsystem	5/15/2020
Maintain Pressure Differential	Sealing Subsystem	5/8/2020
Mechanically Driven	N/A	N/A
Reduce # of Components	N/A	N/A
Ease of Design Scalability	Modify Entry/Exit/Frame Geometry	2/29/2020 COMPLETED

Instead, with the time and limited resources available, our team focused on improving these two criteria as much as possible with only one design-test cycle. After the final prototype had been finalized in CAD, extensive coordination was required to send print files to Zurn and rely upon Mr. Yale and other Zurn engineers to remotely print, assemble, test, and record data. After the data Mr. Yale recorded for the final test run was sent to us, we were able to confirm that the most recent version of the double-disk proved capable of reducing pressure drop across a wide range of flow rates, comparative to Zurn's existing 350 XL inline check valve.

While our final design-test iteration was not able to confirm all of the engineering specifications laid out for this project, including backflow prevention, significant progress was made in providing quantitative evidence that pressure loss reduction is possible with the double-disk.

8.2 Completed Deliverables – Modified

Table 10 details the project deliverables that were agreed upon after considering the impacts of COVID-19. The purpose of these deliverables was, despite not producing a fully functioning prototype, to provide Zurn with a solid proof-of-concept that can be further expanded and improved upon to progress towards a functioning product.

Table 10. Modified Key Deliverables of Project

Key Deliverables
Final Design Report
All 3D models (CAD) associated with final double-disk version
CFD results and analysis (in FDR)
Test data collected during physical "wet" testing (in FDR)
Recommendations for future improvements of double-disk prototype, such as sealing, backpressure testing, and manufacturing (in FDR)

8.3 Reflection of Project Management Strategies

In general, a long-term project will allow team members to identify weaknesses, strengths, and areas for improvement in the dynamics of the team. With that said, a successful project includes the identification of those targets for improvement and acting on them, rather than simply playing a game of balance between strengths and weaknesses. Our project demanded a considerable dedication of time and skill from each team member for the entire duration of the 9-month project.

What did appear to work well was our team's interest in moving the project along at a rapid pace, and becoming comfortable with making major design changes, so long as there is sound judgement to do so. Sometimes, making these large changes in a plan, especially later on in the design process, can be daunting. One aspect of this project that tested our comfort in making such changes was the nature of our test results feeding back into the re-design process; if our test data were to show a major flaw in our current design version, or even our entire design foundation, we would be forced to change course regardless of the progress already made "down the wrong road." So long as we worked diligently and delegating tasks to each other, our team has had to catch back up to speed after making these large changes in design direction.

With a project only bounded by the engineering specification that were to be met with a final design, it was challenging for our team to move past certain parts of the design process for fear of having "left too much out on the table." Whether this took the form of continuously refining modeled components or continuing certain design aspects that proved to be very time intensive with little reward towards accomplishing the engineering specifications, we kept in mind that these can be learning opportunities for the future. This is not to say that a project is expected to go according to plan all the time, but rather, it is important in such open-ended projects like ours to identify scope-creep when it occurs, and for the team to identify what actions and time investments are worth taking given their expected return. As the effects of COVID-19 began to constrict the forward momentum of our design process, our team was forced to re-evaluate our project goals and carefully measure the time and resources available, and whether it would even make sense to pursue all of the originally intended goals for the remainder of the

project. As a result, our team was able to adopt a modulated set of deliverables and project scope with the guidance of Mr. Yale and others at Zurn.

9.0 Conclusions & Recommendations

The original goal of this senior project was to provide Zurn Wilkins with a new check valve design that uses mechanical advantage and fluid dynamic principles to reduce pressure loss comparative to their existing product. Due to the COVID-19 pandemic, the scope of work was altered to accommodate the sudden shift in workflow and the new virtual work environment. We shifted our project to focus on the following: delivering a semi-functional final prototype, providing final test data, and conducting additional support research including CFD analysis.

9.1 Key Results

In this report, our team has provided information regarding the development of our design from the early stages of understanding the problem to discussing our proposed final design. In the last nine months of working on this project, we have had both successes and challenges as a team as well as many lessons learned.

We were successful in establishing an iterative design process. Our design process was built on six primary components: ideate, design, prototype, test, evaluate, and iterate. Using this method, we were able to determine the advantages and disadvantages of various design decisions as well as the level to which they affected overall performance. We had originally planned on a longer design cycle, but this was severely truncated due to the aforementioned COVID-19 pandemic. Despite having this iterative process interrupted, we were able to establish an effective design cycle.

Our design was ultimately successful in reducing the pressure loss in comparison to the existing design. However, the prototype tested was not a complete check valve and lacked proper sealing to provide positive shutoff. This is a key area to improve for our final design.

Having a project team of three individuals had both its benefits and drawbacks. While an additional team member would have provided greater scrutiny and creative input, we were still effective in operating as a cohesive group. Having a smaller team naturally led to us having to specialize in different areas of the project, which proved to be a challenge to manage in the virtual work environment. This created an environment in which we had to rely on and trust one another. Effective communication and team meetings were essential in keeping us all on the same page and moving in the same direction.

Despite the setbacks created by the COVID-19 pandemic, we were still able to deliver a semi-functional final prototype, provide final test data, and conduct additional support research.

9.2 Future Recommendations

In the final stage of development, the final design still lacks some features and does not meet all the necessary design requirements for a complete check valve. We recommend continuing the iterative design process with our proposed final design in order to push the threshold for minimizing pressure loss. We also recommend that, in order to function as a proper check valve, refining the design so that it seals properly and provides positive shutoff.

Despite the setbacks encountered, we believe that the work we have completed serves as a strong foundation and proof-of-concept design that Zurn Wilkins' engineers will be able to build on in order to integrate the double disk into an operational check valve for water supply lines.

9.3 Acknowledgments

We would like to express our appreciation for providing such an excellent learning opportunity. We would like to thank our sponsor Zurn Wilkins as well as Brian Yale, Reuben Westmoreland, and Chris Corral for support in this project. We would also like to thank our senior project advisor Sarah Harding for guiding us through this process and being a pillar of support.

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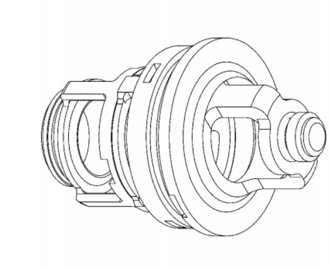
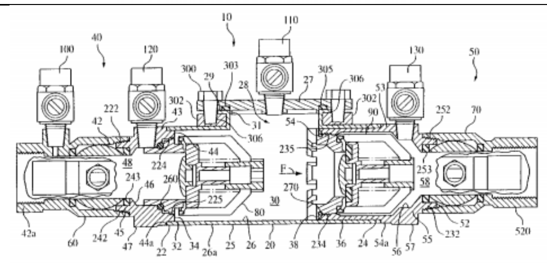
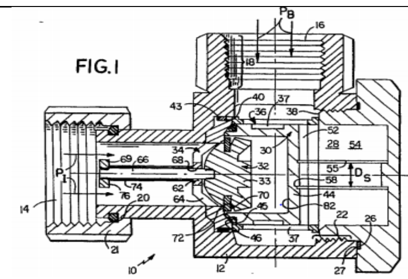
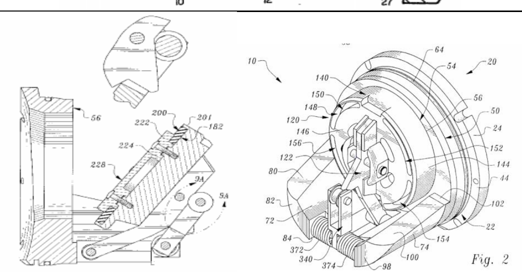
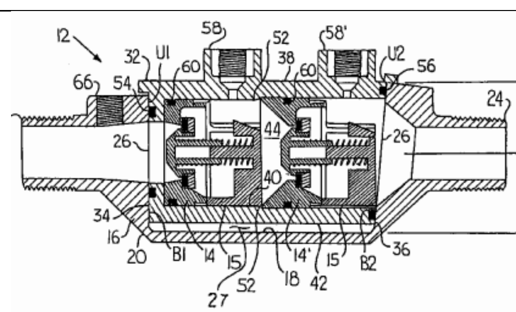
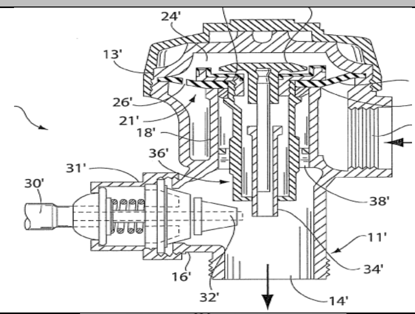
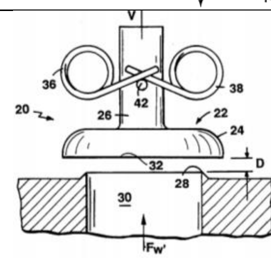
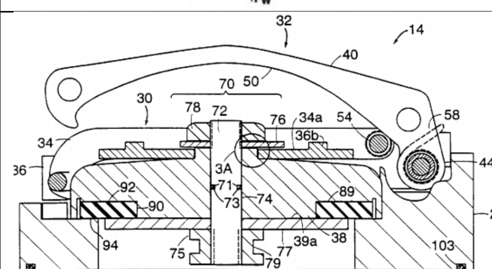
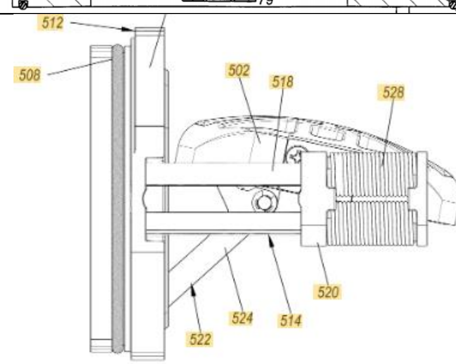
U.S. Valve LLC. "U.S. Valve LLC." *Check Valves, Ball Valves, & Low Pressure Valve Manufacturer U.S. Valve Manufacturer*, http://www.usvalve.com/.

Appendix A – Patent Table

Patent Number	Patent Title	Description	Drawing
US D721789 S	Cartridge Check Assembly	Patent from Watts Water Technologies, Inc. that details a cartridge check assembly	
US 6446652 B1	Backflow Preventer Assembly	Alternative valve mounting method that makes it less expensive to manufacture and permits easy service	
0 418 580 A1	Dual Check Valve	Dual check valve backflow preventer with independently operating check valves in a smaller form factor	
US 6648013 B1	Check Valve Having Variable Opening-Force Threshold	This check valve reduces the amount of force required to hold it open as it opens more; the more open the valve is the greater the reduction in pressure loss	
US 7434593 B2	Double Check Valve Assembly	This is Zurn's patent for the current double check we are working with; it uniquely features a swappable check valve cartridge	

Appendix A - Patent Table - Continued

Patent Number	Patent Title	Description	Drawing
US 2009/0072177 A1	Flush Valve Handle and Check Valve Assembly	A Zurn patent that improves upon the conventional design by designing a valve that utilizes polymeric components in place of brass. This reduces the cost of manufacturing.	
5564467	Poppet Check Valve	This valve uses torsional springs to load the poppet both axially and laterally. This allows the valve to close in two positions, meaning it has two positions to open or close.	
6050293	Flapper Check Valve	This design utilized two lever arms to drive the flapper. The lever arms utilize torsional springs to produce mechanical advantage.	
8,875,733	Check Valve in Backflow Prevention Device	This design utilizes a compound movement for its valve that first separates the valve from the valve seating in a linear fashion, and then rotates the valve so that its aspect ratio relative to the direction of water flow is reduced.	

Appendix B – Applicable Industry Codes, Standards, & Regulations

2.5.1 ASME B16.34

ASME B16.34 is an ASME standard that governs the design of valves. It covers pressure-temperature ratings, dimensions, tolerances, materials, nondestructive examination requirements, testing, and marking for valves of various end connections and materials of construction. It is a part of the ASME B16 family of standards, which cover regulations for valves, flanges, fittings and gaskets. [*ASME B16.34*]

2.5.2 ASSE 1015

ASSE 1015 covers performance requirements for double check backflow prevention assemblies for potable water and fire protection. ASSE (American Society of Sanitary Engineers) is a non-profit organization that represents all disciplines of the Plumbing Industry. Its purpose is to continually improve the performance, reliability, and safety of plumbing systems by developing and maintaining standards, developing and maintaining qualification programs, and promoting public awareness about the importance of safe and correct plumbing practices. [Performance Requirements]

2.5.3 CSA B64.5

CSA (Canadian Standards Association) B64.5 is a standard that applies to double check valve type backflow prevention devices. It is a section of the CSA B64 standard which outlines criteria for backflow preventers and vacuum breakers. [*CSA B64.5*]

2.5.4 AWWA C510

AWWA (America Water Works Association) C510 is a standard that describe the double check valve backflow prevention assembly. This standards purpose is to provide the minimum requirements for double check-valve backflow prevention assemblies for potable water applications. [AWWA]

2.5.5 Cal OSHA Title 8, Subchapter 7, Group 2, Article 9, §3363(h)

Title 8 is a part of the California Code of Regulations (CCR). It outlines Industry Regulations and Subchapter describes general industry safety orders. This specific section outlines the requirements for installation of backflow prevention devices in California.

"Non-potable water systems or systems carrying any other non-potable substance shall be installed so as to prevent backflow or back-siphonage into a potable water system." [Giso]

2.5.6 USC Foundation for Cross-Connection Control and Hydraulic Research Manual of Cross Connection Control, Ninth Edition

The USC FCCCHR provides the Manual of Cross-Connection Control to serve 3 main purposes [USC FCCCHR, 1]:

1. To provide minimum standards for public water supply protection.
2. Contain any potential contamination within the source consumer's premises that results from water backflow.
3. Prevent contamination of private water supplies as a result of cross-connections and backflow between domestic and industrial water supplies

Appendix C – Glossary

***Approved Check Valve:** a check valve that is drip-tight in the normal direction of flow when the inlet pressure is at least one psi and the outlet pressure is zero. A check valve shall permit no leakage in a direction reverse to the normal flow. The closure element shall be internally loaded to promote rapid and positive closure.*

***Backflow:** the undesirable reversal of flow of water or a mixture of water and other liquids, gases or other substances in the distribution pipes of the potable supply of water from any source.*

***Backflow Prevention Assembly:** an assembly that is used to prevent backflow into a potable water system. The type of assembly used is based on the degree of hazard and backflow conditions.*

***Back Pressure:** any elevation of pressure in the downstream piping system (by pump, elevation of piping, or steam/air pressure) above the supply pressure at the point of consideration which would cause a reversal of the normal direction of flow.*

***Contamination:** an impairment of the quality of water which creates an actual hazard to public health through poisoning or through spread of disease.*

***Cracking Pressure:** The amount of pressure necessary to lift the closed valve off the valve seat, allowing forward flow.*

***Cross-Connection:** any unprotected actual or potential connection or structural arrangement between a public or a consumer's potable water system and another system through which it is possible to introduce into used water, industrial fluid, gas, or substance into the potable water system.*

***Double Check Valve Backflow Prevention Assembly (DC):** an assembly composed of two independently acting, approved check valves. It includes tightly closing resilient seated shutoff valves attached at each end of the assembly and fitted with properly located resilient seated test cocks. It should only be used to protect against a non-health hazard (i.e. pollutant).*

***Pressure Loss:** The difference in pressure between two different points due to friction forces. In check valves, the pressure loss occurs between the valve inlet and outlet.*

***Reclaimed Water:** water which, as a result of treatment of wastewater, is suitable for a direct beneficial use or a controlled use that would not otherwise occur and is not safe for human consumption.*

***Water Hammer:** a series of pressure pulsations above and below the normal pressure of water in the pipe. The amplitude and period of the pulsation depend on the velocity of the water as well as the size, length, and material of the pipe. Shock loading from these pulsations occurs when any moving liquid is stopped in a short time. In general, it is important to avoid quickly closing valves to minimize the occurrence of water hammer. Water hammer is often accompanied by a sound resembling a pipe being struck by a hammer (hence the name). Water hammer can rupture pumps, valves, and pipes within the system.*

Appendix D – House of Quality

QFD House of Quality

Project: CP Check Valve

Revision Date: 10/15/19

Column #	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Direction of Improvement	▼	◇	▲	▲	▲	▼	◇	▼								
HOW: Engineering Specifications (Tests)	Pressure loss	Size	MAWP	MAWT	Backpressure	Assembly time	Cracking pressure	# of Components								
HOW MUCH: Target Values	# ≤ Current design	# = Current design	# ≥ Current design	# ≥ Current design	# ≥ Current design	t ≤ Current Design	# = Current design	# ≤ Current design								
Max Relationship	9	9	9	9	9	9	9	9								
Technical Importance Rating	169.9	153.4	134	134	111.7	167	232	176.7	0	0	0	0	0			
Relative Weight	13%	12%	10%	10%	9%	13%	18%	14%	0%	0%	0%	0%	0%	0%	0%	0%
Curr. Products: Our Current Product	2	5	4	4	4	3	4	3								
Curr. Products: Competitor #1: AMES Silver Bullet	2	1	5	5	4	3	4	3								
Curr. Products: Competitor #2: US-Valve	4	4	3	3	2	4	4	2								
Curr. Products: Competitor #3: Watts 719QT	2	4	4	4	4	3	4	3								
0																
Column #	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16

WHO: Customers / WHAT: Customer Requirements (Needs/Wants) / NOW: Curr. Products

Row #	Weight Chart	Relative Weight	Zurn	Maximum Relationship	WHAT: Customer Requirements (Needs/Wants)	1 Pressure loss	2 Size	3 MAWP	4 MAWT	5 Backpressure	6 Assembly time	7 Cracking pressure	8 # of Components	Our Current Product	Competitor #1: AMES Silver Bullet	Competitor #2: *US-Valve*	Competitor #3: Watts 719QT	Row #
1		10%	10	9	Reduce pressure loss	●								2	2	4	2	1
2		10%	10	9	Static pressure differential	○				●		●		5	5	5	5	2
3		9%	9	6	Mechanically driven	○				▽	○	○	○	5	5	5	5	3
4		8%	8	3	High level of reliability			○	○			▽	▽	4	5	4	4	4
5		7%	7	6	Water compliant material									5	5	5	5	5
6		10%	10	9	Meets industry requirements	▽		●	●	▽		●		5	5	5	5	6
7		6%	6	9	Vertical/horizontal operation	○				▽				5	5	3	5	7
8		8%	8	6	Compatible w/ current design		●	○	○			○		5	3	4	4	8
9		7%	7	6	Reduce design complexity						●		●	3	3	2	3	9
10		7%	7	9	Manufacturability		▽				○		●	4	4	4	3	10
11		8%	8	9	Fits w/in 3/4"-2"		●							5	5	5	5	11
12		6%	6	9	Standard tooling						●			5	5	5	4	12
13		7%	7	3	Comparable cost		▽				▽		○	5	4	4	5	13
14		0%																14
15		0%																15
16		0%																16

Correlations	
Positive	+
Negative	–
No Correlation	
Relationships	
Strong	●
Moderate	○
Weak	▽
Direction of Improvement	
Maximize	▲
Target	◇
Minimize	▼

Appendix E – Design Hazard Checklist (Updated)

DESIGN HAZARD CHECKLIST

Team: Check Valve Design – 53 Advisor: Sarah Harding Date: 2/29/20

Y	N	
☐	■	1. Will the system include hazardous revolving, running, rolling, or mixing actions?
☐	■	2. Will the system include hazardous reciprocating, shearing, punching, pressing, squeezing, drawing, or cutting actions?
☐	■	3. Will any part of the design undergo high accelerations/decelerations?
☐	■	4. Will the system have any large (>5 kg) moving masses or large (>250 N) forces?
☐	■	5. Could the system produce a projectile?
☐	■	6. Could the system fall (due to gravity), creating injury?
☐	■	7. Will a user be exposed to overhanging weights as part of the design?
☐	■	8. Will the system have any burrs, sharp edges, shear points, or pinch points?
☐	■	9. Will any part of the electrical systems not be grounded?
☐	■	10. Will there be any large batteries (over 30 V)?
☐	■	11. Will there be any exposed electrical connections in the system (over 40 V)?
☐	■	12. Will there be any stored energy in the system such as flywheels, hanging weights or pressurized fluids/gases?
☐	■	13. Will there be any explosive or flammable liquids, gases, or small particle fuel as part of the system?
☐	■	14. Will the user be required to exert any abnormal effort or experience any abnormal physical posture during the use of the design?
☐	■	15. Will there be any materials known to be hazardous to humans involved in either the design or its manufacturing?
☐	■	16. Could the system generate high levels (>90 dBA) of noise?
☐	■	17. Will the device/system be exposed to extreme environmental conditions such as fog, humidity, or cold/high temperatures, during normal use?
☐	■	18. Is it possible for the system to be used in an unsafe manner?
☐	■	19. For powered systems, is there an emergency stop button?
☐	■	20. Will there be any other potential hazards not listed above? If yes, please explain on reverse.

Appendix F – Revisited Final Design Decision Matrix (Pugh Matrix)

CRITERIA (ENGINEERING SPECIFICATIONS)	WEIGHT	ZURN 350XL (DATUM)	SPLIT INLINE (OPTION #1)	DOUBLE-DISK (OPTION #2)
REDUCE PRESSURE LOSS	5	0	-1	1
MAINTAIN PRESSURE DIFFERENTIAL	4	0	1	0
MECHANICALLY DRIVEN	3	0	1	1
REDUCE # OF COMPONENTS	2	0	0	-1
EASE OF DESIGN SCALABILITY	1	0	1	1
TOTAL:		0	3	7

Appendix G – Indented Bill of Materials

Assembly Level	Part Number	Description Lvl0	Lvl1	Lvl2	Lvl3	Vendor	Qty	Drwg. Type
0	1000	Final Assembly				----	----	----
1	1100		Modified Backflow Housing [354-1A]			Zurn Wilkins	1	Custom
2	1110			Double Disk Check Assembly		----	2	-----
3	1111				Check Hinge	Custom	2	Custom
3	1112				Torsion Spring	Springs International	2	Custom
3	1113				Poppet Disk	Custom	4	Custom
3	1114				Poppet O-ring Arcs [040N]	Zurn Wilkins	4	Vendor/Custom
2	1120			Check O-ring Seals [273N]		Zurn Wilkins	2	Vendor
2	1130			Stainless Pins		----	----	----
3	1131				3/16 Pin	McMaster-Carr	4	Vendor/Custom
3	1132				3/8 Pin	McMaster-Carr	2	Vendor/Custom
2	1140			Check Spacer		Custom	1	Custom
						Total	**24**	

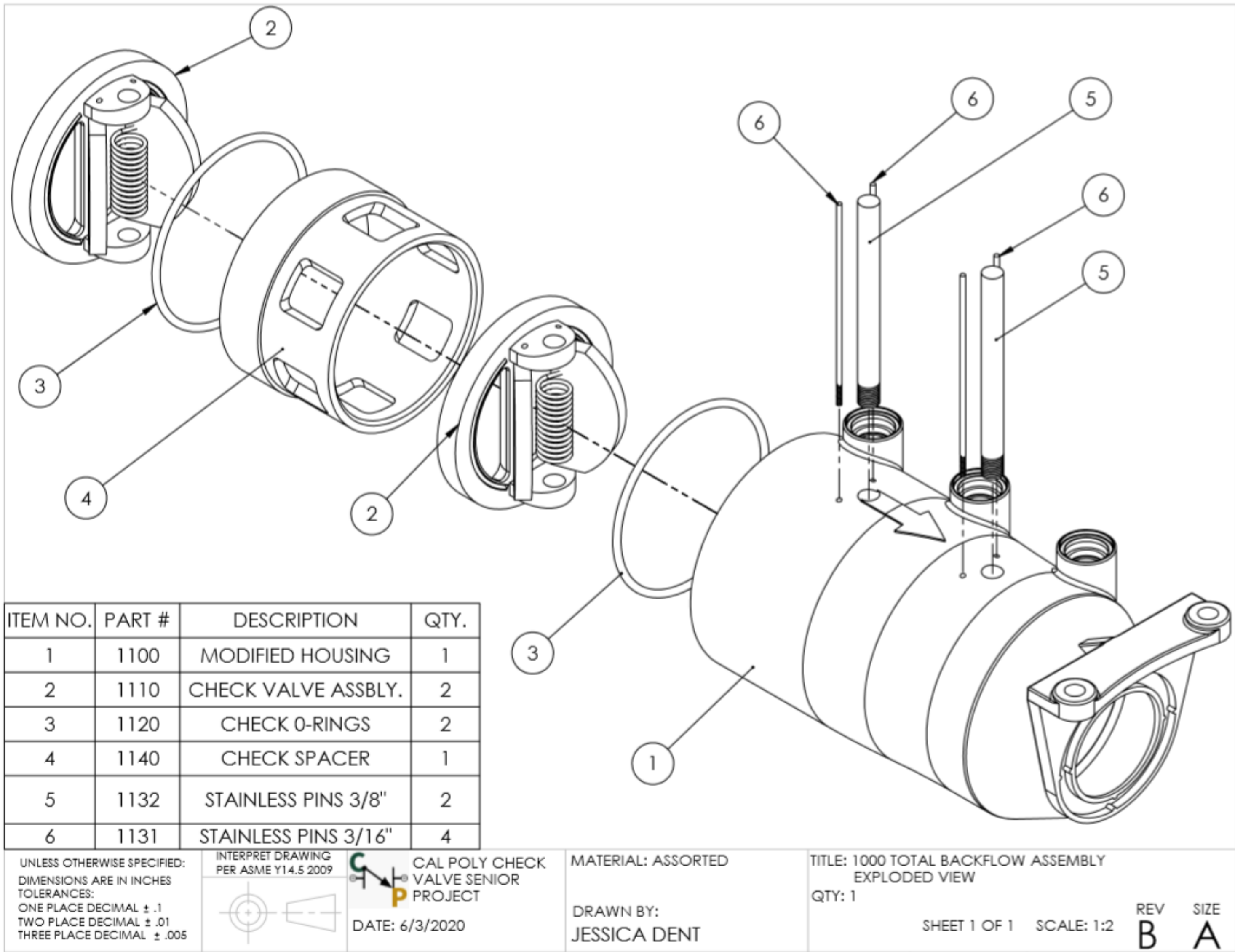
Appendix H – Failure Modes & Effects Analysis (FMEA)

System / Function	Potential Failure Mode	Potential Effects of the Failure Mode	Severity	Potential Causes of the Failure Mode	Current Preventative Activities	Occurrence	Current Detection Activities	Detection	Priority	Recommended Action(s)
Valve Disk/Close in reverse flow condition	valve does not re-seat properly	valve is not 100% shut in reverse condition and water source is contaminated	3	1) sediment or imperfections on sealing surfaces 2) valve disk/plug becomes misaligned 3) lack of closing force	1) Proper use of GD&T to ensure design accuracy 2) Ensure seat gasket material is of sufficient thickness	2	Motion Study/CAD simulation of dynamic model	7	42	Run tests to cycle mechanism for opening and closing
	Valve disk/plug becomes detached	valve is not 100% shut in reverse condition and water source is contaminated	3	1) faulty fasteners 2) impact from water hammer or sediments etc. 3) improper installation/maintenance	1) Built-in water hammer arrestor 2) Reinforcing of connecting member for valve disk	1	1) Dynamic stress analysis 2) Impact analysis	2	6	Apply a factor of safety for fasteners
	mechanical closing mechanism fails	valve is not 100% shut in reverse condition and water source is contaminated	3	1) fatigue failure 2) fouling or rust	1) Anti-corrosion coating 2) Reinforce joints & cycling components	1	1) Lifetime/Life Cycle Analysis 2) Deployment of necessary replacement/servicing routines	2	6	Use non-corrosive materials, run fatigue tests
Mechanical element/ lifts at pre-set pressure	Valve gets stuck/does not lift	Check valve does not allow water to flow through	2	Valve disk/plug gets caught or stuck	Design smooth mating surface for check valve	1	Monitor pressure differential and flow rate	2	4	MInimize friction & snag points during design
	Valve lifts at lower pressure than pre-set pressure	spring cannot hold disk closed	2	Spring stiffness is too low	Spring rate analysis	1	Check for backflow	4	8	Test pressure required to open valve
	Valve lifts at higher pressure than pre-set pressure	valve does not open during normal flow	2	Spring stiffness is too high	Spring rate analysis	1	Check for backflow	4	8	Test pressure required to open valve

Appendix H - Failure Modes & Effects Analysis (FMEA)

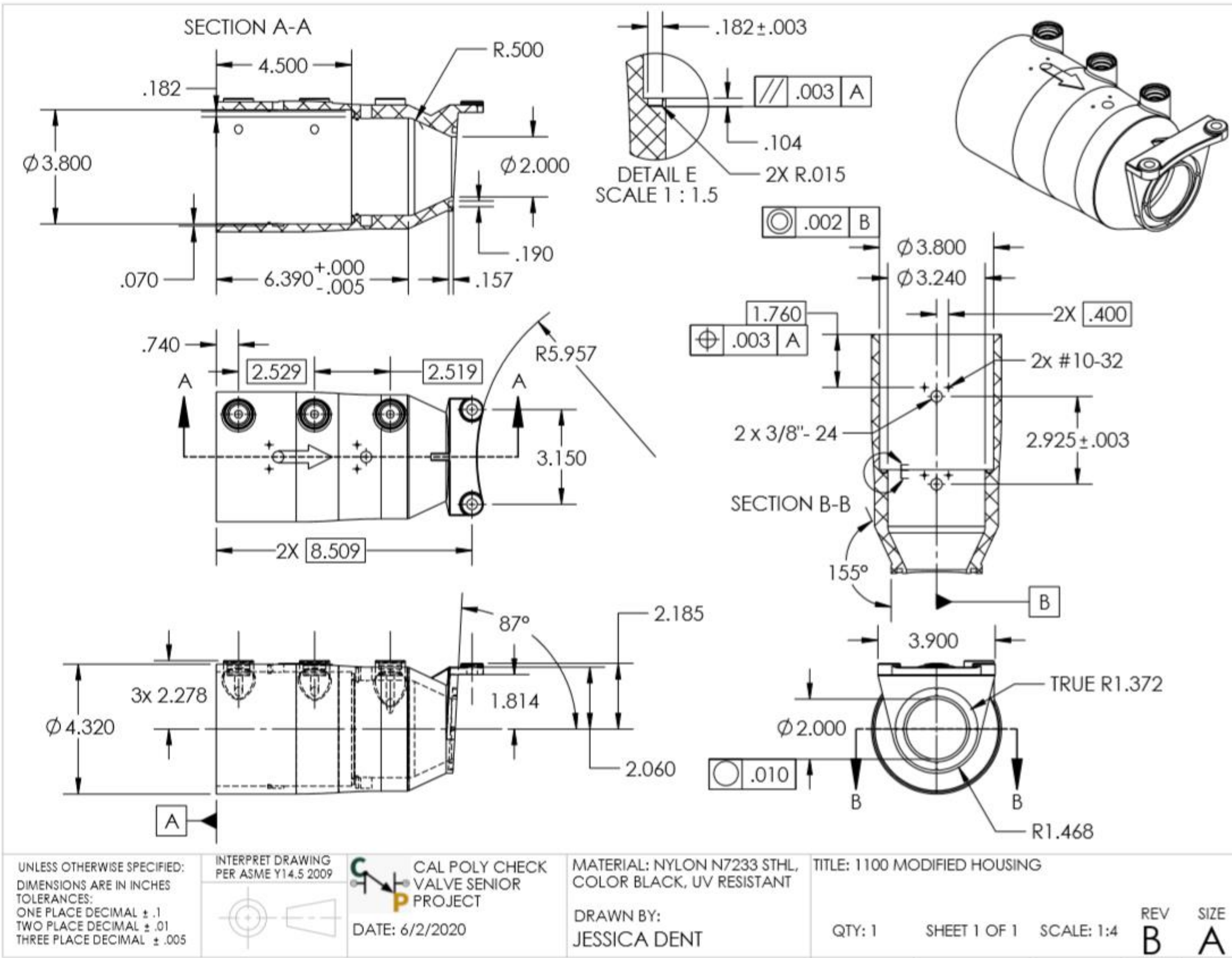
System / Function	Potential Failure Mode	Potential Effects of the Failure Mode	Severity	Potential Causes of the Failure Mode	Current Preventative Activities	Occurrence	Current Detection Activities	Detection	Priority	Recommended Action(s)
O-rings/Flow sealing	Gaskets/ O-rings pinch or lose proper seal	Constant leaking or bursting in high pressure conditions	3	1) O-ring tolerance not tight enough 2) Too sharp of contacting surfaces 3) Too little compression of O-ring	Choose O-ring/O-ring material that provides proper tolerancing	2	Check for leakage under flow conditions	2	12	1) Ensure design specifications smooth contacting surfaces & eliminate sharp edges 2) Source O-rings from manufacturer that provides sufficient tolerance control
	Gaskets/ O-rings damaged or cut	Constant leaking or bursting in high pressure conditions	3	1) Lack of proper service intervals or preventative maintenance 2) Manufacturer defect passed to assembly process	Follow Industry-standard life-cycles for determining service/replacement routines	1	Check for leakage under flow conditions	2	6	1) Specify recommended service intervals and material replacement procedures
Valve body / Reduce pressure loss	Blockage of valve	valve has increased pressure loss	1	Debris is built up on pipe walls or fouling of surfaces	Use materials that do not chip or wear easily and are corrosion resistant	2	Periodically open valve and visually investigate if pipe has debris	2	4	Use ductile materials when possible to reduce chipping and debris creation
Valve housing / mates with adjacent parts	Leaks at mating surfaces	Loses ability to hold water	2	Gaps between sealing surfaces	Use standard thread sizes and tolerances	2	Thread gauge	2	8	Check all new valves for ability to fully seal against neighboring backflow prevention components

Appendix I – Double-disk Technical Drawings #1000 (Assembly Exploded View)

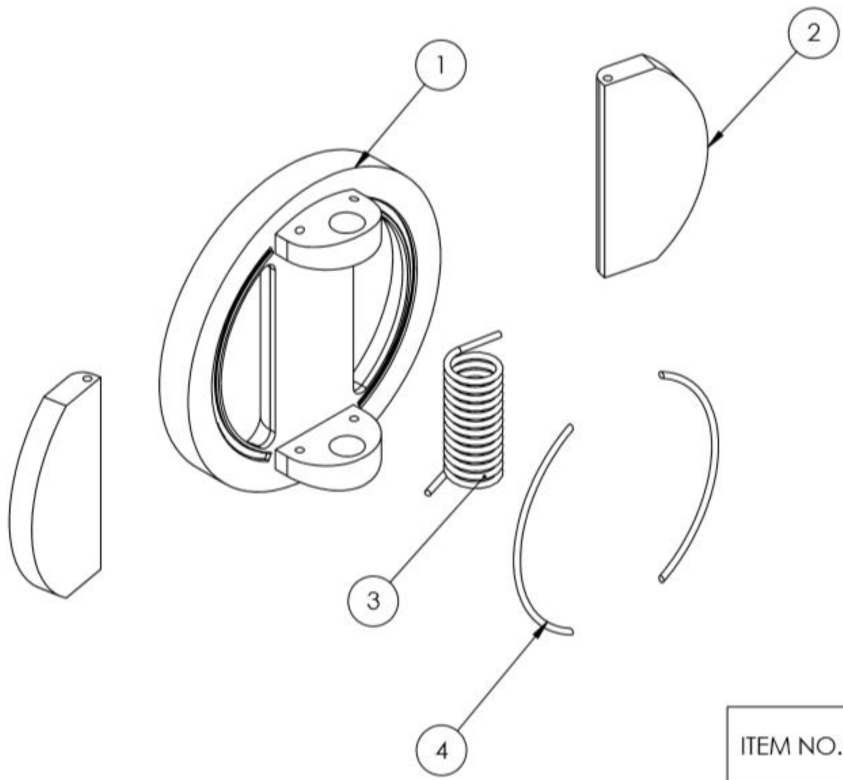


ITEM NO.	PART #	DESCRIPTION	QTY.
1	1100	MODIFIED HOUSING	1
2	1110	CHECK VALVE ASSBLY.	2
3	1120	CHECK 0-RINGS	2
4	1140	CHECK SPACER	1
5	1132	STAINLESS PINS 3/8"	2
6	1131	STAINLESS PINS 3/16"	4

Appendix I – Double-disk Technical Drawings #1100 (Modified Backflow Housing)



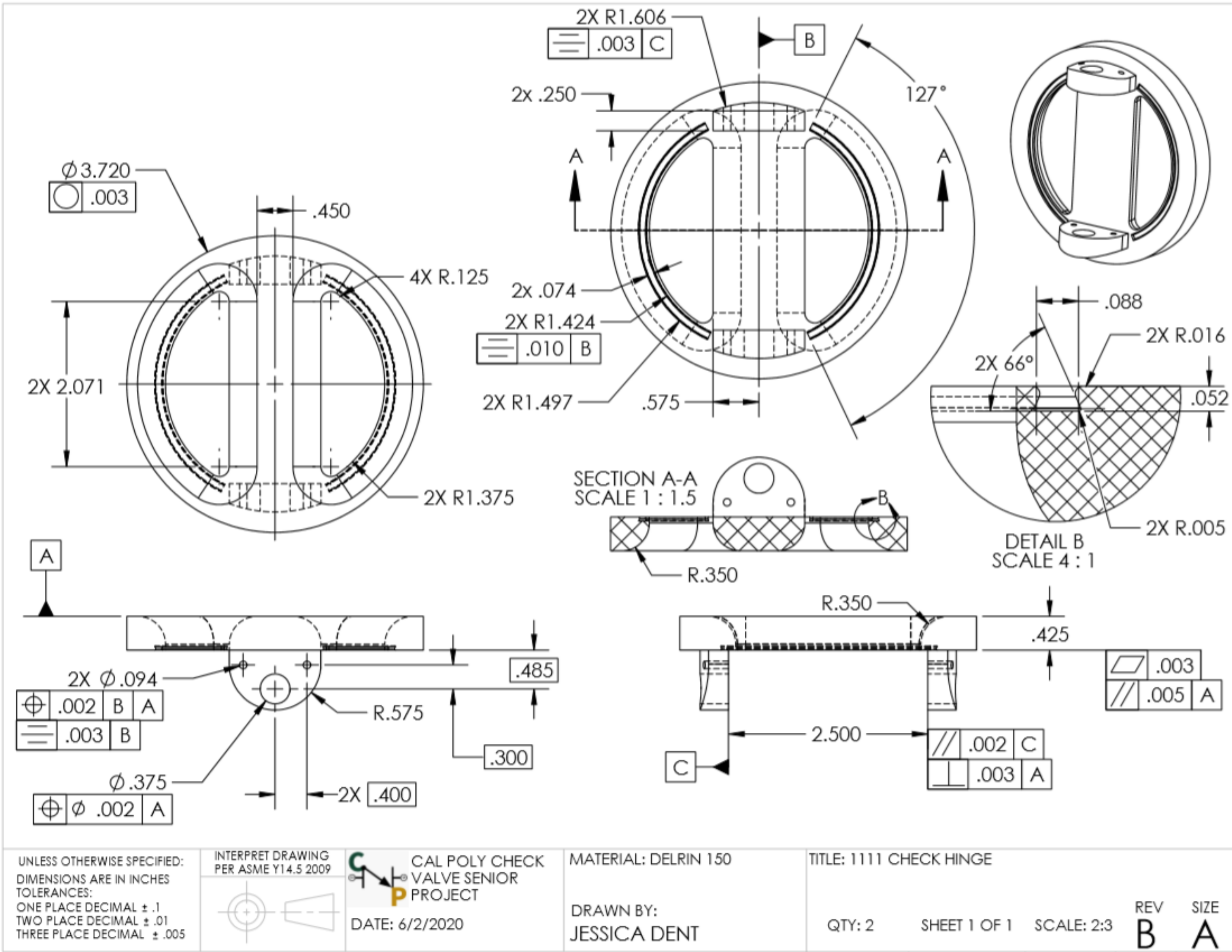
Appendix I – Double-disk Technical Drawings #1110 (Check Assembly Exploded View)



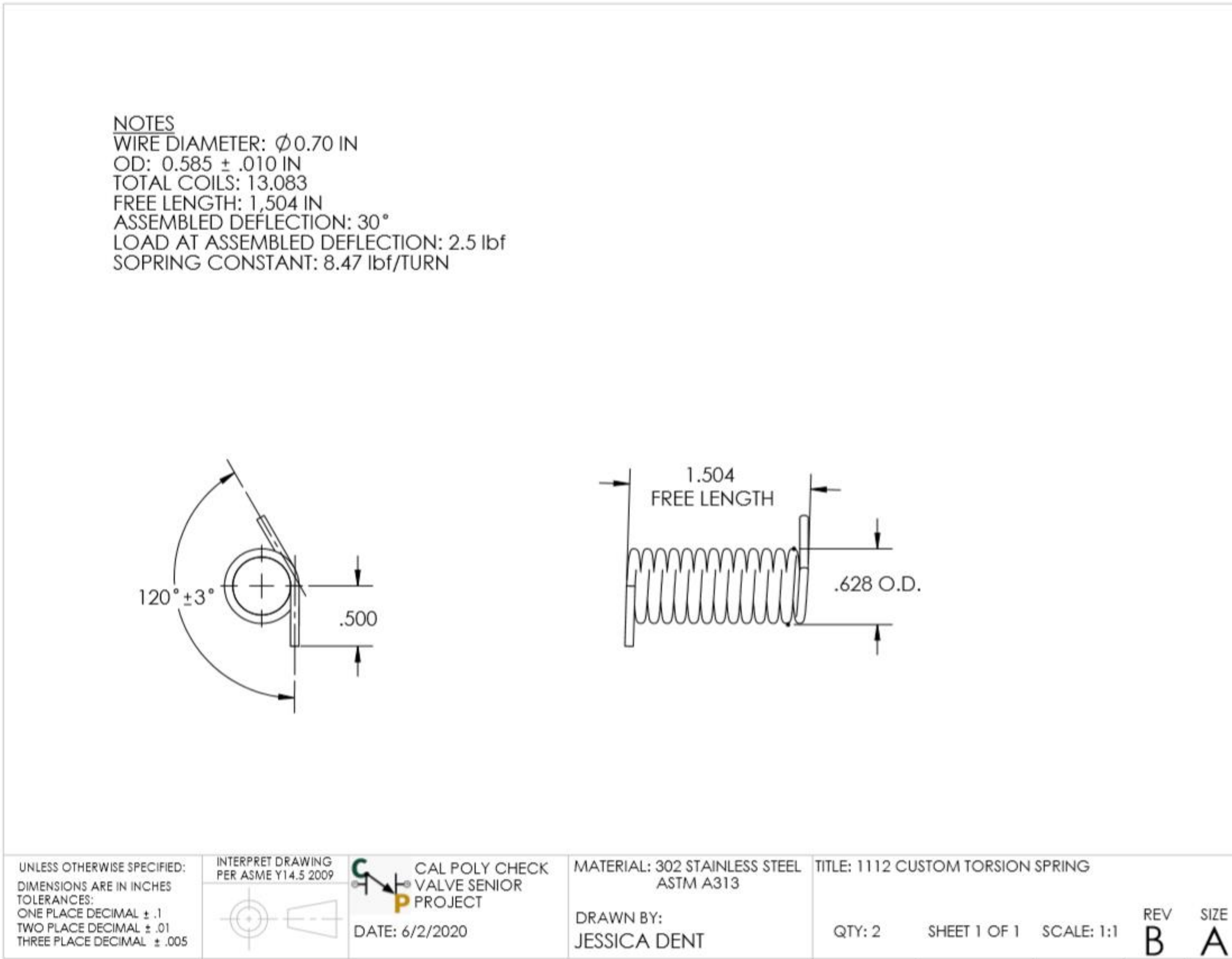
ITEM NO.	PART #	DESCRIPTION	QTY. PER CHECK
1	1111	CHECK HINGE	1
2	1113	POPPET DISK	2
3	1112	CUSTOM TORSION SPRING	1
4	1114	POPPET SEAL ARCS	2

UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN INCHES TOLERANCES: ONE PLACE DECIMAL ± .1 TWO PLACE DECIMAL ± .01 THREE PLACE DECIMAL ± .005	INTERPRET DRAWING PER ASME Y14.5 2009	CAL POLY CHECK VALVE SENIOR PROJECT DATE: 6/3/2020	MATERIAL: ASSORTED DRAWN BY: JESSICA DENT	TITLE: 1100 CHECK ASSEMBLY - EXPLODED VIEW QTY: 2 SHEET 1 OF 1 SCALE: 2:3 REV B SIZE A

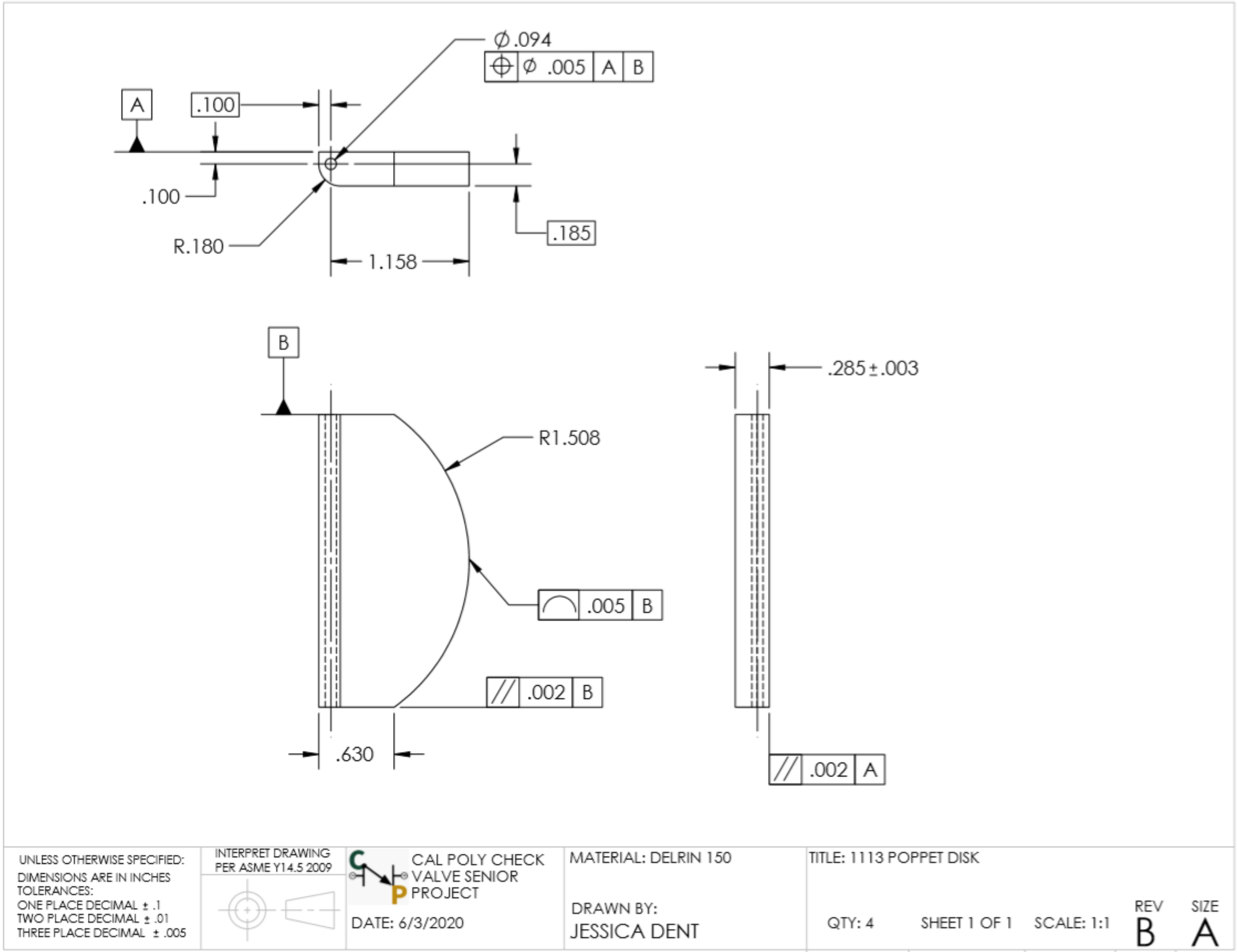
Appendix I – Double-disk Technical Drawings #1111 (Check Hinge)



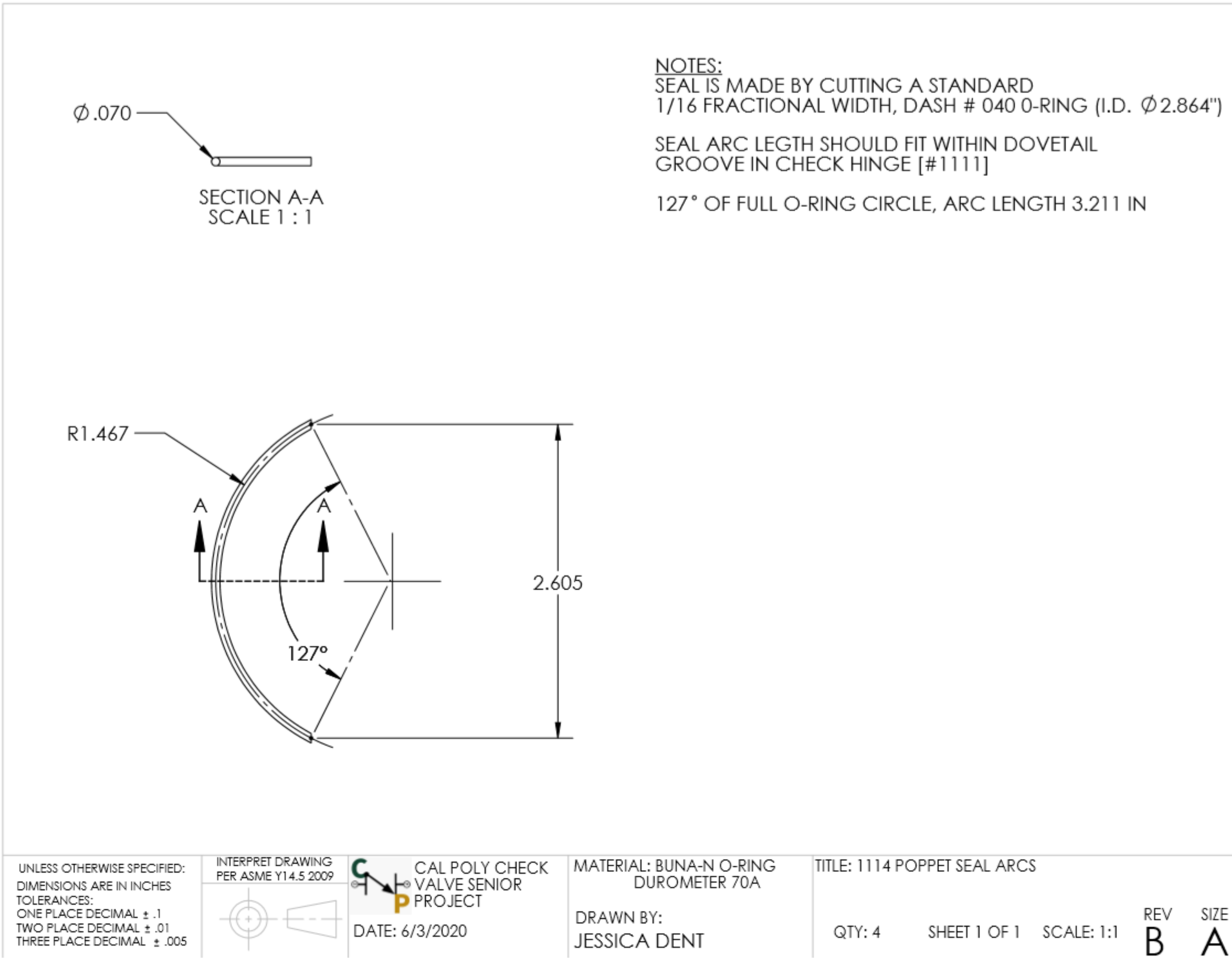
Appendix I – Double-disk Technical Drawings #1112 (Torsional Spring - Custom)



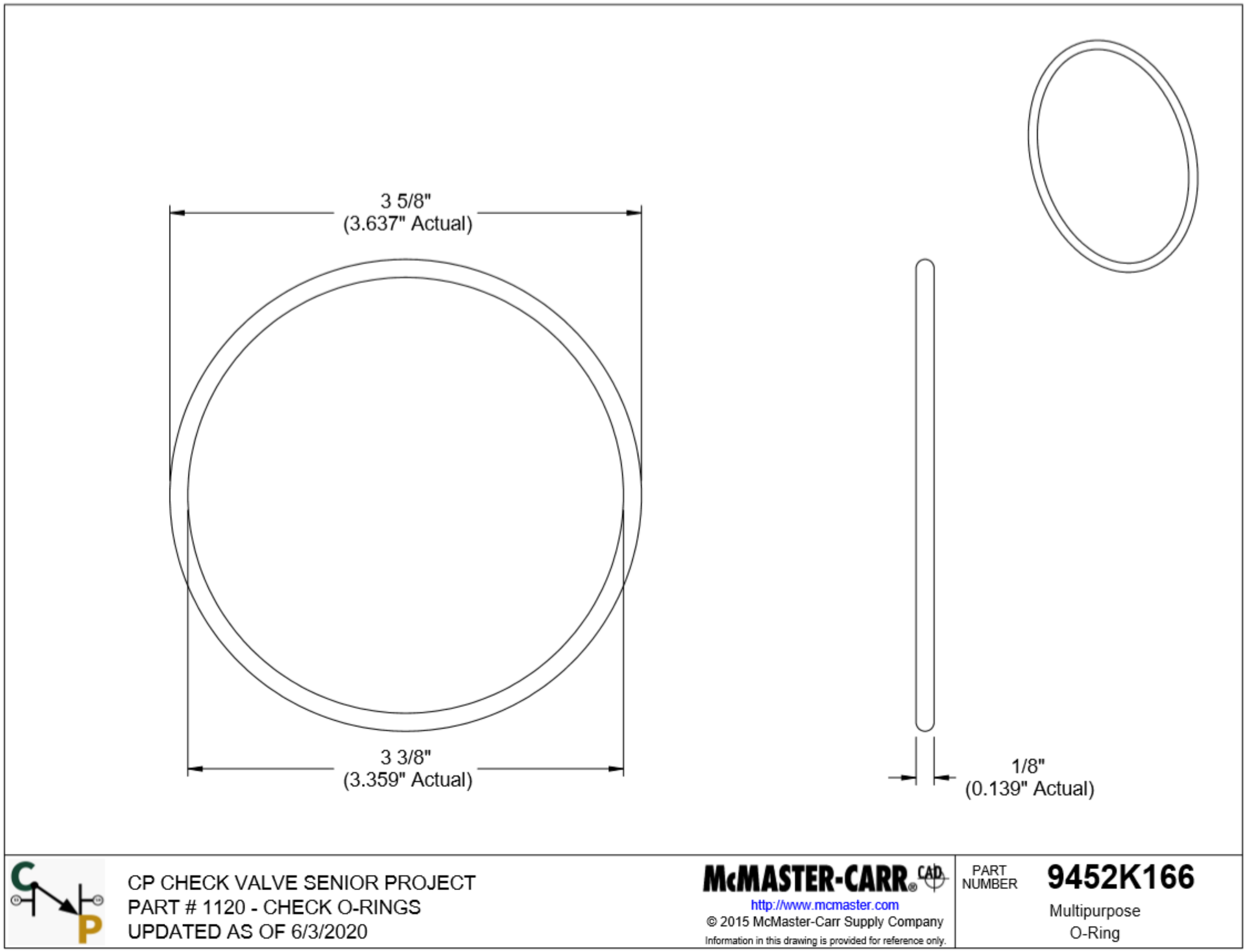
Appendix I – Double-disk Technical Drawings #1113 (Poppet Disks)



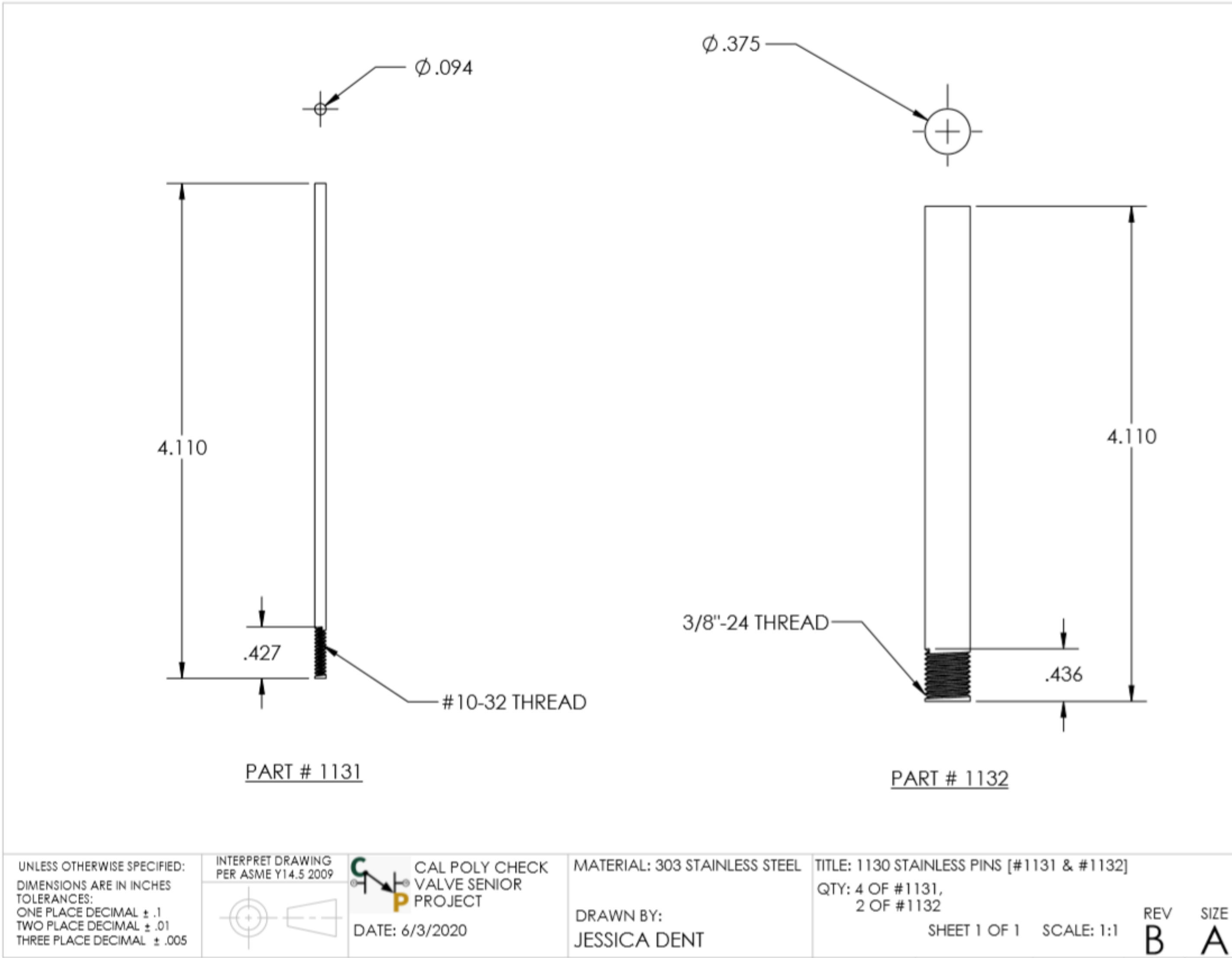
Appendix I – Double-disk Technical Drawings #1114 (Poppet Seal Arcs)



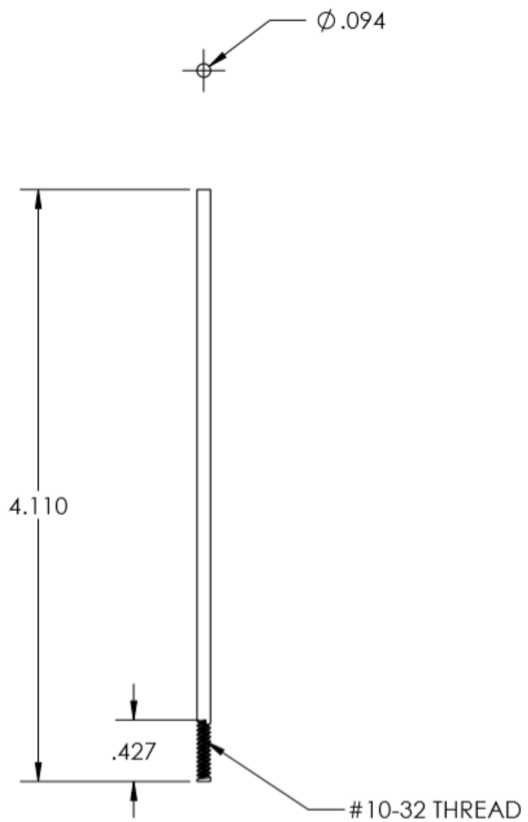
Appendix I – Double-disk Technical Drawings #1120 (Check O-Rings)



Appendix I – Double-disk Technical Drawings #1130 (Stainless Pins)

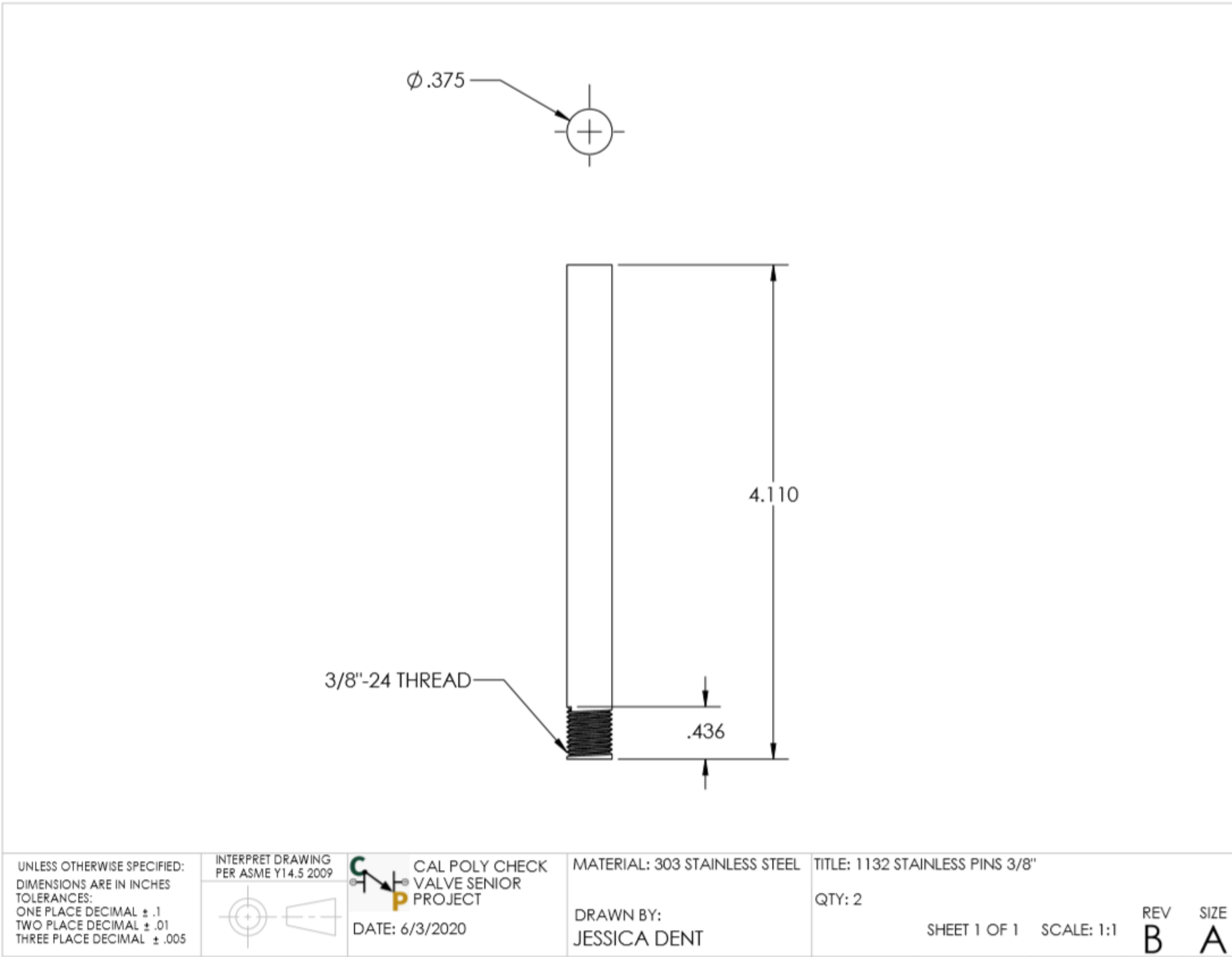


Appendix I – Double-disk Technical Drawings #1131 (Stainless Pin 3/16")

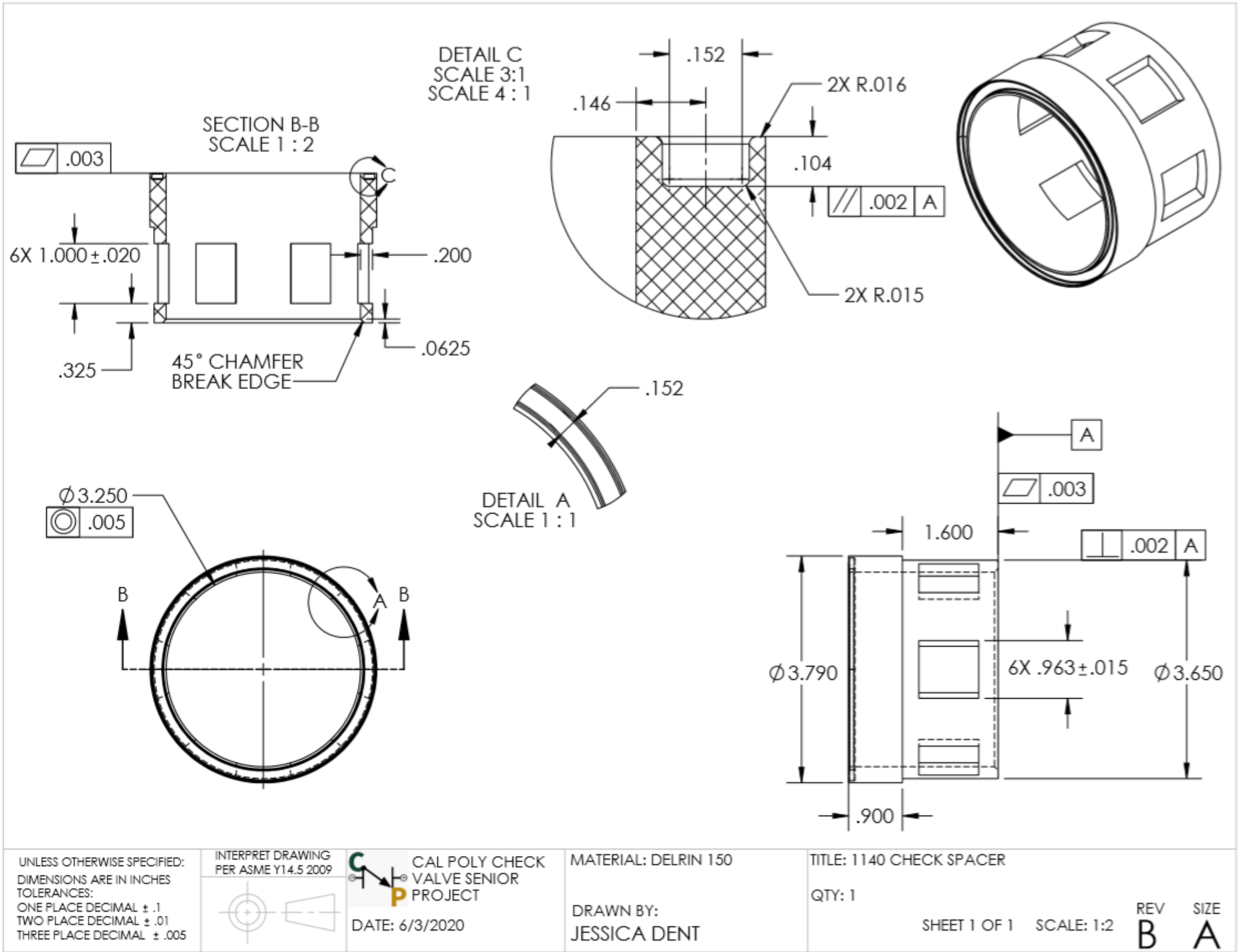


UNLESS OTHERWISE SPECIFIED: DIMENSIONS ARE IN INCHES TOLERANCES: ONE PLACE DECIMAL ± .1 TWO PLACE DECIMAL ± .01 THREE PLACE DECIMAL ± .005	INTERPRET DRAWING PER ASME Y14.5 2009	CAL POLY CHECK VALVE SENIOR PROJECT DATE: 6/3/2020	MATERIAL: 303 STAINLESS STEEL DRAWN BY: JESSICA DENT	TITLE: 1131 STAINLESS PIN 3/16" QTY: 4 SHEET 1 OF 1 SCALE: 1:1 REV B SIZE A

Appendix I – Double-disk Technical Drawings #1132 (Stainless Pin 3/8”)



Appendix I – Double-disk Technical Drawings #1140 (Check Spacer)



Appendix J – Full Budget Breakdown

Name	Description	Used for Part #s	Vendor	Vendor Part Number	Qty	Qty Per Pkg	Cost	Total Cost
Modified Backflow Housing	Nylon N72333 STHL, Black, UV Resistant. Modified from the 350XL, machined to include modifications	1100	Zurn Wilkins	354-1A	1	1	$ 25.09	$ 25.09
Check Hinge	Delrin 150, custom injection molded & post-machined	1111	Protolabs	----	2	2	$ 3.83	$ 7.66
Torsion Spring- Custom	Torsion Spring, 210 Degree Angle, Left-Hand Wound, 0.585" OD, 0.070" Wire Diameter, 0.5" Leg Length, 13.083 Coils	1112	International Industrial Springs	----	1	5	$ 10.00	$ 10.00
Poppet Disk	Delrin 150, custom injection molded & post-machined	1113	Protolabs	----	4	4	$ 0.47	$ 1.88
Poppet O-ring Arcs	Buna-N O-Ring 1/16 Fractional Width, Dash Number 040, [Zurn #040N]	1114	McMaster-Carr	9452K128	1	100	$ 14.26	$ 14.26
Check O-ring Seals	Buna-N O-Ring 1/8 Fractional Width, Dash Number 237. [Zurn #273N]	1120	McMaster-Carr	9452K166	1	50	$ 13.41	$ 13.41
3/16 Stainless Pin	3/16" 303 Stainless Steel Rod, 2 ft length	1131	McMaster-Carr	8984K13	1	1	$ 3.81	$ 3.81
3/8 Pin	3/8" 303 Stainless Steel Rod, 1 ft length	1132	McMaster-Carr	8984K99	1	1	$ 4.54	$ 4.54
Check Spacer	Delrin 150, custom injection molded & post-machined	1140	Protolabs	---	1	1	$ 2.55	$ 2.55
							TOTAL BUDGET	**$ 83.20**

Appendix K – Design Verification Plan (DVP&R)

Senior Project DVP&R													
Date: 3/1/2020		Team: CP Check Valve -	Sponsor: Zurn Wilkins			Description of System: Check valve for dual check backflow preventer						DVP&R Engineer: Skylar Tusting	
TEST PLAN										TEST REPORT			
Item No	Specification #	Test Description	Acceptance Criteria	Test Responsibility	Test Stage	SAMPLES		TIMING		TEST RESULTS			Notes
						Quantity	Type	Start date	Finish date	Test Result	Quantity Pass	Quantity Fail	
1	1	Pressure loss test	< Current Product	ST	SP	1	Sub/Sys	1/23/2020	N/A	Pressure & Flowrate	N/A	Ongoing	
2	2	Size	Fits within current housing	JD	CP	1	Sys	1/1/2020	N/A	Size [in]	N/A	N/A	
3	3	MAWP	175 psi	ST	FP	1	Sys	5/1/2020*	N/A	TBD	N/A	N/A	
4	4	MAWT	180°F	ST	FP	1	Sys	5/1/2020*	N/A	TBD	N/A	N/A	
5	5	Static Differetial	7 psi /valve (RP) 2 psi/valve (DC)	AM	SP	1	Sub/Sys	5/1/2020*	N/A	TBD	N/A	N/A	
6	6	Time taken to assemble	≤ current product	JD	FP	1	Sys	5/1/2020*	N/A	Time [s]	N/A	N/A	
7	7	Cracking Presure	Equivalent to Current product	AM	SP	1	Sub/Sys	5/1/2020*	N/A	TBD	N/A	N/A	

*Open to change as progess is made

March 18th 2020: all testing and manufacturing operations were suspended due to COVID-19

Appendix L – Instrumentation Documentation

OMEGA

14. Specifications

The flowmeter specifications and the type specification code used when ordering the flowmeter are described in this chapter.

14.1 Flowmeter Specifications

■ **Overall Specifications**

Measurement range in terms of flow velocity:

0–0.3 m/s to 0–10 m/s

System accuracy: See the following table.

Table 14.1 System accuracy

Flow rate as a percentage of range	Accuracy	
	0.3 – 1.0 m/s	**1.0–10 m/s**
0 to 20%	————	±0.1% FS
20 to 100%	————	±0.5% of rate
0 to 50%	±0.25% of FS	————
50 to 100%	±0.5% of rate	————

Note: The accuracy above is measured under standard operating conditions OMEGA 's calibration facility.

Fluid conductivity: 5 µS/cm minimum

Fluid temperature: -10 to +120 °C (in the case of teflon PFA lining)

-10 to +80 °C (in the case of EPDM rubber lining)

Ambient temperature: –10 to +60 °C

Dimensions and Mass: See Chapter 15, "Outline Dimensions."

■ **FMG400 Series flange type Detector**

Meter size: 15, 25, 40, 50, 80, 100, 150, and 400mm

Fluid pressure: –0.1 MPa {–1 kgf/cm^2} to the 2 MPa {20kgf/cm^2}

Connection flange standard: See the Specification Code table.

Appendix M – Pressure Loss Testing Procedure

Purpose

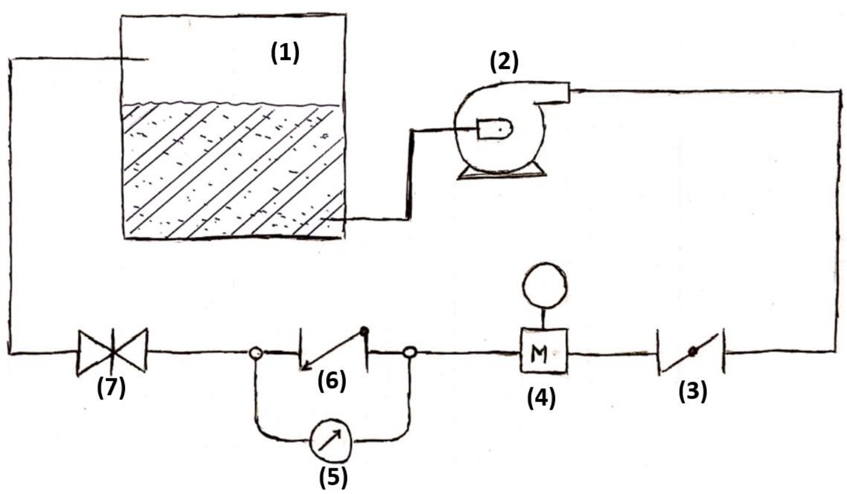
This testing procedure outlines the process of running a pressure loss test at Zurn's wet lab facility located in Paso Robles California.

Disclaimer

This testing procedure was written after testing operations were suspended due to the constantly changing events surrounding COVID-19. The information contained within this procedure has not been physically verified for accuracy and should not be used as a sole resource for running tests. Any use of information in this procedure is done at the risk of the user. These risks may include failure of the testing equipment and bodily harm to operators.

Testing Apparatus Process Flow Diagram

This diagram shows a general layout of essential testing equipment but does not identify all components involved in the testing procedure.



(1) Underground storage tank

(2) Pump system

(3) Isolation Valve

(4) Magnetic flow meter

(5) Differential pressure gauge

(6) Test stand

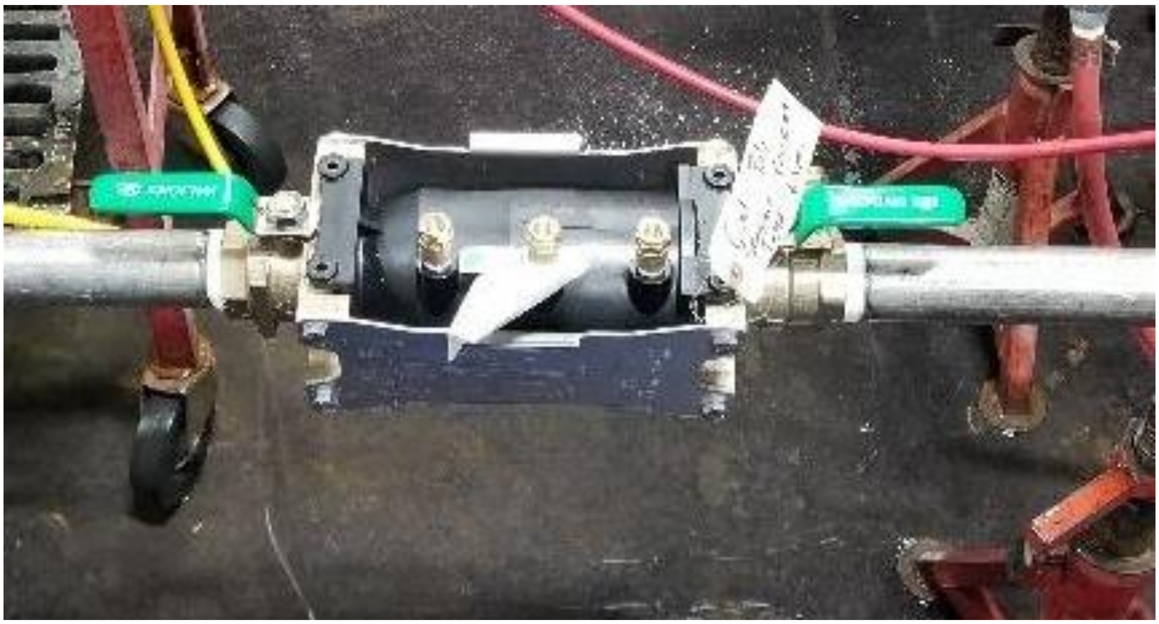
(7) Gate valve

Water is stored in a large underground water tank **(1)** that holds water that circulates through all the various piping manifolds and test stands. The water is pressurized and circulated by several centrifugal pumps **(2)** capable of being set up in a parallel flow configuration to allow for larger flow capacity conditions. Before the water circulates through the test stand its flow path is stopped by the butterfly isolation valve **(3).** This valve is used to prevent the test stand from receiving pressure until it is ready and has been properly lined up. Water then flows through the magnetic flow meter **(4)** where flowrate data is recorded. Pressure data is read from the pressure differential digital gauge **(5)** in psi_d. The pressure taps come off of large bulbous expansions in the piping, this is done to reduce the effects of the dynamic pressure and measure the static pressure at these regions as accurately as possible. Water flows through the test stand **(6)** and the desired check valve configuration to be tested. The last critical component for testing is the hand-actuated gate valve **(7)** which is used for flow control.

Test Procedure

A. *Load check valve to be tested into test stand* ***(6)****:*

- **a.** Bleed pressure out of system to ensure it is in depressurized state with bleed valves (not shown in diagram)
- **b.** Remove 4x flange bolts on valve housing **(6)**
- **c.** Install valve components to be tested into valve housing
- **d.** Reinstall complete valve housing assembly ensuring gaskets are in good condition and in place
- **e.** Install the 4x flange bolts and tighten



B. *Verify equipment and valve lineup:*

- **a.** Ensure isolation valve **(3)** is in the full closed position signified by the yellow indicator being perpendicular to the flow path.

- **b.** Ensure minimum level requirements in water storage tank (level indicator not shown in diagram)
- **c.** Ensure all gaskets and connections are leak tight by visual inspection
- **d.** Ensure flow control gate valve **(7)** is in the full closed position by rotating fully clockwise

- **e.** Ensure flow-meter **(4)** is reading 0 GPM prior to starting test

C. *Turn on pump system:*

- **a.** Energize pump and verify it is running and no leaks present

D. *Lineup flow path for test stand:*

- **a.** Slowly open isolation valve **(3)** to the fully open position indicated by the yellow indicator being parallel to flow path inspecting for leaks as you open in
  - **i.** If any leaks are present abort test and close isolation valve and shut off pump
  - **ii.** remediate any leaks and return to step **C.**
- **b.** If no leaks present, crack open flow control valve **(7)** to purge air from piping and equipment
  - **i.** After purging air for not less than 10 seconds close flow control valve **(7)**

c. Purge pressure instrument **(5)** sensing lines of residual air using purge needle valves (not shown in diagram but present in photo of test apparatus)
    i. Purge until flow out of instrument drains are free of entrained air and flow is laminar and stable

E. *Record no flow condition pressure differential across test apparatus:*

a. Record pressure differential from pressure differential gauge **(5)**

b. **Note:** if this is a prototype, it does not provide positive shutoff and therefore it is expected to have no pressure drop at the no flow condition. A true check valve with positive shutoff would be expected to have a pressure drop no less than 1 PSI per check valve.

F. *Record flow data:*

a. Crack open flow control valve **(7)** and open until flowmeter reads ~10 GPM

b. Wait for flowrate and pressure reading to reach a steady state

c. Record value of flowrate from flow-meter **(4)**

d. Record pressure differential from pressure differential gauge **(5)**

e. Increase the flowrate with flow control valve **(7)** by increment of ~10 GPM

f. repeat steps **F. b. - e.** until flowrate is ~140 GPM

G. *Shut down test equipment:*

a. De-energize pumps

b. Close isolation valve **(3)**

c. Bleed pressure from test stand with bleed valves to depressurize system

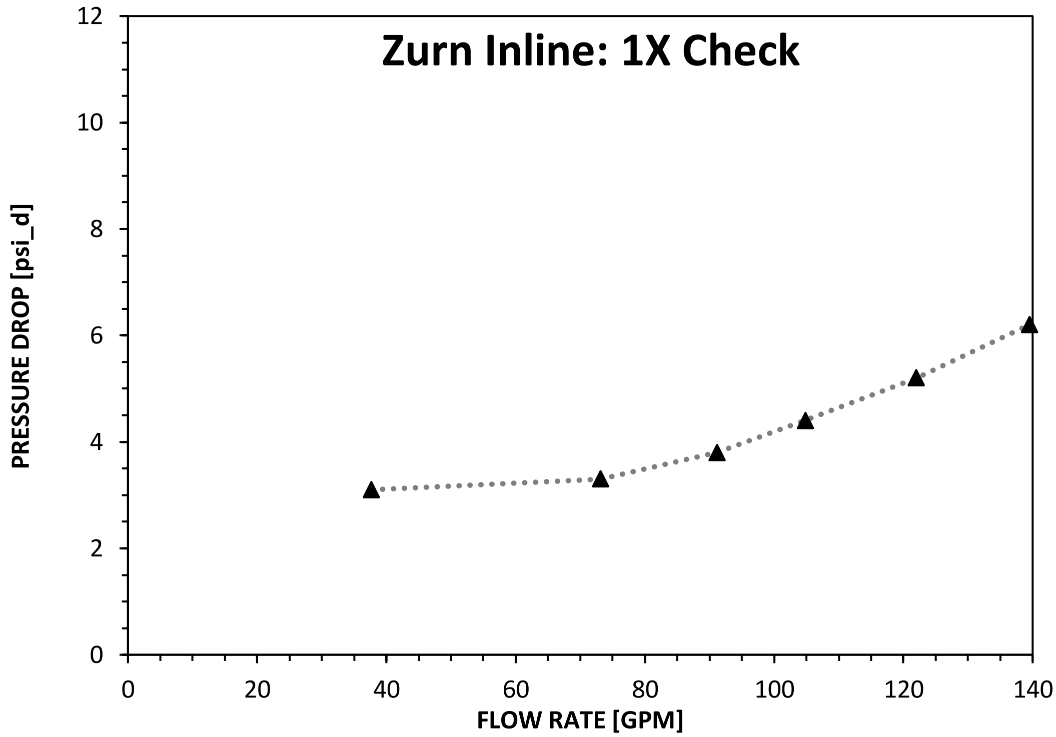
d. Unbolt 4x flange bolts on valve housing **(6)**

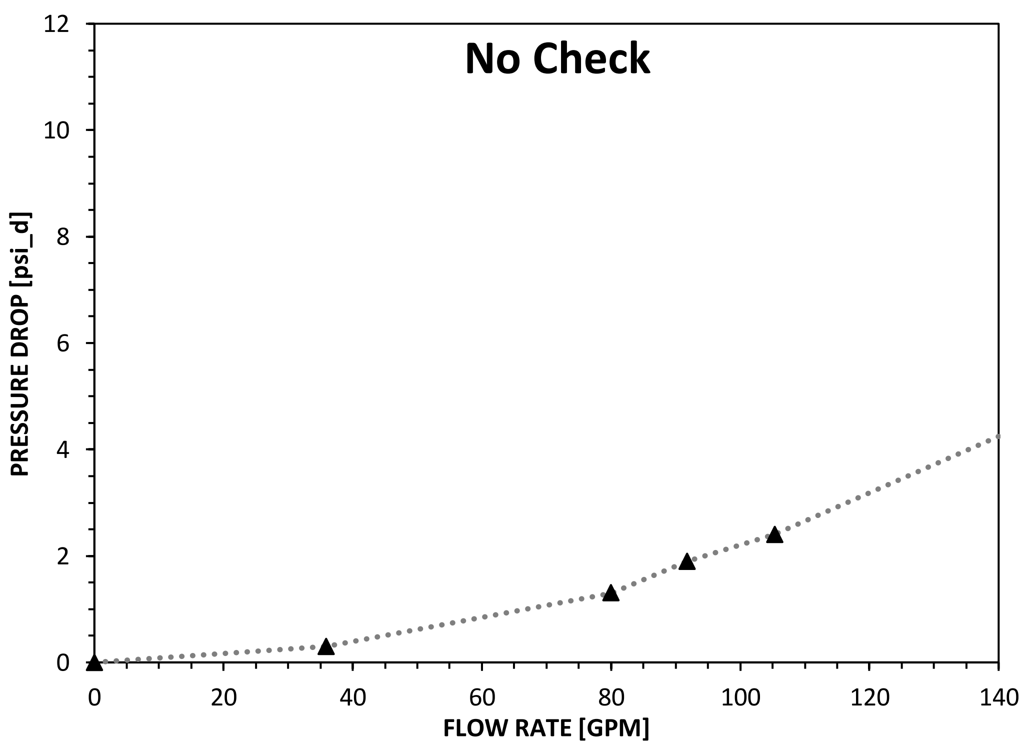
e. Remove check valves from valve housing assembly

f. Reinstall new check valve configuration to be tested

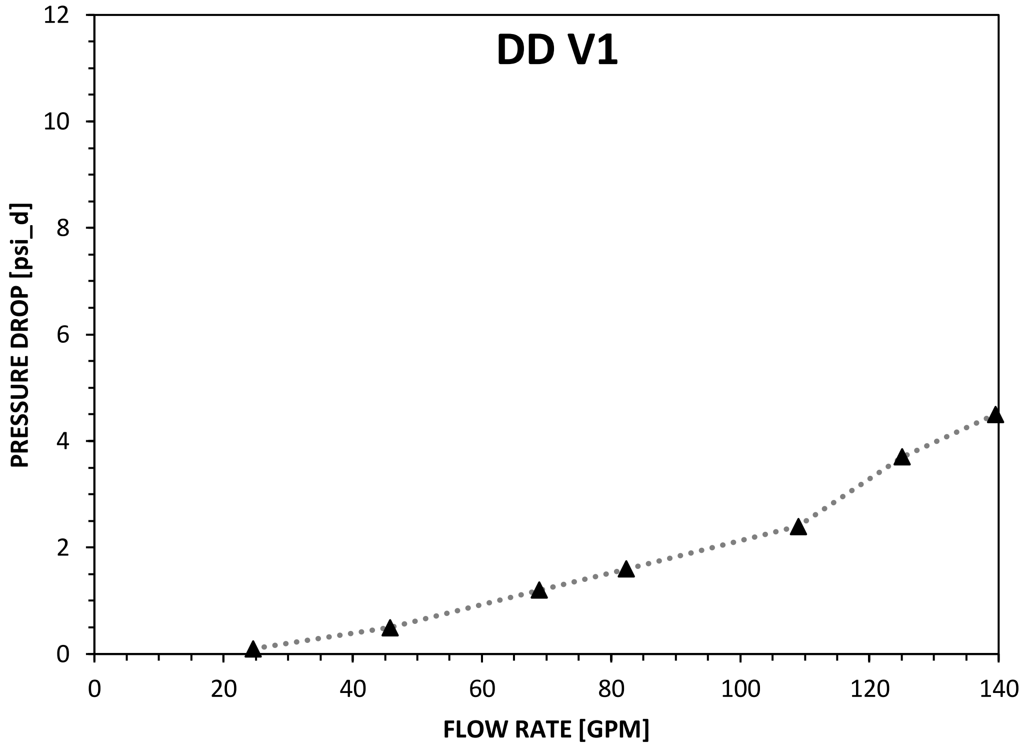
g. Repeat steps **A. - G.** Until all tests are completed

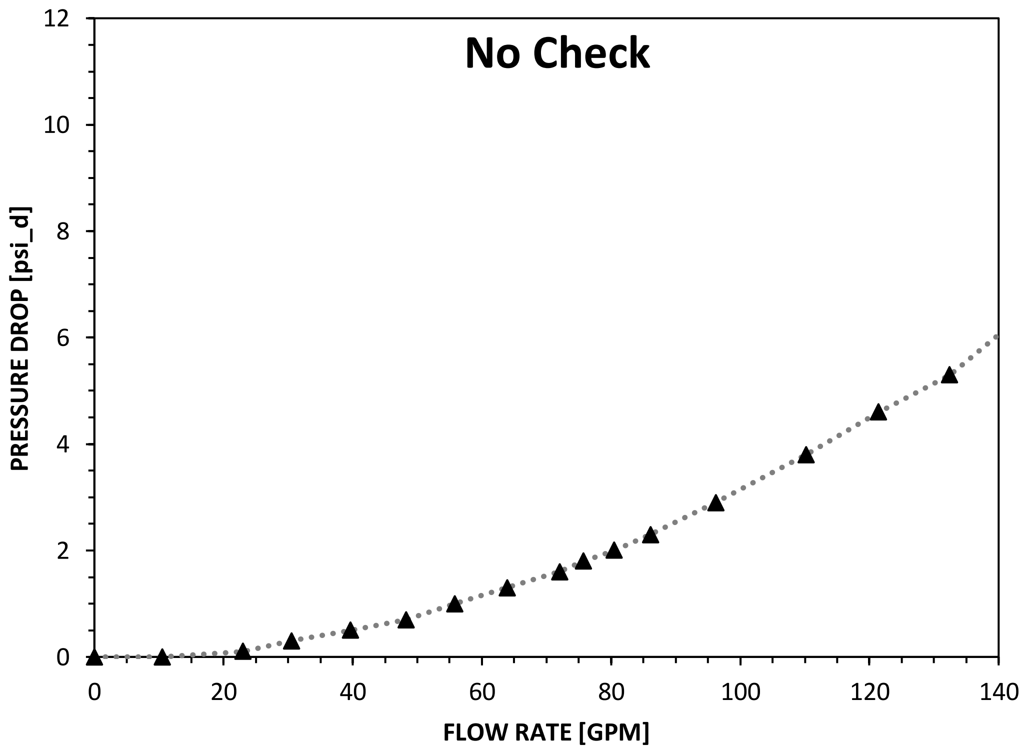
Appendix N – Test Data



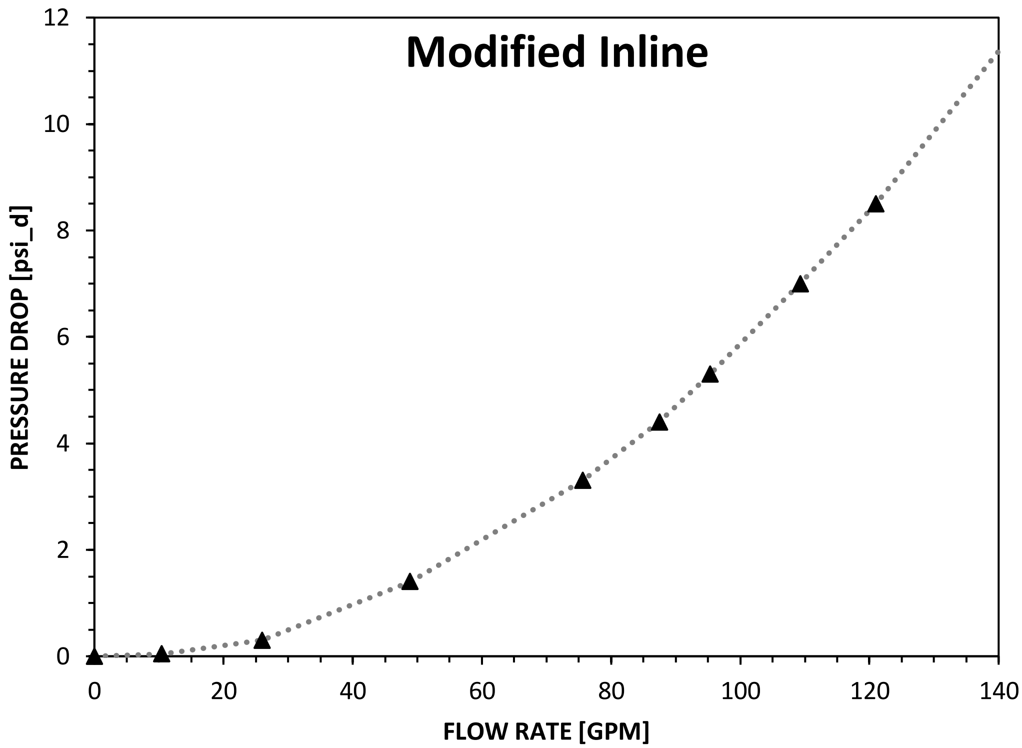


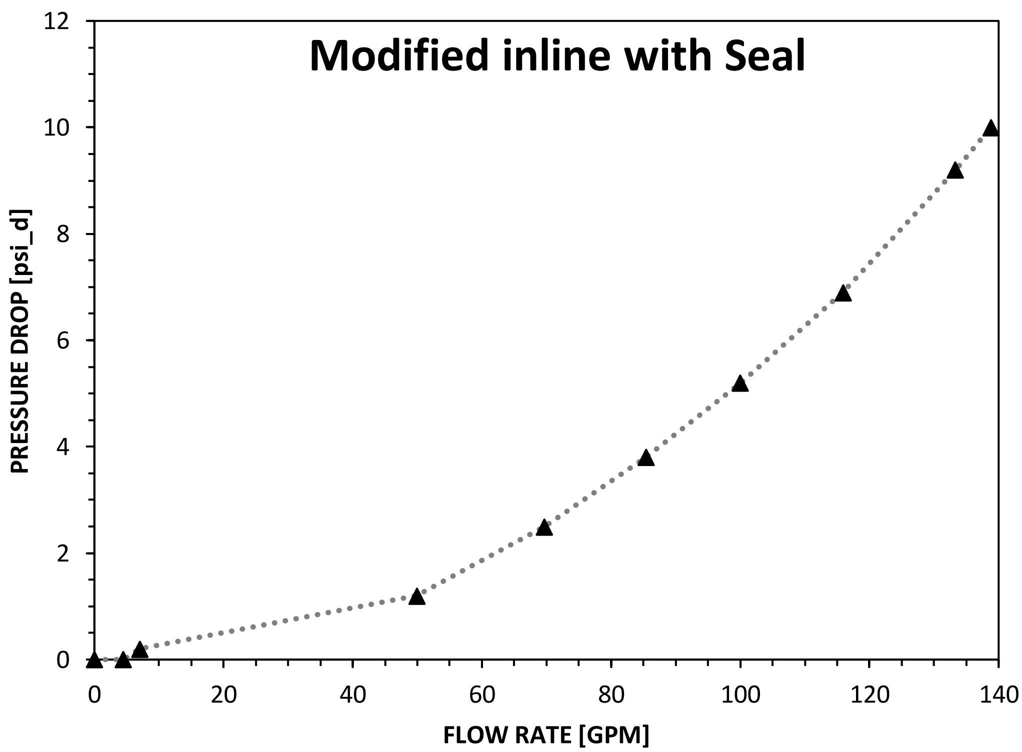
Appendix N – Test Data (Continued)



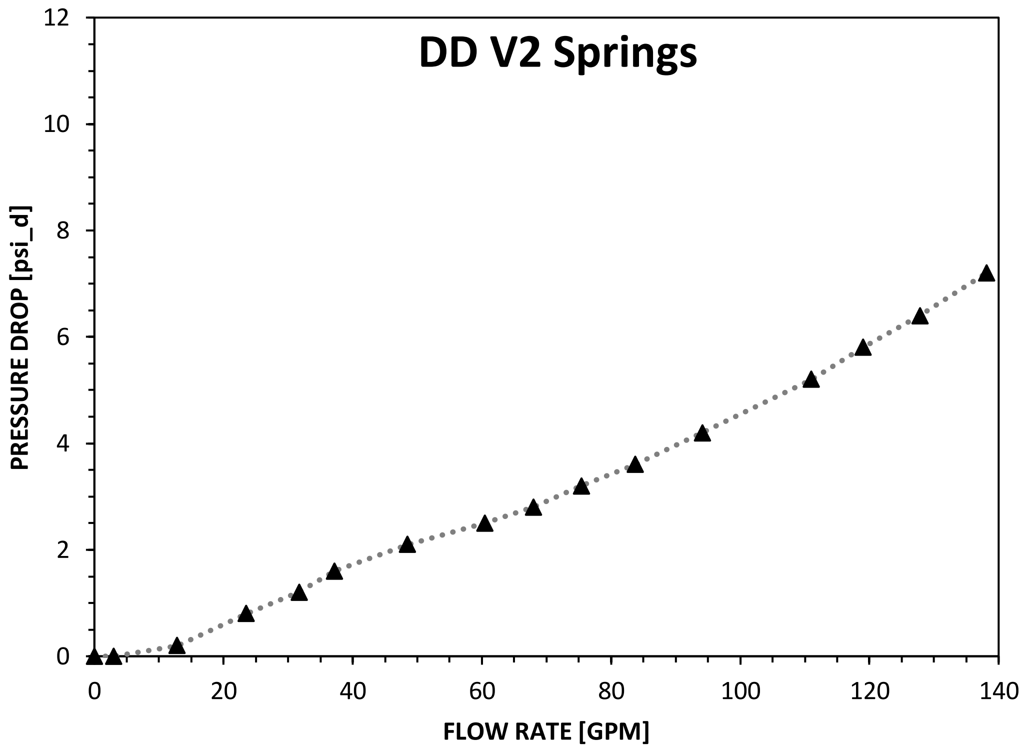


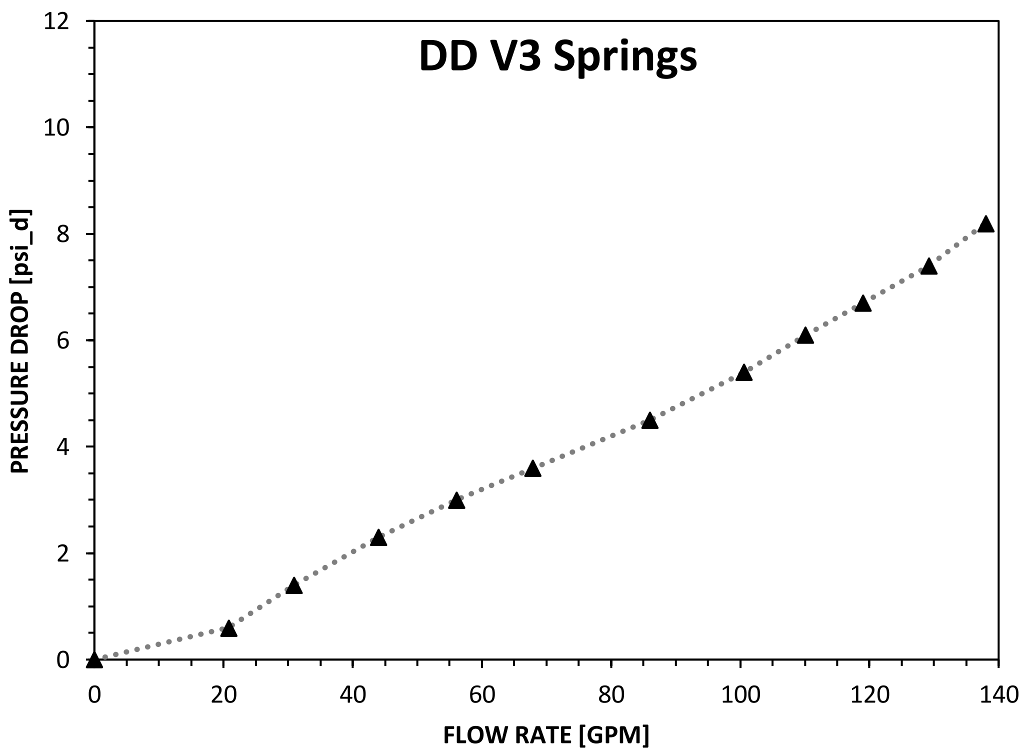
Appendix N – Test Data (Continued)



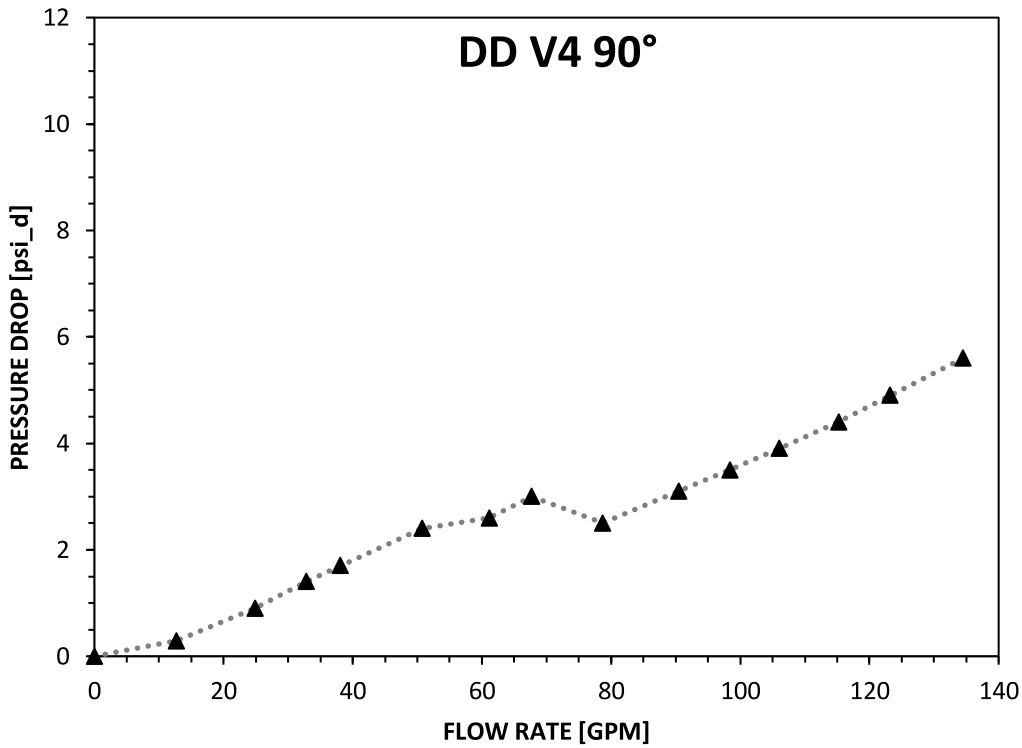


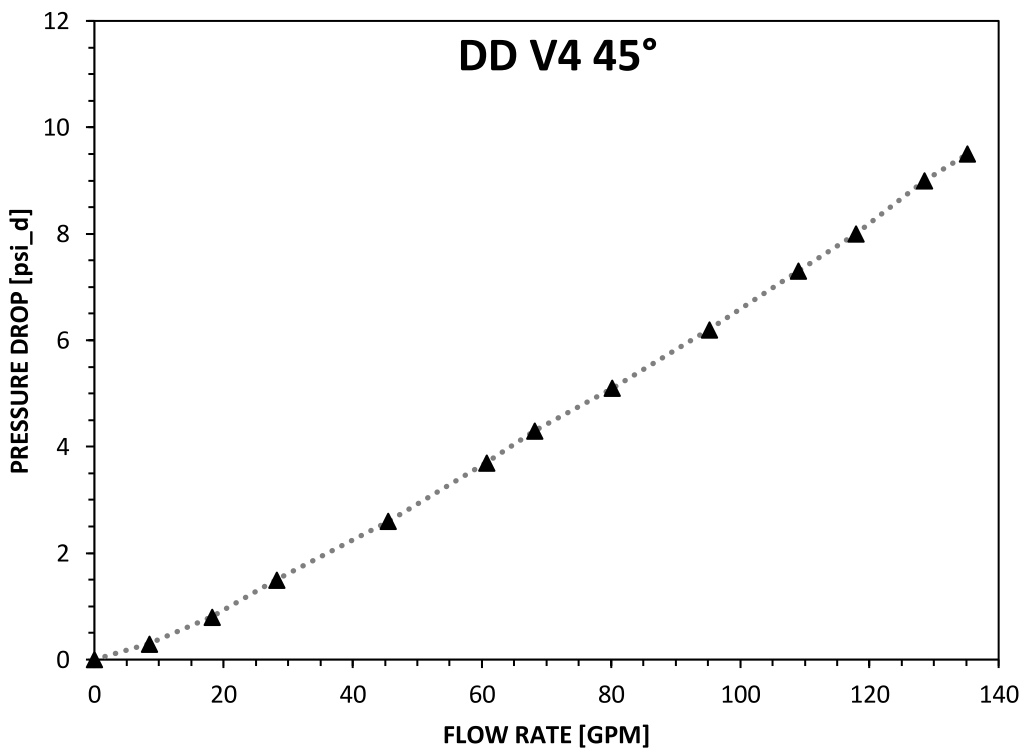
Appendix N – Test Data (Continued)



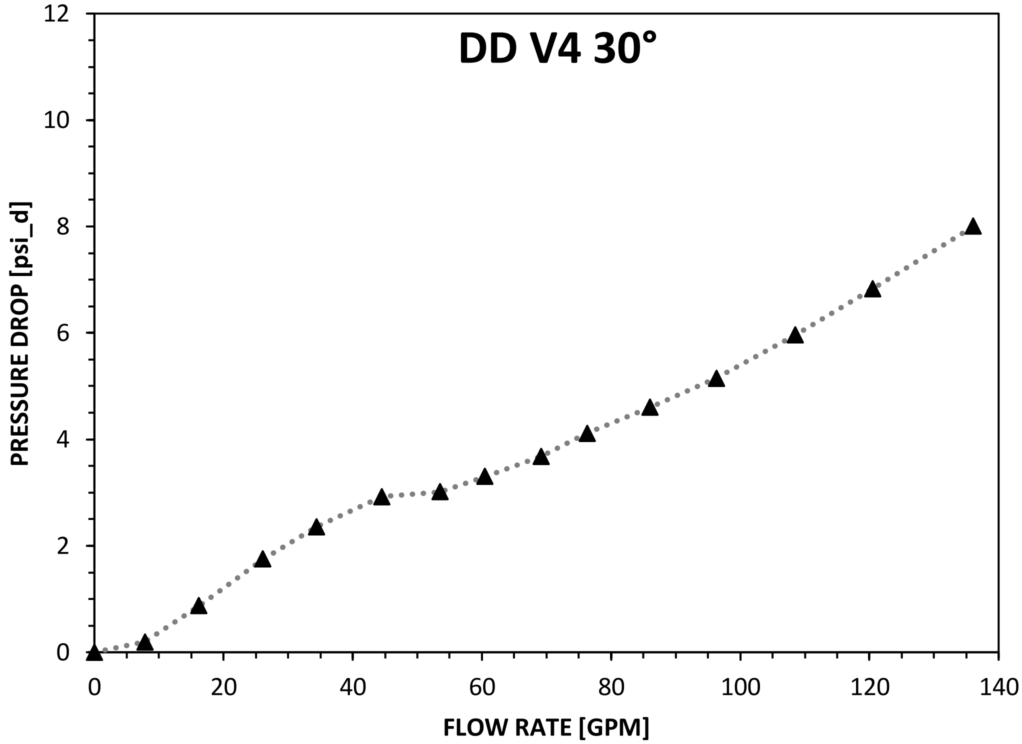


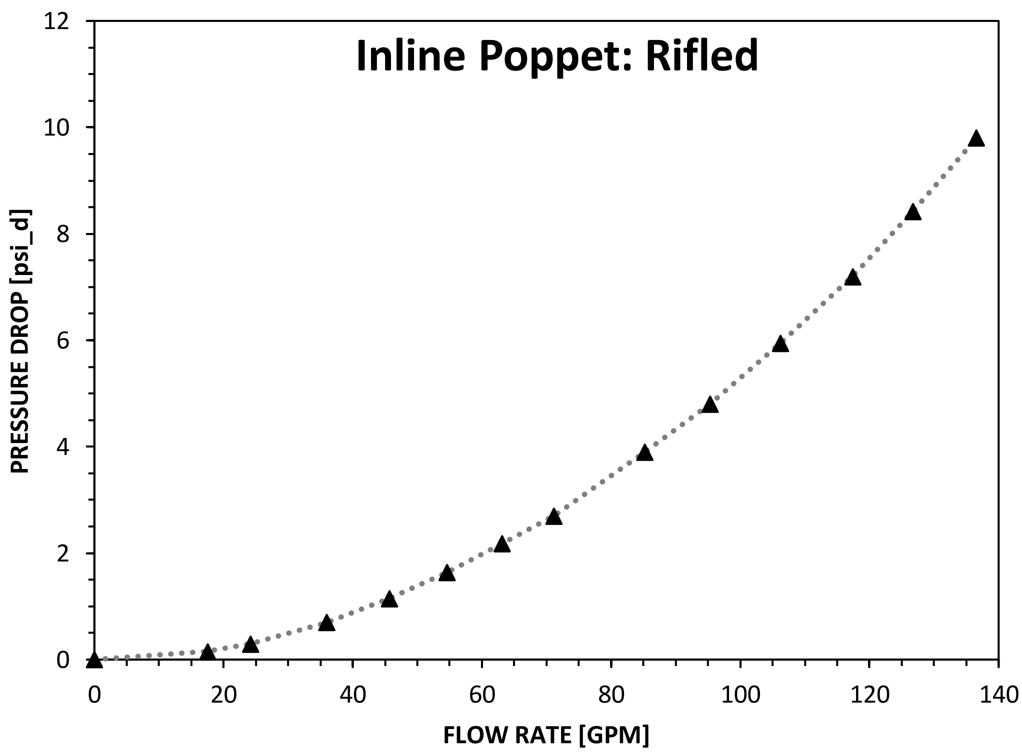
Appendix N – Test Data (Continued)



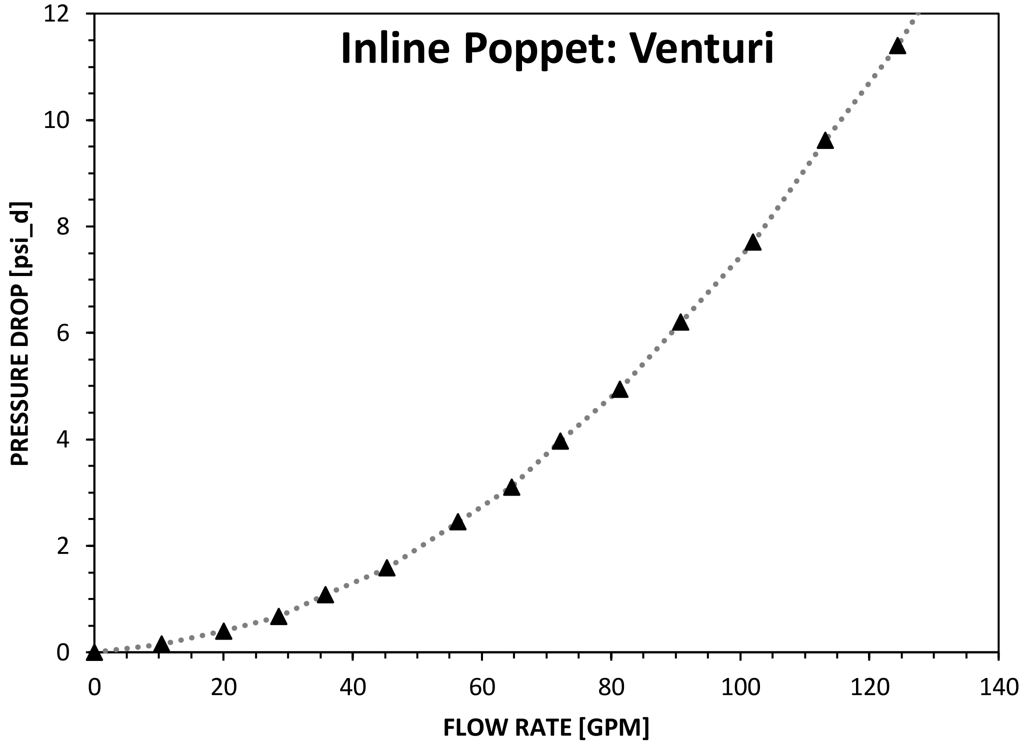


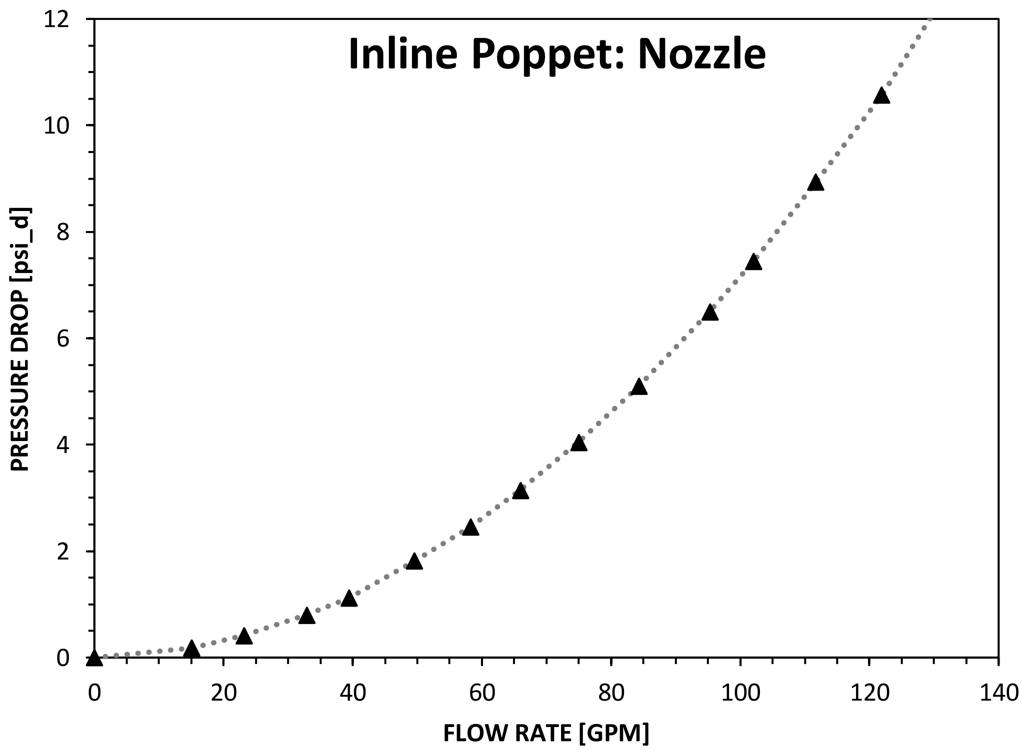
Appendix N – Test Data (Continued)



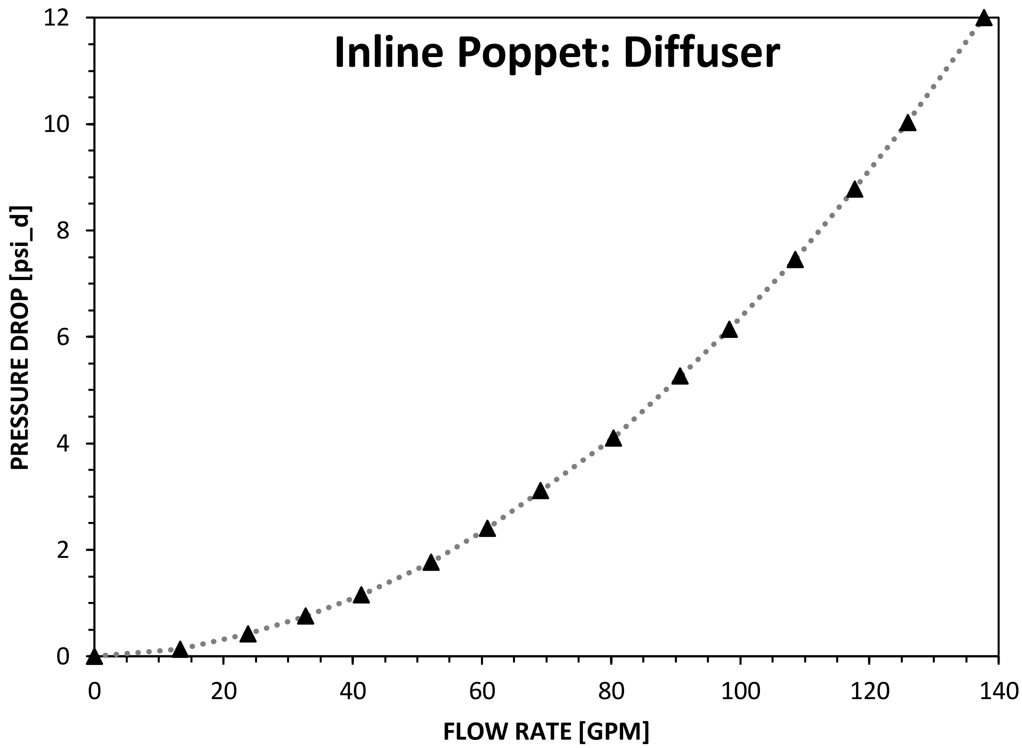


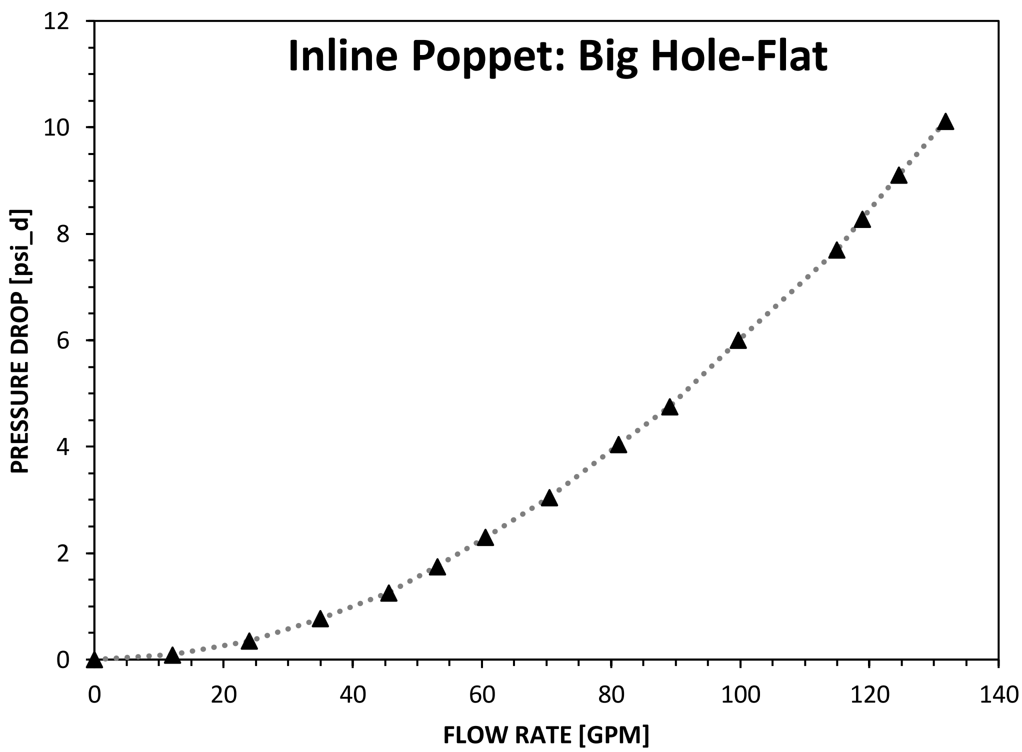
Appendix N – Test Data (Continued)



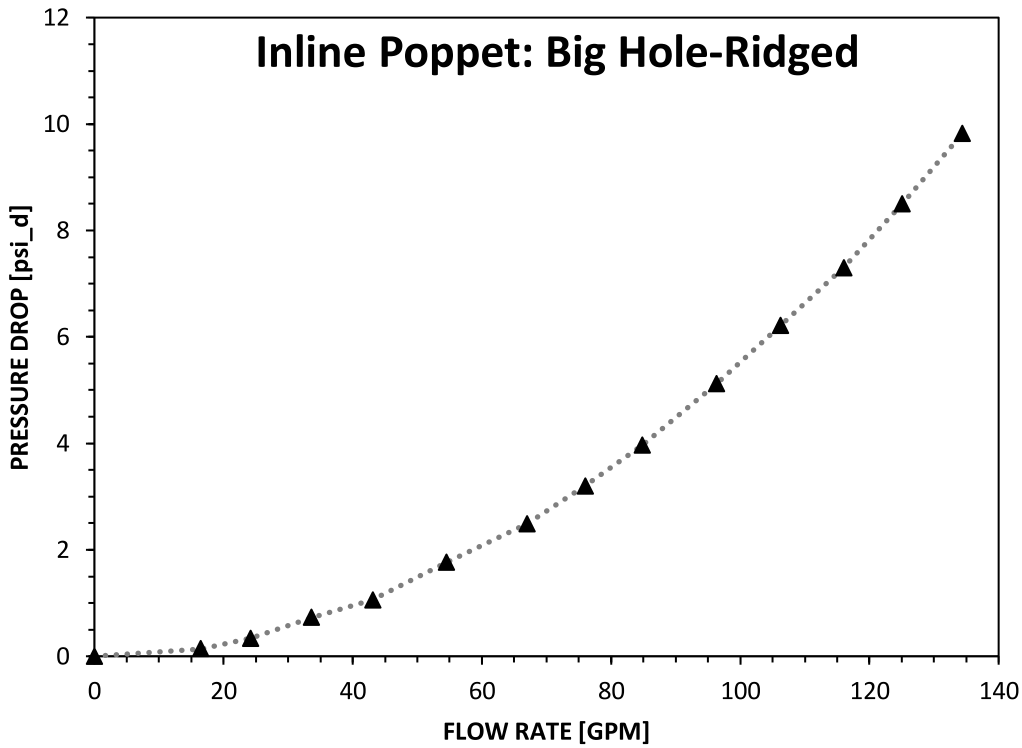


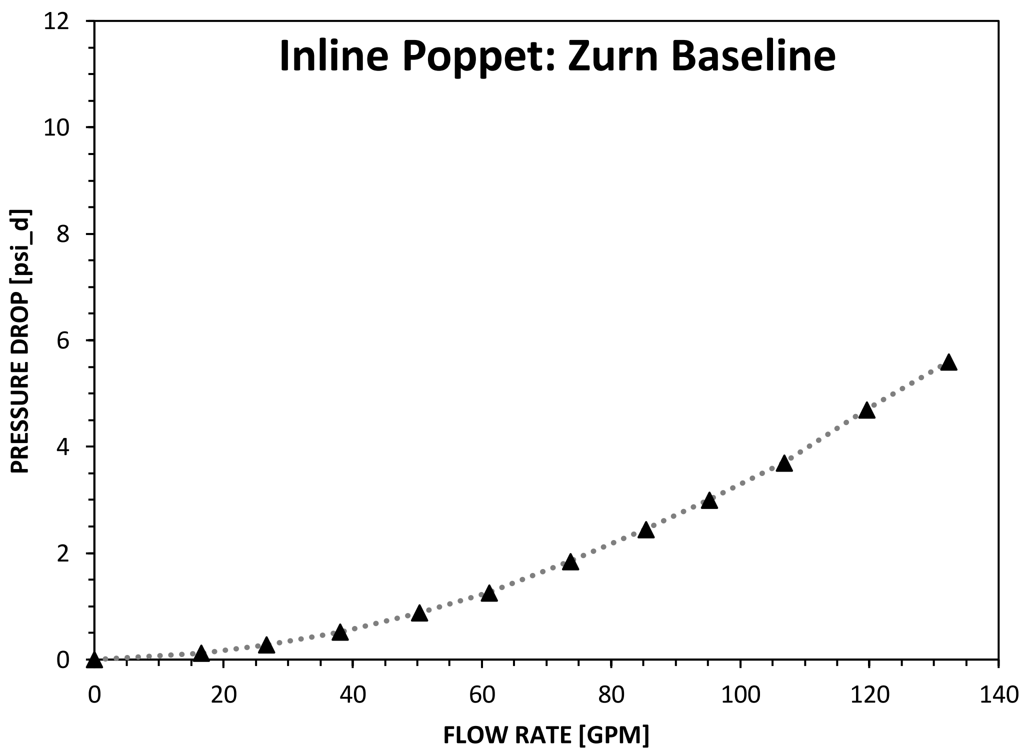
Appendix N – Test Data (Continued)



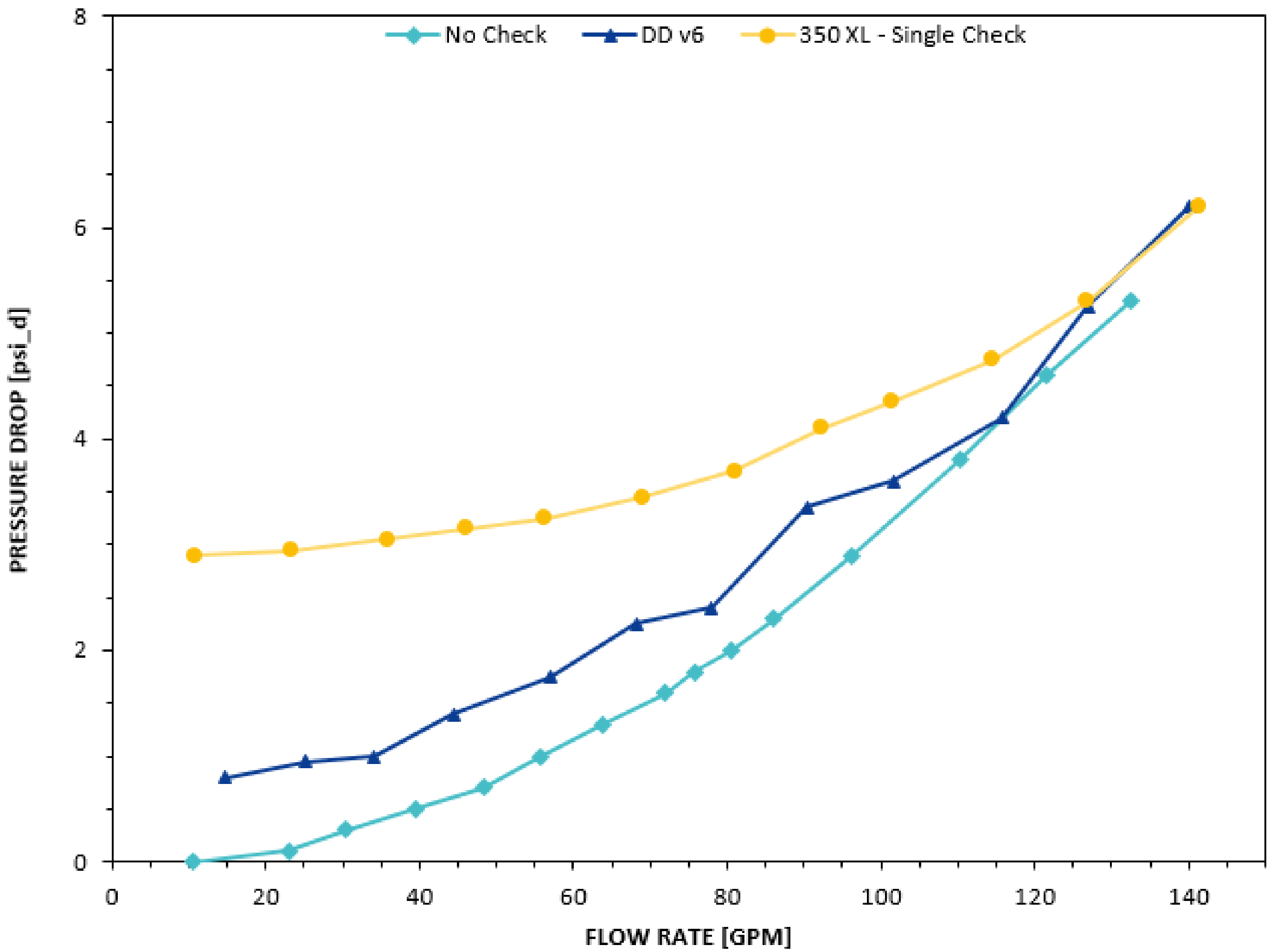


Appendix N – Test Data (Continued)





Appendix N – Test Data (Continued) – Final Test Data with DD Version 6



Appendix O – CFD Results

HOUSING_BASELINE_1

Prepared by: Skylar

Date: Saturday, June 06, 2020

HOUSING_BASELINE_1

Summary

Description

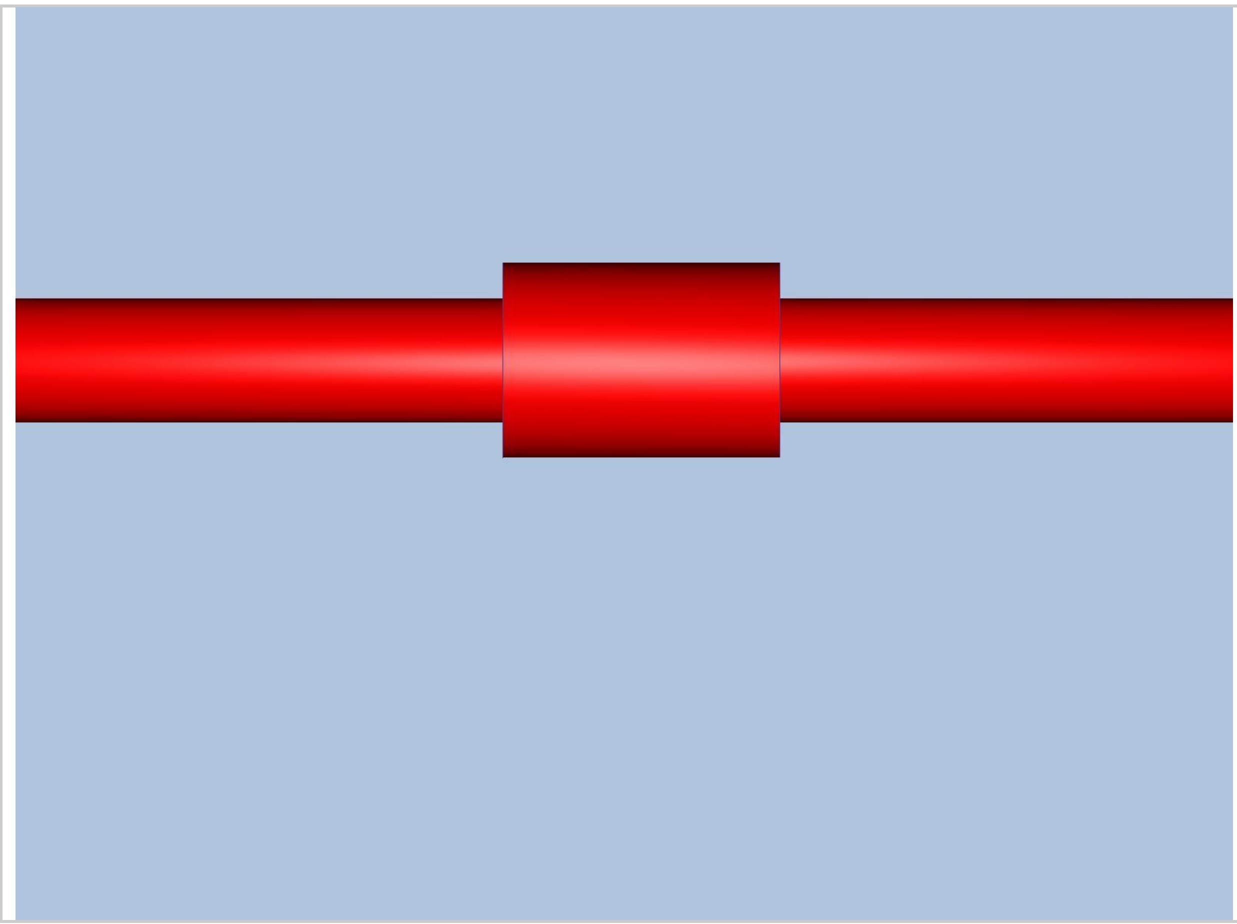
This simulation is of the test stand with no check valve. The primary purpose of this simulation is to establish the reliability of the simulation as compared to physical test data.

Design 1

Length units	inch-BTU/s
Coordinate system	Cartesian 3D

Scenario 1

Materials



NAME	ASSIGNED TO	PROPERTIES
Nylon	Part1.Revolve2	X-Direction 0.145 W/m-K Y-Direction Same as X-dir. Z-Direction Same as X-dir. Density 1.05 g/cm3 Specific heat 2.2 J/g-K Emissivity 1.0 Transmissivity 0.0 Electrical resistivity 9.43e+14 ohm-cm Wall roughness 0.0 meter
Water	CFDCreatedVolume	Density Piecewise Linear Viscosity 0.001003 Pa-s Conductivity 0.6 W/m-K Specific heat 4182.0 J/kg-K Compressibility 2185650000.0 Pa Emissivity 1.0 Wall roughness 0.0 meter Phase Linked Vapor Material

boundary conditions

TYPE	ASSIGNED TO
Pressure(0 psi Gage)	Surface:21
Volume Flow Rate(154.4 gal/min)	Surface:22

Initial Conditions

TYPE	ASSIGNED TO

mesh

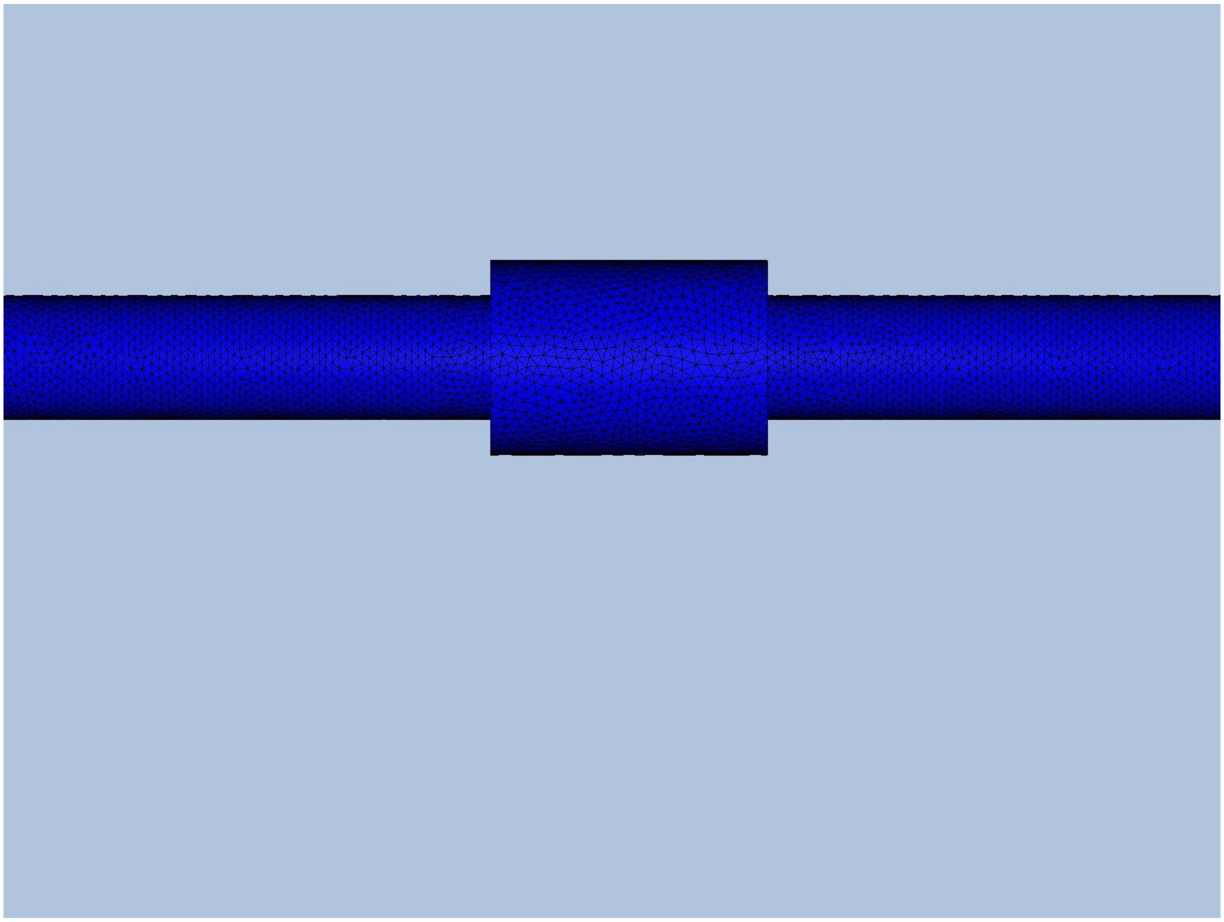
Automatic Meshing Settings

Surface refinement	0
Gap refinement	0
Resolution factor	1.0
Edge growth rate	1.1
Minimum points on edge	2
Points on longest edge	10
Surface limiting aspect ratio	20

Mesh Enhancement Settings

Mesh enhancement	1
Enhancement blending	0
Number of layers	3
Layer factor	0.45
Layer gradation	1.05

Meshed Model



Number of Nodes	45201
Number of Elements	185501

Physics

Flow	On
Compressibility	Incompressible
Heat Transfer	Off
Auto Forced Convection	Off

Gravity Components	0.0, 0.0, 0.0
Radiation	Off
Scalar	No scalar
Turbulence	On

Solver Settings

Solution mode	Steady State
Solver computer	MyComputer
Intelligent solution control	On
Advection scheme	ADV 5
Turbulence model	k-epsilon

Convergence

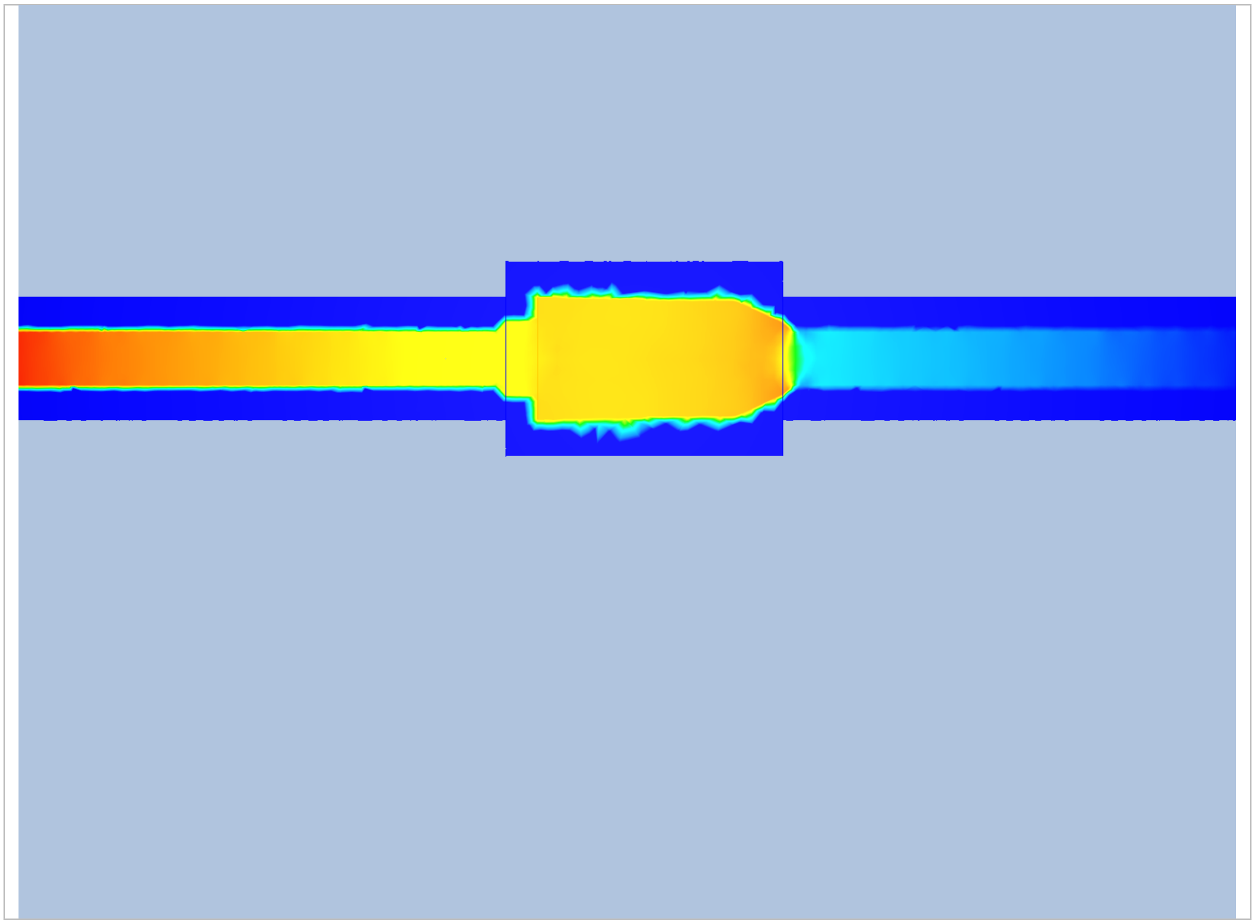
Iterations run	229
Solve time	670 seconds
Solver version	19.2.20190802

Energy Balance

Mass Balance

	IN	OUT
Mass flow	0.0554698	-0.055303
Volume flow	594.402 i^3/s	-592.614 i^3/s

Results



Inlets and Outlets

inlet 1	inlet bulk pressure	7.8016 psi
	inlet bulk	0.0 F
	inlet mach number	5.36053e-05
	mass flow in	0.0554698
	minimum x,y,z of	0.0
	node near minimum	107.0
	reynolds number	289476.0
	surface id	22.0
	total mass flow in	0.0554698
	total vol. flow in	594.402 i^3/s
	volume flow in	594.402 i^3/s
outlet 1	mass flow out	-0.055303
	minimum x,y,z of	0.0
	node near minimum	513.0
	outlet bulk pressure	-0.0 psi
	outlet bulk	-0.0 F
	outlet mach number	4.8265e-05

	reynolds number	288606.0
	surface id	21.0
	total mass flow out	-0.055303
	total vol. flow out	-592.614 i^3/s
	volume flow out	-592.614 i^3/s

Field Variable Results

VARIABLE	MAX	MIN
cond	8.01796e-06 B/in-s-R	1.93767e-06 B/in-s-R
dens	9.81631e-05 lbf-s^2/in^4	9.33204e-05 lbf-s^2/in^4
econd	1.22584 B/in-s-R	0.0 B/in-s-R
emiss	1.0	0.0
evisc	0.00332432 lbf-s/in^2	0.0 lbf-s/in^2
gent	8699.04 1/s	0.0316228 1/s
press	8.01433 psi	-0.02932 psi
ptotl	16.894 psi	-0.02932 psi
scal1	0.0	0.0
seebeck	0.0 V/K	0.0 V/K
shgc	0.0	0.0
spech	385.66 B-in/lbf-s^2R	202.882 B-in/lbf-s^2R
temp	0.0 F	0.0 F
transmiss	0.0	0.0
turbd	733211.0 in^2/s^3	0.141967 in^2/s^3
turbk	3750.16 in^2/s^2	1.45342e-10 in^2/s^2
ufactor	0.0	0.0
visc	1.45342e-07 lbf-s/in^2	0.0 lbf-s/in^2
vx vel	488.511 in/s	-92.0705 in/s
vy vel	73.5947 in/s	-75.0779 in/s
vz vel	74.4286 in/s	-76.0205 in/s
wrough	0.0 in	0.0 in

Component Thermal Summary

PART	MINIMUM TEMPERATURE	MAXIMUM TEMPERATURE	VOLUME AVERAGED TEMPERATURE
Part1.Revolve2	-5.68434E-14	-5.68434E-14	0
CFDCreatedVolume	-5.68434E-14	-5.68434E-14	0

Fluid Forces on Walls

pressx	4.9438 pounds
pressy	0.023812 pounds
pressz	-0.010038 pounds
shearx	2.1095 pounds
sheary	0.00047991 pounds
shearz	-0.0017375 pounds

Decision Center

V6_45

Prepared by: Skylar Tusting

Date: Saturday, June 06, 2020

V6_45

Summary

Description

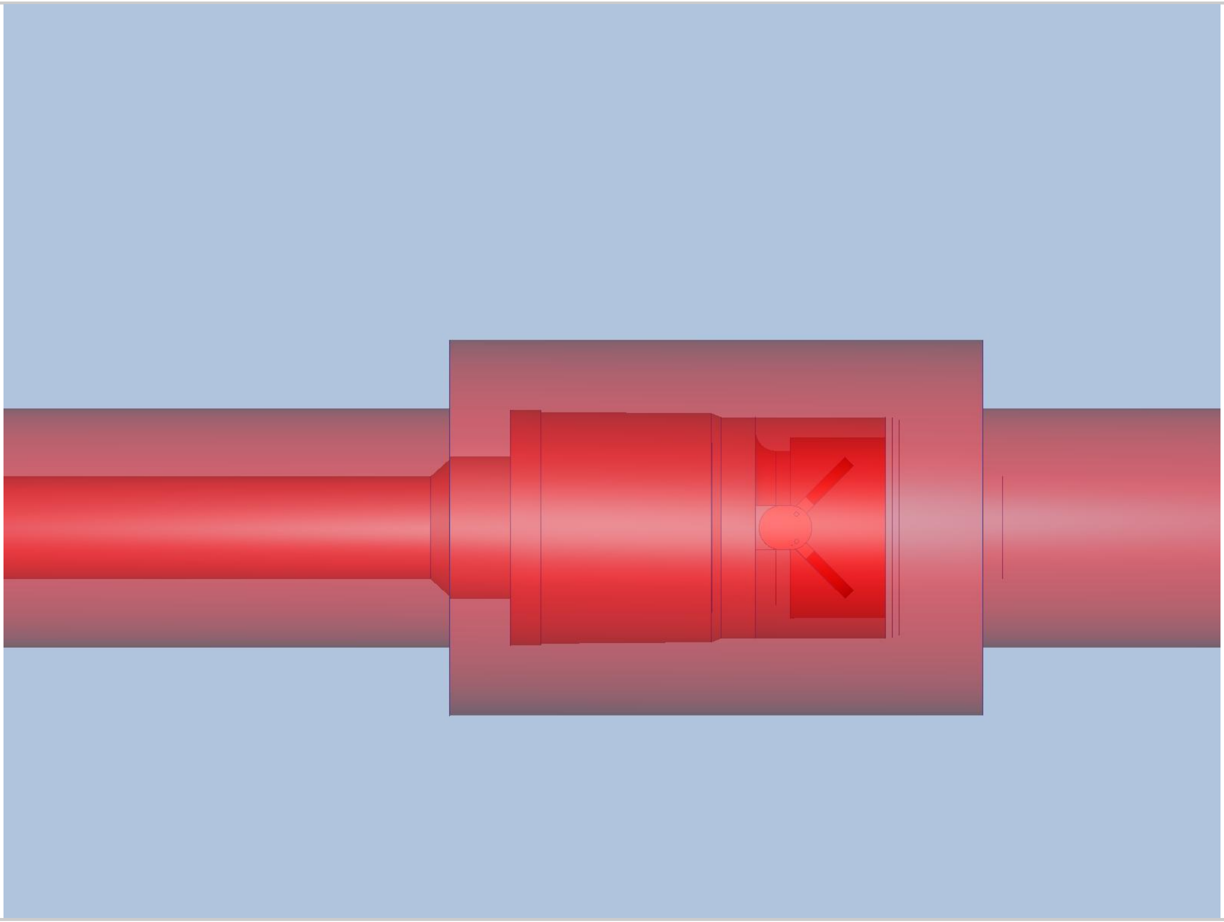
This simulation is of the test stand with the v6 #2 check. This is a static configuration with the disks at 45 degrees.

Design 1

Length units	inch-BTU/s
Coordinate system	Cartesian 3D

Scenario 1

Materials



NAME	ASSIGNED TO	PROPERTIES	
Water	CFDCreatedVolume	Density	Piecewise Linear
		Viscosity	0.001003 Pa-s
		Conductivity	0.6 W/m-K
		Specific heat	4182.0 J/kg-K
		Compressibility	2185650000.0 Pa
		Emissivity	1.0
		Wall roughness	0.0 meter
		Phase	Linked Vapor
Nylon	check2_frame VALVE_BODY_CFD (2) dd_poppet check2_hinge dd_poppet	X-Direction	0.145 W/m-K
		Y-Direction	Same as X-dir.
		Z-Direction	Same as X-dir.
		Density	1.05 g/cm3
		Specific heat	2.2 J/g-K
		Emissivity	1.0
		Transmissivity	0.0
		Electrical resistivity	9.43e+14 ohm-cm
		Wall roughness	0.0 meter

boundary conditions

TYPE	ASSIGNED TO
Pressure(0 psi Gage)	Surface:81
Volume Flow Rate(100 gal/min)	Surface:82

Initial Conditions

TYPE	ASSIGNED TO

mesh

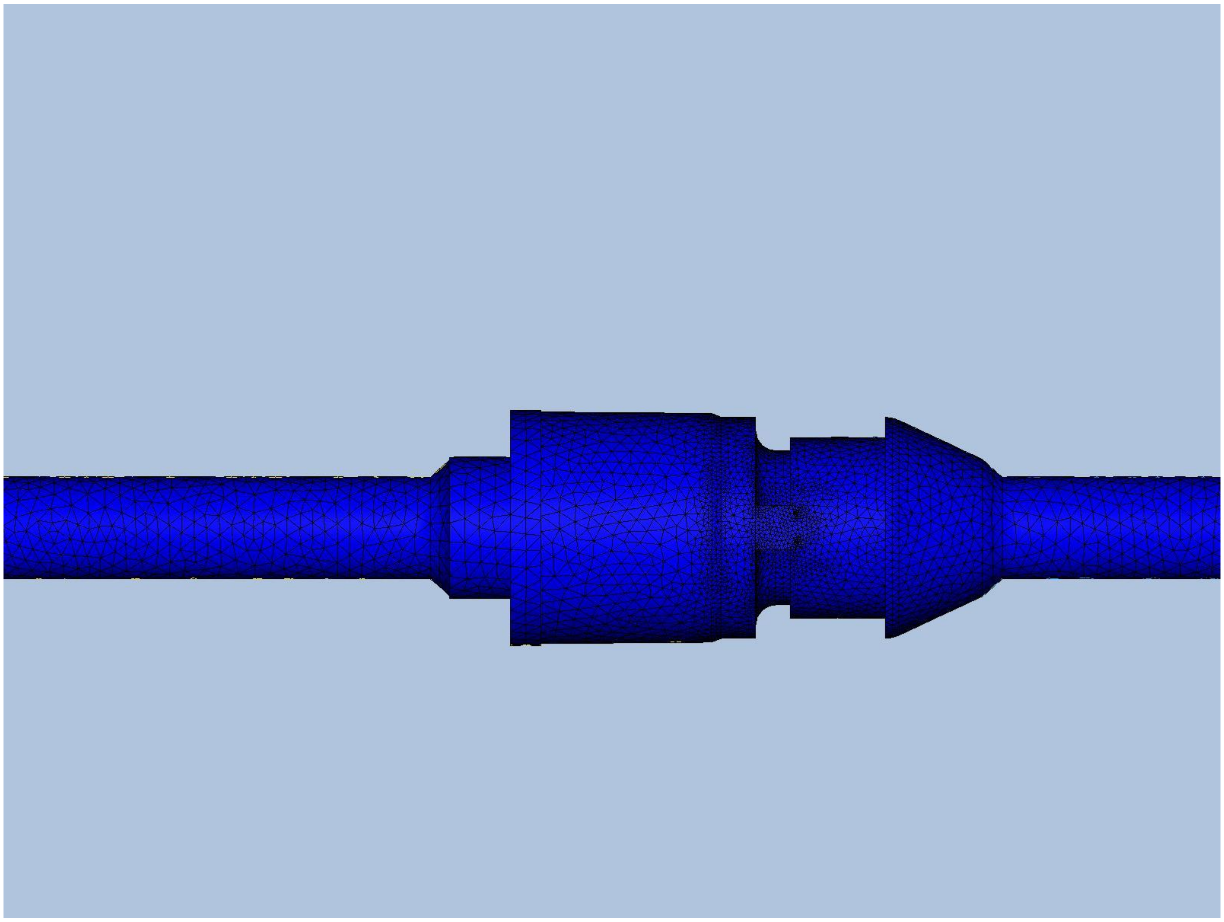
Automatic Meshing Settings

Surface refinement	0
Gap refinement	0
Resolution factor	1.0
Edge growth rate	1.1
Minimum points on edge	2
Points on longest edge	10
Surface limiting aspect ratio	20

Mesh Enhancement Settings

Mesh enhancement	1
Enhancement blending	0
Number of layers	3
Layer factor	0.45
Layer gradation	1.05

Meshed Model



Number of Nodes	113554
Number of Elements	491072

Physics

Flow	On
Compressibility	Incompressible
Heat Transfer	Off
Auto Forced Convection	Off

Gravity Components	0.0, 0.0, 0.0
Radiation	Off
Scalar	No scalar
Turbulence	On

Solver Settings

Solution mode	Steady State
Solver computer	MyComputer
Intelligent solution control	On
Advection scheme	ADV 5
Turbulence model	k-epsilon

Convergence

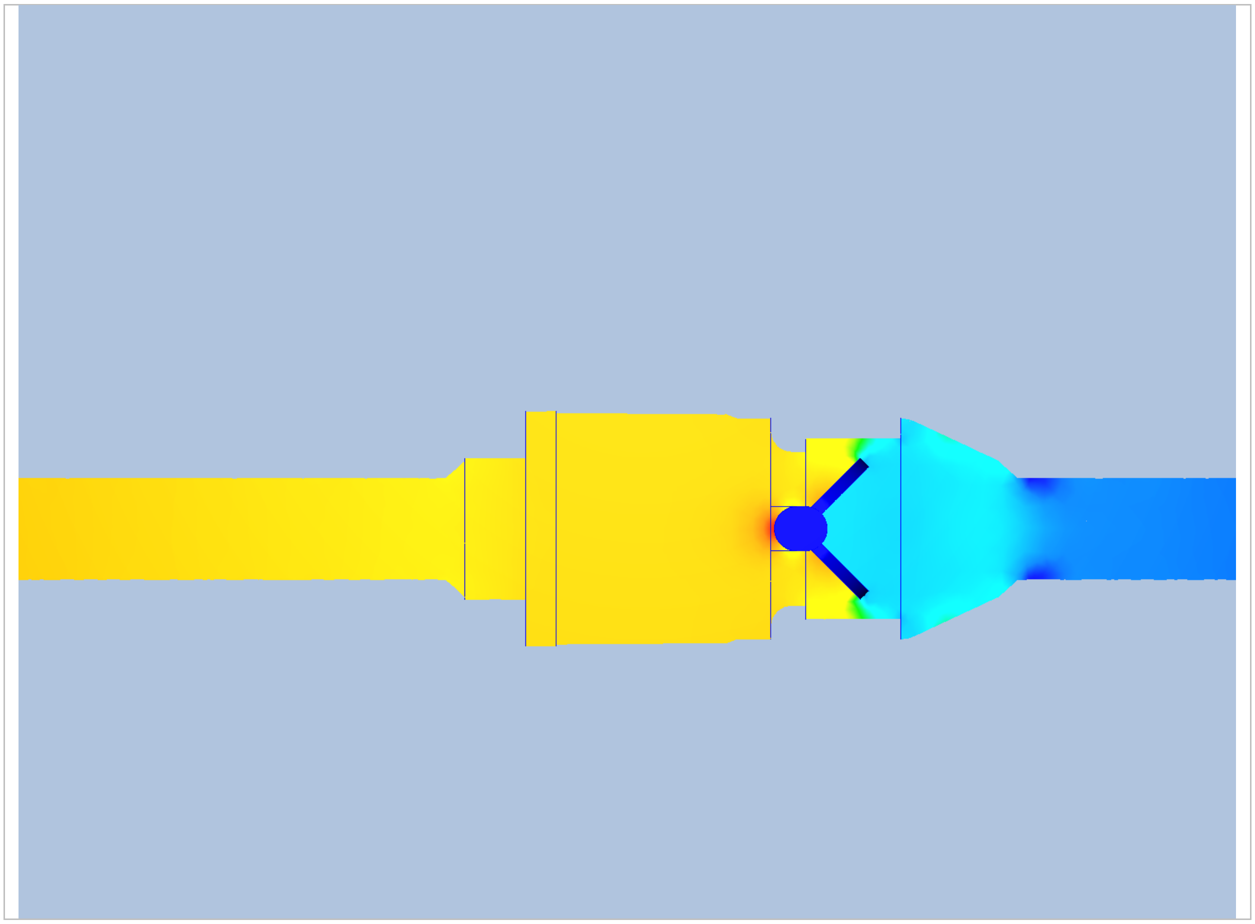
Iterations run	400
Solve time	1316 seconds
Solver version	19.2.20190802

Energy Balance

Mass Balance

	IN	OUT
Mass flow	0.035926	-0.0358429
Volume flow	384.975 i^3/s	-384.085 i^3/s

Results



Inlets and Outlets

inlet 1	inlet bulk pressure	10.0266 psi
	inlet bulk	0.0 F
	inlet mach number	3.50005e-05
	mass flow in	0.035926
	minimum x,y,z of	0.0
	node near minimum	854.0
	reynolds number	187652.0
	surface id	82.0
	total mass flow in	0.035926
	total vol. flow in	384.975 i^3/s
	volume flow in	384.975 i^3/s
outlet 1	mass flow out	-0.0358429
	minimum x,y,z of	0.0
	node near minimum	1437.0
	outlet bulk pressure	-0.0 psi
	outlet bulk	-0.0 F
	outlet mach number	3.10509e-05

	reynolds number	187218.0
	surface id	81.0
	total mass flow out	-0.0358429
	total vol. flow out	-384.085 i^3/s
	volume flow out	-384.085 i^3/s

Field Variable Results

VARIABLE	MAX	MIN
cond	8.01796e-06 B/in-s-R	1.93767e-06 B/in-s-R
dens	9.81631e-05 lbf-s^2/in^4	9.33204e-05 lbf-s^2/in^4
econd	5.43361 B/in-s-R	0.0 B/in-s-R
emiss	1.0	0.0
evisc	0.00287655 lbf-s/in^2	0.0 lbf-s/in^2
gent	11602700.0 1/s	0.0316228 1/s
press	12.6806 psi	-0.668982 psi
ptotl	14.2131 psi	-0.668982 psi
scal1	0.0	0.0
seebeck	0.0 V/K	0.0 V/K
shgc	0.0	0.0
spech	385.66 B-in/lbf-s^2R	202.882 B-in/lbf-s^2R
temp	0.0 F	0.0 F
transmiss	0.0	0.0
turbd	626900000000.0 in^2/s^3	0.00133109 in^2/s^3
turbk	7893460.0 in^2/s^2	1.45342e-10 in^2/s^2
ufactor	0.0	0.0
visc	1.45342e-07 lbf-s/in^2	0.0 lbf-s/in^2
vx vel	392.119 in/s	-406.908 in/s
vy vel	238.063 in/s	-260.103 in/s
vz vel	406.32 in/s	-100.075 in/s
wrough	0.0 in	0.0 in

Component Thermal Summary

PART	MINIMUM TEMPERATURE	MAXIMUM TEMPERATURE	VOLUME AVERAGED TEMPERATURE
check2_frame	-5.68434E-14	-5.68434E-14	0
VALVE_BODY_CFD (2)	-5.68434E-14	-5.68434E-14	0
dd_poppet	-5.68434E-14	-5.68434E-14	0
check2_hinge	-5.68434E-14	-5.68434E-14	0
dd_poppet	-5.68434E-14	-5.68434E-14	0
CFDCreatedVolume	-5.68434E-14	-5.68434E-14	0

Fluid Forces on Walls

pressx	-0.015137 pounds
pressy	0.021737 pounds
pressz	11.159 pounds
shearx	0.0037167 pounds
sheary	-0.00017576 pounds
shearz	1.1936 pounds

Decision Center

summary images

Image 01

Design 1::Scenario 1

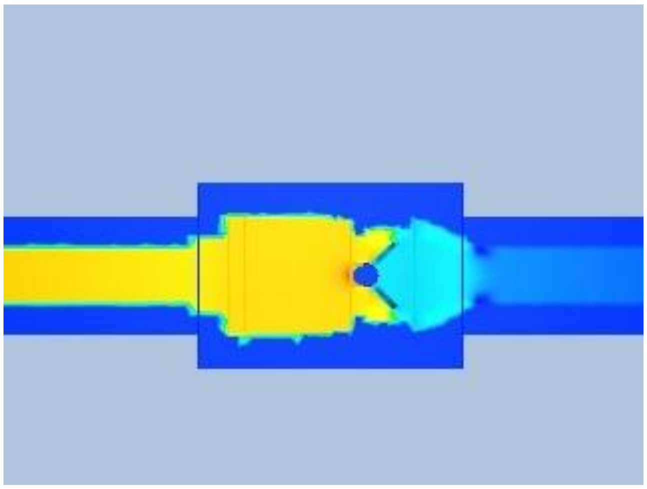


Image 02

Design 1::Scenario 1

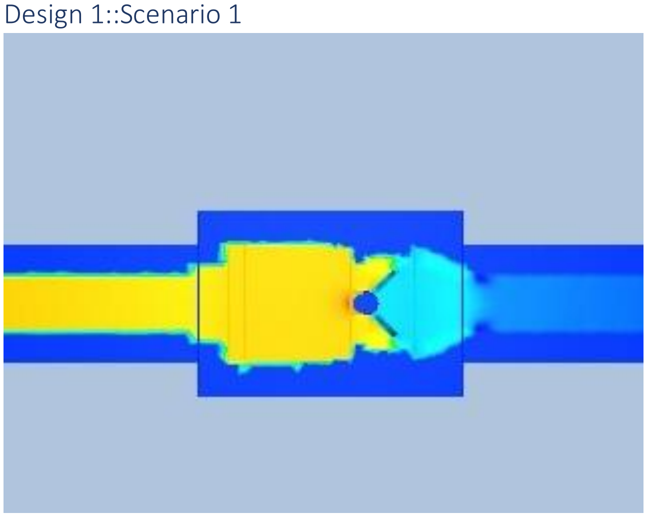
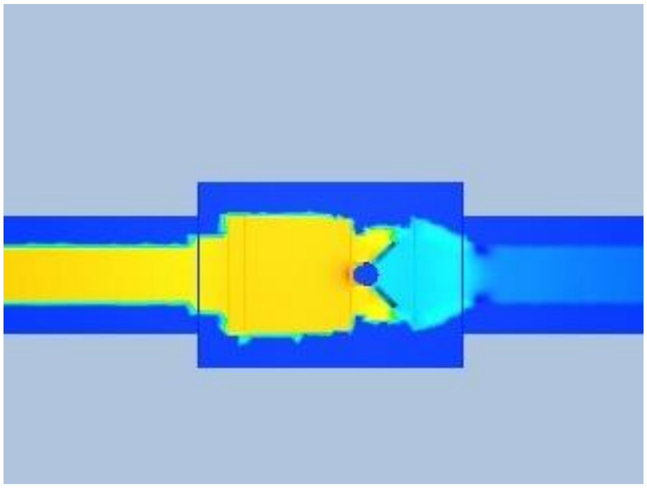


Image 03

Design 1::Scenario 1



V6_90

Prepared by: Skylar Tusting

Date: Saturday, June 06, 2020

V6_90

Summary

Description

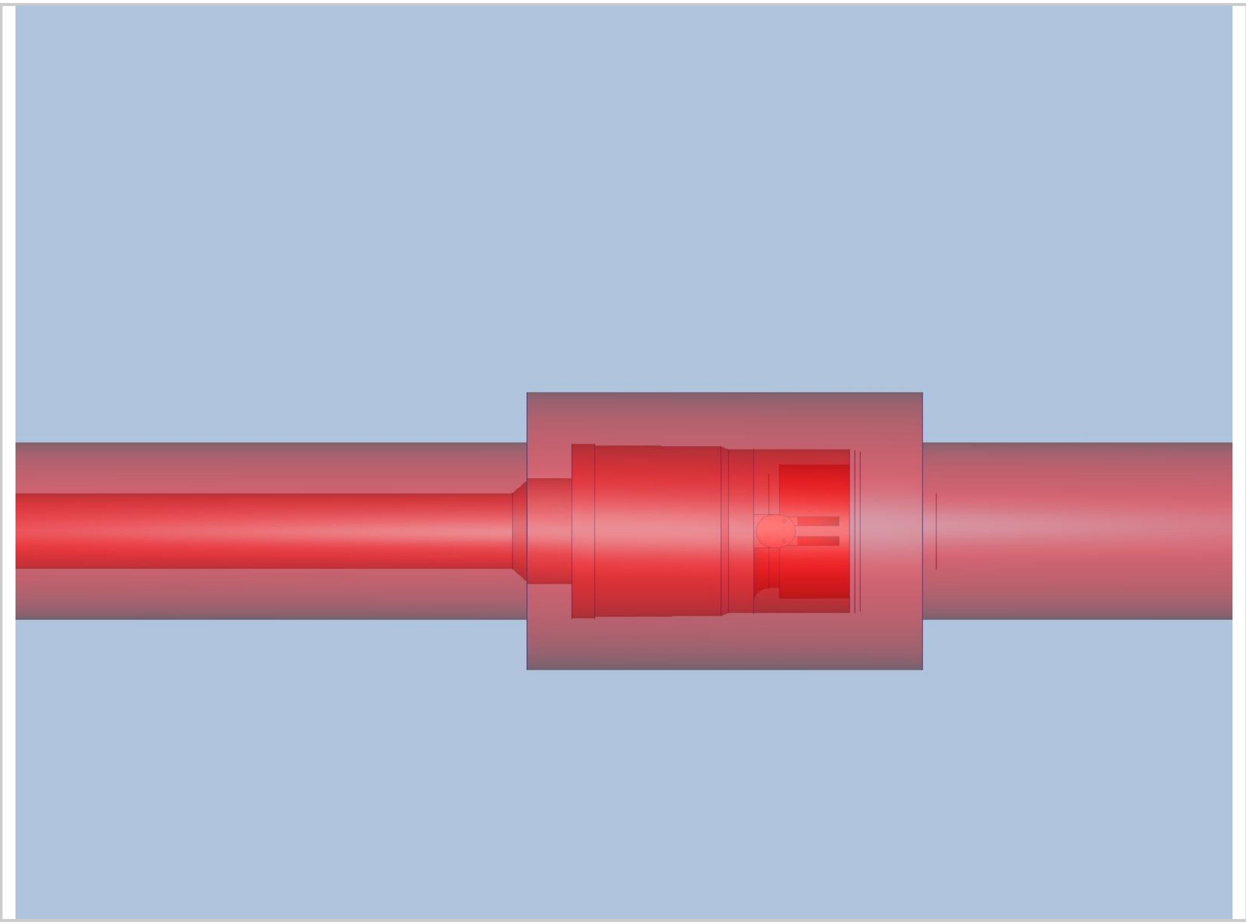
This simulation is of the test stand with the v6 #2 check. This is a static configuration with the disks at 90 degrees.

Design 1

Length units	inch-BTU/s
Coordinate system	Cartesian 3D

Scenario 1

Materials



NAME	ASSIGNED TO	PROPERTIES	
Nylon	check2_frame VALVE_BODY_CFD (2) dd_poppet check2_hinge dd_poppet	X-Direction Y-Direction Z-Direction Density Specific heat Emissivity Transmissivity Electrical resistivity Wall roughness	0.145 W/m-K Same as X-dir. Same as X-dir. 1.05 g/cm3 2.2 J/g-K 1.0 0.0 9.43e+14 ohm-cm 0.0 meter
Water	CFDCreatedVolume	Density Viscosity Conductivity Specific heat Compressibility Emissivity Wall roughness Phase	Piecewise Linear 0.001003 Pa-s 0.6 W/m-K 4182.0 J/kg-K 2185650000.0 Pa 1.0 0.0 meter Linked Vapor Material

boundary conditions

TYPE	ASSIGNED TO
Pressure(0 psi Gage)	Surface:87
Volume Flow Rate(100 gal/min)	Surface:88

Initial Conditions

TYPE	ASSIGNED TO

mesh

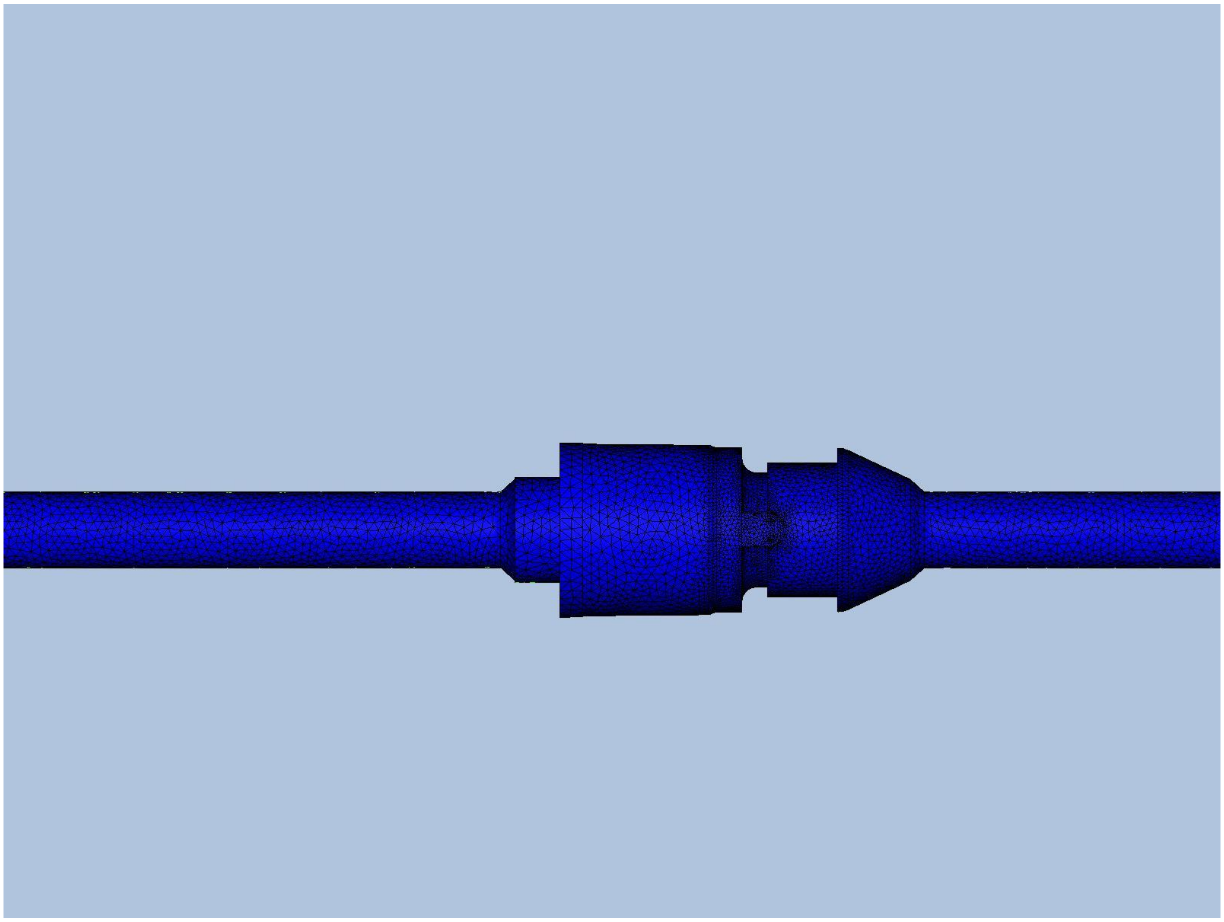
Automatic Meshing Settings

Surface refinement	0
Gap refinement	0
Resolution factor	1.0
Edge growth rate	1.1
Minimum points on edge	2
Points on longest edge	10
Surface limiting aspect ratio	20

Mesh Enhancement Settings

Mesh enhancement	1
Enhancement blending	0
Number of layers	3
Layer factor	0.45
Layer gradation	1.05

Meshed Model



Number of Nodes	136858
Number of Elements	599674

Physics

Flow	On
Compressibility	Incompressible
Heat Transfer	Off
Auto Forced Convection	Off

Gravity Components	0.0, 0.0, 0.0
Radiation	Off
Scalar	No scalar
Turbulence	On

Solver Settings

Solution mode	Steady State
Solver computer	MyComputer
Intelligent solution control	On
Advection scheme	ADV 5
Turbulence model	k-epsilon

Convergence

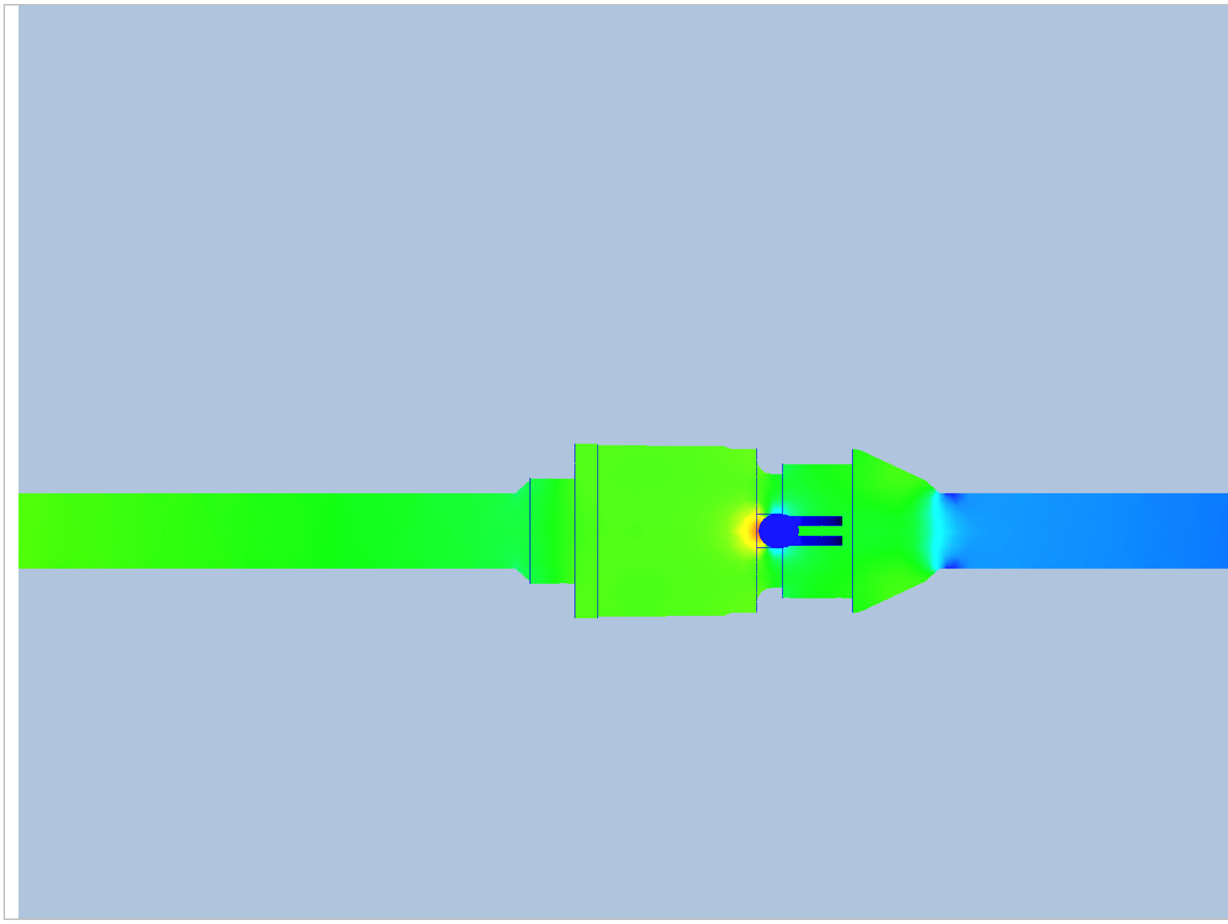
Iterations run	400
Solve time	4233 seconds
Solver version	19.2.20190802

Energy Balance

Mass Balance

	IN	OUT
Mass flow	0.035926	-0.0358869
Volume flow	384.975 i^3/s	-384.556 i^3/s

Results



Inlets and Outlets

inlet 1	inlet bulk pressure	3.69139 psi
	inlet bulk	0.0 F
	inlet mach number	3.49145e-05
	mass flow in	0.035926
	minimum x,y,z of	0.0
	node near minimum	943.0
	reynolds number	186927.0
	surface id	88.0
	total mass flow in	0.035926
	total vol. flow in	384.975 i^3/s
	volume flow in	384.975 i^3/s
outlet 1	mass flow out	-0.0358869
	minimum x,y,z of	0.0
	node near minimum	1600.0
	outlet bulk pressure	-0.0 psi
	outlet bulk	-0.0 F
	outlet mach number	3.16742e-05

	reynolds number 186724.0 surface id 87.0 total mass flow out -0.0358869 total vol. flow out -384.556 i^3/s volume flow out -384.556 i^3/s

Field Variable Results

VARIABLE	MAX	MIN
cond	8.01796e-06 B/in-s-R	1.93767e-06 B/in-s-R
dens	9.81631e-05 lbf-s^2/in^4	9.33204e-05 lbf-s^2/in^4
econd	8.01796 B/in-s-R	0.0 B/in-s-R
emiss	1.0	0.0
evisc	0.0651597 lbf-s/in^2	0.0 lbf-s/in^2
gent	19642500.0 1/s	0.0316228 1/s
press	6.41328 psi	-0.304503 psi
ptotl	6.77423 psi	-0.304503 psi
scal1	0.0	0.0
seebeck	0.0 V/K	0.0 V/K
shgc	0.0	0.0
spech	385.66 B-in/lbf-s^2R	202.882 B-in/lbf-s^2R
temp	0.0 F	0.0 F
transmiss	0.0	0.0
turbd	1.8732e+12 in^2/s^3	0.00527655 in^2/s^3
turbk	18064600.0 in^2/s^2	1.45342e-10 in^2/s^2
ufactor	0.0	0.0
visc	1.45342e-07 lbf-s/in^2	0.0 lbf-s/in^2
vx vel	128.411 in/s	-132.479 in/s
vy vel	122.358 in/s	-103.546 in/s
vz vel	281.121 in/s	-86.0759 in/s
wrough	0.0 in	0.0 in

Component Thermal Summary

PART	MINIMUM TEMPERATURE	MAXIMUM TEMPERATURE	VOLUME AVERAGED TEMPERATURE
check2_frame	-5.68434E-14	-5.68434E-14	0
VALVE_BODY_CFD (2)	-5.68434E-14	-5.68434E-14	0
dd_poppet	-5.68434E-14	-5.68434E-14	0
check2_hinge	-5.68434E-14	-5.68434E-14	0
dd_poppet	-5.68434E-14	-5.68434E-14	0
CFDCreatedVolume	-5.68434E-14	-5.68434E-14	0

Fluid Forces on Walls

pressx	-0.001079 pounds
pressy	-0.025699 pounds
pressz	3.6845 pounds
shearx	-0.00057781 pounds
sheary	-0.00064027 pounds
shearz	1.1111 pounds

V6_MOTION

Prepared by: Skylar Tusting

Date: Saturday, June 06, 2020

V6_MOTION

Summary

Description

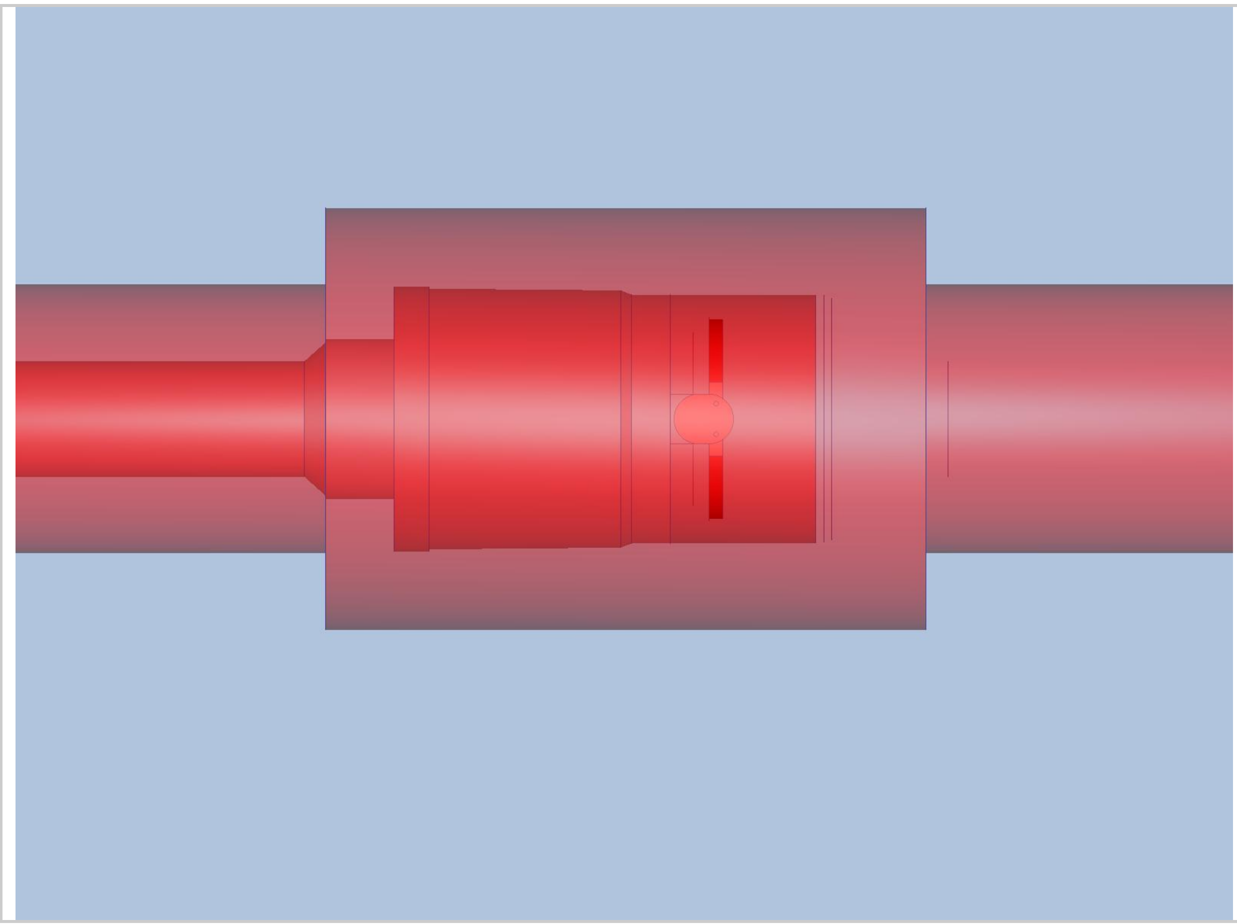
This simulation is of the test stand with the v6 #2 check. This is a transient model with angular motion elements for the poppet disks with torsional springs as resistive torque elements. This was our first successful transient model.

Design 1

Length units	inch-BTU/s
Coordinate system	Cartesian 3D

Scenario 1

Materials



NAME	ASSIGNED TO	PROPERTIES	
Nylon	dd_poppet_CFD VALVE_BODY_CFD dd_poppet_CFD check2_hinge_CFD check2_frame_CFD	X-Direction Y-Direction Z-Direction Density Specific heat Emissivity Transmissivity Electrical resistivity Wall roughness	0.145 W/m-K Same as X-dir. Same as X-dir. 1.05 g/cm3 2.2 J/g-K 1.0 0.0 9.43e+14 ohm-cm 0.0 meter
Water	CFDCreatedVolume	Density Viscosity Conductivity Specific heat Compressibility Emissivity Wall roughness Phase	Piecewise Linear 0.001003 Pa-s 0.6 W/m-K 4182.0 J/kg-K 2185650000.0 Pa 1.0 0.0 meter Linked Vapor Material

boundary conditions

TYPE	ASSIGNED TO
Pressure(0 psi Gage)	Surface:87
Volume Flow Rate(100 gal/min)	Surface:88

Initial Conditions

TYPE	ASSIGNED TO

mesh

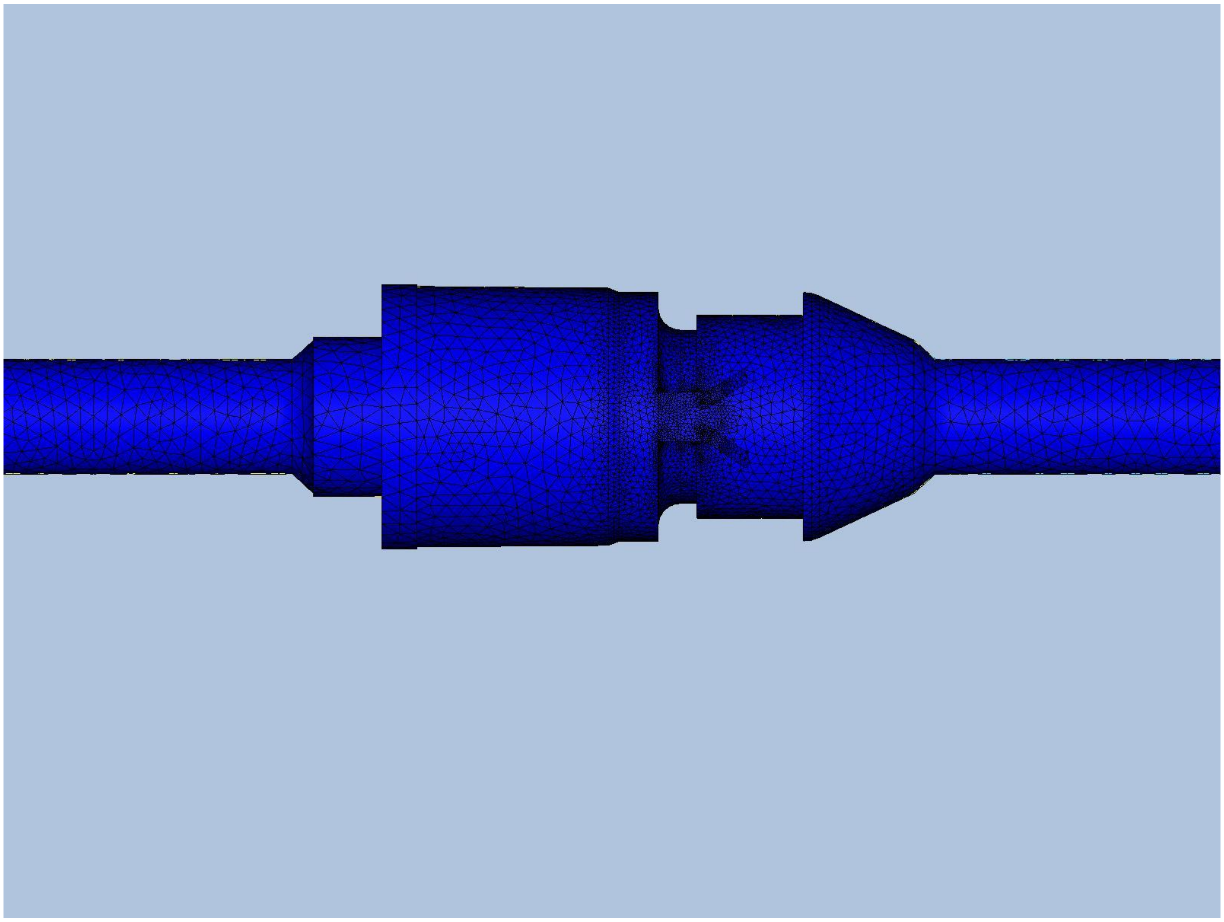
Automatic Meshing Settings

Surface refinement	0
Gap refinement	0
Resolution factor	1.0
Edge growth rate	1.1
Minimum points on edge	2
Points on longest edge	10
Surface limiting aspect ratio	20

Mesh Enhancement Settings

Mesh enhancement	0
Enhancement blending	0
Number of layers	3
Layer factor	0.45
Layer gradation	1.05

Meshed Model



Number of Nodes	95004
Number of Elements	516073

Physics

Flow	On
Compressibility	Compressible
Heat Transfer	Off
Auto Forced Convection	Off

Gravity Components	0.0, 0.0, 0.0
Radiation	Off
Scalar	No scalar
Turbulence	On

Solver Settings

Solution mode	Transient
Solver computer	MyComputer
Intelligent solution control	Off
Advection scheme	ADV 5
Turbulence model	k-epsilon

Convergence

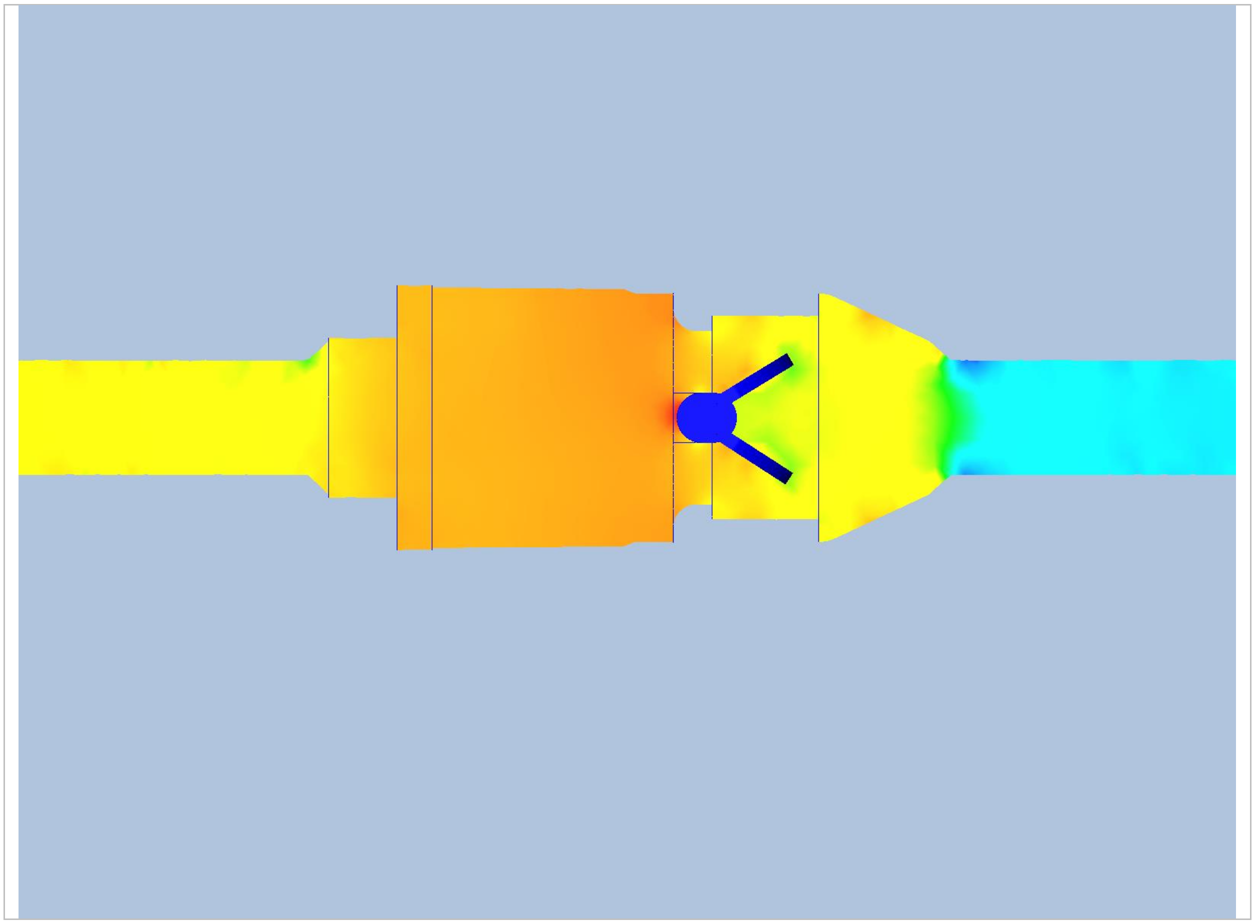
Iterations run	1000
Solve time	5354 seconds
Solver version	19.2.20190802

Energy Balance

Mass Balance

	IN	OUT
Mass flow	0.035926	-0.0351084
Volume flow	384.975 i^3/s	-376.213 i^3/s

Results



Inlets and Outlets

inlet 1	inlet bulk pressure	4.41764 psi
	inlet bulk	0.0 F
	inlet mach number	3.73302e-05
	mass flow in	0.035926
	minimum x,y,z of	0.0
	node near minimum	711.0
	reynolds number	187652.0
	surface id	88.0
	total mass flow in	0.035926
	total vol. flow in	384.975 i^3/s
	volume flow in	384.975 i^3/s
outlet 1	mass flow out	-0.0351084
	minimum x,y,z of	0.0
	node near minimum	1285.0
	outlet bulk pressure	-0.0 psi
	outlet bulk	-0.0 F
	outlet mach number	3.64883e-05

	reynolds number	183382.0
	surface id	87.0
	total mass flow out	-0.0351084
	total vol. flow out	-376.213 i^3/s
	volume flow out	-376.213 i^3/s

Field Variable Results

VARIABLE	MAX	MIN
cond	8.01796e-06 B/in-s-R	1.93767e-06 B/in-s-R
dens	9.81631e-05 lbf-s^2/in^4	9.33204e-05 lbf-s^2/in^4
econd	0.18242 B/in-s-R	0.0 B/in-s-R
emiss	1.0	0.0
evisc	0.00047317 lbf-s/in^2	0.0 lbf-s/in^2
gent	8150.98 1/s	0.0316228 1/s
press	7.21554 psi	-2.42482 psi
ptotl	9.17512 psi	-2.42482 psi
scal1	0.0	0.0
seebeck	0.0 V/K	0.0 V/K
shgc	0.0	0.0
spech	385.66 B-in/lbf-s^2R	202.882 B-in/lbf-s^2R
temp	0.0 F	0.0 F
transmiss	0.0	0.0
turbd	956760.0 in^2/s^3	1.90377e-14 in^2/s^3
turbk	5843.23 in^2/s^2	0.0 in^2/s^2
ufactor	0.0	0.0
visc	1.45342e-07 lbf-s/in^2	0.0 lbf-s/in^2
vx vel	161.393 in/s	-134.654 in/s
vy vel	115.084 in/s	-131.551 in/s
vz vel	341.177 in/s	-150.799 in/s
wrough	0.0 in	0.0 in

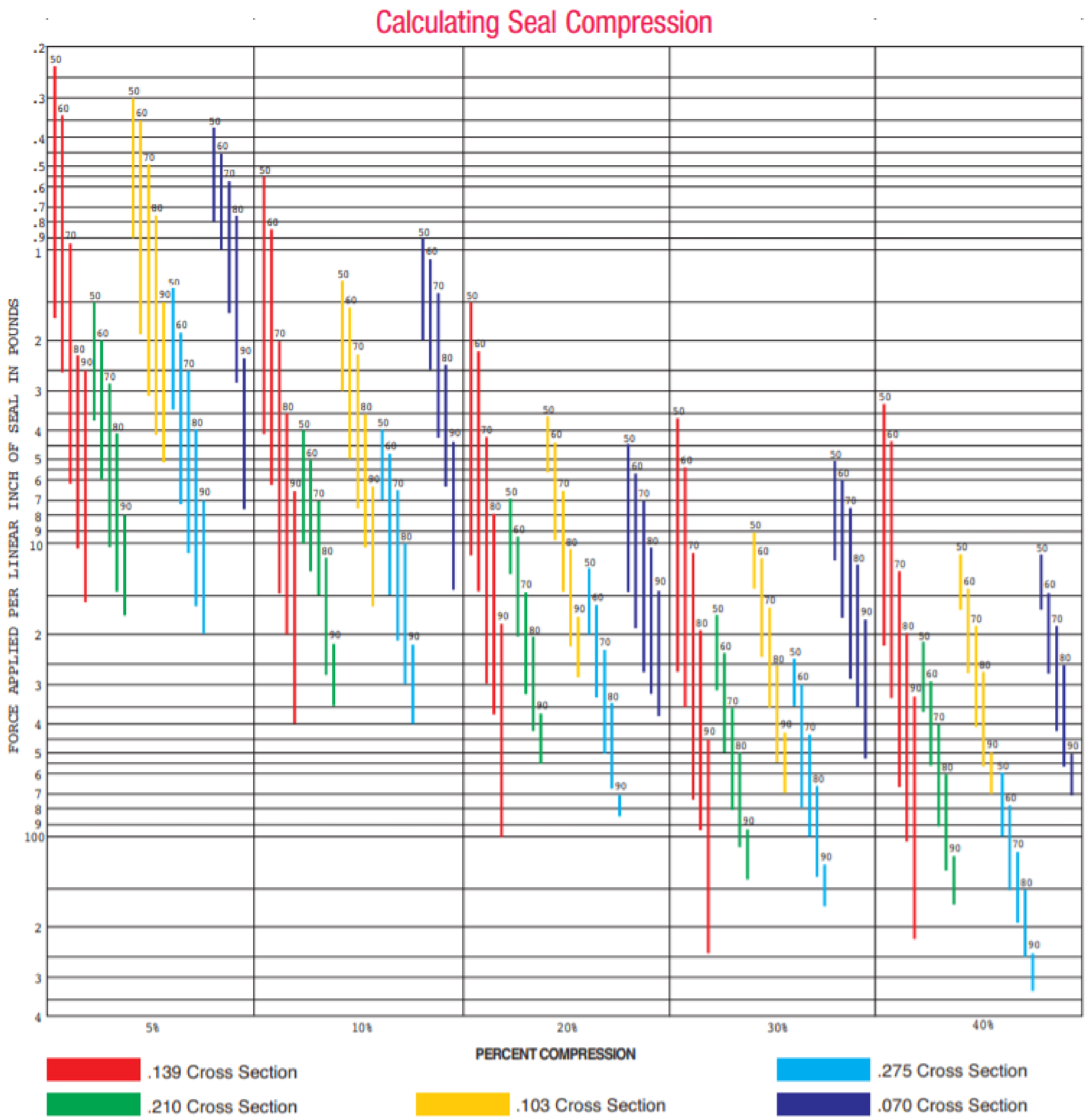
Component Thermal Summary

PART	MINIMUM TEMPERATURE	MAXIMUM TEMPERATURE	VOLUME AVERAGED TEMPERATURE
dd_poppet_CFD	-5.68434E-14	-5.68434E-14	0
VALVE_BODY_CFD	-5.68434E-14	-5.68434E-14	0
dd_poppet_CFD	-5.68434E-14	-5.68434E-14	0
check2_hinge_CFD	-5.68434E-14	-5.68434E-14	0
check2_frame_CFD	-5.68434E-14	-5.68434E-14	0
CFDCreatedVolume	-5.68434E-14	-5.68434E-14	0

Fluid Forces on Walls

pressx	-0.74662 pounds
pressy	0.37856 pounds
pressz	6.4285 pounds
shearx	0.002123 pounds
sheary	-0.001909 pounds
shearz	1.9338 pounds

Appendix P – Seal Design Charts: Compression Calculations & Rules of Thumb (Adapted from Apple Rubber Products)



Appendix P (Continued) – Seal Design Charts: Compression Calculations & Rules of Thumb (Adapted from Apple Rubber Products)

Formulas

$$\text{Maximum \% Compression} = \left[1 - \left[\frac{\left(\dfrac{\text{Min Bore Diameter - Max Groove Diameter}}{2}\right)}{\text{Max O-Ring CS}}\right]\right] \cdot 100$$

$$\text{Minimum \% Compression} = \left[1 - \left[\frac{\left(\dfrac{\text{Max Bore Diameter - Min Groove Diameter}}{2}\right)}{\text{Min O-Ring CS}}\right]\right] \cdot 100$$

$$\text{Maximum O-Ring CS} = \left[\frac{\left(\dfrac{\text{Min Bore Diameter - Max Groove Diameter}}{2}\right)}{1 - \left(\dfrac{\text{Maximum \% Compression}}{100}\right)}\right] - \text{O-Ring CS Tolerance}$$

$$\text{Minimum O-Ring CS} = \left[\frac{\left(\dfrac{\text{Max Bore Diameter - Min Groove Diameter}}{2}\right)}{1 - \left(\dfrac{\text{Minimum \% Compression}}{100}\right)}\right] + \text{O-Ring CS Tolerance}$$

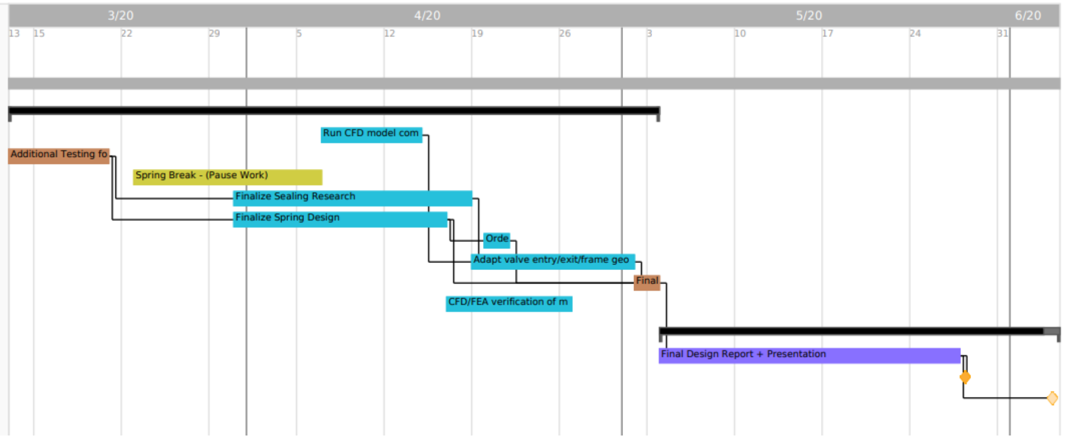
Appendix P (Continued) – Seal Design Charts: Compression Calculations & Rules of Thumb (Adapted from Apple Rubber Products)

Rules of Thumb Summary

- *A stretch greater than 5% on the O-ring I.D. is not recommended because it can lead to a loss of seal compression. (Section 3, p. 7)*
- *A Groove Depth is the machined depth into one surface, whereas a Gland Depth consists of the Groove Depth plus clearance. The Gland Depth is used to calculate seal compression. (Section 3, p. 7)*
- *To create Seal Squeeze, the Gland Depth must be less than the cross section. (Section 3, p. 8)*
- *Static applications are more tolerant of material and design limitations than dynamic applications. (Section 3, p. 9)*
- *The maximum volume of the O-ring should* ***never*** *surpass the minimum volume of the gland. (Section 3, p. 10)*
- *For a static crush seal application, it is recommended that the O-ring volume does not exceed 95% of the gland void. (Section 4, p. 12)*
- *For reciprocating seals – passing O-rings over ports is not recommended. Nibbling and premature wear and seal failure will result. (Section 4, p. 13)*
- *The closer the application is to room temperature, the longer an O-ring can be expected to effectively seal. (Section 4, p. 14)*
- *Avoid using graphite-loaded compounds with stainless steel, as they tend to pit the stainless steel surface over time. (Section 4, p. 15)*
- *Before installation, make sure to lightly coat the O-ring with a lubricant that is compatible with the O-ring material, as well as with system chemicals. (Section 4, p. 16)*
- *When using only one back-up ring, be sure to install it on the low pressure side of the O-ring. (Section 5, p. 55)*
- *Static seal cross sections are generally compressed from 10% to 40%, whereas dynamic seals are from 10% to only 30%. (Section 5, p. 57)*
- *When it is said that an elastomer is good for an application it is meant that some compounds which include that material are acceptable,* ***not all****. For instance, some compounds of* ***EP*** *are good for brake fluid applications, but most are not acceptable. (Section 6, p. 63)*
- *Material cost does not correlate with performance, it depends on the application. (Section 6, p. 70)*
- *You must test all seals in their actual environment because every application is unique. (Section 6, p. 71)*
- *Do* ***not*** *use a lubricant composed of the same material as the O-ring because "like" will dissolve "like." For example, a silicone lubricant should* ***not*** *be used with a silicone O-ring. (Section 7, p. 75)*
- *Resistance of elastomers to chemical attack is greatly reduced at elevated temperatures. (Section 7, p. 76)*

Appendix Q – Gantt Chart – Updated as of 5/31/2020

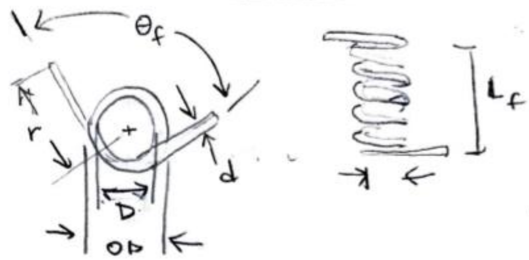
53 - Check Valve Design	start	end	0h	99%
Final Design Approach - Double Disk	**03/13/20**	**05/03/20**	**0h**	**100%**
Run CFD model comparing disk seat angels	04/07	04/14	0	100%
Additional Testing for Double Disk Modifications	03/13	03/20	0	100%
Spring Break - (Pause Work)	03/23	04/06	0	100%
Finalize Sealing Research	03/31	04/18	0	100%
Finalize Spring Design	03/31	04/16	0	100%
Order seals and springs	04/20	04/21	0	100%
Adapt valve entry/exit/frame geometry for full ...	04/19	05/01	0	100%
Final Designs submitted to Brian	05/02	05/03	0	100%
CFD/FEA verification of manufactured valve co...	04/17	04/26	0	100%
Final Design Review	**05/04/20**	**06/04/20**	**0h**	**96%**
Final Design Report + Presentation	05/04	05/27	0	100%
Submit Project Expo Website	05/28	05/28	0	100%
Submit FDR	06/04	06/04	0	0%



Appendix R – Hand Calculations: Torsion Spring Calculator

TORSION SPRING CALCULATOR - V3 5/2/2020 1

VERIFY FINAL SPRING SELECTION



USING SYSTEM VARIABLES

P_{cr} [CRACKING PRESSURE] $P_{cr} = 2$ psi

A_{op} [DISK AREA, OPEN] $A_{op} = 0.3125\ in^2$] FROM SOLIDWORKS

A_d [DISK AREA, CLOSED] $A_d = 1.250\ in^2$

r [LEG LENGTH] $r = 0.5$ in

θ_f [FREE LEG ANGLE] $\theta_f = 210°$

D [AVERAGE DIAMETER] $D = 0.550$ in

OD [OUTSIDE DIAMETER] $OD = 0.620$ in

d [WIRE DIAMETER] $d = 0.070$ in

N_b [NUMBER OF COILS (UNLOADED)] $N_b = 13.083$

d_{pin} [PIN DIAMETER] $d_{pin} = 0.375$ in

* 303 STAINLESS STEEL TO PREVENT CORROSION

$S_{UT} = 1000$ Ksi

$E = 28{,}000$ Ksi

$S_y = 610$ Ksi

USING SHIGLEY'S TABLE 10-4

$A = 90\ \text{kpsi} \cdot \text{in}^m$

$m = 0.478$

CRACKING FORCE APPLIED TO EACH DISC

$F_{crd} = A_d P_{cr}$

$= (1.25 in^2)(2 psi)$

$F_{crd} = 2.5$ lbf

[$F_{cr_{TOT}} = 5$ lb ← FOR BOTH DISCS APPLIED BY THE SAME SPRING]

SPRING RATE REQUIRED FOR CRACKING AT $P_{cr} = 2psi$

$$K = \frac{d^4 E}{10.8 D N_a} \; ; \quad \text{WHERE } N_a = N_b + \frac{l_1 + l_2}{3\pi D}$$

$$N_a = (13.093) + \frac{(0.5in) + (0.5in)}{3\pi(0.55in)}$$

$$N_a = 13.276 \text{ COILS}$$

$$K = \frac{(0.07in)^4 (28{,}000ksi)}{10.8(0.55in)(13.276)}$$

$$\left[\begin{array}{l} K = 8.525\,lbf/turn \cdot \frac{turn}{360°} \\ K = 0.023\,lbf/deg \end{array}\right]$$

PRESSURE REQUIRED TO HOLD VALVE AT FULL OPEN

$$\theta_{op} = \theta_f + 90°$$
$$\theta_{op} = -210° + 90° = -120°$$

$$K = \frac{F_{op} r}{\theta_{op}} \rightarrow F_{op} = \frac{K\theta_{op}}{r}$$

$$P_{op} = \frac{F_{op}}{A_{op}}$$

$$P_{op} = \frac{K\,\theta_{op}}{r A_{op}}$$

$$P_{op} = \frac{(0.023\frac{lbf}{deg})(120°)}{(0.5in)(0.3125in^2)}$$

$$\left[P_{op} = 18.07\,lbf\right]$$ ← DOES NOT ACCOUNT FOR PRESSURE LOSSES OR TURBULENT FLOW

BENDING STRESS IN THE SPRING

SPRING INDEX:

$$C = \frac{D}{d} = \frac{0.55in}{0.070in}$$

$$C = 7.857$$ ← ACCEPTABLE FROM A MFG. STANDPOINT

$$\sigma_B = K_i \frac{32 F \cdot r}{\pi d^3} \quad \text{WHERE } K_i = \frac{4C^2 - C - 1}{4C(C-1)}$$

INTERNAL STRESS CORRECTION FACTOR

$$K_i = \frac{4(7.857)^2 - (7.857) - 1}{4(7.857)(7.857-1)}$$

$$K_i = 1.1047$$

BENDING STRESS WHEN VALVE IS CLOSED

$$\sigma_{CL} = \frac{(1.1047)\,32\,(5\,lbf)(0.5in)}{\pi\,(0.70)^3}$$

$$\sigma_{CL} = 82.017\ ksi$$

BENDING STRESS WHEN VALVE IS FULLY OPEN

$$\sigma_{OP} = \frac{(1.1047)\,32\,(11.29\,lbf)(0.5in)}{\pi\,(0.70)^3}$$

$$\sigma_{OP} = 185.29\ ksi$$

SHOULD BE HIGHER ✓

ALLOWABLE BENDING STRESS FOR 303 STAINLESS SPRING

- FOR STATIC LOADING

$S_y = 0.61\,S_{UT}$ [SHIGLEY'S]

WHERE $S_{UT} = 1{,}000\ ksi$ [MATERIAL PROPERTIES]

$S_y = 0.61(1000ksi)$

$S_y = 610\ ksi$

STATIC SAFETY FACTOR

$$n_{STATIC} = \frac{S_y}{\sigma_{max,\,STATIC}}$$

$$n_{STAT} = \frac{610ksi}{185.29ksi}$$

$$\boxed{n_{STATIC} = 3.29}$$ ACCEPTABLE

- FOR FATIGUE LOADING

SAFETY FACTOR

$$n_{FATIGUE} = \frac{S_a}{\sigma_a}$$

ASSUMING 10^5 CYCLES (LIKELY THE VALVE WOULD EXPERIENCE FEWER CYCLES)

$$S_a = \frac{R^2 S_{UT}^2}{2S_e}\left[-1 + \sqrt{1 + \left(\frac{2S_e}{R\,S_{UT}}\right)^2}\,\right]$$

→ R IS STRENGTH AMPLITUDE FROM SHIGLEY'S GERBER TABLE 6-7

$$R = \frac{M_a}{M_m}$$

AVERAGE MOMENT

$$M_a = \frac{(M_{max} - M_{min})}{2} = \frac{F_{op}\cdot r - F_{CL}\cdot r}{2}$$

$$M_a = 1.574\ lbf\text{-}in$$

MAX MOMENT

$$M_m = \frac{M_{max} + M_{min}}{2} = \frac{F_{op} \cdot r + F_{cr} \cdot r}{2}$$

$$[\; M_m = 4.074 \text{ in-lb}_f \;]$$

$$R = \frac{M_a}{M_m} = \frac{1.5739}{4.074}$$

$$[\; R = 0.386 \;]$$

ALSO NEED ENDURANCE STRENGTH

$$S_e = \frac{S_r/2}{1 - \left(\frac{S_r/2}{S_{UT}}\right)^2}$$, WHERE S_r COMES FROM SHIGLEYS TABLE 10-10

$$S_r = 0.53 S_{UT}$$

$$S_r = 0.53 (1000 ksi)$$

$$[\; S_r = 530 ksi \;]$$

$$S_e = \frac{(530 ksi)/2}{1 - \left(\frac{530/2}{1000 ksi}\right)^2}$$

$$[\; S_e = 285.015 ksi \;]$$

RETURNING TO S_a EQN:

$$S_a = \frac{[0.386)^2 (1000 ksi)^2}{2(285.015 ksi)} \left[-1 + \sqrt{1 + \left(\frac{2(285.015 ksi)}{(0.386)(1000 ks)}\right)^2}\right]$$

$$[\; S_a = 204.97 ksi \;]$$

FATIGUE STRESS

$$\sigma_A = \frac{k_i \, 32 M_a}{\pi d^3}$$ AVERAGE FATIGUE STRESS

$$\sigma_A = \frac{(1.1047)(32)(1.579 \text{ in-lb}_f)}{\pi (.070 in)^3}$$

$$[\; \sigma_A = 51.8 ksi \;]$$

$$\sigma_m = \frac{M_m}{M_a} \sigma_a = \frac{(4.074)}{(1.574)} (51.8 ksi)$$

$$[\; \sigma_m = 185.7 ksi \;]$$

MAX FATIGUE STRESS

FATIGUE FACTOR OF SAFETY

$$\eta_{FATIGUE} = \frac{S_{max}}{\sigma_{max}}$$

σ_{max} FOR FATIGUE LOADING

$$\sigma_{max} = S_a = 204.87 \text{ ksi}$$

$$\eta_{FATIGUE} = \frac{S_a}{S}$$

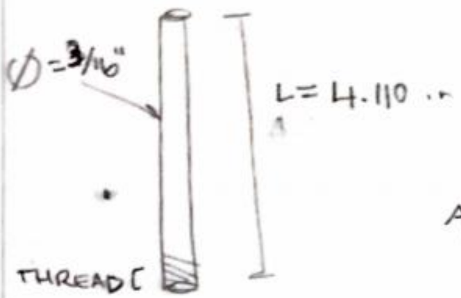
$$\eta_F = \frac{204.87 \text{ ksi}}{185.29 \text{ ksi}}$$

$$\boxed{\eta_{FATIGUE} = 1.106}$$

ACCEPTABLE, VERY HIGH # CYCLES ANALYZED

Appendix S – Hand Calculations: Poppet pin diameter

LOADING OF PINS — 5/4/2020 — 1

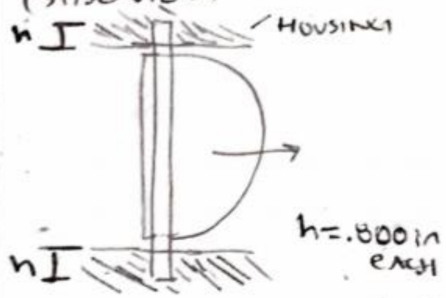


ANALYSIS IS OF SMALLER 3/16" DIAMETER PINS

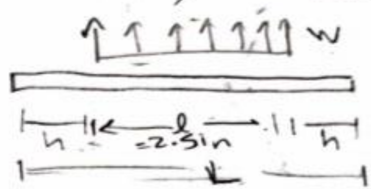
→ EXPERIENCE MORE STRESS

ASSUME:
- PIN IS A CYLINDER
- STP
- DRAG FORCE IS DRIVING LOAD
- AT HIGHEST FLOW RATE (150 gal/min)

LOADING (SIDEVIEW)



ASSUME CONSTANT DISTRIBUTION



$W = F_D$

ASSUME AT MAX FLOWRATE

$\dot{m} = 150\ \text{gal/min}$ (ASSUME STP)

$V = \dot{m}\rho A = \dot{m}\rho_{H_2O}\pi\left(\frac{l}{2}\right)^2$

$V = \left(150\,\frac{\text{gal}}{\text{min}}\cdot\frac{\text{min}}{60\,\text{s}}\cdot\frac{.00378\,\text{m}^3}{\text{gal}}\right)\left(1000\,\frac{\text{kg}}{\text{m}^3}\right)\left(\pi\left(2.5\,\text{in}\cdot\frac{.0254\,\text{m}}{\text{in}}\right)^2\right)$

$V = .1197\ \text{m/s}$

$V = .3927\ \text{ft/s}$

NEED DRAG COEFF, FIND REYNOLDS

$Re = \frac{V l}{\nu}$ ASSUMING STP $\nu = 1.1337\times10^{-5}\ \text{ft}^2/\text{s}$

$Re = \frac{(.3927\,\frac{\text{ft}}{\text{s}})(2.5\,\text{in}\cdot\frac{\text{ft}}{12\,\text{in}})}{1.1337\times10^{-5}\ \text{ft}^2/\text{s}}$

$Re = 6.1\times10^3$

LOOKING AT DRAG TABLE FOR CYLINDER DRAG

$C_D \approx 1.1$ ← (NOT A VERY INFLUENTIAL FACTOR)

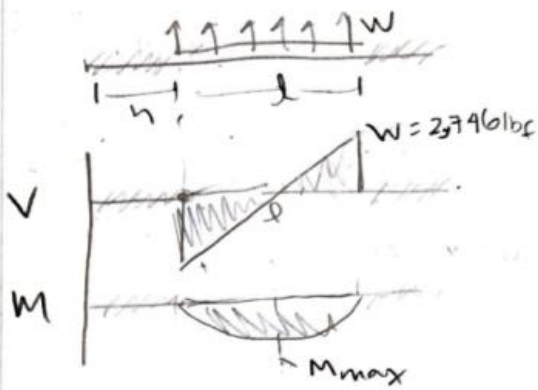
$$w = \frac{F_D}{l} = \frac{C_D \frac{1}{2} \rho V^2}{l}$$

$$W = \frac{(1.1)\ \frac{1}{2}\ (1000 \frac{kg}{m^3})(.1197 \frac{m}{s})^2}{(2.5\,in \cdot \frac{.0254\,m}{in})}$$

$$W = 124.1\ N/m \cdot \left(\frac{8.85\,lb/in}{1\,N/m}\right)$$

$$W = 1098.4\ lb/in$$

BEAM DIAGRAM



POINT OF HIGHEST LOADING AT

$$x = h + l$$

M_{max} AT

$$x = h + \frac{l}{2}$$

USING BEAM TABLES

$$M(x) = \frac{wx}{2}(l - x)$$

$$M_{max}(h + \tfrac{l}{2}) = \frac{w}{2}(h + \tfrac{l}{2})(l - (h + \tfrac{l}{2}))$$

$$M_{max} = \frac{w}{2}(h + \tfrac{l}{2})(\tfrac{l}{2} - h)$$

$$M_{max} = \frac{(1098.4\,lb/in)}{2}(.8\,in + \tfrac{2.5}{2}\,in)(\tfrac{2.5}{2}\,in - .8\,in)$$

$$\longrightarrow M_{max} = 506.6\ in\text{-}lb$$

CALC MAX STRESS - BENDING

$$\sigma_{max} = \frac{Mc}{I}$$ WHERE $c = \frac{d}{2} = .09375\,in$

AND $I = \frac{\pi D^4}{64}$

$$I = \frac{\pi}{64}(3/16\,in)^4 = 6.067 \times 10^{-5}\ in^4$$

$$\sigma_{max} = \frac{M_{max} C}{I}$$

$$\sigma_{max} = \frac{(506 \text{ in-lb})(.09375 \text{ in})}{(6.067 \times 10^{-5} \text{ in}^4)}$$

$$\sigma_{max} = 78.28 \text{ ksi}$$

$$n = \frac{S_{ut}}{\sigma_{max}}$$ ← FOR 303 STAINLESS $S_{ut} = 100$ ksi

$$n = \frac{100 \text{ ksi}}{72.18 \text{ ksi}}$$

$\boxed{n = 1.37}$ ← FAIRLY LOW, ESPECIALLY CONSIDERING ADDITIONAL TORSIONAL LOADS THAT MIGHT BE APPLIED

ROUGH ESTIMATE