

Design of a Miniature Flow Control Valve

MAE 156B - Team 26

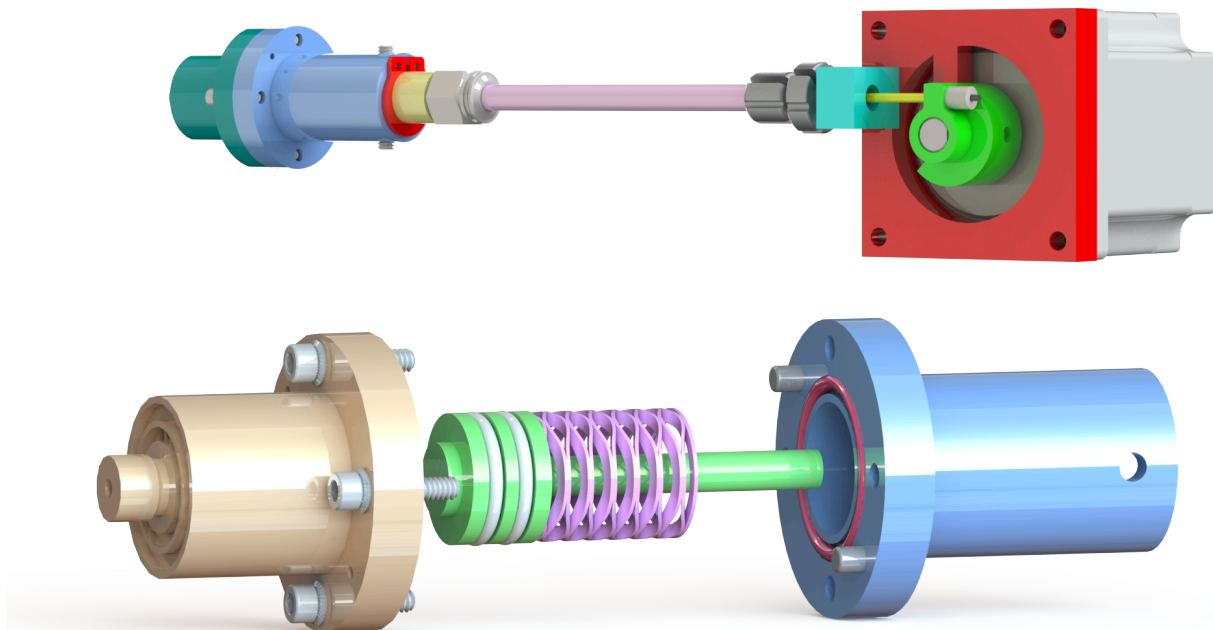
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Abstract

This project, in collaboration with and sponsored by Cohu, focuses on development and subsequent integration of a flow valve into a self developed test stand in order to test semiconductors, thermal subsystems, and semiconductor automated test equipment to be used by a wide range of industries, including but not limited to automotives and aerospace. In order to expand their semiconductor testing capabilities, Cohu requires an alternative design to their current refrigerant control valve. As a project team, our main goal is to provide Cohu with a flow control valve that accommodates for a broader testing temperature range. The solenoid actuator driving the existing flow valve experiences demagnetization with exposure to higher temperatures. Further, secondary electrical components sustain considerable damage when exposed to high temperatures in excess of 150°C. Recognizing the design limitations of the current flow valve, the objective of this study is to redesign the current iteration of the flow valve by separating the electronic actuator from the flow valve and developing an alternative remote connection that would not only withstand the greater range of temperatures, but also prevent the demagnetisation of the solenoid and allow actuation from a further distance. In order to enable this remote actuation system, our team has utilized a Bowden cable design, with a Teflon coated wire rope and an outer Teflon sheath. A number of secondary challenges must also be addressed. These include finding materials that can withstand the corrosive nature of the refrigerant fluid flowing through the valve, increasing usage of commercial off-the-shelf alternatives for valve body components, and reducing the overall manufacturing capital. These are the challenges that we have identified and embarked to tackle as a team of engineers, along with Cohu.

Executive Summary

Semiconductors have a wide variety of applications in contemporary industrial-sector and commercial products, including in many extreme environments. Some examples of high performance semiconductors are found in automobiles where they are exposed to high temperatures adjacent to combustion engines.

Cohu Inc., the sponsor of this design project, is an industry leader in semiconductor testing and validation. The current semiconductor chip testing systems utilized by Cohu depend on liquid cooling for the thermal regulation of computer chips during testing. However, the existing automatic flow control valves that regulate the flow rate of the refrigerants are likely to fail at higher temperatures. These flow valves utilize solenoid actuators that experience demagnetization at higher temperatures, inhibiting their functionality. Similarly, at high temperatures the electronics that drive these valves can experience damage. The testing environment needed to test high performance semiconductors requires elevated temperatures that are unsuitable for the existing AFC valves.

The objective of this design study is to develop a flow valve that isolates the sensitive electronics from the extreme temperatures in the testing environment. The proposed solution decouples the valve body from the actuator and connects both subsystems with a flexible mechanical linkage. The actuator can be operated from outside the test chamber in suitable temperatures while the redesigned flow valve is built to withstand the increased temperature range in the test environment. Furthermore, this advanced actuator is designed to automatically respond in real time to control feedback during the semiconductor testing process. As more cooling is necessary, the flow valve will open to increase the refrigerant flow rate while closing as less cooling is needed.

In this proposed design solution, four main subsystems can be identified:

1. Valve Body
2. Mechanical Linkage
3. Actuator
4. Feedback Control System

The proposed design solution integrates a stainless steel valve body capable of interfacing with the existing AFC valve port. An armature (also referenced as “plunger”) behaves as a linear actuator within the housing, regulating the flow of fluid through the valve as it opens and closes against the inlet. The armature is driven by a Bowden cable acting as a mechanical

linkage to the actuator. This flexible cable can be extended up to one meter from the test environment to protect sensitive electronics from the extreme temperatures experienced within the testing chamber. A feedback control system spins a stepper motor which utilizes a custom attached wheel to develop linear motion in the Bowden cable.

Table of Contents

Abstract.....	1
Executive Summary.....	2
Table of Contents.....	4
List of Figures.....	5
List of Tables.....	7
Chapter 1.0: Project Description.....	8
Chapter 1.1: Background.....	9
Chapter 1.2: Review of Existing Solutions.....	10
Chapter 1.3: Statement of Requirements.....	12
Chapter 1.4: Deliverables.....	15
Chapter 2.0: Description of Final Design Solution.....	16
Chapter 2.1: Design Solution Overview.....	16
Chapter 3.0: Design of Key Components.....	18
Chapter 3.1: Valve Body.....	18
Chapter 3.2.1 Linkage Mechanism (Bowden Cable).....	29
Chapter 3.2.2 Linkage Mechanism (End Fittings).....	35
Chapter 3.3: Actuator.....	47
Chapter 4.0: Prototype Performance.....	50
Chapter 4.1: Performance Testing Overview & Results.....	50
Chapter 5.0: Design Recommendations, Conclusions, and Supplementary Statements.....	57
Chapter 5.1: Future Design Recommendations.....	57
Chapter 5.2: Conclusions.....	57
Chapter 5.3: Applicable Standards.....	58
Chapter 5.4: Impact on Society.....	59
Chapter 5.5: Professional Responsibility.....	61
Acknowledgements.....	62
References.....	63
Appendix.....	64
Appendix A: Project Management.....	64
Appendix B: Individual Component Reports.....	66
Appendix C: Designs Considered.....	102
Appendix D: Other Calculations.....	103
Appendix E: Bill of Materials for Designed Flow Valve.....	104
Appendix F: Budget.....	108

List of Figures

Figure 1.1.0: Chip Testing Rig. <i>Source: Cohu Inc.</i>	9
Figure 1.2.0: Annotated Flow Valve CAD Model.	10
Figure 1.3.0: Flow Valve Front and Top View	10
Figure 1.4.0: Exploded view of existing AFC flow valve spring housing	11
Figure 1.5.0: Valve Connection Interface on Chip Testing Rig	12
Figure 2.1.1: Final Assembly of Motor and Valve Housing	17
Figure 3.1.1: Annotated Valve Housing Assembly in Exploded View	18
Figure 3.1.2: Valve Housing Section View	19
Figure 3.1.3: Valve Housing Dimensional View	19
Figure 3.1.4: Face and Radial Seal Diagrams	22
Figure 3.1.5: Differences Between Linear and Nonlinear Springs	24
Figure 3.1.6: Smalley CS062-M8-S17 Wave Spring	25
Figure 3.1.7: Valve Spring Force Diagram	25
Figure 3.1.8: Valve Housing Armature Prototype	27
Figure 3.1.9: Valve Housing Rear Prototype	27
Figure 3.1.10: Valve Housing Front Prototype	27
Figure 3.1.11: Valve Housing Assembly	28
Figure 3.2.1: Animation of the Bowden Cable in Use in the Current Valve Housing	30
Figure 3.2.2: Cross-Sectional View of 7x7 Wire Rope	30
Figure 3.2.3: Wire Rope Fatigue Life and Sheave Pressure Relationship Graph	31
Figure 3.2.4: Cable Cross-Section FBD with infinitesimal angle $d\theta$	32
Figure 3.2.5: Illustrations of a) cable backlash, and b) conduit deflection	33
Figure 3.2.6: Threaded end fitting	35
Figure 3.2.7: Terminology of a Male Thread	35
Figure 3.2.8: Depiction of Bending Stresses in Threaded Components	36
Figure 3.2.9: Strength Requirements of Thread by Grade	38
Figure 3.2.10: Part Specs for Wire Rope Crimp Stopper	39
Figure 3.2.11: Solidworks Drawing of Step-Down Crimp Ferrule	40
Figure 3.2.12: Solidworks Drawing of Sheath Stopper	40
Figure 3.2.13: Push-to-Connect Spec Sheet	41
Figure 3.2.14: Close-up of (Current) Sheath/Valve Interface and Terminal Fitting	42
Figure 3.2.15: Close-up of (Former) Sheath/Valve Interface and Terminal Fitting	42
Figure 3.2.16: FBD Model of Set Screw in Valve Housing Back	42
Figure 3.2.17: Graph of Set Screw Loadings	43
Figure 3.2.18: Surface Factor Statistical Estimation	43

Figure 3.2.19: Size Factor Statistical Estimation	44
Figure 3.2.20: Load Factor Statistical Estimation	44
Figure 3.2.21: Reliability Factor Statistical Estimation	45
Figure 3.2.21: Stress Concentration Factors for Filleted/Grooved Shafts	46
Figure 3.3.1: Isometric view of final CAD actuator design	47
Figure 3.3.2: Front view of final CAD actuator design	48
Figure 3.3.3: Free body diagram of spool	48
Figure 3.3.3: Nema 23 Stepper Motor	49

List of Tables

Table 1: Statement of Requirements	11
Table 2: Deliverables	14
Table 3: Valve Housing Design Requirements	16
Table 4: Housing and Spring Material Selection Chart	18
Table 5: O-Ring Material Compatibility Chart	20
Table 6: Spring Properties	23
Table 7: Valve Housing Manufacturing Tooling Chart	24

Chapter 1.0: Project Description

The scope of this project is defined by the industry sponsor, Cohu Inc. Cohu is seeking to develop a new iteration of their automatic flow control (AFC) valve for refrigerant delivery in their new semiconductor testing platforms. The current valve functions well, but is not suitable for the high temperature conditions of the new testing environment. The high temperatures present in the testing environment would result in demagnetization of the solenoid actuator and damage to the electronics in the current valve design. Therefore, Cohu has asked the team to design and manufacture a new valve that separates the flow valve from the electronic actuator and connects the two with a mechanical linkage over a nonlinear distance of one meter.

Chapter 1.1: Background

Cohu Inc. is an industry leader in the semiconductor testing industry. Semiconductors have a wide variety of applications in contemporary industrial-sector and commercial products, including in many extreme environments. Some examples of high performance semiconductors are found in automobiles where they are exposed to high temperatures adjacent to combustion engines. The testing environments for these high performance semiconductors require high temperatures that are unsuitable for the current flow valve in use by Cohu.

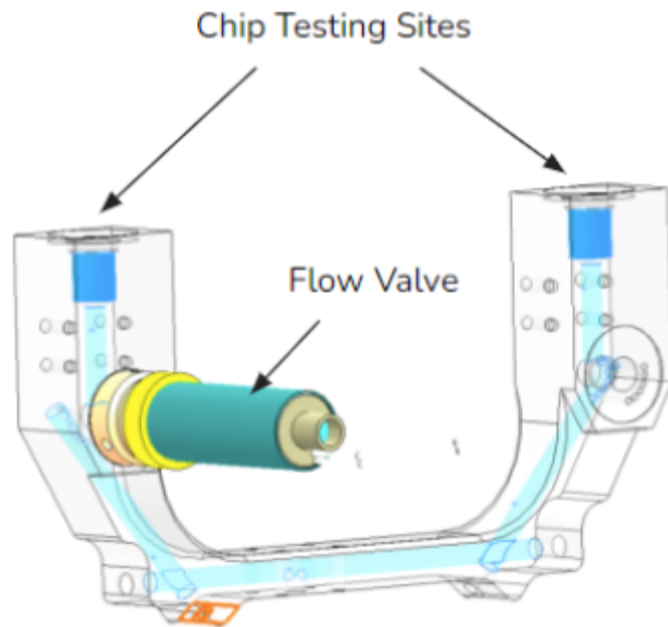


Figure 1.1.0: Chip Testing Rig. Source: Cohu Inc.

Chapter 1.2: Review of Existing Solutions

In Figure 1.1.0, the chip testing sites are shown with respect to the thermal control unit that uses a refrigerant fluid to regulate temperature at the testing sites. The flow valve is the component that sets the rate of fluid flow to the test sites which regulates heat transfer in that area.

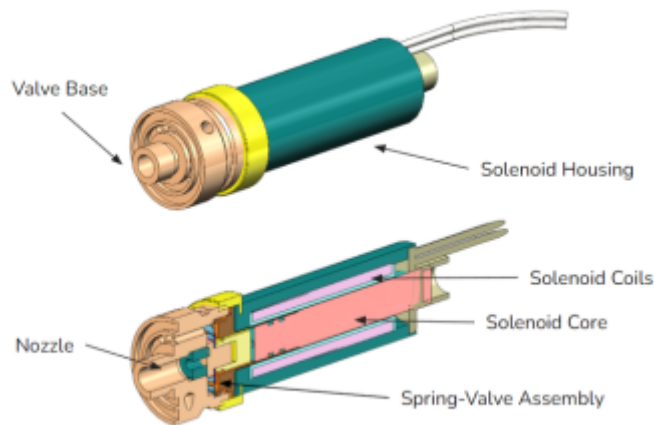


Figure 1.2.0: Annotated Flow Valve CAD Model



Figure 1.3.0: Flow Valve



The CAD model of the current flow valve is shown in Figure 1.20. The flow valve is shown in Figures 1.3.0a and 1.3.0b. A solenoid coil is attached directly to the valve mechanism which is present in the testing environment. A detailed breakdown of the valve housing is shown below in Figure 1.4.0.

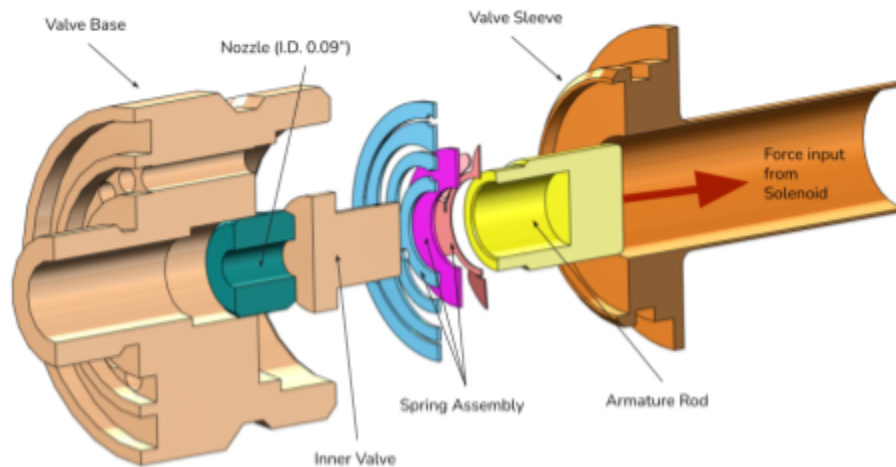


Figure 1.4.0: Exploded view of existing AFC flow valve spring housing

The spring assembly shown in Figure 1.4.0 controls the flow rate of the process fluid through the valve. As the PWM duty cycle of the solenoid coils is increased, the magnetic force generated by the coils increases. This magnetic force produces a backward force on the armature (shown in Figure 4), which pulls the inner valve that compresses the spring assembly. As the springs are compressed, the inner valve is retracted and flow is permitted through the flow valve. A pulse width modulated duty cycle in the coils produces a variable force on the armature which controls the flow rate.

Testing semiconductors in high temperature environments requires mitigating concerns related to demagnetization and electrical degradation in the solenoid and electronics in the valve. This poses a major challenge in the current design, preventing the flow valve from being used to test in high-temperature environments. By redesigning the flow valve that interfaces with the testing rig to separate electromechanical actuators from the valve mechanism, it is possible to isolate the vulnerable subsystems from the extreme temperatures experienced within the testing chamber.

The cost of the existing flow valve is approximately \$270. Although a secondary design objective, it is desirable to reduce the cost further using commercially sourced components.

Chapter 1.3: Statement of Requirements

The comprehensive list of design requirements listed in Table 1 are provided by the project sponsor.

Table 1a: Design Footprint

- The actuator is to be of a similar size to the existing AFC valve assembly.
- The valve is to be half the size of the existing AFC valve assembly.
- The valve must work with similar M8 threaded ports or similarly sized metal or hard plastic tubes.

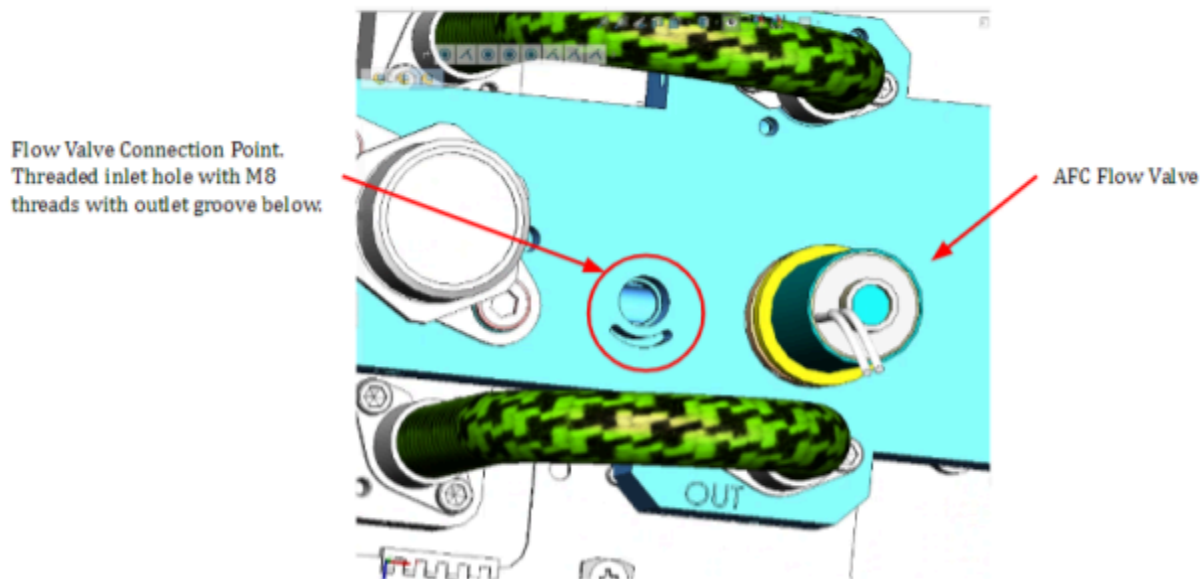


Figure 1.5.0: Valve Connection Interface on Chip Testing Rig

Table 1b: Mechanical Separation

- Separate the AFC assembly into a valve section and electronic actuator section connected by a mechanical linkage.
- The sections must be separated by 1 meter. The linkage must not be a straight path in order to simulate other components in the way for Cohu's real-world application.

Table 1c: Valve Performance

- The valve must be able to fully shut off fluid flow.
- The valve must be able to go from fully open to fully closed in less than 1 second.
- The valve must be able to go from fully closed to fully open in less than 1 second.
- The valve must be able to be controlled with PWM to a position between fully open and fully closed and held in place by the actuator.
- The valve defaults to closed when unpowered.
- At 50 psi, the flow across the valve is more than 1.5 L/min of water.
- The leak rate of process fluid is not to exceed 5 mL/min.
- The response time must be less than 10 ms from input to the beginning of valve actuation.
- The position is repeatable +/- 2.5% at a given position.

Table 1d: Temperature

- The valve process fluid will range from -80°C to 100°C.
- The environment the valve and linkage will be placed in will range from -40°C to 175°C.
- The environment the actuator will be placed in will range from 0° to 40°C.

Table 1e: Pressure

- The valve will experience 70 psi during normal operation.

Table 1f: Process Fluid

- The design can be tailored for liquid or gas mediums.
- The design primarily considers HFE 7100 & SF10 liquid coolants.
- The design is to be tested with water & air at ambient temperatures.

Table 1g: Voltage

- The design utilizes 24VDC for PWM control.

Table 1h: Actuator

- The actuator must be able to control the valve proportionally.

Table 1i: Test Profile

- Beginning with the valve closed, measure flow and open the valve in 10% increments until fully open. Measure flow returning to closed again moving in 10% increments.
- Record the valve PWM and fluid flow at each step.
- Repeat this test 5 times.
- Repeat the test for both air and water mediums.

Table 1j: Cost

- Secondary priority. Lower the price to produce the valve from \$270 by reducing usage of custom manufactured parts and implementing more COTS parts.

Chapter 1.4: Deliverables

The intended deliverables at the completion of the project are as follows:

Table 2a: Primary Deliverables:	Table 2b: Reach Deliverables:
• Fully operational prototype of new AFC valve design. • Supporting documentation with valve report and accompanying experimental data for use case. • Test bench used for air and water testing of the new AFC valve. • Data from the air and water flow tests as outlined in the Statement of Requirements.	• Fatigue cycle data to validate life cycle assessment of prototype mechanism

Chapter 2.0: Description of Final Design Solution

Chapter 2.1: Design Solution Overview

The final design solution integrates a modified aluminum valve housing compatible with the process fluid and testing chamber with a stepper motor actuator system using a Bowden cable mechanical linkage. The Bowden cable is a mechanical linkage that, in this case, is composed of a FEP coated steel inner wire that transfers loading from the driving end to the driven end. The path of the wire is determined by an outer PTFE tubing, which is statically fixed at both ends. This design solution separates the sensitive electronics in the actuator from the extreme temperatures experienced within the test chamber.

The stepper motor actuator directly interfaces with the steel cable in the Bowden Cable through a wheel mount. Incremented turns by the stepper motor pull on the Bowden cable which transmits the force to the valve housing armature. The armature is pulled against the spring in the valve, causing the stop to be pulled away from the valve nozzle and permit the process fluid to flow through. As the force of the actuator is increased, a greater force is exerted on the armature causing it to displace further from the nozzle and increasing the flow rate through the valve.

A flowmeter measures volumetric flow rate through the system to provide real-time feedback on refrigerant fluid moving through the valve. Based on a user or system specified cooling rate that is proportional to the flow rate of fluid, the measured flow rate can be adjusted to match the desired flow rate. In the final design solution, two separate Arduino controllers are used, one to receive digital output from the flowmeter, and another to manually direct the stepper motor's movement. The final hardware prototype does not incorporate the real-time feedback system. However, it does possess the hardware required to do so.

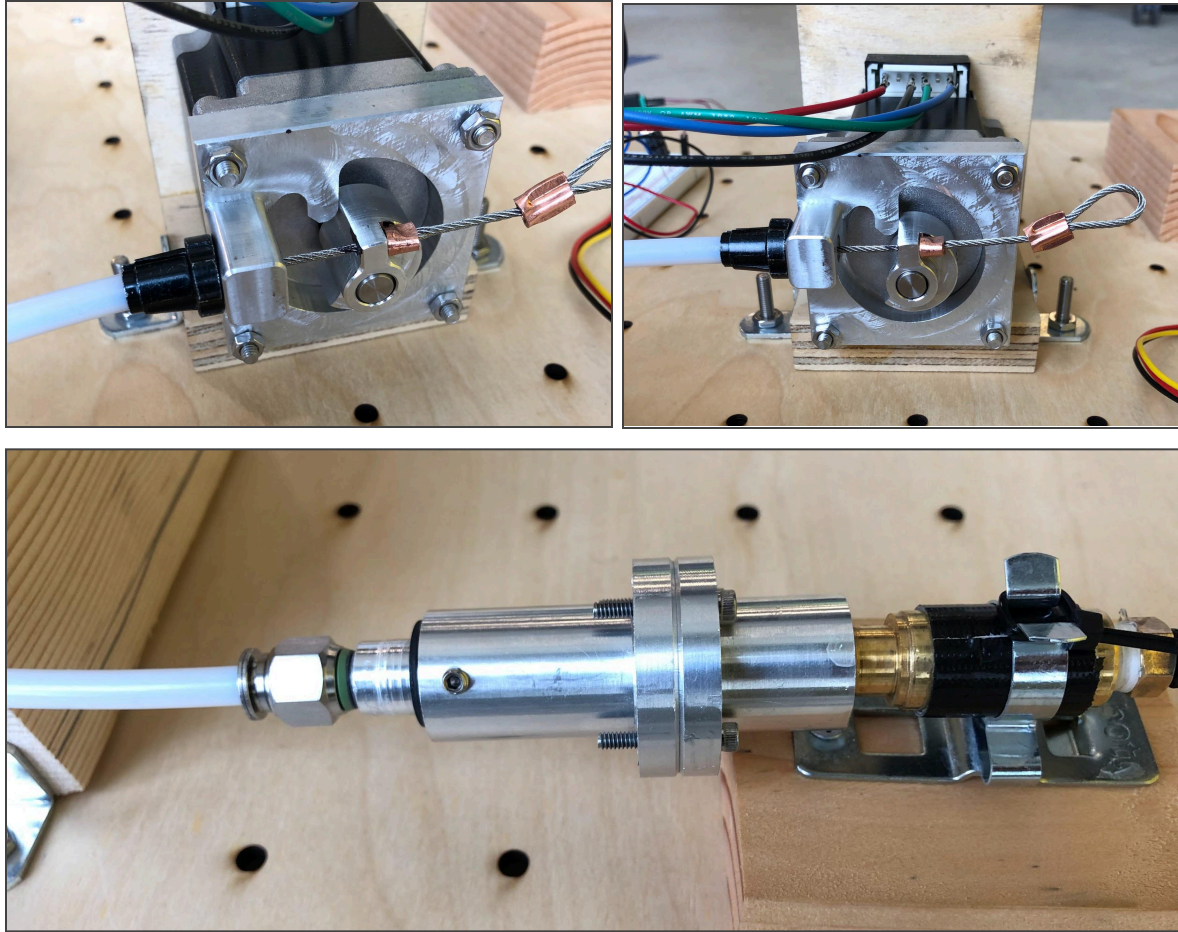


Figure 2.1.1: Final Assembly of Motor (top) and Valve Housing (bottom)

Chapter 3.0: Design of Key Components

Chapter 3.1: Valve Body

Table 3: Valve Housing Design Requirements

- Inlet and outlet interfaces are compatible with the current testing rig.
- The valve remains closed with zero force input from the actuator.
- Valve footprint remains within approximately half of the current AFC flow valve. Less than 7.5 cm (3 in.) in length. Approximately 2.6 cm (1 in.) in diameter.
- Material compatibility with temperature range of testing environment and process fluid specified to be -80°C to 175°C (-110°F to 350°F).
- Material compatibility with fluorinated refrigerant fluids. Acceptable corrosion and degradation resistance.
- Sufficient structural strength for 70 psi fluid with leakage specification at less than 5 mL/min
- Accommodates maximum flow rate of 1.5 L/min
- Valve housing separate from actuator system, connected by linkage mechanism

Valve Housing Design

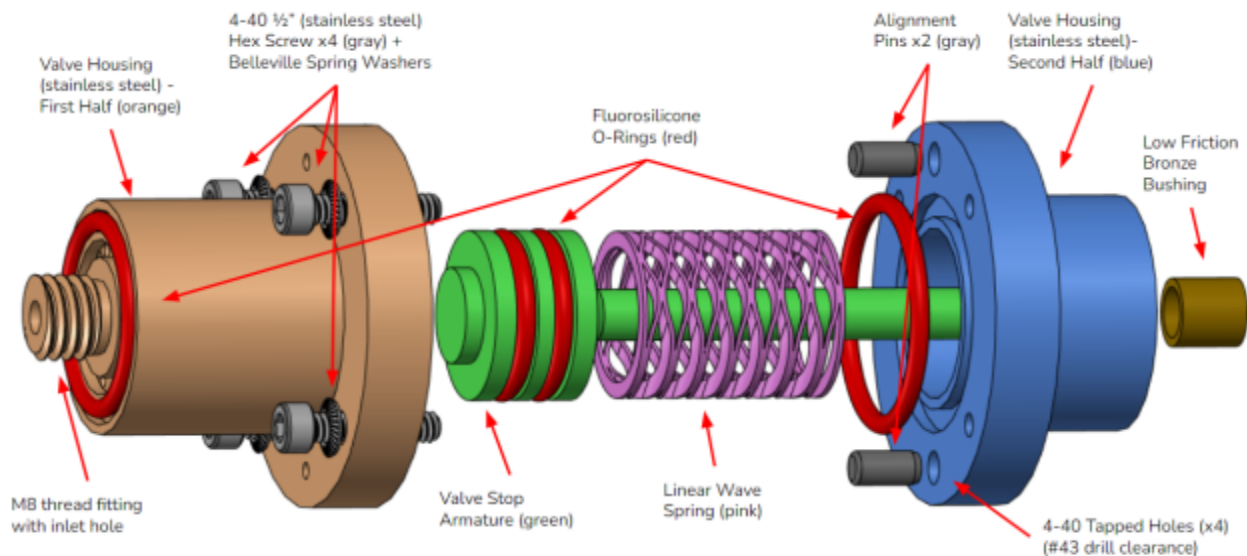


Figure 3.1.1: Annotated Valve Housing Assembly in Exploded View

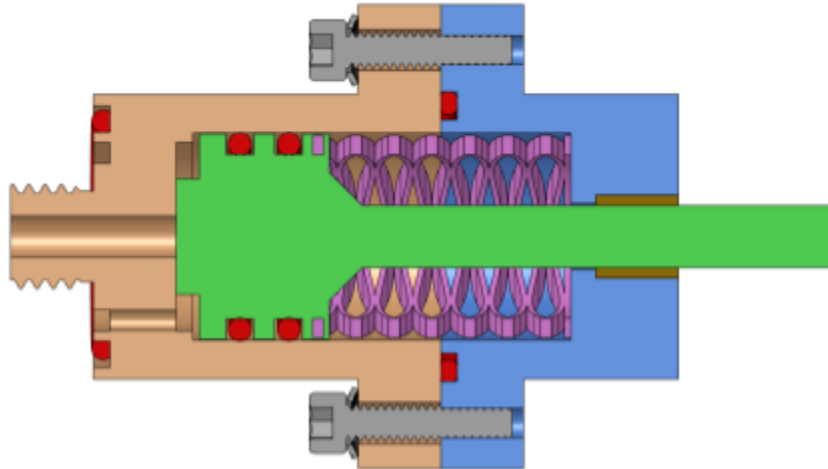


Figure 3.1.2: Valve Housing Section View

In this chapter, a redesign of the existing flow valve is discussed. The primary objective is to design a housing that can be attached to a mechanical linkage capable of joining an actuator to the armature. This design must fit the performance requirements outlined in Table 3. To ensure interface compatibility, the valve front is kept identical to the existing AFC valve while the remaining form factor is kept to a minimum.

Figures 3.1.1 and 3.1.2 depict the final flow valve concept design. The valve body has a length of approximately 51mm (2 in.) with a maximum external diameter of 35.5 mm (1.4 in.). The armature rod extends approximately 10mm (0.4 in.) from the rear of the housing where it interfaces with the Bowden cable.

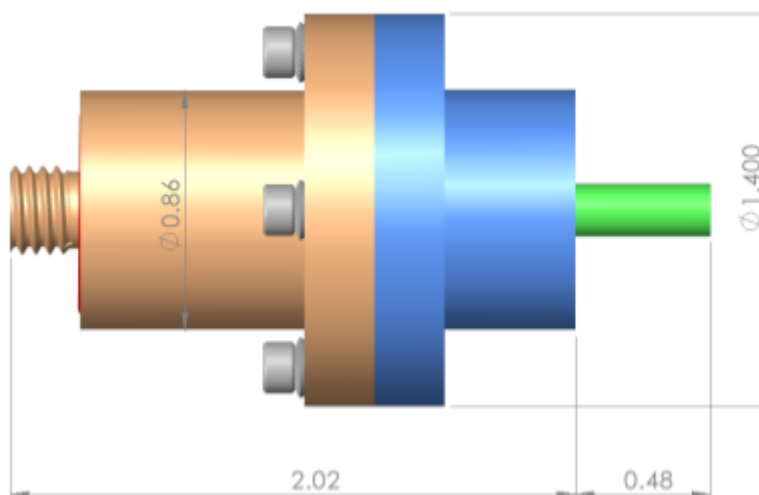


Figure 3.1.3: Valve Housing Dimensional Drawing

Material selection was a key consideration in the valve body design. A substantial temperature range necessitates materials with a melting point at least three times the maximum temperature of 175°C and a glass transition temperature well above the lowest temperature of -80°C. Furthermore, in the use of metals, alloys must be resistant to brittle transition at below freezing temperatures that is characteristic of certain alloys.

Further material considerations were contingent on chemical compatibility with the process fluid as well as other materials in the flow housing. Components that come into contact with process fluids and other valve housing materials must resist oxidation, corrosion, and other forms of chemical degradation appropriate to the life cycle of the flow valve. The process fluids used with this valve are compressed dry air, liquid water, HFE 7100 and Opteon SF10. The latter two fluids are fluorinated engineered fluids, described in detail in the accompanying data sheets located in appendix C1 and C2. Datasheets for these fluids [References] identify chemical compatibility with many metals and polymers, notably stainless steel, aluminum, and brass.

Ease of manufacture and costs associated with valve component sourcing were other driving considerations in materials. It is preferable to source components in the valve housing commercially as opposed to manufacturing them in house due to reduced costs and lead times. Exotic materials such as inconel steel that have favorable material and temperature compatibility were eliminated from the selection process due to difficulty in machining and higher sourcing costs.

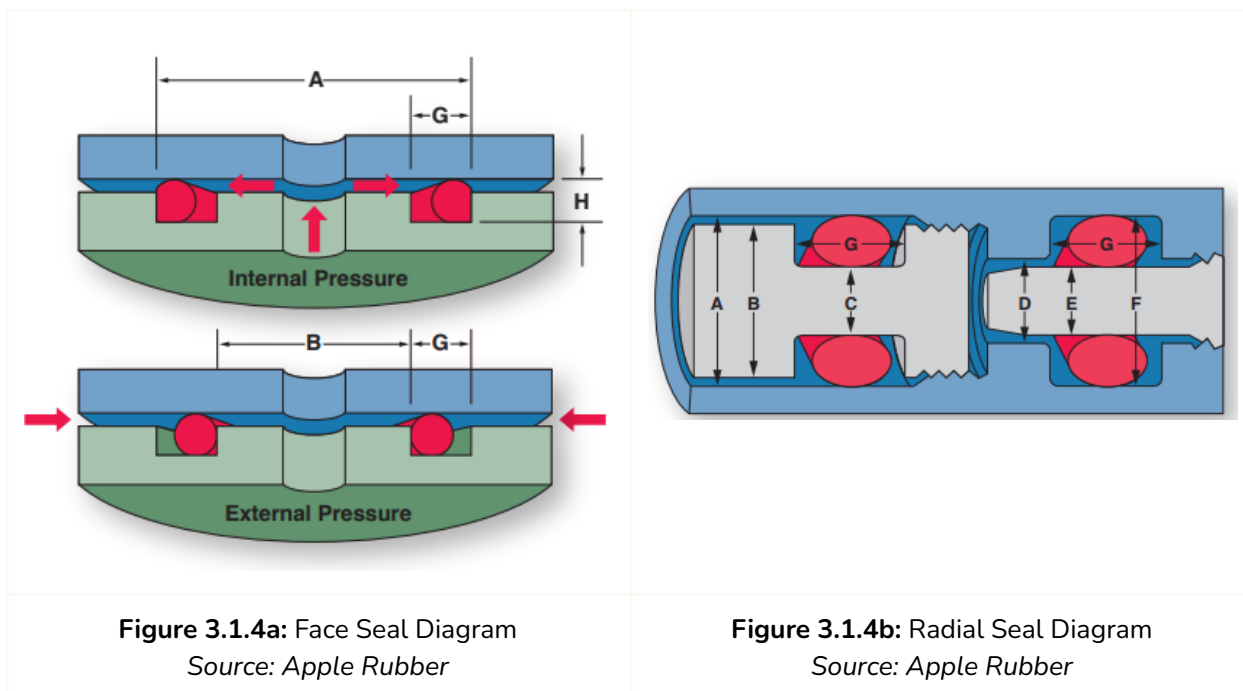
Table 4: Housing and Spring Material Selection Chart						
Material	Approx. Melting Temp.	Cost 1 - low 5 - high	Spring Availability	Low Temp. Compatible -80°C	Refrigerant Compatible & Chemically Stable	Ease of Machining 10 - hardest 1 - easiest
Aluminum 6061-T6	600°C	2	No	Yes	Yes	5
<i>Polymers</i>	-	-	-	-	-	-
-PP	160°C	1	No	No	Yes	N/A
-PETE	260°C	1	No	No	Yes	N/A
-PVC	85°C	1	No	No	No	N/A
-ABS	100°C	1	No	No	Yes	N/A
Brass 360	900°C	4	No	Yes	Yes	6

17-7 Stainless Steel	1400°C	3	Yes	Yes	Yes	7
304 Stainless Steel	1400°C	4	No	Yes	Yes	8
316 Stainless Steel	1400°C	4	No	Yes	Yes	9
1018 Steel	1400°C	3	Yes	Yes	No	7
Inconel Steel	1400°C	5	No	Yes	Yes	10

304 and 17-7 Stainless Steel, Brass 360, and Aluminum 6061-T6 metals appear to be most ideal in satisfying the requirements outlined in this section. These metals are explicitly stated to be chemically compatible with all process fluids in their respective data sheets and are resistant to thermal changes in the test environment and fluid. Furthermore, these metals are readily available commercially for material sourcing. Although chemically compatible and low cost, polymers are not an ideal material due to poor mechanical properties at high temperatures. 316 Stainless Steel and Inconel Steel are discarded from consideration due to a higher price point and higher expected machining difficulty for improved, but unnecessary, material strength.

This investigation will utilize aluminum 6061-T6 for design prototype due to low cost and ease of machining. For long-term strength retention and fatigue resistance, it is recommended to use 304 stainless steel.

To maintain an effective seal within the valve housing, several o-rings have been incorporated into the geometry. Two primary styles of o-rings are used for seals, face seals and radial seals. Face seals incorporate a gland housing perpendicular to the two faces of the surfaces being sealed, depicted in figure 3.1.1a. Radial seals provide a seal between two concentric sliding shafts, depicted in figure 3.1.1b.



Design constraints such as operating temperature range and the necessity of chemical stability in exposure to process fluids dictate o-ring material selection. Common o-ring materials and key material properties are listed in table 5 to aid in selection. O-rings must withstand temperature exposure to the HFE 7100 and Opteon SF10 process fluids and testing environment from -80°C to 175°C without degradation.

Table 5: O-Ring Material Compatibility Chart						
Material	Min Temp.	Max Temp.	COTF Availability	Cost 1 - low 5 - high	Refrigerant Compatible HFE 7100 / SF10	Temp. Compatible
PTFE (Teflon)	-180°C	260°C	Yes	4	No / No	Yes
Fluorosilicone	-70°C	200°C	Yes	3	Yes / Unsure	Yes*
Fluorocarbon (Viton)	-18°C	200°C	Yes	1	No / No	No
Silicone	-50°C	200°C	Yes	2	No / No	No
Nitrile (Buna)	-30°C	120°C	Yes	1	Yes / No	No
EPDM	-55°C	150°C	Yes	1	Yes / Yes	No

*Fluorosilicone o-rings are not rated for operation at -80°C , but sponsor guidance resulted in narrowing the scope of the design temperature requirements to a minimum of -70°C .

Sources: McMaster Carr O-Ring Listings & 3M/Opteon Datasheets

Initially, Polytetrafluoroethylene (PTFE also known by its trade name, Teflon), was identified as a suitable o-rings material. PTFE has desirable properties permitting use at both low and high temperatures as well as broad chemical stability. However, a review of HFE 7100's datasheet identified concerns about its compatibility with PTFE. Sponsor guidance suggested that contact with HFE 7100 caused swelling in PTFE o-rings, undermining its sealing capability.

Furthermore, PTFE plastic is hard with limited capability for deformation as needed for a housing seal.

To mitigate known concerns about swelling in PTFE, fluorosilicone o-rings were selected. However, there are concerns about fluorosilicone's chemical compatibility with Opteon SF10 because it is not explicitly identified as a compatible material on the corresponding datasheet. Using fluorosilicone o-rings necessitates modifying the design requirements for a minimum of -70°C in the process fluid.

In finalizing material selection, other steel fasteners and bronze bushings are used in the valve housing, but are not expected to come into contact with the process fluid.

Subsequent to sourcing an acceptable material, it was necessary to locate an acceptable spring for the valve mechanism. Compression springs are a commonly used mechanical component in industrial design with wide ranging variances in design specifications. Varying material choice and geometry will affect spring characteristics such as spring load and spring rate. The spring rate is the amount of force needed to compress a spring by a certain distance, typically expressed in pounds per inch or newtons per mm¹. This is also commonly expressed as the spring constant, K . This is determined by the shear modulus of the spring material and the specific geometric characteristics of the spring such as the number of coils and the diameter of the coil wire. A stiffer material, such as steel, is a better choice for a spring designed for large forces. The spring load is the force produced by the spring as a function of spring displacement, typically given by the following formula:

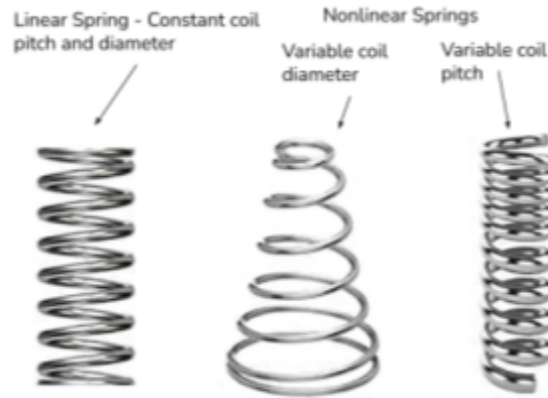


Figure 3.1.5: Differences Between Linear and Nonlinear Springs

$$\text{Equation 1: } F_{spring} = K \cdot \Delta x$$

Equation 1 is only valid for linear springs, or springs for which the spring load is proportional to spring displacement. Nonlinear springs do not follow this relationship. Although less commonly used than linear springs, nonlinear springs can be very useful in a wide variety of applications.

Springs represent the most predictable and cost effective way to meet the design requirements posed earlier. Linear springs can be easily modeled to predict valve behavior and can be paired with a solenoid actuator to easily implement increment displacements for valve flow. Linear springs are recommended for use in this design solution for the provided factors.

Based on the exit area of the nozzle and the pressure of the fluid, analytical methods were used to determine that a stainless steel wave spring with a maximum load of approximately 8N (2 lbf) is appropriate for a 70 psi fluid. The spring selected for this design solution is the CS062-M8-S17, sourced commercially from Smalley. A specification sheet for this spring is available in the appendix. The spring properties are described in Table 6 below.

¹ <https://www.collinsdictionary.com/us/dictionary/english/spring-rate>

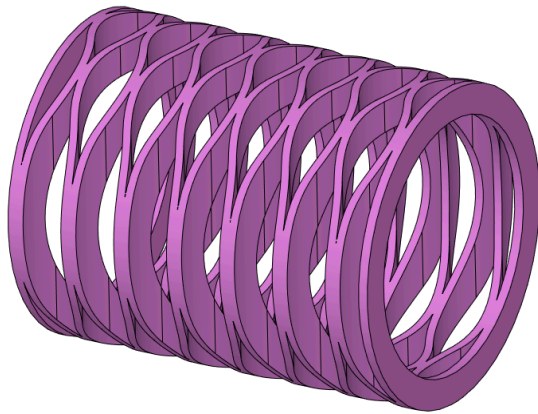


Figure 3.1.6: Smalley CS062-M8-S17 Wave Spring

Table 6: Spring Properties	
Spring Property	Description
Material	17-7 PH Condition CH900 Stainless Steel
Spring Constant (K)	6.5 N/mm (37 lbf/in.)
Maximum Load	53.4 N (12 lbf)
Outer Bore Diameter	15.875 mm (0.625 in.)
Inner Clearance Diameter	11.43 mm (0.450 in.)
Free Height	19.81 mm (0.780 in.)

To ensure that the zero-energy off-state of the valve is maintained, the spring must have an initial compression state such that the force is equivalent to the desired pressure holding force plus a safety factor. The force diagram can be visualized in figure 3.1.7.

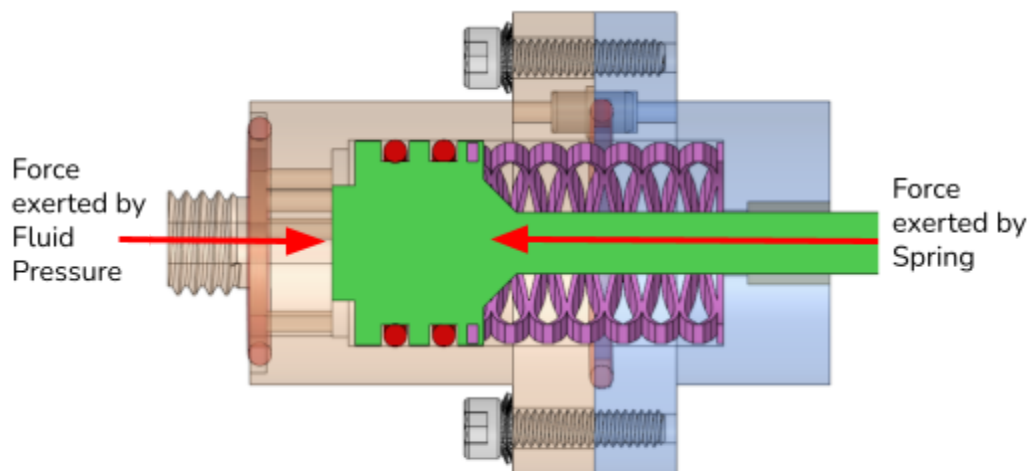


Figure 3.1.7: Valve Spring Force Diagram

The calculations for the force balance can be found in Appendix D1. The internal geometry of the housing must then be precision machined to ensure that the zero-energy state of the system compresses the spring to around 8N (1.8 lbf).

A bronze bushing is used at the back end of the valve to ensure the flow control armature is concentric to the valve housing.

Manufacturing

For the final hardware test, various aluminum prototypes were produced using CNC mill, CNC lathes, and manual finishing operations. A variety of tooling, ranging from end mills, drill bits, taps, and lathe tools is necessary for complete manufacturing. The CAM programs shall be included in the appendix. The individual aluminum prototypes are depicted in figures 3.1.8-3.1.10. The valve housing rear shown has the modified secondary housing which contains the interface between the plunger and bowden cable. Overall, the manufacturing process was highly successful in producing usable prototypes and validating manufacturing methods. The tooling necessary is shown in the table below:

Table 7: Valve Housing Manufacturing Tooling Chart		
Tooling Necessary (Milling)	Tooling Necessary (Lathing)	Tooling Necessary (Ancillary)
1/16" Flat End Mill	55° Turning Tool	4-40 Tap
1/8" Flat End Mill	35° Turning Tool	M8 Tap (If Necessary)
1/4" Flat End Mill	60° Outer Diameter Turning Tool	
3/4" Flat End Mill	0.125" Parting Groove Tool	
1/16" Drill Bit	0.070" Groove Tool	
3/16" Center Drill	0.25" Inner Diameter Boring Tool	
1/8" Drill Bit		
1/4" 45° Chamfer Mill		
#33 Drill Bit (0.089")		
#7 Drill Bit (0.201")		



Figure 3.1.8a: Valve Housing Armature (Isometric)



Figure 3.1.8a: Valve Housing Armature (Side)



Figure 3.1.9a: Valve Housing Rear (Back Isometric)



Figure 3.1.9a: Valve Housing Rear (Front Isometric)



Figure 3.1.10a: Valve Housing Front (Top)



Figure 3.1.10a: Valve Housing (Back)

A fully assembled valve housing can be visualized below in figure 3.1.11, incorporating the fasteners and internal components.

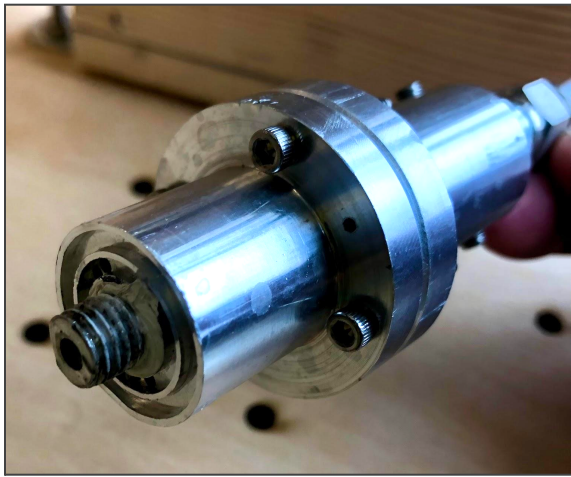


Figure 3.1.11a: Valve Housing Assembly

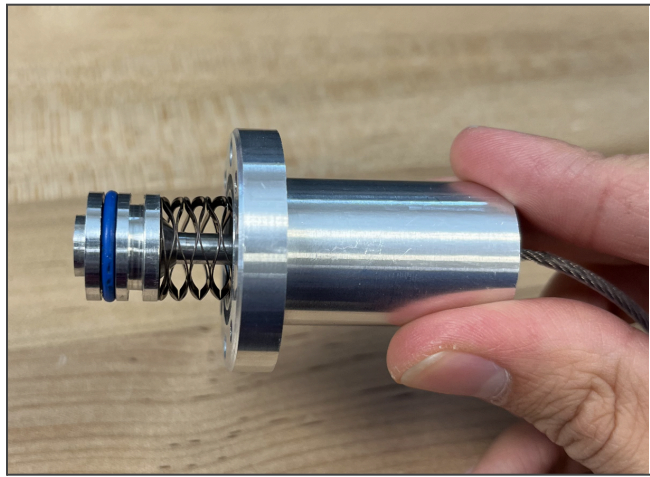


Figure 3.1.11b: Valve Housing Internal View

CAM Files:

https://drive.google.com/drive/u/3/folders/1QvcNeibxSXLeixYz1m_pYmUq4wqDbRpa

Chapter 3.2.1 Linkage Mechanism (Bowden Cable)

To meet the design requirements listed above, a Bowden cable was selected as the flexible mechanical linkage separating the actuator and valve housing. A Bowden Cable consists of three parts.

1. A wire rope transmits the forces from the actuator to the valve spring. It is typically made of stainless or galvanized steel due to those materials' high tensile and compressive strengths. It is connected to the actuator and valve terminal ends through an end fitting, typically an eye bolt or crimped ferrule connection.
2. A polymer tubing houses the wire rope to prevent any interference between the linkage and the environment. Unlike the wire, it is fixed at both ends and does not move or transmit force. A polymer is used for the purpose of low friction with steel wire, maximizing efficiency of the system and reducing wear.
3. A flexible metal sheath houses the other two components to prevent damage to the polymer tubing and wire rope. Like the polymer tubing, the metal sheath is fixed at both ends. As the name implies, the flexible metal sheath is composed of either galvanized or stainless steel due to their high durability and malleability.

This design was chosen due to the extreme flexibility of the linkage between both terminal ends, ease of manufacturing, and low cost. The system is to operate in a closed, predictable environment, so durable polymers are capable of fulfilling the role of both the tubing and flexible sheath. A hard plastic PTFE tubing was chosen for a multitude of reasons:

- As a hard plastic polymer, PTFE is more durable than softer polymers in given environmental conditions.
- It has a high operating temperature range of -200°C to 260°C.
- It has a low coefficient of friction.
- It can be ordered in bulk, notably for much cheaper than a typical metal sheath.

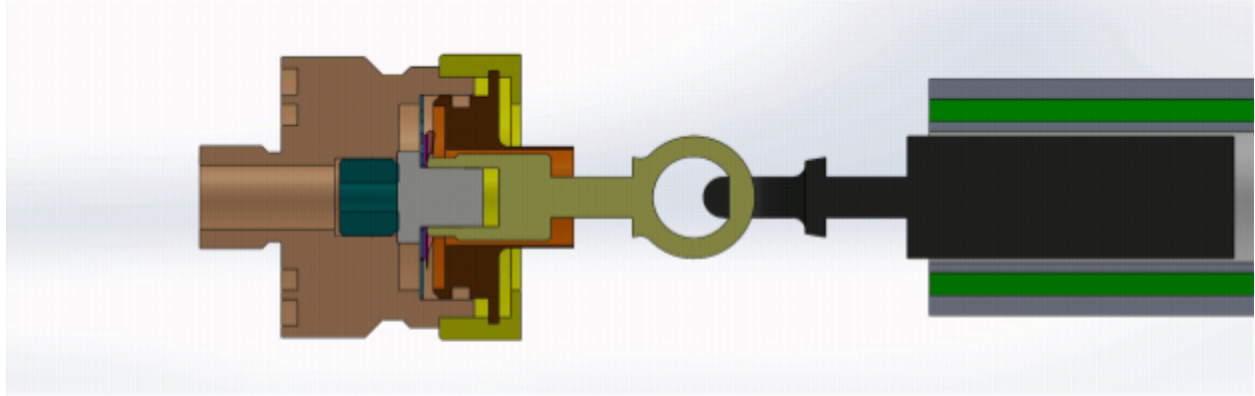


Figure 3.2.1: Animation of the Bowden Cable in Use in Old Valve Housing
(Black: Wire Rope, Green: Polymer Tubing, Gray: Spiral Sheathe)

Since the valve housing will be in constant cyclic motion and the forces are relatively small, fatigue will be the failure mode of the cable. The wire rope will not be tensioned in a straight line, so there will be two major forces that contribute to the fatigue loading on the wire: equivalent bending loading and tension under flexing.

The equivalent bending loading (F_b) is defined as the tension experienced by the wire when it is bent around a sheave. The equation for the equivalent bending loading is given below.

$$\text{Equation 2: } F_b = \frac{E_r d_w A_m}{D}$$

E_r = elastic modulus of the WIRE ROPE

d_w = diameter of the WIRE

A_m = area of metal in the ROPE (ratio of sum total of wire diameters to rope diameter)

D = sheave diameter

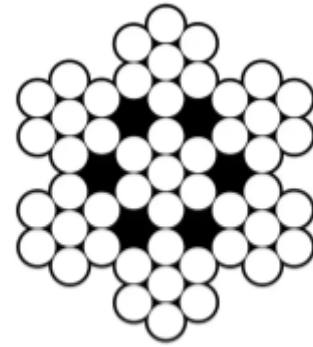


Figure 3.2.2: Cross-Sectional View of 7x7 Wire Rope

For a 7x7 lang lay wire rope we can determine the diameter of the wire and the area of metal in the rope relative to the diameter of the wire rope. As seen in Figure 3.2.2, the diameter of each wire (d_w) should be $1/7$ of the diameter of the whole wire rope. As such, the area of metal in the rope (A_m) can be found by taking the area of each wire and multiplying by 49, the number of individual wires. Thus, Equation 2 can be rewritten in terms of the diameter of the wire rope (d) as follows.

$$\text{Equation 3: } F_b = \frac{E_r \left(\frac{d}{7}\right) \left[\pi \left(\frac{d}{14}\right)^2\right]}{D} = \frac{\pi}{2916} \left(\frac{E_r}{D}\right) d^3$$

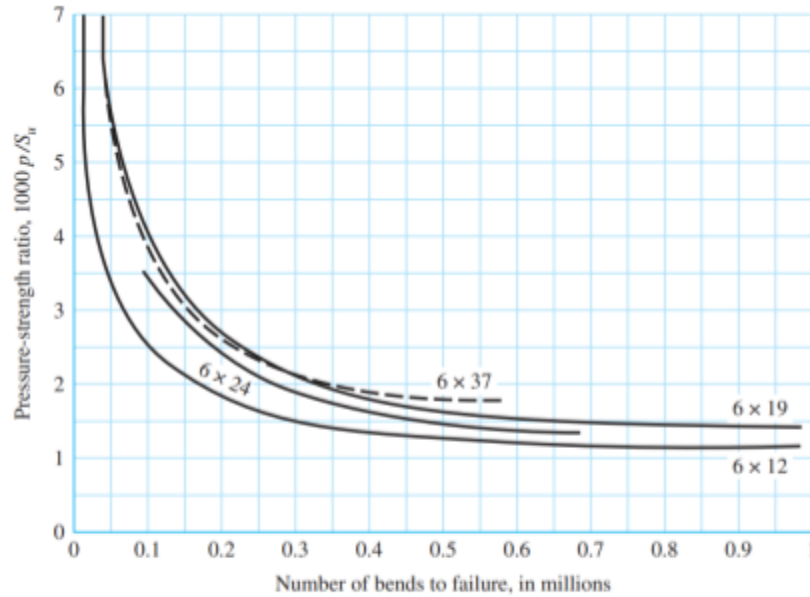


Figure 3.2.3: Experimentally determined relation between the fatigue life of wire rope and the sheave pressure
Source: Shigley's Mechanical Engineering Design

The allowable fatigue tension in the wire (F_f) measures the load on the wire “as it is flexed a number of times corresponding to the ratio of the sheave pressure to the ultimate tensile strength” (Shigley's Mechanical Engineering Design). As seen in Figure 3.2.3, the pressure-strength ratio seems to asymptotically approach 0.001 as the wire tends towards infinite cycle lifetime, which is what the system should be designed for. The equation used to measure the allowable fatigue tension is given below.

$$\text{Equation 4: } F_f = \frac{(p/S_u)S_u D}{2} d$$

p = sheave pressure to the ultimate tensile strength

S_u = ultimate tensile strength

D = sheave diameter

d = diameter of the WIRE ROPE

The tension of the wire under flexing (F_f) is the maximum tension the terminal ends of the wire rope will experience when the external loading is applied. The spring in the valve housing is rated for up to 12 lbf before plastic deformation, as specified above. The maximum tension in the wire will be estimated using this value.

Efficiency losses due to frictional forces will occur between the loading end of the wire (actuator) and the holding end of the wire, as seen in Figure 3.2.4. Deriving the equation based

on the FBD gives the following relationship between the tensions on the two ends of the pulley when the wire rope is not slipping.

$$\text{Equation 5: } T_L = T_H e^{\mu_s \Theta}$$

T_L = loading tension

T_H = holding tension

μ_s = coefficient of static friction

Θ = TOTAL bend angle of the wire rope (in radians)

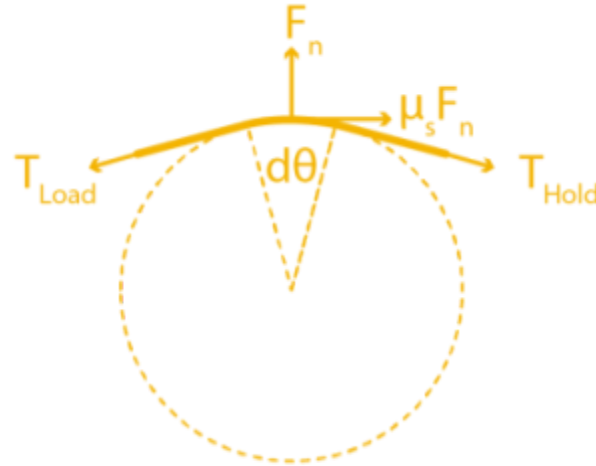


Figure 3.2.4: Free body diagram of an infinitesimal cross section of cable measuring with infinitesimal angle $d\theta$
(Source: Hackaday)

These friction losses are proportional to the total bend angle of the wire rope in contact with the sheave. Because of the unpredictability of Cohu's working environments, this bend angle can not be quantified, so the worst case scenario of a full 360° (equivalent to four 90° bends) is assumed. Because the outer sheath was chosen to be teflon, the friction coefficient between the wire rope and the outer tubing is only 0.04. Therefore, the maximum tension of the wire rope under flexing at the actuated will be at most 1.286 times the tension in the wire at the holding end (15.43 lbf).

The relationship between these forces and the fatigue factor of safety in the wire rope is provided by the following equation.

$$\text{Equation 6: } n_f = \frac{|F_f - F_b|}{F_t}$$

n_f = fatigue factor of safety

F_f = allowable fatigue tension in the wire

F_b = equivalent bending loading

F_t = maximum tension in the wire under flexing

Rearranging the equation to solve for the rope diameter gives us the following cubic equation.

$$\text{Equation 7: } \frac{\pi}{2916} \left(\frac{E_r}{D} \right) d^3 - \frac{S_u D}{2000} d - F_t n_f = 0$$

Because the design is unique to Cohu's specific testing environment and the loads are relatively light, a minimum FOS of 8 was chosen for the current design. For the FEP (teflon) coated, 304 stainless steel wire rope chosen for our system, the minimum desired nominal

diameter for the wire rope comes out to 0.0553". A diameter of 0.0625 (1/16") was chosen based on these calculations as it is the closest to the calculated nominal diameter and it is easy to source fittings for.

Because the intermediate linkage connecting the actuator to the valve plunger will not be a direct linear path, there is bound to be some motion losses between the motion source

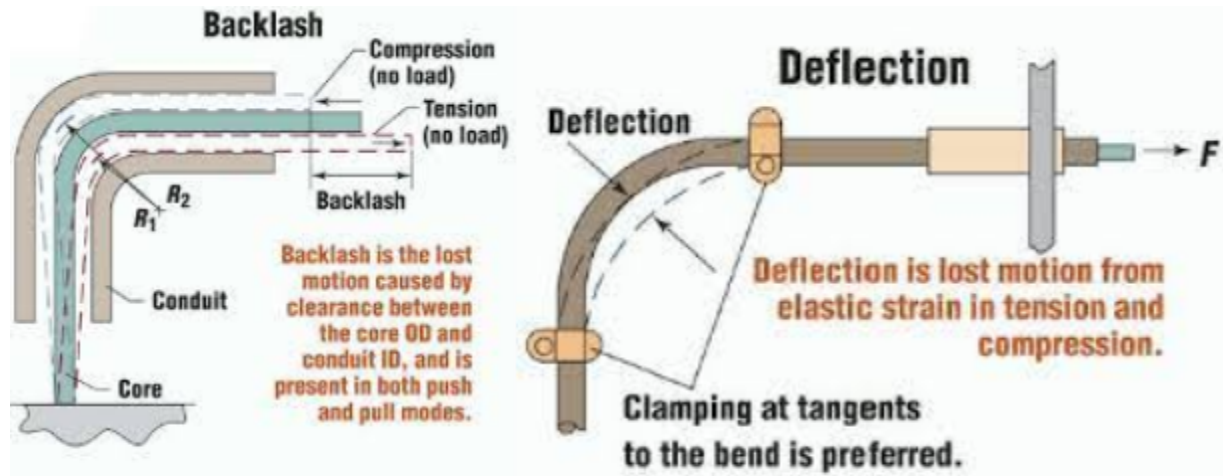


Figure 3.2.5: Illustrations of a) cable backlash, and b) conduit deflection.
Source: Cable Manufacturing & Assembly Co. Inc

(actuator) and the receiving end (valve plunger) known as backlash. Cable backlash is caused by the “clearance between the core OD and conduit ID, and is present in both push and pull operating modes” (Cable Manufacturing & Assembly Co. Inc.).

There are three key factors that contribute to cable backlash (referencing Figure 3.2.5):

1. Core Centerline in Tension (R_1)
2. Core Centerline in Compression (R_2)
3. Total Angular Degrees of Bend in Routing (X)

This can be calculating using the following formula:

$$\text{Equation 8: } B = R_2 \frac{X\pi}{180^\circ} - R_1 \frac{X\pi}{180^\circ}$$

As per Cohu’s design requirements, the linkage is supposed to be flexible and fit within a wide variety of testing environments for their applications. This means that the number and angle of the bends will be unpredictable and, as such, so will the resulting backlash. To minimize the backlash and maintain a consistent control scheme, there must be sufficient clearance between the polymet tubing (conduit) and the inner wire (core). Alternatively, we could have a fixed

number of bends with a fixed radius to generalize the backlash among each of the testing environments.

Another source of motion loss is cable deflection, which is caused by tensile or compressive loads in the cable itself. This can be calculated using the following formula:

$$\text{Equation 9: } \Delta L = \frac{FL}{AE}$$

F = average load ($\frac{1}{2}$ input load + $\frac{1}{2}$ output load)

L = length of active inner core

A = core cross-sectional area

E = Young's Modulus of Core

With our wire length of 1/16" the deflection measures on the order of 10^{-7} m, which is negligible compared to the maximum cable retraction (~2mm) and the length of each step of the cable.

Chapter 3.2.2 Linkage Mechanism (End Fittings)

The terminal ends of the wire and the polymer sheath will need to be fitted onto the valve body and the actuator with some sort of fitted connection. An example of a typical threaded end fitting in a Bowden cable is shown in Figure 3.2.6.

Because the diameter changes between the Bowden cable and the valve components are so large, some of these components cannot be secured using commercially available fittings and thus, must be custom made.

There are four total fittings within the system. The fittings must connect both the wire and the sheath to the valve housing and the actuator.

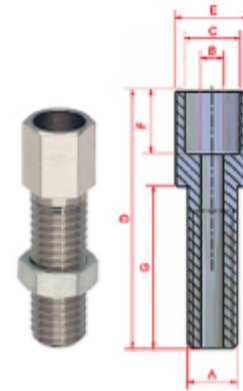


Figure 3.2.6 Threaded End Fitting
(Source: CarlStahl Technocables)

The fitting for the wire rope at the valve terminal is a custom made step-down ferrule fitting made from aluminum stock. One end of the ferrule will have a 10-32 female threading that will be attached to the valve stop armature depicted in Figure 3.1.3. The failure mode of the threads on both the ferrule and the armature can be determined by measuring the applicable stresses at the interface and determining the factor of safety. All four types of stresses (axial, bending, torsional, and shear) contribute to the load bearing capacity of the threads.

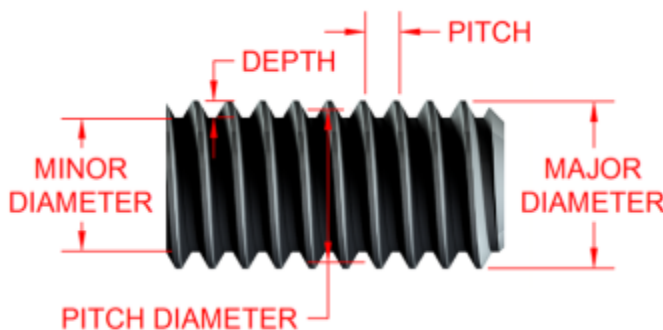


Figure 3.2.7: Terminology of a Male Thread
(Source: Tormach Blog)

Axial stress occurs in the main body of the threaded components. It is caused by the system actuation and can be calculated by assuming each of the components as shafts with a diameter equal to the minor diameter of the thread and a nominal load equivalent to the maximum tensile load the actuator provides. Thus, the axial stress can be determined by the following equation.

$$\text{Equation 9: } \sigma_{axial} = \frac{F_{motor}}{\pi(d_r/2)^2}$$

σ_{axial} = axial stress caused by an applied loading

F_{motor} = applied loading from the motor (positive because its a tensile load)

d_r = minor (root) diameter of the thread

Bending stress occurs in the individual threads themselves due to the reaction forces between the male and female threaded components. The distribution of these reaction forces depends on the number of total threads engaged in the system, which is why a 10-32 threading was chosen over a 10-24 threading. An example of the bending stresses due to the thread engagement can be shown in Figure 3.2.8. As shown

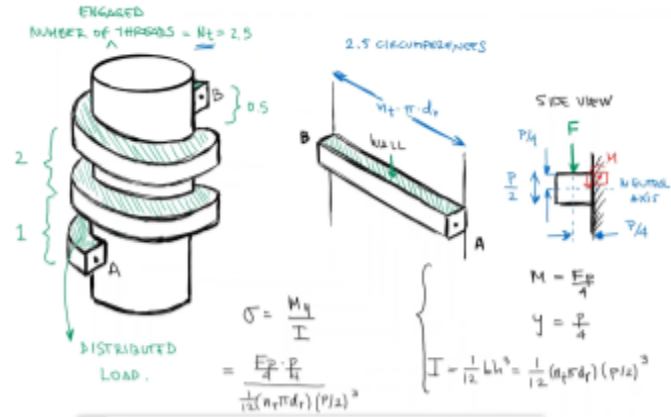


Figure 3.2.8: Depiction of Bending Stresses in Threaded Components
(Source: Less Boring Lectures)

in the figure, the application of the force occurs at exactly the pitch diameter of the thread. These reaction forces are responsible for bending the threads slightly as if it were a very short, thick cantilever beam. The area moment of inertia can be modeled as a rectangle with a height equal to half the pitch of the thread and a base equal to half the height of the thread. Thus, the bending stress can be determined by the following equation.

$$\text{Equation 10: } \sigma_{\text{bending}} = \frac{6F_{\text{motor}}}{\pi d_r n_t p}$$

σ_{bending} = bending stress caused by an applied loading

F_{motor} = applied loading from the motor (positive because its a tensile load)

d_r = minor (root) diameter of the thread

n_t = number of threads engaged in the system

p = pitch distance between the threads

Shear stress occurs between the threads of one component and the root of the other due to the sliding friction that occurs between them when the bolt is screwed into the ferrule. The torque needed to tighten the bolt must be enough to overcome the friction necessary to screw the armature into the ferrule. The same rectangle can be assumed for the applied area of this torque. Thus the shear stress between the male and female threads can be determined by the following equation.

$$\text{Equation 11: } \tau_{transverse} = \frac{4T}{\pi d_r^2 n_t p}$$

$\tau_{transverse}$ = transverse shear stress caused by bolt preload torque

T = bolt preload torque (minimum torque needed to overcome friction to screw components together)

d_r = minor (root) diameter of the thread

n_t = number of threads engaged in the system

p = pitch distance between the threads

Torsional stress occurs around the root of the engaged component as a result of the same bolt preload torque that causes the shear stress between the male and female threaded components. The bolt preload torque required to overcome the friction between the threaded components also causes an internal moment within the threads that twists these components. The torsional stress between the male and female threads can be determined by the following equation.

$$\text{Equation 12: } \tau_{torsion} = \frac{16T}{\pi d_r^3}$$

$\tau_{torsion}$ = torsional shear stress caused by bolt preload torque

T = bolt preload torque (minimum torque needed to overcome friction to screw components together)

d_r = minor (root) diameter of the thread

n_t = number of threads engaged in the system

p = pitch distance between the threads

Using these four quantities, we can determine the von Mises stress, which relates the in plane and out of plane stresses experienced by the bolt into one singular stress quantity. Each of the stresses act on the bolt in a specific direction or plane. With the assumption that the bolt is facing downwards (pointed in the negative y-direction) the von Mises stress within the bolt is determined by the following equation.

$$\text{Equation 13: } \sigma_{vm} = \sqrt{(\sigma_x - \sigma_y)^2 + (\sigma_x - \sigma_z)^2 + (\sigma_y - \sigma_z)^2 + 6(\tau_{xy} + \tau_{xz} + \tau_{yz})}$$

$\sigma_x = \sigma_{bending}$ = normal stress acting on the bolt in the x-direction

$\sigma_y = \sigma_{axial}$ = normal stress acting on the bolt in the y-direction

$\sigma_x = 0$ = normal stress acting on the bolt in the x-direction

$\tau_{xy} = 0$ = shear stress acting on the bolt in the xy-plane

$T_{xz} = T_{\text{transverse}}$ = shear stress acting on the bolt in the xz-plane

$T_{xy} = T_{\text{torsion}}$ = shear stress acting on the bolt in the yz-plane

Because of the precision of the displacement in the system, the proper failure mode will be yielding (not ultimate tensile failure) within either of the threaded connections. The yield strength of the bolt is determined by two factors: the minimum yield strength of the ASTM grade and the stress area of the specific diameter of the thread type. The minimum yield strength of the ASTM can be obtained from external sources, as shown in Figure 3.2.9.

Grade Marking		Specification	Material and Treatment	Nominal Size (In.)	Mechanical Properties					Hardness  Rockwell	
					Proof Load Min (ksi)	Yield Strength Min (ksi)	Tensile Strength Min (ksi)	Elong % Min ^[8]	RA % Min ^[10]	Min	Max
☐		A193/A320 Grade B8 Class 1	AISI SS304 Stainless	No Restrictions	-	30	75	30	50	-	B96

Figure 3.2.9: Strength Requirements of Thread by Grade
(Source: Portland Bolt and Manufacturing Company)

The calculated stress area in a standard 10-32 UNF bolt type is 0.020 in² and the minimum yield strength in our 304 stainless steel plunger is 30 ksi. The potential switch from aluminum to steel means that we also need to determine the minimum yield strength of an aluminum grade bolt. Aluminum and steel have very similar yield and ultimate tensile strengths, with the minimum yield strength of an Aluminum 2024 grade bolt being 34 ksi. For this case, the yield strength of the 304 stainless steel will be used since it is the worst possible case, but the allowable yield strengths will be relatively similar.

$$\text{Equation 14: } n_s = \frac{S_y}{\sigma_{vm}}$$

With the previously established values, the static factor of safety in the ferrule is calculated to be around 1.257, which suggests that the component will not yield under the maximum loading, but is still relatively low for the system.

The static factor of safety is high enough that the armature will not yield when there are no external factors affecting the system. However, as with the wire rope, the constant cyclic loading and low maximum loading suggests that the system will suffer fatigue failure. It can be assumed that the system will not have infinite life and that the armature will wear due to fatigue loadings, so added supports are necessary to ensure a high life cycle.

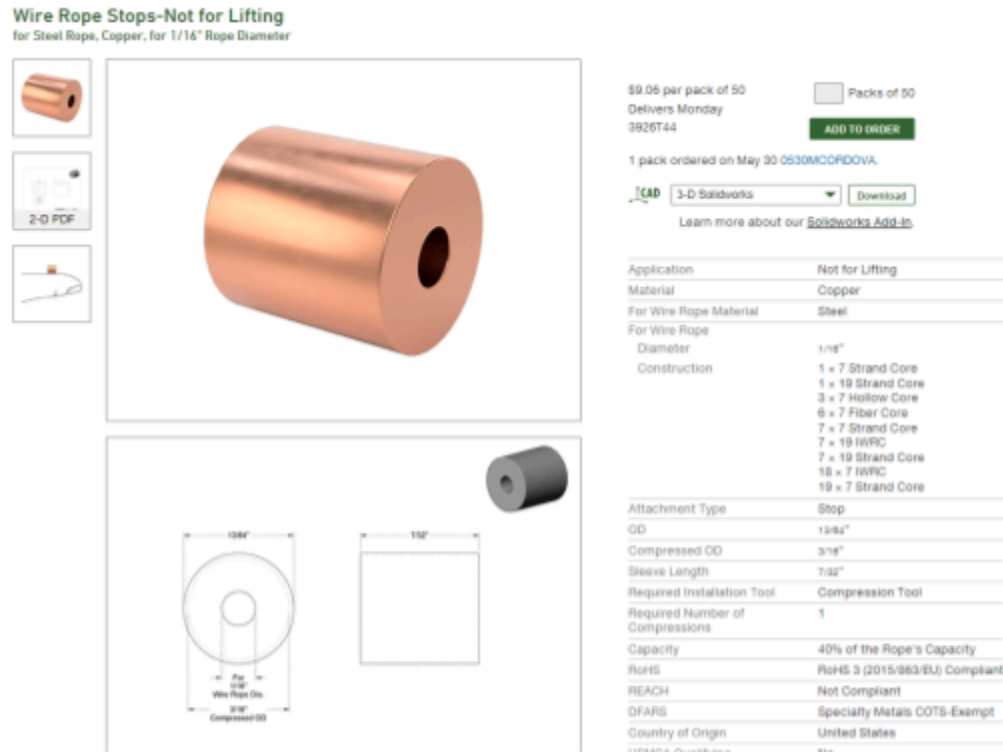


Figure 3.2.10: Part Specs for Wire Rope Stop
(Source: McMaster Carr)

These added supports could include solder between the armature and the ferrule, which would make the connection permanent and part replacement exceedingly difficult. Another method of support might include adding a slight chamfer to the armature to increase the thread size at the interface, since the current failure mode is in the threads, not the shaft of the armature. The major constraint for this design would be assembly of the mechanism, since the armature would now have to be split into two parts to properly run it through the bushing. Unfortunately with the added support to the armature/ferrule interface due to the relatively low static FOS, it will not be possible to properly quantify the fatigue FOS alone.

The other end of the ferrule will be crimped to the stripped wire. The unpredictability of crimp strength makes it difficult to determine the failure point of the ferrule/wire interface, but it can be assumed to be a bit weaker than the threaded fitting. Because the fitting is custom-made, the dimensions of the crimp should be modeled after commercially available crimp fittings to ensure the best possible fit. The sleeve length was increased to $\frac{3}{8}$ " to increase engagement with the wire rope and increase the fatigue FOS of the crimp fitting. The compressed OD of the crimp fitting must be $\leq \frac{5}{32}$ " to ensure a proper sliding fit between the crimped end of the step-down ferrule and the ID of the push-to-connect inlet that connects the polymer sheath

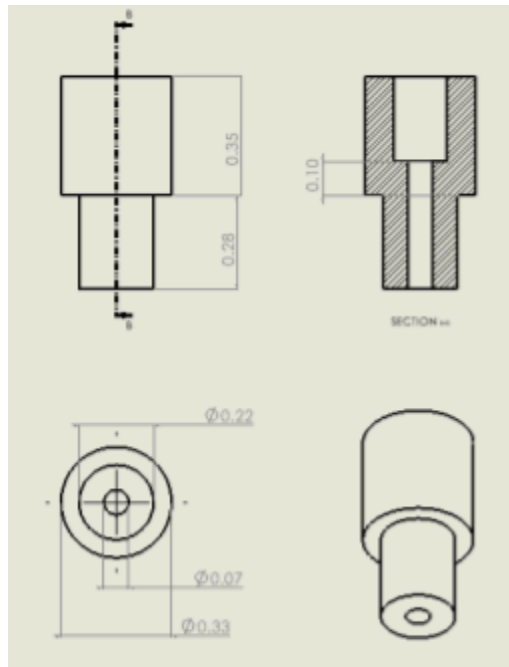


Figure 3.2.11: Solidworks Drawing of Step-Down Crimp Ferrule
Note: wider section will have a 10-32 threading

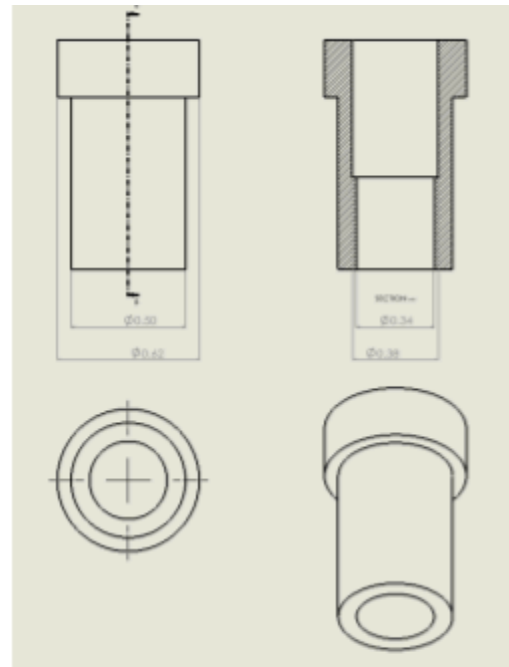


Figure 3.2.12: Solidworks Drawing of Sheathe Stopper
Note: thinner section will have a 1/4" NPT threading that will be 0.25-0.3" long to connect to the push-to-connect fitting on the polymer sheathe

to the sheath stopper shown in Figure 3.2.12. An improper crimp can cause this problem, which means precautions need to be taken to ensure that the ferrule can slide in and out of the stopper.

To resolve this issue, the crimp fitting can be filed down to fit the sizing of the push-to-connect inlet. For the best possible fitting, the ferrule should be attached to the rope using a swage press so the load is more evenly distributed across the wire since it compresses the fitting radially rather than tangentially like a typical crimping tool. Alternatively, the diameter of the crimped-end can be reduced to fit within the push-to-connect inlet and can be soldered to the wire using high temperature brazing flux (like silver) to better secure the wire. The dimensions of the push-to connect fitting are shown in Figure 3.2.13 and will be discussed more regarding the terminal end fitting for the polymer sheath at the valve end.

The fitting for the wire rope at the actuator end is far simpler than the step-down ferrule connecting the wire rope to the valve. A compressive stop, like the one shown in Figure 3.2.11, will be crimped onto the wire rope. As seen in Figure 3.2.13 the stopper will be tensioned against the motor wheel to reduce any backlash in the wire rope. These sleeves are COTS and will also be a critical point of failure since it directly takes the full 12 lbf from the wire rope while experiencing shear between the wire rope. Unfortunately, the strength of this crimp fitting is also difficult to quantify, so the same methods can be used to ensure a stronger

attachment method to the wire rope, without all the temperature constraints at the valve terminal.

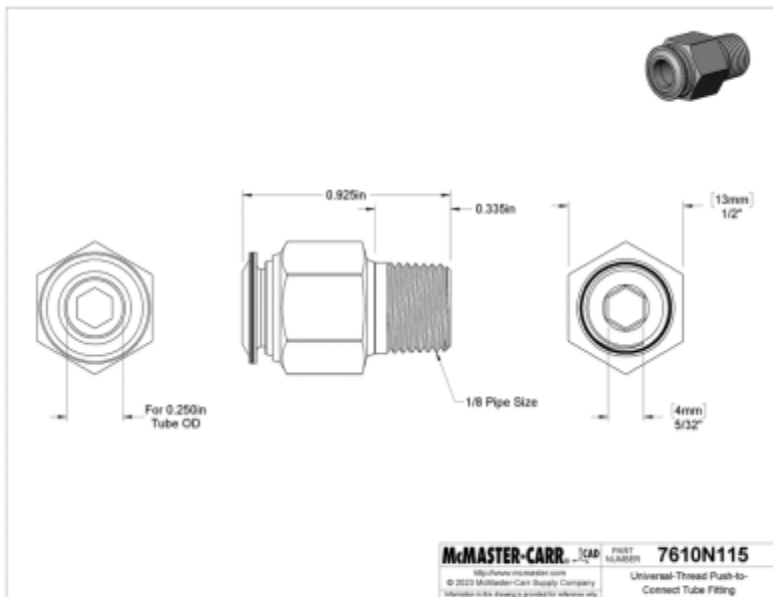


Figure 3.2.13: Push-to Connect Spec Sheet
Source: McMaster Carr

The fitting for the polymer sheath at the stepper motor is an M7 barrel adjuster. This component's purpose is to terminate the sheath while also allowing for adjustment of tension in the wire rope. By screwing the barrel adjuster into the standoff that is attached to the motor plate, the effective length of the conduit is reduced, reducing the amount by which the wire rope is stretched, reducing tension. The converse is true for unscrewing the barrel adjuster. The further it

protrudes from the conduit side, the greater the effective length of conduit and thus the tension. This way, the tension does not have to be perfectly set when crimping the wire rope stop at the motor wheel. A slight tension of 0.5 lbf was found desirable, measured from the loop ferrule on the actuator side past the stop via force gauge.

The fitting for the polymer sheath at the valve housing is modeled very similarly to the threaded end fitting shown in Figure 3.2.6, however it will not contain a female threaded connection. If both terminals are threaded and the sheath experiences any torque, the threads can come undone, which is problematic design. Instead, the terminal end will act more like a stopper mechanism that will be held in place by a set screw and an external retaining ring. An external retaining ring will be wrapped around the stopper itself which will prevent the stopper from being pushed further into the valve housing as the wire is tensioned at the barrel adjuster. The set screw will prevent the stopper from being pulled out of the valve body as depicted in Figure 3.2.14. An internal retaining ring was recommended instead of the set screw for securing the stopper, however the method of assembly proved too difficult and inconsistent for retaining the stopper mechanism. A section view of the internal retaining ring assembly can be seen in Figure 3.2.15, and may still be considered if assembly proves to be less complicated.

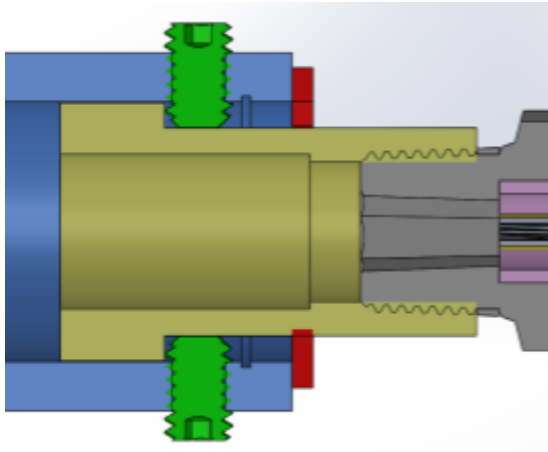


Figure 3.2.14: Close-up of (Current) Sheathe/Valve Interface and Terminal Fitting

Blue: Back of Valve Housing
Pink: PTFE Polymer Sheathe
Yellow: Sheathe Terminal Plug
Gray: Push-to-Connect Fitting
Red: External Retaining Ring (attaches to plug)
Green: 8-32 Set Screws

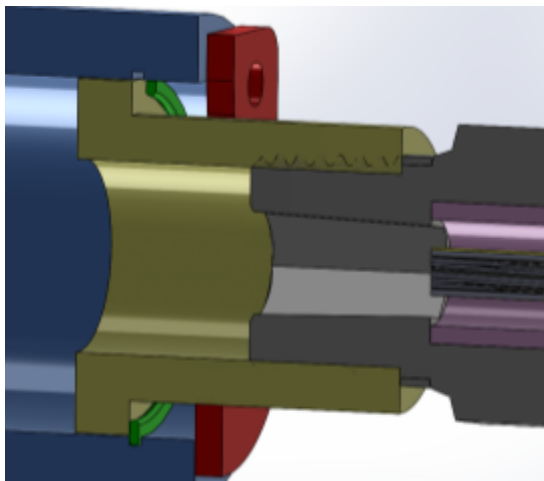


Figure 3.2.15: Close-up of (Former) Sheathe/Valve Interface and Terminal Fitting

Blue: Back of Valve Housing
Pink: PTFE Polymer Sheathe
Yellow: Sheathe Terminal Plug
Gray: Push-to-Connect Fitting
Red: External Retaining Ring (attaches to plug)
Green: Internal Spiral Retaining Ring

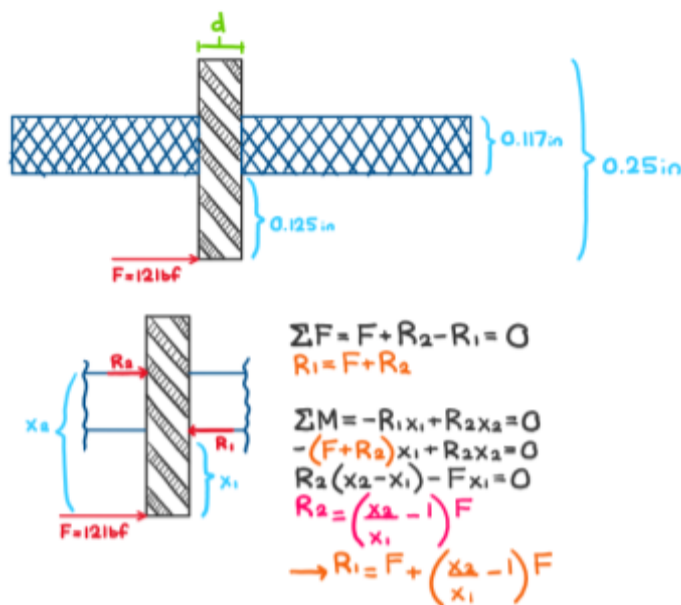


Figure 3.2.16: Model and FBD of the Set Screw in the Back Valve Housing

The dimensions of the set screw were chosen assuming the screw as a solid shaft with a diameter equal to the root diameter of the screw. For the worst-case scenario, we assume that the sheath is loaded with the same force (12 lbf) as the wire. A diagram of the component can be seen in the figure below.

As seen in Figure 3.2.14, the two reaction forces cause the main bending moment in the shaft. A diagram of the shear forces and the bending moments can be seen in Figure 3.2.15 below. From the graph, we determine that the

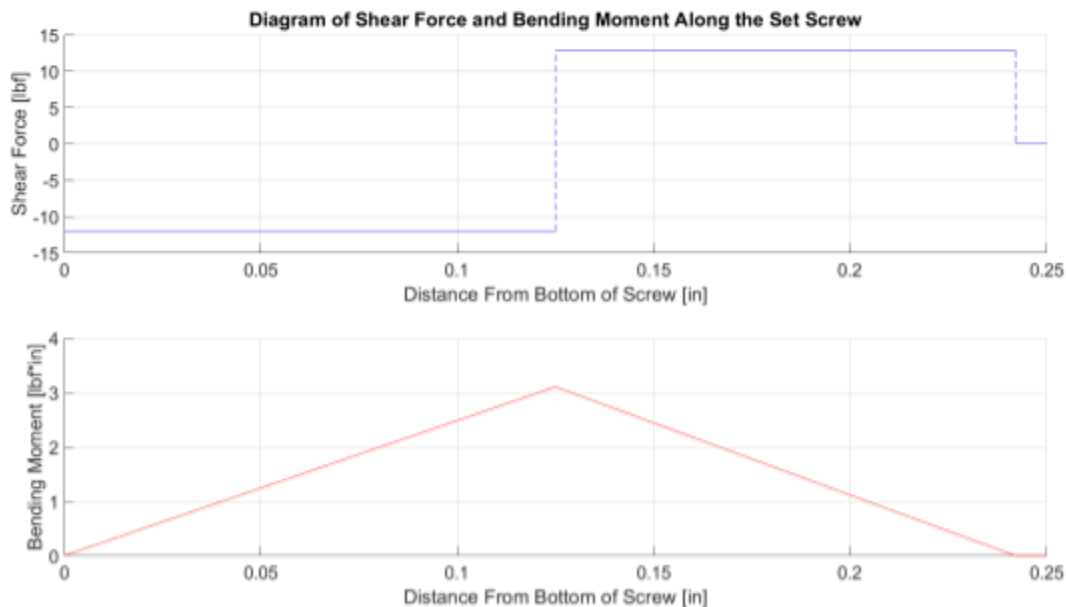


Figure 3.2.17: Graphs of Loadings on the Set Screw

location of maximum stress will be at the point where the first reaction force is applied, and thus the fatigue calculations will be done at that location.

To determine the required diameter for the set screw to have theoretical infinite life, we must determine the endurance limit of the material. The endurance limit of a part or system is defined as the stress level at which the material can have infinite life. The endurance limit is determined by a variety of modification factors known as Marin factors, which are unique to the material selection and the part.

The first Marin factor is the surface factor (k_a) which has an impact on cracks forming at areas of high stress concentration. The surface factor is defined by the relationship given in equation 15, which has been determined from the experimental testing of various steel components. The empirical data has been exponentially modeled for each surface finish, and the constants used for each finish are given in Figure 3.2.16 below. For our component, we will assume a machined finish for an 18-8 stainless steel set screw.

Surface Finish	Factor a		Exponent b
	S_{ulr} kpsi	S_{ulr} MPa	
Ground	1.34	1.58	-0.085
Machined or cold-drawn	2.70	4.51	-0.265
Hot-rolled	14.4	57.7	-0.718
As-forged	39.9	272.	-0.995

Figure 3.2.18: Empirical Data for Determining Surface Factor
Source: Shigley's Mechanical Engineering Design

Equation 15: $k_a = aS_{ut}^b$

k_a = surface factor of component

S_{ut} = ultimate tensile strength of component

The second Marin factor is the size factor (k_b), which relates the surface area to the likelihood of crack formation. Larger parts have greater surface area at high stress levels, so their size factor is much lower than it would be for smaller parts. For bending loads and torsional loads, the size factor for components with circular cross-sections are given based on empirical testing data.

$$k_b = \begin{cases} (d/0.3)^{-0.107} = 0.879d^{-0.107} & 0.11 \leq d \leq 2 \text{ in} \\ 0.91d^{-0.157} & 2 < d \leq 10 \text{ in} \\ (d/7.62)^{-0.107} = 1.24d^{-0.107} & 2.79 \leq d \leq 51 \text{ mm} \\ 1.51d^{-0.157} & 51 < d \leq 254 \text{ mm} \end{cases}$$

Figure 3.2.19: Equations for Determining Size Factor
Source: Shigley's Mechanical Engineering Design

$$k_c = \begin{cases} 1 & \text{bending} \\ 0.85 & \text{axial} \\ 0.59 & \text{torsional} \end{cases}$$

Figure 3.2.20: Equations for Determining Load Factor
Source: Shigley's Mechanical Engineering Design

The third Marin factor is the loading factor (k_c) which accounts for the changes in the endurance limit for different types of fatigue loadings. If a component experiences multiple loads, then

the largest of the loading factors are chosen. The choice of loading factor is provided by the following diagram. For our assumption, the set screw only experiences bending loads, though the threads will experience shearing as a result of the screw bending

The fourth Marin factor is the temperature factor (k_d) since many high cycle loading applications occur at high temperatures. At higher temperatures, the material might weaken, so we need to factor this into the fatigue loading using the following equation. We will assume our maximum operating temperature of 350°F for the worst-case scenario, however, these temperatures will also go below room temperature given the nature of the refrigerants we are using.

Equation 16:

$$k_d = 0.975 + (0.432 * 10^{-3})T_F - (0.115 * 10^{-5})T_F^2 + (0.104 * 10^{-8})T_F^3 - (0.595 * 10^{-12})T_F^4$$

k_d = temperature factor

T_F = operating temperature (in degrees Fahrenheit)

The fifth Marin factor is the reliability factor (k_e) which accounts for the standard deviations of endurance strength based on these empirical results. This measures the

Reliability, %	Transformation Variate z_o	Reliability Factor k_e
50	0	1.000
90	1.288	0.897
95	1.645	0.868
99	2.326	0.814
99.9	3.091	0.753
99.99	3.719	0.702
99.999	4.265	0.659
99.9999	4.753	0.620

Figure 3.2.21: Statistical Data for Determining Reliability Factor
Source: Shigley's Mechanical Engineering Design

probability that the material will meet or exceed the expected

endurance strength based on the z-score for reliability, which can be seen in Figure 3.2.19. For our case, we will assume a reliability factor of 99% for safety purposes.

The final Marin factor is the miscellaneous factor (k_f) which accounts for all the other specific factors of the material like corrosion, residual stresses, cyclic frequency, etc. For our purposes, we will assume this Marin factor to be 1 since further testing is required for this component.

Using the listed Marin Factors, we can determine the endurance limit of the set screw using the following equation.

$$\text{Equation 17: } S_e = \left(\frac{S_{ut}}{2}\right) k_a k_b k_c k_d k_e k_f$$

Fatigue factor of safety is employed when a system experiences fluctuating stresses. The easiest way to model these fluctuating stresses is with a sine wave. Given that the set screw is meant to keep the stopper from falling out of the assembly, the maximum stress it will experience is due to the maximum loading (12 lbf) from the stepper motor. The minimum stress it will experience is zero, since the screw will experience no load when the wire mechanism is not tensioned. With these two stresses, we can obtain the midrange stress (σ_m) and the alternating stress (σ_a) which correspond to the mean and the amplitude of the sine wave respectively. The equation for any shaft undergoing fluctuating loads is modeled below using the ASME elliptic standard for fatigue factor of safety.

Equation 18:
$$d = \left\{ \frac{16n_f}{\pi} \left[4\left(\frac{K_f M_a}{S_e}\right)^2 + 3\left(\frac{K_{fs} T_a}{S_e}\right)^2 + 4\left(\frac{K_f M_m}{S_y}\right)^2 + 3\left(\frac{K_{fs} T_m}{S_y}\right)^2 \right]^{1/2} \right\}$$

n_f = fatigue factor of safety (chosen to be 4 for our standards)

K_f = stress concentration factor (see Figure 3.2.20) (negligible since no fillets or chamfers)

K_{fs} = stress concentration factor (see Figure 3.2.20) (negligible since no fillets or chamfers)

M_a = alternating moment loading

M_m = mean moment loading

T_a = alternating torsional loading (assume negligible torsional effects from bolt preloading)

T_m = mean moment loading (assume negligible torsional effects from bolt preloading)

S_e = endurance limit

S_y = yield strength

Given that the size factor relies on the diameter of the shaft, this becomes a polynomial equation, which can be solved numerically. From these calculations, we determine that a screw with a root diameter of 0.1386" is necessary for our requirements. Sizing up to the nearest ANSI inch standard, the best screw for our purposes is an 8-32 set screw, which has a minimum (root) diameter of 0.139".

	Bending	Torsional	Axial
Shoulder fillet—sharp ($r/d = 0.02$)	2.7	2.2	3.0
Shoulder fillet—well rounded ($r/d = 0.1$)	1.7	1.5	1.9
End-mill keyseat ($r/d = 0.02$)	2.14	3.0	—
Sled runner keyseat	1.7	—	—
Retaining ring groove	5.0	3.0	5.0

Figure 3.2.22: Stress Concentration Factors for Shafts with Fillets or Chamfers
Source: Shigley's Mechanical Engineering Design

Chapter 3.3: Actuator

When choosing the actuator for this project, we had to keep in mind the requirements specific for the actuator given by the sponsor, which are: to be able to fully open and fully close valve in under 1 second, have a response time of under 10ms, limited to 24V and be able to withstand temperatures from 0°C to 40°C. There were several options that were considered for the actuator of the fluid control valve. The first one considered was a solenoid actuator that is able to pull with a force of 21 lbf. Unfortunately, we were unable to find a solenoid actuator strong enough to pull that much force. The second option we considered was a servo motor, however, they lack holding torque capabilities. The last option considered and the one chosen was a stepper motor. While larger in size, the model we chose provides an FOS = 2 torque to open and close the valve. The wire is terminated at the motor wheel by a crimped end, visible in Figure 3.3.1. The front plate interfaces with the motor wheel with a stopper hanging down from the top. It also serves as an attachment point for the standoff that the barrel adjuster screws into. This standoff aligns the barrel adjuster, which serves as the sheath end terminal, to the motor wheel hole at 0% actuation, which serves as the wire rope termination point.

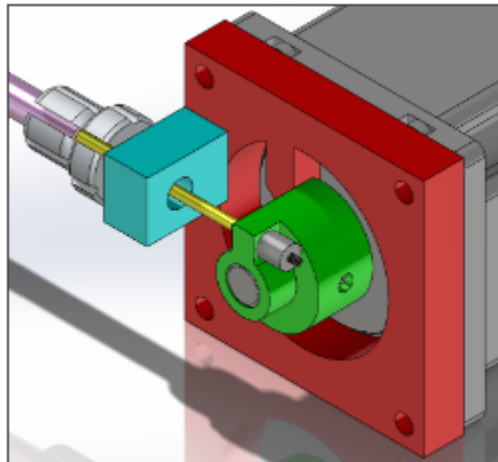


Figure 3.3.1: Isometric view of final CAD actuator design

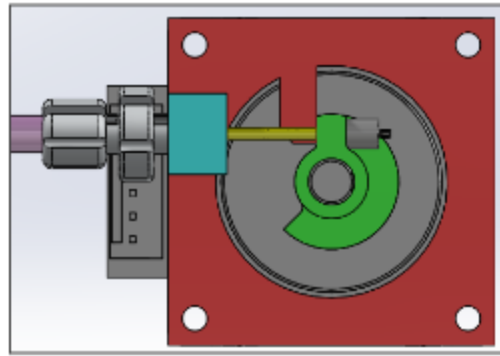


Figure 3.3.2: Front view of final CAD actuator design

The maximum displacement of the plunger will be 0.11811 in, and the distance between the wire and the center of the shaft is 0.25 in. The holding force used for the system is 21 lbf and the friction force in consideration is 3.5 lbf. In order to choose the right stepper motor, an analysis was carried out with these parameters.

Equation 19: $Torque = Force * Radius$

$$T_h = (F_h R) / 0.0625 = 84 \text{ ozf} - \text{in}$$

T_h = holding torque

F_h = holding force

R = radius

$$T_f = (F_f R) / 0.0625 = 14 \text{ ozf} - \text{in}$$

T_f = friction torque

F_f = friction force

R = radius

$$T_s = T_h + T_f = 98 \text{ ozf} - \text{in}$$

T_s = system torque

T_h = holding torque

T_f = friction torque

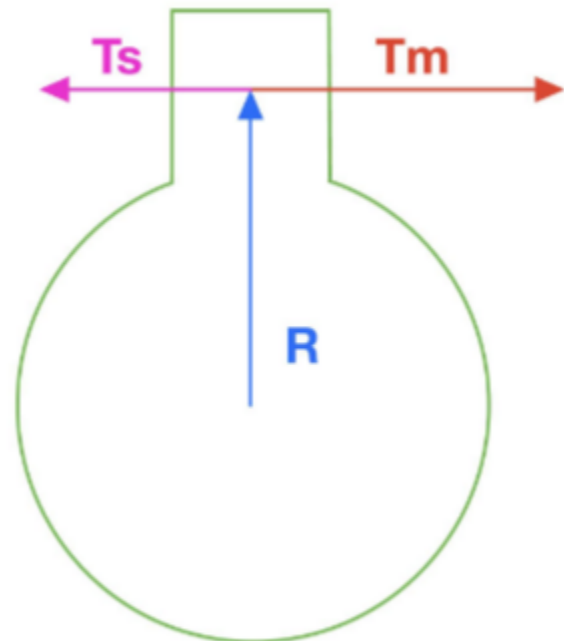


Figure 3.3.3: Free body diagram of spool

The desired factor of safety for the stepper motor was 2 to account for any loss motion since the geometry in which the cable will be subjected to is unpredictable.. A strong enough

stepper motor was chosen to satisfy this requirement, which was SCA5618M4204-A – STEPPER MOTOR – NEMA 23 from Nanotec Plug & Drive. The holding torque of this model is 198.26 ozf-in and it has a step size of 1.8°. To calculate the factor of safety and steps required to open the valve we used

$$FOS = T_m/T_s = 2.0231$$

FOS = factor of safety

T_m = motor torque

T_s = system torque

$$\theta = (d_{max}/R) * 180/\pi = 27.0688^\circ$$

Θ = angle (in degrees)

d_{max} = maximum displacement

R = radius

$$steps = \theta/1.8^\circ = 15.0382$$



Figure 3.3.4: Nanotec Plug & Drive

SCA5618M4204-A – STEPPER MOTOR – NEMA 23

With these numbers, we had a baseline and generated an arduino code to control the stepper motor accordingly. The code controls the stepper motor using an arduino board. It allows us to specify a desired motor position through the serial monitor, and the motor moves accordingly. There is a step array that holds positions for different percentage openings of the motor from 0% to 100% open.

Chapter 4.0: Prototype Performance

Chapter 4.1: Performance Testing Overview & Results

Testing a prototyped flow valve is crucial for several reasons. It allows engineers to verify the design and ensure that the valve performs as intended and meets the specified requirements. Secondly, testing evaluates the valve's performance under different operating conditions, identifying potential issues and ensuring it functions reliably. Thirdly, safety considerations are addressed through testing, identifying any risks or failure modes that could pose hazards in critical systems. Additionally, testing provides feedback for design optimization and iteration, improving performance, reliability, and safety. It also ensures compliance with industry standards and regulations. In summary, testing plays a vital role in validating the design, assessing performance and reliability, ensuring safety, optimizing functionality, and meeting industry requirements for a prototyped flow valve.

The tests stand validation that the prototypes valve is subjected to is used to validate various criteria such as:

1. **Flow Rate:** Testing verifies the actual flow rate achieved by the valve under different operating conditions. It ensures that the valve can effectively regulate and control the desired flow rates within the specified range.
2. **Pressure Drop:** Testing helps assess the pressure drop across the valve during flow regulation. It ensures that the valve maintains acceptable pressure differentials while allowing the desired flow rates.
3. **Leakage:** Testing can detect and quantify any leakage through the valve when it is in the closed or fully closed position. It ensures that the valve provides effective shut-off and minimizes any undesired flow when not intended.
4. **Control Accuracy:** Testing evaluates the valve's ability to accurately and precisely control the flow rate within a specified range. It assesses the valve's responsiveness and control characteristics, ensuring that it maintains the desired flow conditions as intended.
5. **Response Time:** Testing measures the time it takes for the valve to reach a stable flow rate after adjustments to the control mechanism. It ensures that the valve responds quickly and efficiently to changes in control inputs or system requirements.
6. **Turndown Ratio:** Testing examines the valve's ability to handle a wide range of flow rates. It evaluates the valve's turndown ratio, which is the ratio between the maximum and minimum flow rates it can effectively regulate.

7. Cavitation and Choking: Testing helps identify the occurrence of cavitation or choking phenomena, which can negatively impact the valve's performance. It ensures that the valve design and material selection are suitable to mitigate these issues under specified operating conditions.
8. Reliability and Durability: Testing assesses the valve's reliability and durability under various operating conditions, such as pressure, temperature, or corrosive environments. It ensures that the valve can withstand these conditions without degradation or failure over its intended lifespan.

In order to ensure that all these tests were conducted, and the most accurate data was collected after the fact, a custom testing rig was built with various components, including a pressure gauge, a flow meter, and obstacles, to simulate testing conditions. The importance of each of these components in the overarching testing of our flow valve and to validate the expected data from our design is as follows:

1. Pressure Gauge: A pressure gauge is an indispensable tool for measuring the pressure exerted by the fluid passing through the valve. It allowed us to monitor the valve's response under different pressure levels, providing insights into its functionality and performance across a range of operating conditions. By accurately measuring the pressure, we identified any potential issues such as leaks, pressure drops, or excessive pressure, which helped us in optimizing the valve design and ensuring its safe operation. The leaks across the system were corrected with the help of teflon tape and custom made brass fittings to counter the loss in pressure that the leaks in the system cause.
2. Flow Meter: A flow meter is used to measure the volumetric flow rate of the fluid passing through the valve. It enabled us to quantify the amount of fluid flowing per unit of time, allowing for the assessment of the valve's capacity to handle specific flow rates. By measuring the flow rate accurately, we determine whether the valve is delivering the expected flow and identify any deviations or restrictions that might affect its performance. This information was crucial for validating our valve's internal structural geometry and ensuring optimal flow characteristics per the requirements of our sponsor.
3. Obstacles: Incorporating obstacles within the testing rig simulated real-world conditions that the flow valve may encounter during operation. These obstacles included bends, elbows, filters, or other types of restrictions that mimic the flow path in an actual system and enabled us to validate our actuation system, which, as has been mentioned above, is the Bowden cable setup. By introducing these obstacles, we

assessed the valve's ability to handle changes in flow direction, negotiate through narrow passages, and cope with flow disturbances. The presence of obstacles allows for a comprehensive evaluation of the valve's performance under realistic conditions, providing valuable insights into its efficiency, pressure recovery, and flow stability.

Selecting the right pressure gauge for testing purposes involved considering specific requirements such as the desired pressure range, compatibility with the fluids being tested, and cost-effectiveness. In this scenario, where a pressure gauge was needed to test both water and compressed air up to at least 70 psi, while also keeping the cost in mind, a suitable gauge needed to be chosen. While the final tests did not implement any compressed dry air testing, the pressure gauge source was sought after keeping that requirement in mind. Therefore, the pressure gauge sourced was one that was compatible with water, and hence would be compatible with air. The opposite however would not have been true because of the difference in their physical properties.

Water is denser and less compressible than air. This means that the force exerted by a given pressure of water is typically higher than the force exerted by the same pressure of air. Pressure gauges designed specifically for air may not have a sensing element that is capable of withstanding the higher forces exerted by water. The gauge may not be calibrated to accurately measure the higher pressures associated with water, or the materials used in the gauge may not be resistant to corrosion or damage caused by water. Considering the cost factor, it is possible to find pressure gauges that meet the specified requirements without being excessively expensive. Exploring different brands, models, and suppliers can help identify cost-effective options without compromising on quality and accuracy.

Considering the cost factor, it was completely possible to find pressure gauges that met the specified requirements without being excessively expensive. Exploring different brands, models, and suppliers helped identify cost-effective options without compromising on quality and accuracy.

Checking for features like corrosion resistance, appropriate connection types (such as NPT or BSP), and the ability to withstand the conditions of the testing environment further ensured a suitable choice.

Flow Meter Selection: *insert picture of flowmeter and the amazon link as well”

When searching for a flow meter that could work with both water and air while having electrical outputs, it was challenging to find specific products that met all the desired criteria on platforms like Amazon.

In such cases, the solution we came up with was to use a flow meter that provides mechanical outputs, such as a pulse output or a mechanical rotor, and then integrate it with an additional component like an Arduino microcontroller to convert the mechanical signal into an electrical output that can be processed and recorded by a data acquisition system.

The process typically involved connecting the mechanical output of the flow meter to the appropriate input pins of the Arduino. For example, if the flow meter provides a pulse output, it can be connected to one of the digital input pins of the Arduino. Using the Arduino programming language, the code can be written to count the number of pulses within a specific time frame, allowing for the calculation of the flow rate based on the known characteristics of the flow meter.

By utilizing an Arduino, it became possible to interface the flow meter with the data acquisition system and obtain the flow rate readings in an electrical format that was further processed, recorded, and transmitted for analysis or monitoring.

This approach enabled the conversion of the mechanical signal into an electrical output compatible with the data acquisition system, allowing for flow rate monitoring and analysis.

A custom brass fitting was created to connect our aluminum valve housing with M8 threading to a standard NPT (National Pipe Thread) threading. This presented a rather unique challenge as off-the-shelf fittings for this specific connection are not available on sites like McMaster or Amazon. The reason behind this is that M8 threading is part of the metric thread size standard, while NPT threading is an American taper thread standard primarily used for pipe fittings. These two threading standards have different thread profiles, pitches, and tolerances, making it difficult to find a direct fitting that transitions from M8 to NPT.

To overcome this limitation, a custom brass fitting was crafted to accommodate the transition between M8 threading and NPT threading. The process involved several steps. Firstly, a suitable brass material compatible with both aluminum and NPT threading was selected.

Next, the dimensions of the M8 threading on the valve body and the specifications of the desired NPT threading are measured, these measurements are the outer diameter of the threads on the aluminum valve.

Based on these measurements, a custom design was created for the brass fitting. Machining processes such as turning, threading, and milling were then employed to shape the brass material into the desired fitting, adhering to the design specifications.

The M8 threading on one end of the brass fitting was machined or cut to match the threading on the valve body. The thread pitch and dimensions are replicated accurately to ensure a proper fit.

On the other end of the brass fitting, NPT threading was created using appropriate threading tools like taps and dies, and by machining the threads. It was crucial to conform to the NPT standard and ensure compatibility with the desired pipe size.

After machining the custom brass fitting, thorough cleaning was conducted to remove any metal debris or contaminants. This step ensured a clean and secure connection.

To assemble the custom fitting, it was screwed tightly onto the M8 threading of the valve body. Then, the NPT fitting is attached to the threaded end of the custom fitting.

The connection was checked for tightness and proper alignment. Testing was performed to verify a leak-free and robust connection between the valve body and the NPT fitting.

The tests we ran to validate our valve before we did a full cycle test were as follows:

1. Preliminary Leak Check: The preliminary leak check is vital for safety, reliability, efficiency, compliance, and preventive maintenance. It ensures the safety of the system by identifying and addressing leaks, enhances reliability by verifying proper operation, saves resources by preventing costly repairs, ensures compliance with regulations, and serves as preventive maintenance by detecting weaknesses early on. Conducting this check established a solid foundation for subsequent testing and operation, optimizing performance and minimizing risks.

To conduct a preliminary leak check for the valve, we started by setting up the water source to deliver 70 psi pressure. We installed a pressure gauge capable of measuring up to 70 psi and connected it to the system. The aluminum valve was connected to the pressure gauge and to activate the actuation system, someone from Team 26 hand pulled the Bowden cable in order to ensure an output flow from our valve. Gradually,

the water pressure was increased while monitoring the gauge and visually inspecting for leaks. We verified leakage-free operation by observing all components and using a suitable leak detection method, and if needed, readjusted some of the fittings, added teflon tape, and switched out certain fittings for newer ones/ones we thought might work out better with the system. This process ensured the integrity of the valve assembly before further testing or operation was conducted with the whole system.

2. Flowmeter Testing: We started by connecting the flowmeter to the arduino and generating a code that will display the flow rate going through the device. To make sure the circuit would work, we checked the voltage of all the pins and tested with different water sources. Once a reliable water source was found, we were able to calculate the calibration factor by manually measuring the flow per minute.
3. Stepper Motor Testing: Once the stepper motor arrived, we included it in a circuit that will allow us to control it using an arduino board. The pins needed to be paired correctly in order to have the shaft rotate correctly. During the first try, the stepper motor did not run. We needed to adjust the current limit on the driver to match the NEMA 23. Once the circuit was done, we tried running a simple code that would test the motor rotating back and forth. With the full system assembled, we could start opening the valve using the stepper motor. Ideally, the number of steps needed to open the valve would be 15 steps, but we needed to run tests to find the right amount of steps accounting for lost motion due to the bends in the system.

Overall Testing Procedure:

The testing rig encompassed various components that worked together to facilitate accurate testing and measurement. The setup began with a connection to a high-pressure water source, supplying pressurized water for testing purposes. This water source was integrated into the system and connected to a pressure gauge, allowing for precise monitoring and measurement of the applied pressure.

Continuing along the testing rig, the pressurized water flows into a flow meter. The flow meter serves as a vital instrument for measuring the rate of water flow through the system. By capturing flow data, it enables the evaluation of performance and efficiency.

To facilitate data acquisition and control, an Arduino microcontroller is incorporated into the setup. The Arduino receives inputs from the pressure gauge and flow meter, processing and

analyzing the data. This allows for real-time monitoring of pressure and flow parameters, as well as enabling control functionalities.

Moving closer to the valve housing, the front side of the valve assembly is connected to the output of the flow meter. This connection ensures that the flow of pressurized water is channeled through the valve, providing a controlled environment for testing purposes. The valve housing is designed to accommodate the required pressure and flow specifications. At the rear end of the valve housing, a Bowden cable is attached. This Bowden cable serves as the actuation mechanism for the valve. It connects to a stepper motor, which allows for precise and controlled actuation of the valve. By actuating the valve, the Bowden cable enables the adjustment of flow conditions and provides control over the fluid flow rate.

Together, the integrated components of the testing rig create a comprehensive setup for testing and measurement. The connection to a high-pressure water source, coupled with the pressure gauge and flow meter, ensures accurate monitoring of pressure and flow parameters. The Arduino microcontroller facilitates data acquisition, analysis, and control. The valve housing, driven by the Bowden cable actuated by a stepper motor, allows for precise control over the fluid flow conditions. This testing rig provides a versatile platform for conducting experiments, analyzing performance, and gathering valuable data for evaluation and optimization.

Chapter 5.0: Design Recommendations, Conclusions, and Supplementary Statements

Chapter 5.1: Future Design Recommendations

- Further study of aluminum's long-term chemical compatibility with HFE 7100/7500 and Opteon SF10.
- Further study of fluorosilicone o-ring chemical compatibility with Opteon SF10
- Utilizing 304 stainless steel for valve housing for improved durability and corrosion resistance.
- Characterization of lost motion using chosen cable components via repeated testing at different lengths and total bend angles.
- Implementation of higher resolution flow meter.
- Implementation of higher resolution stepper motor.

Chapter 5.2: Conclusions

- The Bowden cable is a viable solution to the necessity of separating valve and actuator.
- The internal valve geometry and seal designs were successful in delivering a leak free actuation.
- Care must be taken to select quality parts, especially in terms of electronics.

Chapter 5.3: Applicable Standards

One commonly used standard that was referred to for valve body material selection was the American Society of Mechanical Engineers (ASME) standard B16.34. This standard provides guidelines for the selection of materials for valve bodies, bonnets, and trim components based on the intended service conditions, pressure-temperature ratings, and corrosion resistance requirements. To make sure that the valve material is appropriate for the given service conditions, Team 26 looked at the various grades of materials specified in the standard, such as carbon steel, stainless steel, and alloy steel.

Another important standard that was considered for valve body material selection was the American Petroleum Institute (API) specification 6D. This standard provides guidelines for the design, manufacture, and testing of pipeline valves for the petroleum and natural gas industries. The standard specifies the requirements for valve body materials, such as carbon steel, low-alloy steel, and austenitic stainless steel, that are suitable for use in pipeline applications.

Besides these industry-specific standards, international standards, such as those developed by the International Organization for Standardization (ISO), for valve body material selection were also referenced. For instance, ISO 15761 provides guidelines for the selection of metallic materials for valves, and ISO 14313 specifies requirements for pipeline valves suitable for petroleum and natural gas industries.

Another applicable standard we use is Aerospace Standard AS568 which defines standardized sizes for O-Ring seals. This allows us to define geometries compatible with the o-rings selected.

Chapter 5.4: Impact on Society

Semiconductors have become ubiquitous in modern industrial settings, finding usage in a wide range of applications. They power everything from basic household appliances to high-performance cars and sophisticated computer systems. When designing any product, engineers must take into account a multitude of factors, including public health and safety, environmental impact, cultural and social considerations, and global economic concerns. In light of these factors, it is essential that the flow valve in question meets the most stringent standards.

When designing this product, the primary focus needs to be on safety. The refrigerant fluid used in the product can pose a significant danger if not handled properly, making it crucial to incorporate safety features into the valve's design. For instance, the valve should be designed to prevent any leaks or spills and should be straightforward to operate and maintain. Additionally, it is essential to consider the corrosive nature of refrigerant fluid and its potential environmental impact. Therefore, the valve should also be designed to be environmentally friendly.

It is crucial to take into account the significant impact that the valve's failure could have on the semiconductor industry. Precise testing of chips is a vital aspect of this industry, and any malfunction of the valve could potentially lead to production delays or even defects in the chips themselves. Such issues could negatively affect the performance and dependability of electronic devices that rely on these chips, ultimately having adverse consequences on society as a whole.

Another critical consideration is the impact that the valve's failure could have on industries heavily reliant on electronic devices, such as aerospace or medical fields. In such cases, a malfunctioning valve could have severe implications for public health and safety. For instance, if a medical device utilizes a chip that wasn't adequately tested due to valve failure, it could potentially result in severe health complications for patients.

Last but not least, it is crucial to consider the environmental implications of a valve that undergoes frequent demagnetization and necessitates regular replacement. Such a scenario could contribute to heightened waste and pollution levels, potentially having adverse effects on both the local environment and the global issue of climate change.

In summary, the consequences of a flow valve that is constantly demagnetized as a result of exposure to high temperatures would vary depending on the specific application of the valve.

However, it is evident that such a scenario could have considerable adverse effects on society, especially in fields where dependable electronic devices are critical for public health and safety.

Chapter 5.5: Professional Responsibility

In designing our solution, we have an obligation to deliver a product that is designed to last. Our sponsor, Cohu, has a reputation of producing quality products. Additionally, producing short lived components is an action in bad faith to the environment. Therefore, we have an obligation to produce a design that lasts through repeated use and to verify these expectations through experimental evaluation.

We also have a responsibility to make our design as affordable as it can be, both in consideration of Cohu, who must manufacture it, and the consumer, who must purchase it.

Our project also contains a Teflon component that is intended to operate at a high temperature. Since Teflon can produce toxic fumes when overheated, we are also obligated to make sure that the temperatures at which our system will operate will be safe and not pose health risks to the consumer that purchases Cohu's system.

Acknowledgements

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Thank you to the MAE 156B instructional team, including the dedicated professors, teaching assistants, and staff at the MAE Machine Center.

Thank you to our peers in MAE 156B for their thoughtful feedback.

References

Team 26 Website

https://sites.google.com/d/1FQOPGKGd_Hkzgd7VZSNnJmM_Vw1uE6Bj/p/1QFB5LLXsNVKmayJ8nLgPji8M7H1Oxs2f/edit

HFE 7100 Datasheet

<https://multimedia.3m.com/mws/media/1998180/3m-novec-7100-engineered-fluid.pdf>

SF 10 Datasheet

<https://www.opteon.com/en/-/media/files/opteon/opteon-sf10-technical-information.pdf?la=en&rev=92043d889d94408e9c875589cbb77642>

<https://frasertech.co.uk/wp-content/uploads/2018/05/opteon-sf10-heat-transfer-technical-information2.pdf>

CarlStahl TechnoCables

<https://www.carlstahl-technocables.com/gb/en/>

Cable Manufacturing and Assembly Co.

<https://www.wire-rope.org/cable-manufacturing-assembly-co-inc/>

Nanotec Plug & Drive

<https://us.nanotec.com/products/2766-sca5618m4204-a>

Appendix

Appendix A: Project Management

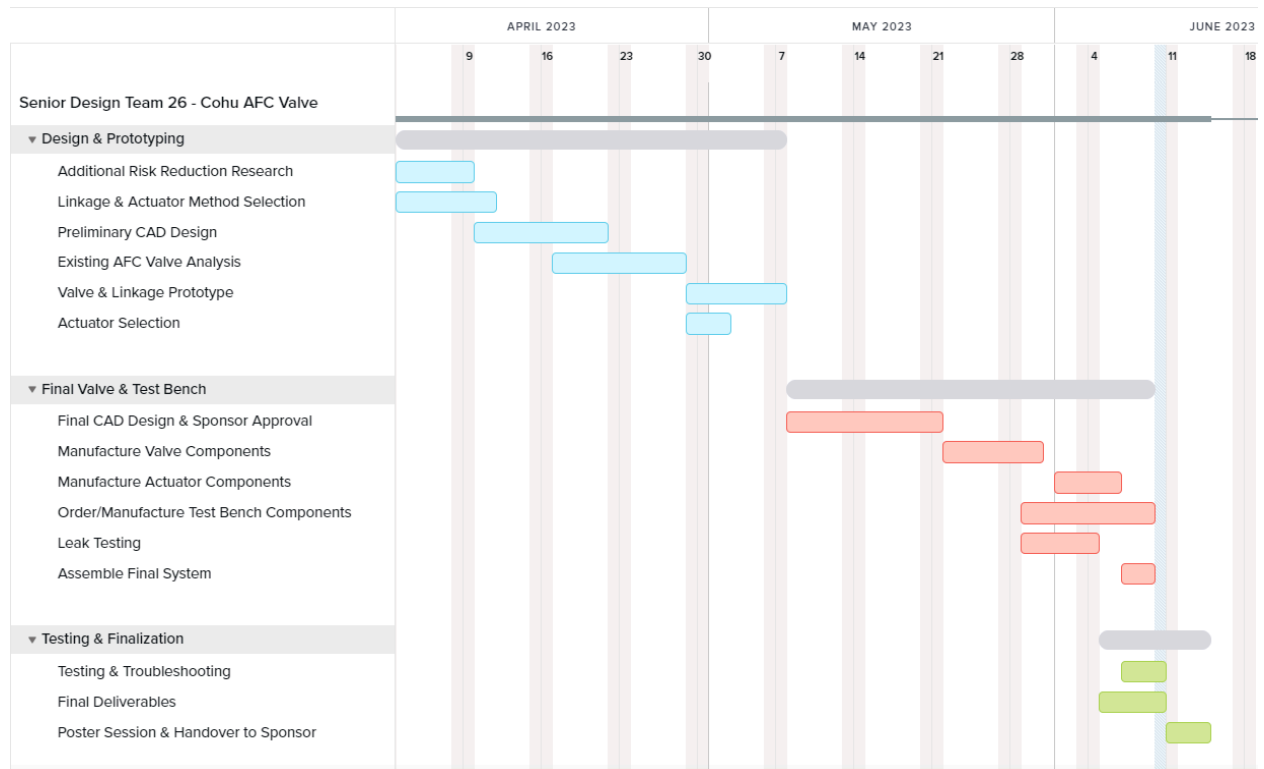
A1: Task Distribution

- Bora Gursel - Valve Housing Design, Manufacture. Motor Assembly Manufacturing. Financial Coordinator.
- Manoel Aguirre Lara - Actuator System Research, Design, Assembly, Programming. Test Bench Design and Assembly. Website Development.
- Matthew Cordova - Mechanical Linkage Research and Assembly. Flowmeter and Actuator Programming and Assembly
- Michael Gigoux - Mechanical Linkage Design, Research, Manufacturing, and Assembly
- Anannaya Vij - Fluid Systems Integration. Materials Research

A2: Risk Reduction Effort

- Bora Gursel - Valve Housing
- Manoel Aguirre Lara - Actuator Research
- Matthew Cordova - Mechanical Linkage Research
- Michael Gigoux - Mechanical Linkage Research
- Anannaya Vij - Materials Research

A3: Intermediate Deadlines & Gantt Chart



Appendix B: Individual Component Reports

Individual Component Reports are available in [this folder](#).

Individual Component Report #1: Valve Housing

By Bora Gursel

Section 1: Background

The spring assembly is a key component in the flow valve that regulates the flow rate of refrigerant fluid through the valve. As the PWM duty cycle of the solenoid coils is increased, the magnetic force generated by the coils increases. This magnetic force produces a backward force on the armature (shown in figure 2), which pulls the inner valve that compresses the spring assembly. As the springs are compressed, the inner valve is retracted and flow is permitted through the flow valve. A pulse width modulated duty cycle in the coils produces a variable force on the armature which controls the flow rate.

A key requirement for the spring is to maintain a default closed state, necessitating a resting position of the spring such that the spring force is greater than the pressure force created by the fluid at 70 psi. It must also fit within the enclosure volume of the valve base, corresponding to a bore diameter of approximately 16mm with a 4mm length. The spring must also be hollow, able to accommodate an 11.2mm inner clearance diameter. With a secondary objective to minimize cost, it is desirable to purchase a commercial off the shelf (COTS) spring.



Figure 1: Flow Valve.
Front View

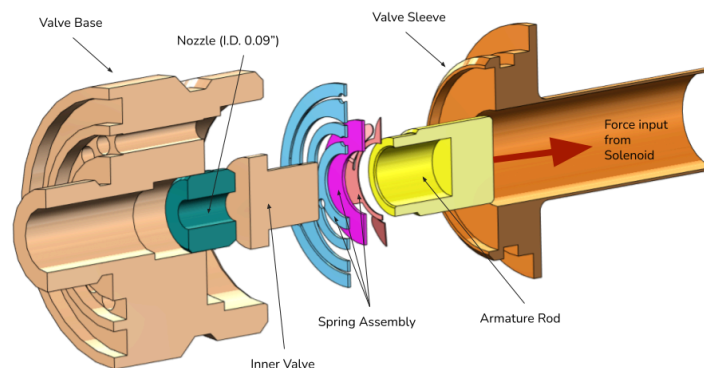


Figure 2: Flow Valve Model. Annotated view of exploded valve-spring assembly section. Source: Cohu Inc.

Section 2: Summary of Key Findings and Relevant Applications

Compression springs are a commonly used mechanical component in industrial design, and as such have wide ranging variances in design specifications. Varying material choice and geometry will affect spring characteristics such as spring load and spring rate. The spring rate is the amount of force needed to compress a spring by a certain distance, typically expressed in pounds per inch or newtons per mm². This is also commonly expressed as the spring constant, K . This is determined by the shear modulus of the spring material and the specific geometric characteristics of the spring such as the number of coils and the diameter of the coil wire. A stiffer material, such as steel, is a better choice for a spring designed for large forces. The spring load is the force produced by the spring as a function of spring displacement, typically given by the following formula:

$$\text{Equation 1: } F_{\text{spring}} = K \cdot \Delta x$$

Note that Equation 1 is only valid for linear springs, or springs for which the spring load is proportional to spring displacement. Nonlinear springs do not follow this relationship. Although less commonly used than linear springs, nonlinear springs can be very useful in a wide variety of applications.

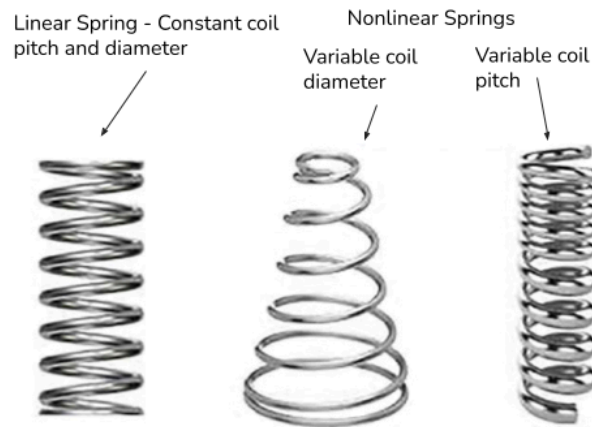


Figure 3: Nonlinear vs. Linear Springs.²

In addition to traditional coil springs, it is possible to produce machined springs with unique geometries and high tolerances. Some sources claim that machined springs are capable of achieving a nearly infinite life cycle without experiencing permanent deformations that undermine the stated spring performance. The tradeoff for these machined springs is a higher cost.

In general, springs represent the most predictable and cost effective way to meet the design requirements posed earlier. Linear springs can be easily modeled to predict valve behavior and can be paired with a solenoid actuator to easily implement increment displacements for valve flow.

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² <https://www.collinsdictionary.com/us/dictionary/english/spring-rate>

³ <https://www.motioncontroltips.com/what-are-linear-springs-and-how-do-they-work/>

Section 3: Discussion of Information Gathering Approach

Search Keywords:

- “Nonlinear springs”, “Spring Rate Calculation”, “Low temperature compatible materials”, “Machined springs”, “Largest U.S. Industrial Spring Manufacturers”, “Low Form Factor Spring”

List of Articles, Design Guides, and Webpages

- <https://www.collinsdictionary.com/us/dictionary/english/spring-rate> - definitions for key spring characteristics
- <https://www.designworldonline.com/wire-springs-or-machined-springs/> - Overview of Machined vs. Wire Springs
- <https://www.smalley.com/wave-springs> - Spring Vendor (recommended by sponsor)
- <https://www.ctspring.com/> - Spring Vendor
- <https://www.rlspring.com/services/products/> - Spring Vendor
- <https://trc.nist.gov/cryogenics/materials/materialproperties.htm> - National institute of Standards on cryogenic compatible materials. (also applicable for low temperatures with temperature dependent material property curves)
- <https://www.opteon.com/en/-/media/files/opteon/opteon-sf10-heat-transfer-bulletin.pdf?la=en&rev=945712c6c20e4f10bb9f5550754c607b#:~:text=Opteon%E2%84%A2%20SF10%20specialty%20fluid%20is%20compatible%20with%20most%20metals.and%20aluminum%20showed%20good%20stability> - Opteon refrigerant datasheet. See material compatibility
- <https://www.opteon.com/en/-/media/files/opteon/opteon-sf10-heat-transfer-bulletin.pdf?la=en&rev=945712c6c20e4f10bb9f5550754c607b> - Opteon refrigerant heat transfer bulletin
- <https://multimedia.3m.com/mws/media/1463476O/3m-novec-engineered-fluids-white-paper-for-thermal-management-fv-pdf.pdf> - 3M HFE 7100 / 7500 fluid material compatibility datasheet.
- <http://www.kvastainless.com/17-7.html> - Information on 17-7 PH Stainless Steel

Summary of Phone Calls & Emails

- Phone call with Smalley sales representative to receive verbal quote on low form factor spring SSB-0063-S17. This spring costs \$25 for a batch of four springs. The spring is currently out of stock, with a backorder time of 28 days. Advised to order earlier than needed.
- Phone call with Smalley engineering regarding properties of SSB-0063-S17 spring related to maximum allowable force. Discovered that exceeding maximum allowable force will lead to spring damage. May need to explore other springs.
- Email with Sponsors regarding spring characterization. Established key performance criteria for spring behavior - related to holding force needed to resist fluid expansion. For 70 psi flow behavior, spring must resist opening with an F.O.S. = 4.

Remaining Knowledge Gaps

- Research COTS valve housing options for spring compatibility.

Section 4: Detailed Description of Findings

The following section will discuss materials that are acceptable for use in the spring, the desired spring behavior and characteristics, and an overview for vendors with acceptable options.

Material Compatibility in Springs

Based on the design requirements, it is expected that the springs may come into contact with refrigerant fluids and other potentially corrosive elements. The refrigerants specified by the sponsor are: 3M HFE 7100, 3M HFE 7500, Opteon SF10, and water. Furthermore, these fluids are expected to reach temperatures between -80°C to 100°C , while environmental temperatures may vary between -40°C to 175°C .

The datasheet for Opteon SF10 states that the refrigerant shows stability with most metals, including stainless steels. In general, the fluid is stated to not oxidize or degrade during storage, indicating that it is unlikely to pose a corrosive threat to the spring. The datasheet for 3M's Novec fluid products, including HFE 7100 and HFE 7500, indicates that they are compatible with most metals, including stainless steels. Specifically, the datasheet calls out 302 stainless steel. Stainless steels are also highly resistant to corrosion caused by water, unlike other alloy steels and metals. They are a preferable choice for corrosion properties.

In addition to analyzing material stability in contact with fluids, this case study also examines material behavior at low temperatures, reaching -80°C . The mechanical properties of metals change as they reach low temperatures. Certain low-carbon steels experience a ductile to brittle transition that significantly undermines resistance to crack propagation in those materials and leaves them susceptible to a brittle failure. This transition is not particularly extreme for austenitic steels such as 304 stainless steel. 17-7 steel is noted to behave as a semi-austenitic behavior. It is necessary to continue a literature review to determine its suitability.

In general, the findings of this study suggest that corrosion resistant steels are the preferred choice for the spring. The specific kind of stainless steel is best determined by vendor availability so as not to not impose unrealistic expectations for spring sourcing. The table below summarizes the findings of the literature review for two acceptable materials.

Material	Melting Point	Opteon SF10 Compatibility	3M HFE 7100 / 7500 Compatible	Low Temperature Compatible?	Spring available in material?
17-7 Stainless Steel ⁴	1400°C	High Likelihood	High Likelihood	Medium-High Likelihood	Yes - Smalley, CSS,
304 Stainless Steel ⁵	1400°C	High Likelihood	High Likelihood	Yes	No

Spring Behavior

The primary design factor dictating the spring load and spring rate is the “defaults to closed” requirement which requires the valve to remain shut off without any input from the solenoid. The inner valve must be able to stay closed against the force generated by the fluid pressure acting at 482,633 Pa (70 psi). The nominal nozzle opening has a diameter of 2.286mm (0.090 in.). Hence for a safety factor of 4, the spring force necessary can be calculated:

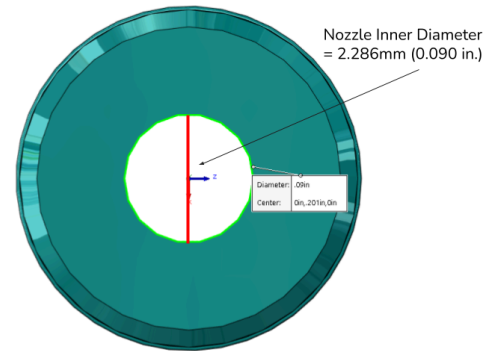


Figure 5: Flow Valve Nozzle Inner Diameter

$$Force = Pressure \times Area = (482,633) \times \pi(2.286/2 \times 10^{-3})^2 = 1.98N$$

$$Adjusted\ Force = Force \times F.O.S. = 1.98N \times 4 \approx 8N$$

The baseline loading for the spring must be able to accommodate approximately 8N of force. Subsequently, the spring must also be further compressed by the actuator mechanism, necessitating a maximum load greater than 8N. Further research must be completed to determine the upper limit for the necessary spring displacement and spring load.

⁴ <http://www.kvastainless.com/17-7.html>

⁵ <https://www.matweb.com/search/DataSheet.aspx?MatGUID=abc4415b0f8b490387e3c922237098da>

Spring Geometry and Key Vendors

A market search has identified several suitable vendors for springs. Smalley, Connecticut Spring & Stamping, and R&L Spring company are vendors that provide compression springs for valves.

Smalley stands out in particular for their proprietary unique low form factor wave springs. Figure 6 shows the Smalley SSB-0063-S17 wave spring which has desirable characteristics for this project. The material is 17-7 PH steel, which is compatible with the design requirements. The spring has a maximum load of 45N, in reasonable excess of the minimum load needed. The spring behaves linearly, with a spring rate equal to 65N/mm.



The spring dimensions are as follows:

- Outer diameter: 16mm
- Inner Clearance Diameter: 11.28mm
- Free Height: 2.29mm

Based on a phone inquiry, the spring cost was determined to be \$25 for 4 springs. The cost per spring scales down as additional springs are ordered which improves its basis for selection. The spring is currently on backorder with a lead time of 28 days.

Further research into CSS and R&L Spring Company determined that these companies produce springs to customer specifications, resulting in higher costs, ranging for approximately \$100 per batch of 8 springs, with additional cost savings at scale. At the current stage of the investigation, these vendors will not be considered to maximize commercial off the shelf purchasing.

In summary, further changes to spring selection will be needed as the valve housing is developed. The basis for spring selection outlined in this analysis will serve as a guideline for the design process as it evolves. The spring selected shall be available as a commercially off the shelf product to minimize cost. Any spring selected shall be an appropriate material for the refrigerant fluids while also maintaining an acceptable form factor that fits inside the valve housing.

Section 5: Appendix

*Marked up copies of material found including articles, specification sheets, and book pages.
Notes of phone calls and copies of email correspondence.*

Phone Calls & Emails:

- *Phone call with Smalley sales representative to receive verbal quote on low form factor spring SSB-0063-S17. This spring costs \$25 for a batch of four springs. The spring is currently out of stock, with a backorder time of 28 days. Advised to order earlier than needed.*
- *Phone call with Smalley engineering regarding properties of SSB-0063-S17 spring related to maximum allowable force. Discovered that exceeding maximum allowable force will lead to spring damage. May need to explore other springs.*
- *Email with Sponsors regarding spring characterization. Established key performance criteria for spring behavior - related to holding force needed to resist fluid expansion. For 70 psi flow behavior, spring must resist opening with an F.O.S. = 4.*

Individual Component Report #2: Bowden Cable as the Transmitting Linkage

By Matthew Cordova

Section 1: Background

The subject of my research is the main creative area of our project, the mechanical transmitting linkage. Due to this area's essential nature to our project and the relative simplicity of the project's other aspects, Michael Gigoux and myself will be addressing different research within this field. This mechanical linkage serves to transmit a mechanical signal from the solenoid actuator somewhere outside of the testing environment, to the flow valve, which is situated at the heart of the testing environment. This signal must be transmitted accurately and quickly so that the flow valve can responsively adjust the flow of refrigerant to the chip.

The mechanical transmitting linkage (henceforth referred to as "the linkage") has only a few concrete requirements. The components that make up the linkage must remain reliable in an environment ranging in temperature from -40°C to 175°C . The high temperature range eliminates the possibility of having electronics within the linkage itself. It must also separate the actuator and valve by a distance of up to 1 meter in a nonlinear path. This requirement is meant to simulate the use case of the flow valve system, as the geometry of the testing environment will vary by model of testing machine. The machine geometry will be varied, and as of yet is undecided. In order to maximize compatibility, the linkage must also be easily modifiable without requiring a redesign of the essential principle. This necessitates that the design scales easily independent of linkage length.

Due to the nature of these requirements, I have elected to conduct in depth research on the viability of a Bowden cable to serve as the linkage. Since the cable transmits a linear mechanical signal, it will easily interface with the solenoid actuator and flow valve, and it would be flexible to fit around any potential geometry.

Section 2: Summary of Findings

Throughout the course of my research, I learned about the functions, components, and material selection of the Bowden cable. Beginning first by looking at both the Bowden and push-pull cables, then narrowing my selection to only Bowden cables. Bowden cables consist of a specialized cable and fittings on each end depending on how motion needs to be transmitted. This specialized cable has an innermost cable used to transmit force typically made of stainless steel or another metal. This cable is housed in a polymer inner tube to reduce friction and wear as the steel cable slides through the inner tube. This inner tube is contained within a flexible, longitudinally incompressible, outer metal sheath that may have a

polymer coating on the outside to prevent corrosion and other forms of wear. The fitting on the pulling end of the cable attaches to an actuator that pulls the fitting and thereby the inner cable through the housing.

One major concern in the utilization of this mechanism for our project's design is the temperature constraint of -40°C to 175°C. Most polymers melt or deform plastically at high temperatures, severely limiting the selection of premade cables available for use to those utilizing Teflon or another high temperature polymer. Cables available as COTS through McMaster-Carr or other websites use materials intended for much lower temperatures, since high temperature applications are relatively specialized or uncommon. Secondly, choice of cable is heavily dependent on the independent analysis conducted by my teammates. Decisions regarding friction minimization versus cost can only be made once estimates of required travel distance, force from the solenoid actuator, and precision requirements have been determined. Despite these missing factors, I was able to find suppliers for different cable options, depending on the construction method chosen (purchase of a completed custom assembly vs. purchase of custom parts and self assembly).

Section 3: Information Gathering Approach

Since the emergence of the Bowden cable as a viable linkage option began as a comment about bicycle brake systems, I first had to find the correct terminology. This first involved a quick series of google searches to find the term Bowden cable and an alternative type of cable, the push-pull cable. After reading, I determined that a Bowden cable would be better for our applications than a push-pull cable. Once I had the correct terminology, I began searching to learn more about the different components in Bowden cables, what they are good for, and what materials are used to make them.

Here is a list of keywords I used in my search:

- Bicycle brake cable mechanical system
- Bowden cable
- Push-pull cable
- Bowden cable construction
- Teflon temperature properties
- Teflon bowden cable
- Bowden cable backlash

Here is a list of the articles, design guides, and other webpages I found:

- Research Articles:
 - [Analysis of Bowden cable Transmission performance for orthosis applications | IEEE Conference Publication](#)

- [Position control using adaptive backlash compensation for bowden cable transmission in soft wearable exoskeleton | IEEE Conference Publication](#)
- Design Guides:
 - [Push Pull Cable Controls - Industrial Quick Search](#)
 - [Mechanical Push Pull Cable Assemblies - Motion Control Technologies Inc.](#)
 - [Push-Pull Cable Design | Push-Pull Control Cable Assemblies - Tyler Madison Inc.](#)
- Other Webpages:
 - [Bowden cable - Wikipedia](#)
 - [Push Pull Cables |Throttle Control | Shift - Control Cables Inc.](#)
 - [Commercial Stainless Steel Cable | Engineering & Coating - Motion Control Technologies Inc.](#)
 - [Custom Bowden Cable Assembly | Stranded Wires - Tyler Madison Inc.](#)
 - [Bowden cable manufacturer Germany - Carl Stahl TECHNOCABLES](#)
 - [Spiral sheaths for bowden cables - Carl Stahl TECHNOCABLES](#)
 - [Coated Mechanical Cable - Sava*](#)
 - [Industrial PTFE Cable Liners - GREMCO GmbH**](#)
 - [FITCO® Liner Natural PTFE - GREMCO**](#)

In the course of research for this report, I contacted both Carl Stahl Sava Industries Inc. under the Other Webpages section above. The link to their website has been marked with an asterisk.

I made a phone call and after going through the operator spoke to inside sales representative, Dave Balister for about 7 minutes. Dave was able to inform me that while pricing for the cable can vary a lot depending on the construction and performance specifications, Sava would be able to provide a 25 ft sample free of charge, which would be sufficient length for our entire project. Dave was able to email me the data sheet for their available FEP coated cables, giving me access to data not available online. I asked if there were further specifications concerning temperature available besides the 400°F ballpark, but was informed that due to the variability of use cases, such testing was not conducted. However I can research the type of FEP used to determine its capabilities. I was not given a yes or no answer as to whether the cable would be suitable for my temperature requirement. Lastly, I asked about flexibility and bend angles, but was told that those also vary significantly by construction. A recreated transcript of the call is provided in the appendix.

Due to the concurrent nature of research on the other aspects of the project, concrete parameters for how a Bowden cable would need to interface with the solenoid actuator and flow valve were unavailable. This leaves the following questions:

- What construction and diameter cable is required? This question will be answered by knowing how much force the solenoid actuator will exert, allowing us to choose the most flexible cable that still maintains a strength with an appropriate factor of safety.
- While the environment temperature will reach 175°C, what temperature will the cable realistically reach? This relies on the duration of the 175°C temperature, as well as how quickly the temperature ramps up and down. Some of this information can be gleaned by speaking further with our sponsor, while some of it will require literature research.
- How much will we need to reduce friction in order to maintain our performance criteria? This question is a good candidate for further risk reduction, as building prototypes may be the fastest and easiest way to model the effects of friction in the Bowden cable.
- Lastly, what types of fittings will be used for the ends? This question is low priority, as there are realistically many different interfaces between the valve and the cable and the actuator and the cable that will fulfill our purposes. But, it is still a question that is important to consider due to the potential for losses due to friction.

In concluding my research, I also identified GREMCO as a company to contact in regards to their PTFE cable liners designed for actuation cables. A link to their website and one of their data sheets are marked with two asterisks under the Other Webpages section above. I'll be asking for information regarding their temperature rating, general tolerancing, minimum purchase requirements, and availability of sampling or student discount programs.

Section 4: Detailed Findings

Bowden cables are a type of flexible cable capable of transmitting mechanical actuation through the movement of a typically steel inner cable through a polymer inner tube contained within a steel outer housing. This outermost housing may also have an outer coating. They are a common linkage in the aerospace, automotive, and biomedical industries. The inner cable transmits force linearly, with a pulling force on one end pulling the inner wire through the outer metal housing. The polymer inner tube serves to reduce friction when the inner cable comes into contact with the sides of the outer sheath. It is important for the outer metal housing to be longitudinally incompressible, so that the path of the cable's displacement remains constant. A visualization of the internals of a Bowden cable are shown in Figure 1.

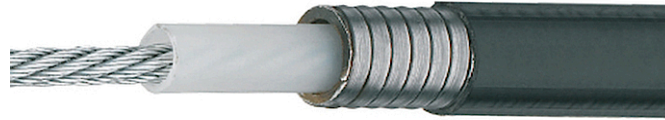


Figure 1: Typical Bowden cable construction. (Source: Carl Stahl Technocables GmbH)

The ends of the Bowden cable can be fixed with many different types of fitting on both the input and output sides. These fittings serve to anchor the cable to its attachment point in order to maintain the pathing of the guiding outer sheath. Through my research I found a variety of fittings, and I find it is reasonable to assume fabricating or purchasing a suitable fitting will be a low risk endeavor. As such, I did not conduct further research on Bowden cable end fittings, because they have little performance impact on the cable itself and they rely heavily on the design considerations of my teammates, which as of now are undefined.

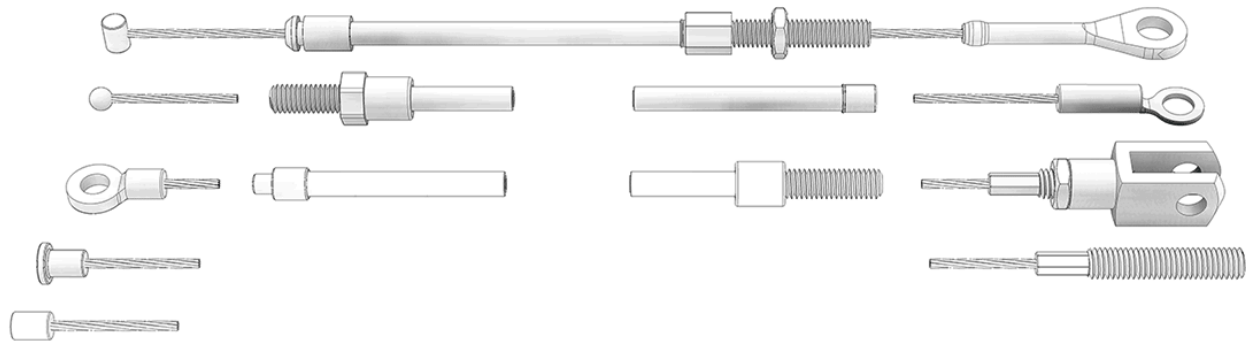


Figure 2: Variety of Bowden cable fitting and end components available for purchase from Carl Stahl Technocables. (Source: Carl Stahl Technocables GmbH)

Initially, I also researched push-pull cables due to their robust capabilities and the fact that manufacturers for these cables gear them to be suitable for high precision control environments. However, I discovered that they typically have higher cost, lower flexibility, and lower fatigue lifespans. This information, combined with our use case only requiring a pull condition, led me to eliminate push-pull cables from consideration early in the research cycle.

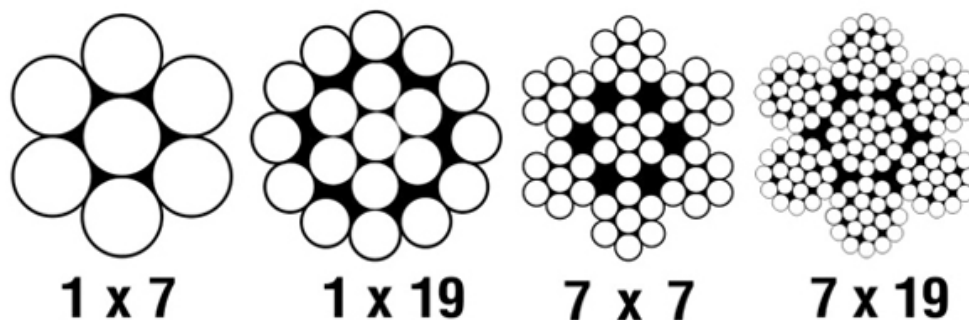


Figure 3: Cable construction configurations. Each circle represents a wire. (Source: Motion Control Technologies Inc.)

The construction of a Bowden cable's inner cable contributes most heavily to the cable's overall performance characteristics. Inner cable material does not vary much, as it is typically stainless or galvanized steel. Figure 3 shows construction configurations for the inner cable. When diameter is maintained, cables with more wires will be more flexible and resistant to tensional fatigue. Cables with fewer wires will resist greater loads and be more resistant to crush or abrasion forces. The main concerns for our project's use of cable is mostly tensional fatigue and flexibility, meaning that a higher wire count cable would fit our purposes best. Since the amount of displacement we will be dealing with is small in relation to the forces required to open the spring loaded valve, tensional fatigue is the main concern, followed by flexibility. However, abrasive forces are a concern due to the low response time the valve and therefore cable will need to have. Abrasive forces are also increased due to bending of the cable where the cable will come into more contact with the polymer inner tubing. Therefore, while more research must be done on the fatigue failure characteristics of Bowden cables, a 7x7 or 1x19 cable would likely be a good choice for initial prototypes.

The inner tube acts to reduce friction and abrasive forces. However, most of the commercially available Bowden cables utilize HDPE, Nylon or other low temperature rated polymers. However, I found a couple of companies, Sava Industries and GREMCO, that produce Teflon coated cable and Teflon cable liners respectively. Teflon is potentially a good choice for inner tubing material due to its high temperature rating of 200°C+, with GREMCO's formulations boasting a rating of 300°C. Sava's 400°F (204°C) rated FEP (Teflon) coated cable is a composite stainless steel core with FEP coating, that comes in constructions of 7x7 or 7x19. It would be used as the inner cable, with the Teflon coating helping to reduce friction against the inner tube. This cable is flexible and comes in diameters conducive to our use case, but I still have concerns about its temperature capabilities after talking to a representative. While the testing environment of Cohu's machines may only reach 175°C, depending on the duration of this environment's temperature setting and other unpredictable/varying factors, the cable itself may exceed its rated 204°C. This is a dangerous possibility, as when Teflon begins to break down, it releases toxic fumes. GREMCO's Teflon cable liners, with the Fitco® Liner Insulating Hose being rated for -40°C to 300°C, would act as an inner tube material. According to their website, their liners offer high flexibility and low friction coefficient, but validation of specifications other than those that are thermally related is still necessary.

The outer sheath should be relatively easy to source. Several of the manufacturer websites listed above have highly flexible, stainless steel Bowden sheaths available for

purchase. It serves as a protective structure that also maintains the shape of the cable. While outer sheaths are typically coated, this is for the protection of the cable from the environment in applications such as automotive, where the cable may come into contact with water or dirt. Our sponsor's testing machines will not have hostile conditions except for the temperature requirement. Therefore, we can omit the outer coating easily by purchasing uncoated steel sheathing. In conclusion, the high priority objectives for implementation of the Bowden cable as our mechanical linkage of choice are:

- **Inner tubing temperature constraints.** A polymer would be best suited due to its smooth surface and flexible nature, but few if any commercially available polymers can withstand the required temperatures.
- **Cable geometry.** Selection will be based on projected relative magnitudes of tensile forces, abrasive forces, and fatiguing. Assigning priority to resistance of the characteristics will allow for a choice to be made. This requires discussion of the system as a whole.
- **Minimum purchasing requirements.** Due to the specialty nature of our desired cable, minimum purchasing requirements exist. Finding vendors with low/no purchasing requirement will help reduce risk of wasted funds and material.

Section 5: Appendix

Phone Call with Dave Balister from Carl Stahl Sava Industries (Duration: ~7 min)

Q: I am interested in your FEP Teflon coated cables, what is the minimum length purchase requirement?

A: Dave stated that particulars of purchase requirement and pricing vary greatly depending on construction and diameters. However, Sava would be able to provide 25 ft of cable free of charge as a sample.

Q: The cable will be placed in an environment of around 350°F (175°C) and the website says the cable is rated to 400°F. Would prolonged exposure to this environment be outside of the cable's specifications, and does Sava have further specifications regarding temperature?

A: I was informed that the cable "might work" for my use case. There was no additional available information, and Dave informed me that they do not conduct in depth testing with regards to temperature, as performance will vary depending on application.

Q: What kind of flexibility can I expect from the cable?

A: Flexibility depends on construction, with higher strength cables having lower flexibility.

Dave emailed me the specification sheet for the types of FEP coated cables available, and invited me to email him back in the future with further questions once I know more design constraints and would like to order cable.

Attached are annotated copies of the following documents in order:

1. [Analysis of Bowden Cable Transmission Performance for Orthosis Applications | IEEE Conference Publication](#)
2. [Custom Bowden Cable Assembly | Stranded Wires - Tyler Madison Inc.](#)
3. [Push-Pull Cable Design | Push-Pull Control Cable Assemblies - Tyler Madison Inc.](#)
4. [Mechanical Push Pull Cable Assemblies - Motion Control Technologies Inc.](#)
5. [FITCO® Liner Natural PTFE - GREMCO](#)

Individual Component Report #3: Solenoid Component Analysis

By Manoel Aguirre Lara

Section 1: Background

The project is to design a miniature fluid flow control valve that meets several specific requirements. The valve must be able to withstand a wide range of extreme temperatures (-55°C to 175°C), have a small volumetric footprint, work with both gas and liquid, and have a remote actuator that is located in a more benign environment. Furthermore, the design must not include any electronics near the valve, but instead use a mechanical method of actuation such as a linkage or bicycle cable. Finally, the valve must have an adjustment range that allows it to go from fully closed to fully open. The preferred component to work the valve will be a solenoid actuator since it can be used remotely. A solenoid actuator is a device that uses an electromagnet to convert electrical energy into mechanical energy. It consists of a coil of wire that carries an electric current, surrounded by a ferromagnetic core, usually in the form of a rod or a plunger. When the coil is energized, it creates a magnetic field that pulls the ferromagnetic core into the center of the coil, or pushes it away, depending on the design of the actuator.

Pull-type Linear Solenoid Actuator

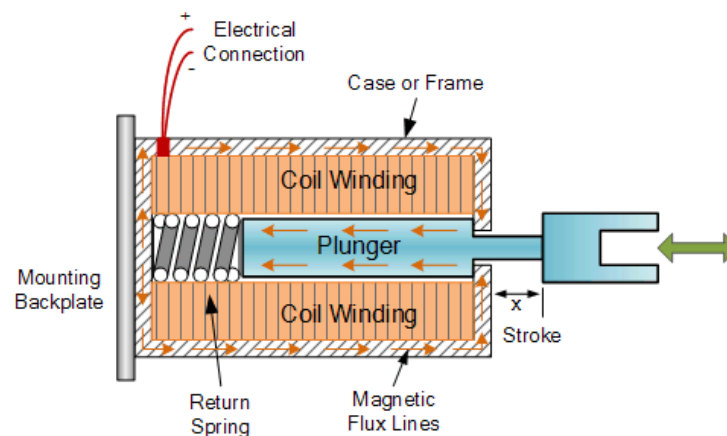


Figure 1: Solenoid actuator segmented view from ElectronicsTutorials

Section 2: Summary of Information learned

Solenoid actuators are commonly used in a wide range of applications, including valves, switches, door locks, and starter motors in automobiles. They are preferred over other types of actuators because they are relatively simple, efficient, and reliable. They can be designed to operate at high speeds and with high forces, making them suitable for applications that require precise control and fast response times. There are many different types of solenoid actuators, including **linear, rotary, and latching types**. Linear solenoid actuators move in a straight line, while rotary solenoid actuators rotate around a central axis. Latching solenoid actuators can maintain their position even when the power is turned off, making them ideal for applications where power consumption is a concern. For our purposes, we will be looking at linear actuators. The different types of linear actuators consist of voice coil motors, stepper motors, tubular solenoids, push-pull solenoids, proportional solenoids and super stroke solenoids. The one that best fits our project is a voice coil motor. They are “highly precise and controllable electrical actuators, suitable for applications needing only limited displacement” (Geeplus.com). As for valves, **there are normally closed, normally open and bi-stable types**. Normally closed and normally open valves are as the name suggests, closed or opened respectively until given a signal to do the opposite. Bi-stable solenoid valves can be switched by a momentary power supply and will stay in that position without power, thanks to permanent magnets instead of a spring. It is not normally open or closed, as it remains in its current position when no power is applied. In addition to these, there are several solenoid valve working principles. These are: **direct acting, indirect acting, semi-direct acting, and 3-way direct acting**. In summary, solenoid actuators are widely used electromechanical devices that use an electromagnetic field to convert electrical energy into mechanical energy. They are reliable, efficient, and can be designed for a wide range of applications.

As a group, we have to design a solenoid valve actuator that is linked to a valve with a distance of 1 m in between the two components. This linkage must be flexible in order to be placed around obstacles or be switched between systems easily. We haven't decided how the linkage will work yet, as it seems to be the most important part of the project. Depending on what we decide on, we will be able to choose the best fitting solenoid actuator. It will most

likely be a normally closed valve with a linear solenoid actuator that will be able to push or pull the mechanism in the linkage in order to fully close or fully open the valve.

Section 3: Information Gathered

Keywords

- Normally closed
- Normally open
- Linear
- Rotary
- Latching
- Tubular
- VCM
- Bi-stable
- Mono stable
- Direct acting
- Semi-direct acting
- Indirect acting
- 3-way direct acting

List of articles

- <https://tameson.com/pages/solenoid-valve-types>
- <https://tameson.com/pages/choosing-an-ac-or-dc-coil-for-a-solenoid-valve>
- https://tameson.com/pages/solenoid-valve-symbols#_skj4ic2ci5us
- <https://www.kendrion.com/en/products/solenoids-actuators/rotary-solenoids>
- <https://www.kendrion.com/en/products/solenoids-actuators/rotary-solenoids/monostable-rotary-solenoids>
- <https://www.kendrion.com/en/products/solenoids-actuators/rotary-solenoids/bistable-rotary-solenoids>
- <https://www.kendrion.com/en/products/solenoids-actuators/rotary-solenoids/compact-rotary-solenoid>
- <https://tameson.com/pages/solenoid-valves-for-pneumatic-systems>
- https://www.electronics-tutorials.ws/io/io_6.html

Section 4: Description of Findings

Item	Price	Availability	Pros	Cons
Solenoid valve ST-DB ¼" NO brass EPDM 0-12 bar 12V DC / 24V AC / 24V DC / 230V AC / 120V AC	\$54.28	In Stock on TAMESON 3-6 business days delivery	- Fast acting - Delivered including coil and connector - EPDM not suitable for (hot) water, neutral fluids and many chemicals - Good protection against the environment thanks to the IP65 rating	- EPDM is not suitable for oils and fats - Sensitive to dirt, can only be used with clean liquids and gasses - Uses energy while activated and the coil may get hot. A well ventilated installation environment is recommended to dissipate the heat
Solenoid Valve ST-DB ¼" NO brass FKM 0-12 bar 12V DC / 24V DC / 24V AC / 230V AC / 120V AC	\$54.28	In Stock on TAMESON 3-6 business days delivery	- Solenoid valves open and close very fast compared to many other valve types - Delivered including coil and connector - Good protection against the environment thanks to the IP65 rating	- Solenoid valves are sensitive to dirt and can only be used with clean liquids and gasses - Uses energy while activated and the coil may get hot. A well ventilated installation environment is recommended to dissipate the heat
Solenoid Valve	\$56.27	In Stock on	- Compact and	- Solenoid

CM-DA 1/8" Stainless Steel FKM 0-10 bar 24V DC / 230V AC / 24V AC / 12V DC / 120V AC		TAMESON 3-6 business days delivery	low energy consumption of 6 VA - Solenoid valves open and close very fast compared to many other valve types - Delivered including coil and connector - Good protection against the environment thanks to the IP65 rating	valves are very sensitive to dirt and can only be used with clean liquids and gasses - Solenoid valves use energy while activated and the coil may get hot. A well ventilated installation environment is recommended to dissipate the heat.
Bi-stable PDM50 (50mm) 24-48V DC, 60-100W, max 40°	Requested	Upon request	- High holding torque (up to 1 Nm) - Long service life (>100 million switching cycles) - Robust design - Short switching times (10-20 ms) - Quiet operation	- Up to a 40° angle of rotation
Monostable Rotary Solenoid Type D 24V DC / 205V DC - Variants D1-D9	Requested	Upon request	- Compact design - High durability - Shaft operating axial stroke - Precise shaft mounting	- Mostly used for mobile machinery and transportation

			- Short switching cycles - Up to 95° of rotation	
Compact Rotary Solenoid 30x30x46mm 24V DC	Requested	Upon Request	- Cost-efficient design - Low space requirements - High reliability	- Dampers have to be installed to reduce noise when the rotary solenoid switches

Table 1: Comparison of products

One requirement for our final experiment is to open the valve in 10% increments, which would be facilitated by a voice coil motor. This type of actuator provides “excellent control for short strokes”. It also needs to withstand temperatures from 0°C to 40°C. The models from Geeplus.com that I am considering are:

- VM75P2

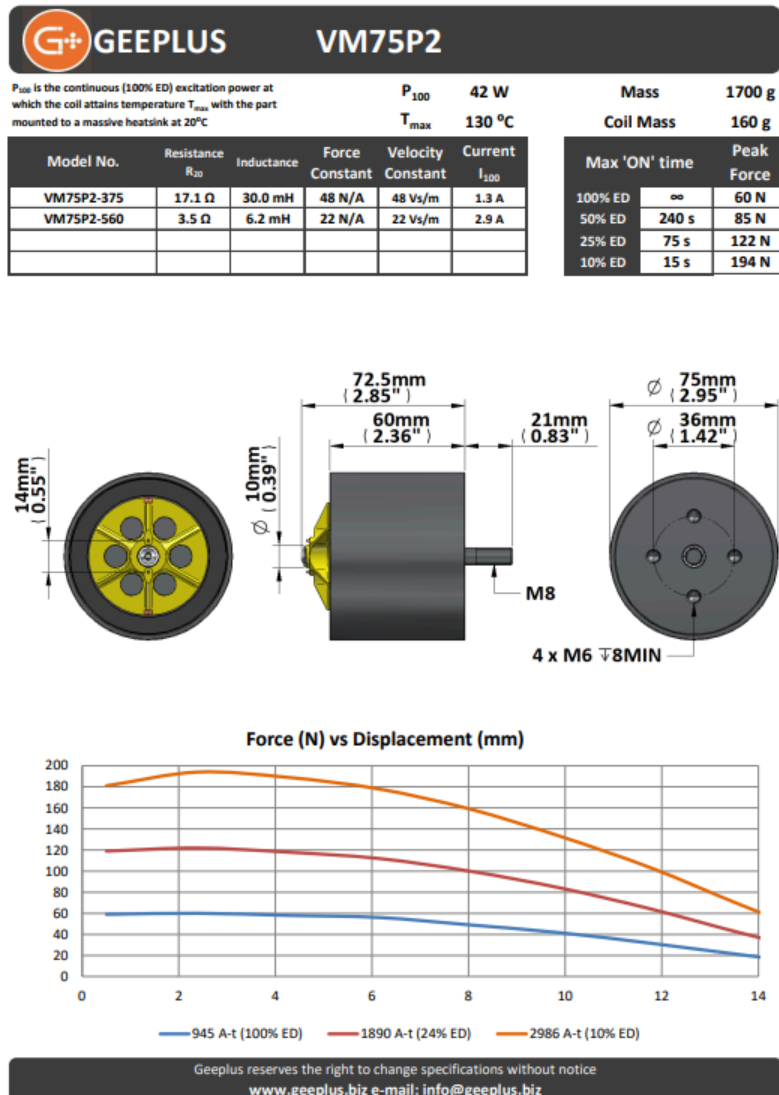


Figure 2: Description of of Voice motor coil model VM75P2

- VM8054 & VM8080

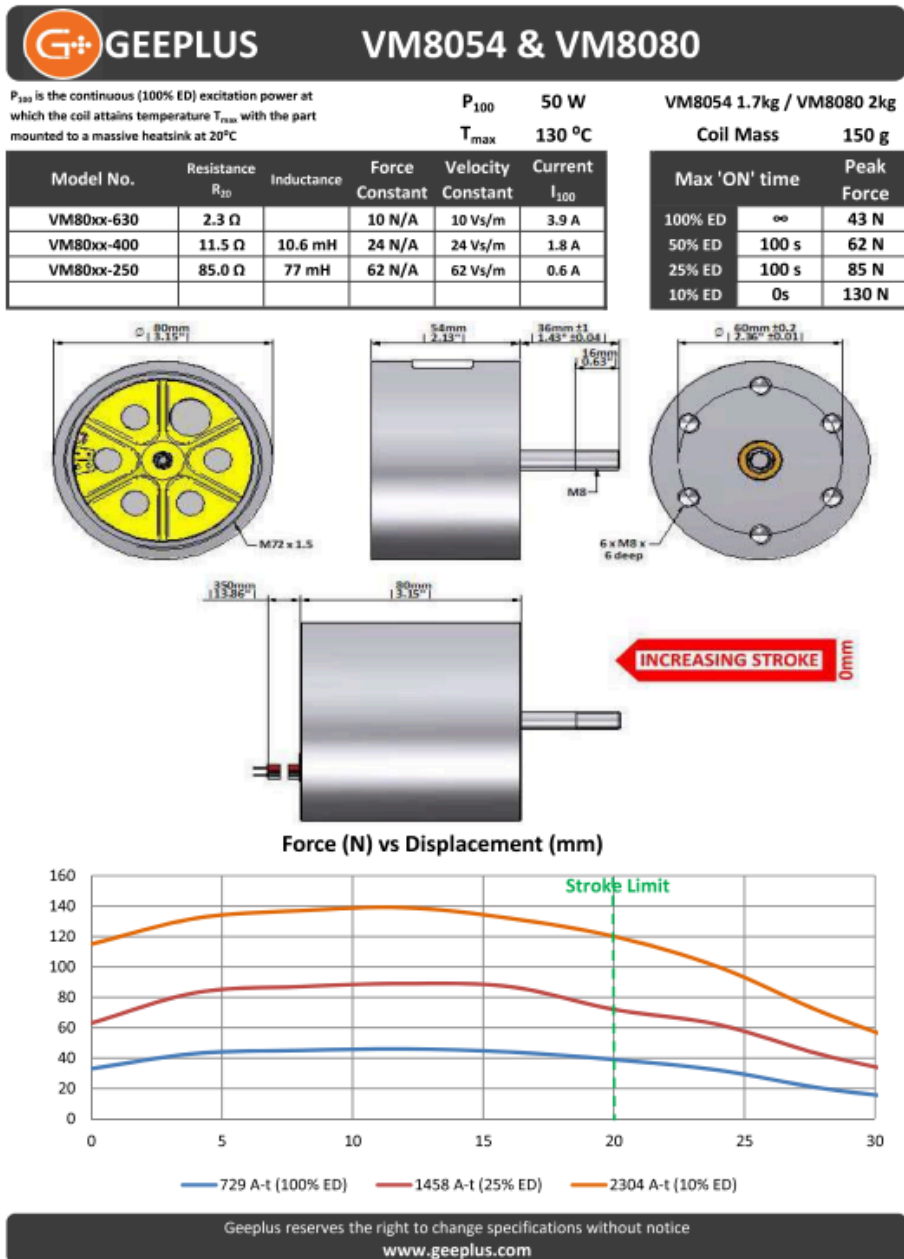


Figure 3: Description of voice coil models VM0854 & VM8080

These models are a good fit because the actuator needs to pull with a force of at least 44N and provide precise control for the small distance we need.

Physics

As stated before, a solenoid actuator can convert electrical energy into mechanical energy.

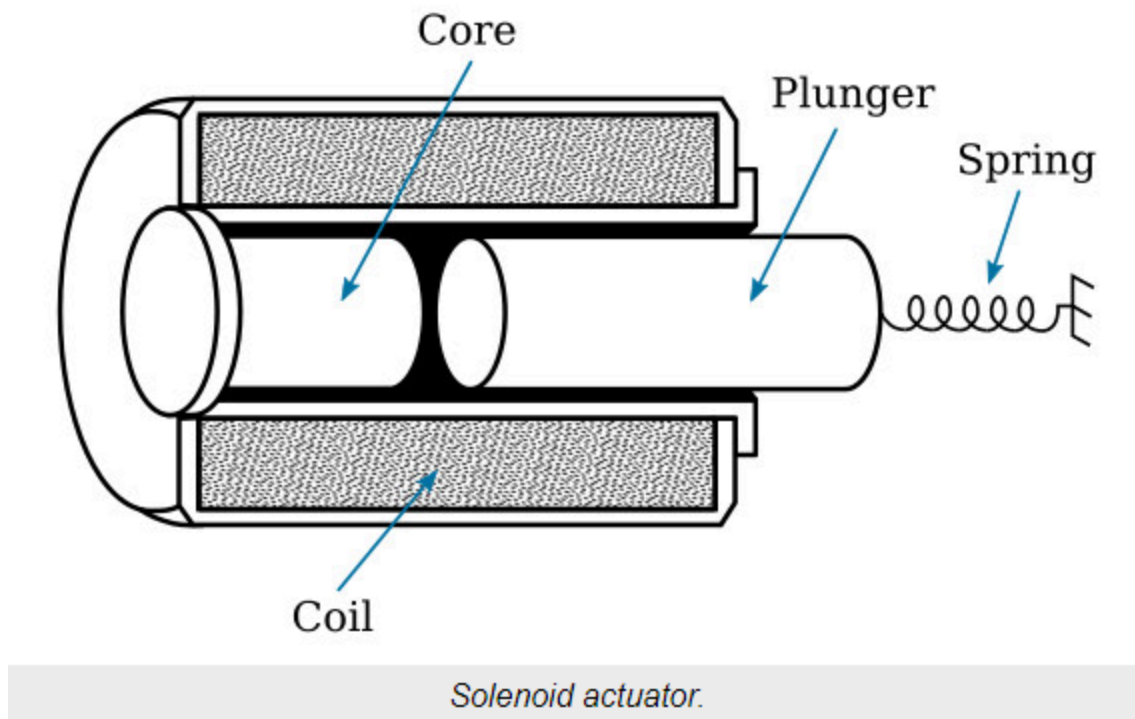


Figure 4: Cutaway picture of a solenoid from ECStudio

The solenoid is composed of an armature in the middle which is wrapped with a coil. This armature can be a plunger or have a plunger inside, these solenoid actuators are most commonly used with spring. When current flows through the coil, it creates a magnetic field that can pull or push the plunger which would result in opening or closing of the valve. The working principles are direct acting, indirect acting, semi-direct acting, and 3-way direct acting.

There are direct acting solenoid valves that work with a plunger and a spring in normally closed or open, which simply pushes or pulls the plunger. Indirect acting solenoid valves use pressure differential of the medium to open and close the valve, and typically require a minimum pressure differential of 0.5 bar. They have a rubber membrane with a small hole that separates the inlet and outlet ports, and a pilot orifice that is larger than the hole in the membrane. When the solenoid is energized, the pilot orifice is opened, causing the pressure above the membrane to drop and allowing the medium to flow from the inlet to the outlet port. The extra pressure chamber above the membrane amplifies the flow rate, allowing a small solenoid to control a large flow rate. These valves are only used for media flow in one

direction and are used in applications with a sufficient pressure differential and a high desired flow rate.

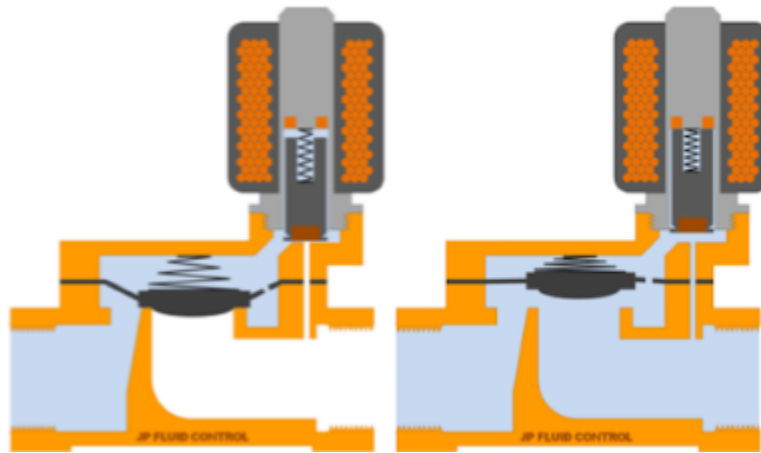


Figure 5: Indirect acting solenoid valve (TAMESON)

Semi-direct acting solenoid valves combine features of both direct and indirect valves. They can operate from zero bar and handle high flow rates. They have a movable membrane with pressure chambers on both sides, but the solenoid plunger is directly connected to the membrane. When the plunger lifts, it opens the valve and opens a second, slightly larger orifice in the membrane, which causes the pressure above the membrane to drop, allowing the membrane to lift not only due to the plunger but also due to the pressure difference. They are also known as assisted-lift solenoid valves and often have more powerful coils than indirect valves.

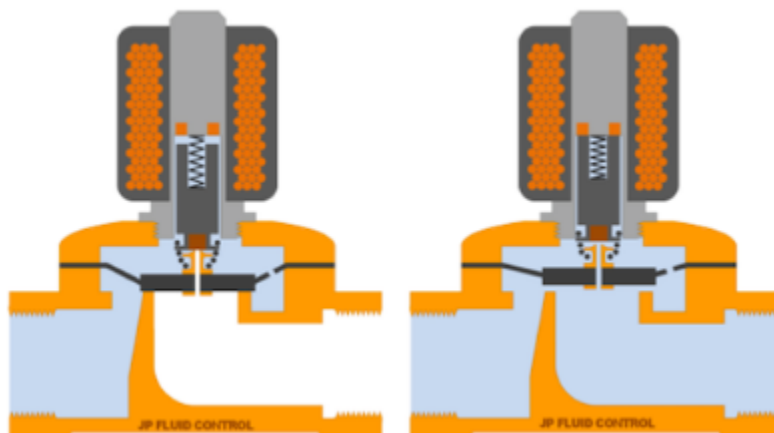


Figure 6: Semi- direct acting solenoid valves (TAMESON)

A 3-way solenoid valve has three ports, and its operation can be either mixing or diverting, depending on the desired inlet and outlet configurations. Some valves can operate in both directions, known as universal circuit function, but only two ports are connected in each state. Only one valve seat is open at a time, and the plunger has an orifice on both the top and bottom. This enables the media to flow in the desired direction.

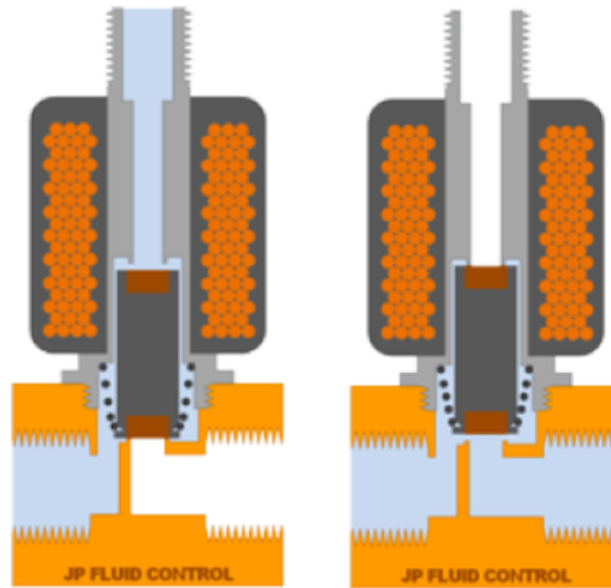


Figure 7: 3-way direct acting solenoid valve (TAMESON)

Some examples of circuit functions for a normally closed valve are:

- **Diverting 3-way solenoid valve:** It has one inlet (bottom left, Fig. 3) and two outlets (top and bottom right, Fig. 3). In a normally closed valve, when there is no power, the plunger blocks the bottom orifice, and the media flows from the inlet to the top outlet. When power is applied, the plunger moves up, closing the top outlet and allowing the media to flow from the inlet to the right outlet.
- **Mixing 3-way solenoid valve:** This instance has two inlets (top and bottom right, Fig. 3) and one outlet (bottom left, Fig. 3). When there's no power, the plunger blocks the bottom orifice, and the media flows from the top inlet to the outlet. When power is applied, the plunger moves up, closing the top outlet and enabling the media to flow from the bottom right inlet to the outlet.

- **Universal 3-way solenoid valve:** These valves work like diverting 3-way solenoid valves, but the media can flow in any direction and still only has two ports.

Section 5: Appendix

Contacts

- Kendrion contact



Manoel Aguirre-lara <maguirrelara@ucsd.edu>
to sales-ims

4:06 PM (0 minutes ago) ☆ ↶ ⋮

Hello Kendrion,

My name is Manoel Aguirre Lara and I'm a mechanical engineering student at UCSD. My group is working with solenoid valves and we would like to know the prices for your bi-stable, monostable and compact rotary solenoids. Please reply to this email.

Thank you,
Manoel Aguirre Lara

- Geeplus contact

<https://drive.google.com/file/d/11qllOI6X1o29gRVvx6oE4zRur-ucA6ND/view?usp=sharing>

Material

- Solenoid valve symbols -
https://drive.google.com/file/d/1vp1IHFv1L14MeQbDg3pvPNdCX_57_gcM/view?usp=sharing
- Solenoid valve - how they work -
<https://drive.google.com/file/d/1V7Wri8126B9-qwoWTvLHlFE2epst9J5p/view?usp=sharing>
- Rotary solenoids -
<https://drive.google.com/file/d/1PkQinhjDt5D46QExeqjJDlwObS9xaCOD/view?usp=sharing>
- Monostable rotary solenoids -
https://drive.google.com/file/d/1hsdSFkY2OMyskMbrkkEr_iqwh7V_L6r/view?usp=sharing
- Directional control valve - how they work -
https://drive.google.com/file/d/1XioszbbW3EGs0K4NPUP6o0E-QL8RGqO_/view?usp=sharing
- Compact rotary solenoid -
https://drive.google.com/file/d/1qnFw3Wh6R-qxt3nnlc4p97Jdtv_1L2yN/view?usp=sharing
- Bistable rotary solenoid -
https://drive.google.com/file/d/1mA1oo5kXzxNJjhwa-Wq_xlPcsIOv0692/view?usp=sharing
- AC or DC solenoid valve coil overview -
https://drive.google.com/file/d/14PT_6wrP3bPHQZKhsiv82436xKZSfLiF/view?usp=sharing
- <https://www.geeplus.com/characteristics-and-selection-of-voice-coil-motors/>

Individual Component Report #4: Flexible Mechanical Linkage

By Michael Gigoux

Section 1: Brief Project Description

Cohu Inc. performs semiconductor validation and design thermal regulation systems to help maintain chip testing temperatures. They have tasked us with redesigning a flow control valve that can handle a higher temperature range for semiconductor testing. The current design does not perform well under high temperature conditions given that the solenoid/actuator mechanism is built into the valve and, under higher temperatures, the solenoid can undergo demagnetization and the other electrical components can also suffer damages.

Our main goal is to redesign the valve by separating the actuator mechanism from the actual valve by a (non-linear) distance of 1 meter. The current operating environmental temperature range of the ACF valve is 0°C to 40°C . The target temperature range for the flow valve is -40°C to 175°C , while the operating range for the actuator mechanism stays the same. Thus, the linkage between the actuator and the flow valve must be fully-mechanical (at least near the flow valve) and the material selection must be able to withstand these operating temperatures. Based on the figure below, the following two design constraints have been provided.

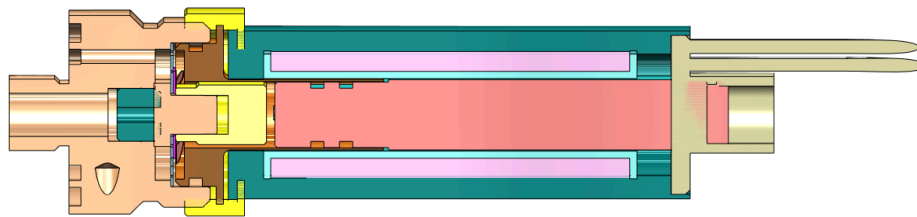


Figure 1: Section View of the Current ACF Valve (note that the solenoid and other wiring, emphasized in pink, is part of the current design)

1. Separate into 2 sections, valve section & actuator section (electronics) linked by a mechanical linkage
2. Separated 2 sections by a 1m distance
 - a. Linkage not allowed to be straight/direct path (Simulating other components in the way in the real world application)

The primary design challenge here is the fact that the linkage cannot be a “straight or direct path”, which removes the obvious option of a straight gearbox mechanism. This means that the degrees of freedom (DOF) of the system will have to be somewhat variable to maneuver around any electronic components in the way of the mechanism on the test site. The

geometry we need to maneuver around has not been specified, so the design should be manufactured to be easily disassembled and reassembled to fit their environment.

Section 2: Summary of Information Learned

Based on the detailed findings (discussed later), we discovered a few mechanical linkages that work around the indirect 1 meter path that we had to connect the actuator and the flow control valve. From my findings, I primarily focused on the development of a worm drive system.

A worm drive is a gearbox that has a typical spur gear that acts as the driven gear and an externally threaded rod (called a “worm”) that drives the spur gear. A sliding mesh gear system is fully planar, as seen in the figures below. A worm gear allows for 90° bends which are not only useful to maneuver around the unknown geometry we will be facing, but also for translating from purely planar motion to 3D.



Clear Distinction Between Helical Gears and Worm Gears ([Image Source](#))

Regardless of the non planar motion, worm gears are also very practical for this application since they can typically have a much larger gear reduction ratio than typical helical gearboxes with just as many parts. Since the output force required to pull the spring in the flow valve is very minimal (max compression of around 1-2 mm) the gearbox needs a high reduction ratio from input to output. To translate the rotational motion of the motor and gear system to a linear compression of the spring, we will employ a rack and pinion system near the valve itself.

The main concern of worm gears is the sliding friction between components, so lubrication of the parts and pitting of the teeth also needs to be accounted for. The lubricants typically used are typically high-viscosity synthetic oils since they provide slightly higher efficiencies than low viscosity lubricants and even mineral oils. At an input speed of 1200 rpm, the overall gearbox efficiency averages around 94-95%. There are ways to numerically estimate the efficiency of these gears using conservation of mass and Navier-Stokes equations. These equations help us model flow resistance and air resistance which can be used to determine the total power loss between gears without any manufacturing required.

Using AGMA recommendations and standards, we can determine the type of gear we need that can be rated for the life cycles and the environmental conditions provided to us. By

using these standards, we can avoid making any custom gear models and save money on the expense of developing a flexible, variable-gear system.

Section 3: Information Gathering Approach

Because the mechanical system couldn't be linear, we couldn't just have a straight shaft or a set of sliding mesh gears connecting the actuator to the flow control valve. Thus, we need some initial idea on what type of mechanical system we want to use before continuing the research into the effectiveness of said system.

Me and Matthew decided to split up this project research among ourselves since this mechanism is the bulk of what the project entails. The initial choice of keywords to determine the types of mechanical linkages we might use were as follows:

- Mechanical linkage
 - Non-linear
 - Variable degree of freedom
- Flexible Linkage
 - Signal Transmission
 - Elastic
- High-temperature rated
 - Materials
 - Mechanical Efficiency
 - Wear

This was our first approach in determining a feasible mechanical system and we came across a few ideas based on this initial research. The following concepts are listed below based on our initial findings.

1. Worm gears
2. Chain drives
3. Flexible drivetrains

The two most feasible contenders were the first two listed: the worm gears and the chain drive. So, we decided to split our efforts among the two mechanical systems; he researches the effectiveness of chain drives while I research the effectiveness of worm gears.

A worm gear system consists of two parts: the spur gear (driven gear) which has the typical cylindrical gear shape, and the worm gear (driving gear), which consists of a shaft wrapped in a spiral thread that engages with the teeth of the spur gear. Because of the mechanism, there is no rolling component between the gears and the movement is entirely sliding motion, thus the main issue for this type of system is lubrication. To minimize sliding friction and sliding wear, proper lubrication needs to be in place between the spur gear and the worm gear. Thus, the main keywords used for this component were as follows:

- Worm gear
 - Manufacturing
 - Lubrication
- Worm gear efficiency
- Worm gear fatigue
 - Material selection
 - High-cycle (or Infinite cycle)

Based on these search results (and textbook entries) I found the following sources of information about worm drives, including lubrication, efficiency in high temperatures, manufacturing methods, and material selection.

Web Pages:

General Overview of Worm Gears

Articles:

Lubrication Information About Worm Gears (Numerical and Empirical Approach)

Efficiency of Worm Gearboxes

Textbook Findings:

Shigley's Mechanical Engineering Design (useful for gearbox manufacturing and determining fatigue failures)

After determining the feasibility of implementing this method, we contacted RushGears, an American Gear Manufacturer that provides a wide range of different gear types from helical gears, to bevel gears, to worm gears. They provide a list of the best materials and provide quotes for their commonly ordered gear specs, but they also offer custom 3D modeled gears that can be quoted as well.

The remaining questions we still have or will need to figure out on our own include the following.

- What is the expected cycle lifetime of these worm gears?
- What a good reduction ratio?
 - Minimize back EMF on the motor
 - Still make sure it's slow enough so it doesn't provide too large of an output force on the spring.
- What is a good number of gears that allow us to move through all the geometries Cohu Inc wants?
 - Don't want too many gears for risk of efficiency
 - Don't want too little gears since the system is supposed to be flexible

Section 4: Detailed Description of Findings

The first thing I needed to determine, of course, was “What is a worm drive?” The web page above gives a general overview of a worm gear and defines it as follows.

“A worm gear is a gear consisting of a shaft with a spiral thread that engages with and drives a toothed wheel... It changes the rotational movement by 90 degrees, and the plane of movement also changes due to the position of the worm on the worm wheel (or simply "the wheel"). They are typically comprised of a steel worm and a brass wheel.”

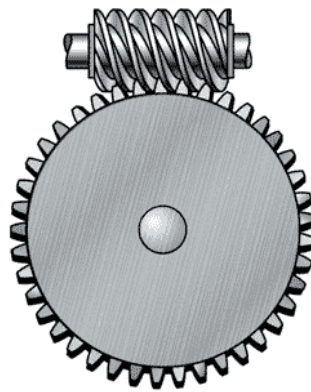


Figure 1. Worm gear. Most worms (but not all) are at the bottom.

The key takeaway here is that a worm drive has a driving threaded worm gear (top gear) and a driven spur gear (bottom gear) that allows the gear system to make 90° bends unlike a typical gear system, which only has planar translation of rotational energy. This is important for our design as the separation distance between the actuator and the flow valve is not linear, and we have to bend around certain geometries to translate the energy from the driving motor to the flow valve. Using a rack and pinion system, that rotational motion from the gearbox can be converted to rectilinear motion near the flow valve itself.

Of course, worm gears come with many pros and cons compared to a regular gear system, so I have listed them below.

Pros:

- High gear-reduction ratio
- Inability to reverse direction of power (friction prevents spur gear acting as the driving gear)
 - Prevents adding a backstop

Cons:

- Movement between the two is entirely sliding (no rolling component)

- **Requires plenty of lubrication to maintain the system**
- Requires high-viscosity lubricant to overcome sliding between gears

The high gear-reduction ratio is important since we want the spring in the actuator to move very minimally. However, we run into a problem with the major con of the mechanical system: efficiency losses and constant lubrication. The need for constant lubrication does increase the cost by a bit, despite the relatively small size of the gears we will be working with. Due to the constant lubrication, we must also quantify the efficiency losses between the gears as the output in the rack and pinion system needs to be very precise to compress the spring enough to regulate flow through the valve.

This leads us to the next source of discussion: maximizing worm gearbox efficiency. Based on the findings of Mautner et al. in their paper *Efficiency of worm gearboxes*, when properly lubricated, these gearboxes maximize at around 96% efficiency and stabilize at around 94% efficiency within a range of 0 to 8 million cycles. The specs of the gearbox, the test parameters, and the results of this experiment are listed below.

Table 2. Test program on large-sized worm gears (CuSn12Ni2-C-GZ worm wheel with 20MnCr5 worm shaft, $a = 315$ mm, $i = 10.25$).

Test	Main test conditions			Lubricants	
	Input speed n_1/min^{-1}	Output torque T_2/kNm	Output power P_2/kW	Lubricants for running-in	Main test lubricant
1	1200	14.7	180	Klübersynth GH6-460 (polyglycol)	Klübersynth GH6-460 (polyglycol)
2	300		45	Klübersynth GH6-460 (polyglycol)	
3	1200		180	Klübersynth GEZ6-220 (polyglycol)	
4	300		45	Fuchs Renolin CLP 460 (mineral oil)	

Table 1. Gear data of investigated test gearboxes.

Flank form ZI according to DIN 3975,² right-handed thread

	Symbol	Value	Unit
Centre distance	a	315	mm
Gear ratio	$i = z_2/z_1$	41/4 = 10.25	—
Pressure angle	α_0	20	°
Axial module	m_x	12.5	mm
Lead angle	γ_m	21.80	°
Profile shift	x_2	−0.3	—
Reference ϕ , worm	d_{m1}	125	mm
Reference ϕ , wheel	d_{m2}	505	mm
Rim width	b_{2R}	90	mm
Nominal torque ^a for $n_1 = 300 \text{ min}^{-1}$	T_{2N}	22,800	Nm
$n_1 = 1200 \text{ min}^{-1}$		11,000	Nm

^aAccording to Henschel Antriebstechnik³ for CuSn12Ni2-C-GZ.

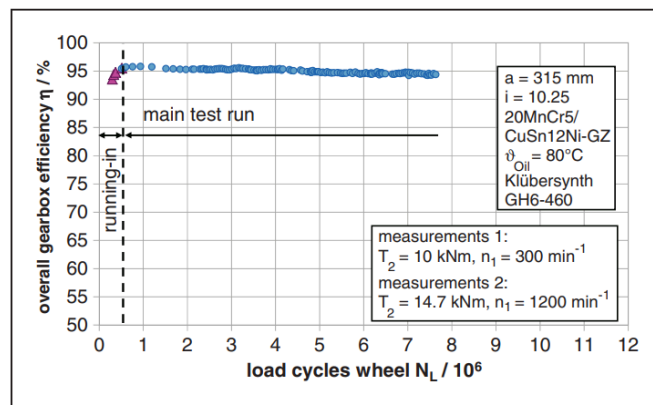


Figure 3. Overall efficiency over load cycles wheel during test I.

The previous flow valve design was rated for 100 million cycles and these tests only maximize at around 8 million cycles, so to quantify this information, we will have to determine a gear ratio that translates the 8 million cycles of the gearbox to roughly 100 million cycles of the flow valve (one flow valve cycle is defined going from full close to full open and back to

full close again). Likewise, the size of the gears used in this experiment are at least 10x larger than the gears we will be using in our system, so we will need to determine how to maximize lubrication efficiency not only for the increased load cycles, but for the decreased pitch diameter.

Deng et al. found ways to quantify lubrication at various speeds, oil depths, viscosity, and worm arrangement in a typical worm drive system. Using a moving particle semi-implicit (MPS) method, we can numerically estimate the efficiency of the system with proper lubrication using the following governing equations

Equation of Continuity (Law of Mass Conservation):

$$\frac{D\rho}{Dt} = 0$$

Navier Stokes Equation (Law of Momentum Conservation):

$$\frac{Du}{Dt} = -\frac{\nabla P}{\rho} + \nu \nabla^2 u + g$$

The rest of the article goes in depth on how to break down the Navier-Stokes equation into an explicit stage (calculating all the terms except for the pressure term since they are given) and an implicit stage (calculating the pressure term). By then using the pressure term to model both flow and air resistance, we can estimate the churning loss torque. This churning loss torque gives us an estimate of the power loss in the system, which can be used to determine the efficiency of the system by taking a ratio between the expected output power and the theoretical max power provided by the driving motor.

RushGears is an American manufacturing company that specializes in gear manufacturing ranging from simple spur gears to bevel gears to all sorts of custom gears. They also develop worm gears and, alongside the typical specs, which I have listed in the figure below, they also take custom quotes when provided a CAD model. Of course, since we want to reduce the number of custom parts, we more than likely want to stick to the specs that they provide, but in the case that these custom parts need to be ordered, they can provide.

CAD	PARTNO	QUOTE	PITCH	TEETH	PD	OD	PA	RATIO	CD	BORE	FACE	STYLE	HD	HP	OAL	HAND	LA	STARTS	WORMLEAD	MATEPD
CAD	WB1016QL	☐	10	16	1.6	1.9	14.5	4:1	1.425	0.75	0.88	B	1.5	0.75	1.63	L	17.75	4	1.2566	1.25
CAD	WB1016QR	☐	10	16	1.6	1.9	14.5	4:1	1.425	0.75	0.88	B	1.5	0.75	1.63	R	17.75	4	1.2566	1.25
CAD	WB1020DL	☐	10	20	2	2.3	14.5	10:1	1.625	0.875	0.63	B	1.63	0.63	1.26	L	9.0833	2	0.6283	1.25
CAD	WB1020DR	☐	10	20	2	2.3	14.5	10:1	1.625	0.875	0.63	B	1.63	0.63	1.26	R	9.0833	2	0.6283	1.25
CAD	WB1020L	☐	10	20	2	2.3	14.5	20:1	1.625	0.5	0.63	B	1.25	0.75	1.38	L	4.5666	1	0.3142	1.25
CAD	WB1020QL	☐	10	20	2	2.3	14.5	5:1	1.625	0.875	0.63	B	1.63	0.63	1.26	L	17.75	4	1.2566	1.25
CAD	WB1020QR	☐	10	20	2	2.3	14.5	5:1	1.625	0.875	0.63	B	1.63	0.63	1.26	R	17.75	4	1.2566	1.25
CAD	WB1020R	☐	10	20	2	2.3	14.5	20:1	1.625	0.5	0.63	B	1.25	0.75	1.38	R	4.5666	1	0.3142	1.25
CAD	WB1030DL	☐	10	30	3	3.3	14.5	15:1	2.125	0.875	0.63	B	1.75	0.63	1.26	L	9.0833	2	0.6283	1.25
CAD	WB1030DR	☐	10	30	3	3.3	14.5	15:1	2.125	0.875	0.63	B	1.75	0.63	1.26	R	9.0833	2	0.6283	1.25
CAD	WB1030L	☐	10	30	3	3.3	14.5	30:1	2.125	0.625	0.63	B	1.75	0.75	1.38	L	4.5666	1	0.3142	1.25
CAD	WB1030QL	☐	10	30	3	3.3	14.5	7.5:1	2.125	0.875	0.63	B	1.75	0.63	1.26	L	17.75	4	1.2566	1.25
CAD	WB1030QR	☐	10	30	3	3.3	14.5	7.5:1	2.125	0.875	0.63	B	1.75	0.63	1.26	R	17.75	4	1.2566	1.25
CAD	WB1030R	☐	10	30	3	3.3	14.5	30:1	2.125	0.625	0.63	B	1.75	0.75	1.38	R	4.5666	1	0.3142	1.25

Rush Gears Quotes for Commonly Ordered Worm Gears within our Specifications (Pitch Diameter from 1.6" to 3")

To determine the right specs for our worm gears, we can use Shigley's Mechanical Engineering Design. This textbook give specifications as to how to characterize the gears pitch diameter in terms of the transverse and axial pitch, characterize the lead angle, and standardize the tooth system. Once characterized, we can use the AGMA standards to determine the allowable tangential force, the sliding velocity, and the allowable torque in terms of these given parameters. The following equations are listed to determine said information.

Spur Gear Pitch Diameter:

$$d_g = \frac{N_g p_t}{\pi}$$

N_g = number of teeth of spur gear

p_t = transverse circular pitch

Lead:

$$L = p_x N_w$$

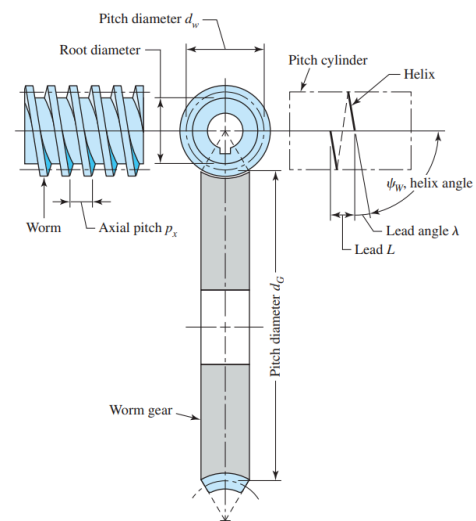
p_x = (axial pitch = 0 if shaft angle is 90°)

N_w = number of teeth of spur gear

Lead Angle:

$$\tan \lambda = \frac{L}{\pi d_w}$$

d_w = pitch diameter of worm



Tangential Force on Worm-Gear Tooth:

$$(W^t)_{all} = C_s D_m^{0.8} F_e C_m C_v$$

C_s = materials factor

D_m = mean gear diameter [in]

F_e = effective face width of the gear (actual face width, but not to exceed $0.67d_m$, the mean worm diameter) [in]

C_m = ratio correction factor

C_v = velocity factor

Friction Force:

$$W_f = \frac{fW^t}{\cos(\lambda)\cos(\phi_n)}$$

f = coefficient of friction

λ = lead angle at mean worm diameter

ϕ_n = normal pressure angle

Sliding Velocity

$$V_s = \frac{\pi n_w d_m}{12\cos(\lambda)}$$

n_w = relative speed of the worm

d_m = mean worm diameter

Torque at Worm Gear

$$T_G = \frac{W^t D_m}{2}$$

D_m = mean gear diameter

We can determine the material parameters from our supplier (RushGears) and use these to best determine the right selection for our component. Given the analysis before and the provided pressure and lead angles from our supplier, we can mathematically calculate the mechanical efficiency of the gears without lubrication to better understand how the teeth/threading of the spur/worm gears interact with each other.

With all this information, we can now manufacture or outsource a gear to our specifications that allows us to work around the variable geometry we are to expect in between the flow valve and the actuator.

Individual Component Report #5: Valve Housing

By Bora Gursel

Appendix C: Designs Considered

PTFE O-Rings

The original temperature range constraint precluded the use of fluorosilicone. The low end of the refrigerant temperature range at -80C was lower than the -70C minimum temperature rating for fluorosilicone o-rings. PTFE was considered, but ultimately not selected due to concerns that exposure to fluorinated refrigerant fluids would cause swelling in the rings.

Appendix D: Other Calculations

The primary design factor dictating the spring load and spring rate is the “defaults to closed” requirement which requires the valve to remain shut off without any input from the solenoid. The inner valve must be able to stay closed against the force generated by the fluid pressure acting at 482,633 Pa (70 psi). The nominal nozzle opening has a diameter of 3.175mm (0.125 in.). Hence for a safety factor of 4, the spring force necessary can be calculated:

$$Force = Pressure \times Area = (482,633) \times \pi(3.175/2 \times 10^{-3})^2 = 3.82N$$

$$Adjusted\ Force = Force \times F.O.S. = 3.82N \times 2 \approx 8N\ (1.8\ lbf)$$

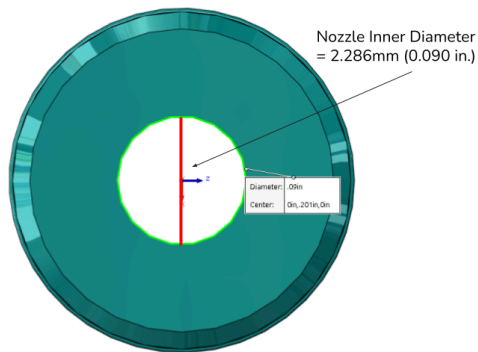


Figure D1: Valve Nozzle Diameter

Appendix E: Bill of Materials for Designed Flow Valve

Description	Vendor	Unit Price	Quantity	Part Number	Total Amount
Multipurpose 6061 Aluminum, 3/8" Diameter, 2 Foot Long	McMaster	\$3.23	1	8974K24	\$3.23
Multipurpose 6061 Aluminum, 5/8" Diameter, 2 Foot Long	McMaster	\$6.36	1	8974K48	\$6.36
Multipurpose 6061 Aluminum, 1" Diameter, 2 Foot Long	McMaster	\$19.67	1	8974K13	\$19.67
Multipurpose 6061 Aluminum, 1-1/2" Diameter, 1 Foot Long	McMaster	\$20.67	1	8974K18	\$20.67
CS062-M8-S17 Wave Spring 0.625in O.D. Link	Smalley	\$25.31	2	CS062-M8-S17	\$50.62
Oil-Embedded Bronze Sleeve Bearing for 3/16" Shaft Diameter and 1/4" Housing ID, 1/4" Long	McMaster	\$0.90	4	6391K122	\$3.60
Dowel Pin, 4037, 4140 Alloy Steel, 1/8" Diameter, 1/4" Long, Packs of 50	McMaster	\$10.29	1	98381A469	\$10.29
McMaster - Chemical-Resistant Low-Temperature Fluorosilicone O-Ring, 1/16 Fractional Width, Dash Number 014, Packs of 5	McMaster	\$6.25	1	8333T124	\$6.25

McMaster - Chemical-Resistant Low-Temperature Fluorosilicone O-Ring, 1/16 Fractional Width, Dash Number 016	McMaster	\$7.12	1	8333T126	\$7.12
McMaster - Chemical-Resistant Low-Temperature Fluorosilicone O-Ring, 1/16 Fractional Width, Dash Number 018, Packs of 5	McMaster	\$8.50	1	8333T128	\$8.50
McMaster - 316 Stainless Steel Socket Head Screw, 4-40 Thread Size, Packs of 50	McMaster	\$5.75	1	92185A110	\$5.75
Stainless Steel, Cable, Coated 7x7, FEP, Commercial Coated Diameter 0.084"	Sava Industries	\$1.58	250	103185	\$395.00
PTFE Standard Tubing - All Sizes - Fractional (in) / 3/16" ID X 1/4" OD	Fluorostore	\$1.43	25	F015001	\$35.75
Universal-Thread Push-To-Connect Tube Fitting For Air And Water, Stainless Steel, Straight, For	McMaster	\$13.76	1	7610N115	\$13.76

1/4" Od, 1/8 Pipe Size					
Self-Locking External Retaining Rings For 1/2" Od, Black-Phosphate 1060-1090 Spring Steel, Packs of 25	McMaster	\$8.41	1	90341A107	\$8.41
Micro Traders 4pcs M7 7mm Brake Lever Adjusting Regulator Screws for Mountain Road Bike MTB Bikes Cable Adjusters Bolts Brake Parts Aluminum Alloy Black M7x26.6x14.6mm	Amazon	\$6.89	1	SP_452_B	\$6.89
Wire Rope Stops-Not For Lifting For Steel Rope, Copper, For 1/16" Rope Diameter, packs of 50	McMaster	\$9.06	1	3926T44	\$9.06
Wire Rope Compression Sleeve-For Lifting For Steel Rope, Copper, For 1/16" Rope Diameter, packs of 50	McMaster	\$9.48	1	3897T22	\$9.48
18-8 Stainless Steel Flat-Tip Set Screws, 8-32 Thread, 1/4" Long, packs of 25	McMaster	\$3.30	1	94355A190	\$3.30
Stepper motor NEMA 23	Nanotec	\$145.19	1	SCA5618M4204-A	\$145.19
DRV8825 Stepper Motor Driver Carrier,	Pololu	\$16.09	2	2982	\$32.18

High Current (Header Pins Soldered)					
TOTAL COST					\$815.08

It is important to note that most of these items are bought in bulk or in large packs, which can be used to assemble multiple flow valves. The only two components that were not ordered in bulk are the motor and the motor driver. Accounting for the amount of material used per valve, the number of individual COTS components used, and shipping costs, the total cost of manufacturing one valve comes out to be \$244.80, which is lower than the cost of Cohu's current AFC valve.

Machining costs have not been factored into the cost of the new valve. The shipping on the motor was also >\$80, meaning that they can be ordered in bulk for a cheaper price, much like most of the other parts listed in the bill of materials. Some components, like the FEP coated wire rope and the PTFE standard tubing, have to be ordered in bulk for the sake of shipping costs. Since our design consists of 6061-T6 aluminum, costs might change if Cohu decides to use another material (i.e. 304 Stainless Steel).

Appendix F: Budget

		Budget Limit: \$2,000.00	Spent: \$1,748.41	Remaining: \$251.59
Date	Description	Vendor	Specified by (team member name)	Total Amount
4/12/2023	1 in. O.D., 2 ft. long Aluminum Round Stock: Link	McMaster	Bora Gursel	\$19.67
4/12/2023	Wire Rope-Not for Lifting (Unlubricated, 3/16" Diameter): Link	McMaster	Michael Gigoux	\$19.50
4/12/2023	High Temperature Soft Rubber Tubing for Air and Water (1/4" ID): Link	McMaster	Michael Gigoux	\$64.00
4/12/2023	Flexible Metal Conduit: Link	McMaster	Michael Gigoux	\$90.25
4/12/2023	McMaster Carr - Sales Tax & Shipping Fees	McMaster	Bora Gursel	\$24.08
4/28/2023	CS062-M8-S17 Wave Spring 0.625in O.D. Link	Smalley	Bora Gursel	\$50.62
4/28/2023	CS062-L3-S17 Wave Spring 0.625in O.D. Link	Smalley	Bora Gursel	\$26.74
4/28/2023	Smalley - Shipping Fee	Smalley	Bora Gursel	\$14.40
5/1/2023	Multipurpose 6061 Aluminum, 1-1/2" Diameter, 1 Foot Long	McMaster	Bora Gursel	\$20.67
5/1/2023	Oil-Embedded Bronze Sleeve Bearing for 3/16" Shaft Diameter and 1/4" Housing ID, 1/4" Long	McMaster	Bora Gursel	\$3.60

5/1/2023	Dowel Pin, 4037, 4140 Alloy Steel, 1/8" Diameter, 1/4" Long, Packs of 50	McMaster	Bora Gursel	\$10.29
5/1/2023	Ultra-Chemical-Resistant Rigid PTFE O-Ring, 1/16 Fractional Width, Dash Number 018, Packs of 25	McMaster	Bora Gursel	\$14.25
5/1/2023	McMaster Carr - Sales Tax & Shipping Fees	McMaster	Bora Gursel	\$12.18
5/10/2023	Ultra-Chemical-Resistant Rigid PTFE O-Ring, 1/16 Fractional Width, Dash Number 014, Packs of 25	McMaster	Bora Gursel	\$10.22
5/10/2023	McMaster Carr - Sales Tax & Shipping Fees	McMaster	Bora Gursel	\$7.71
5/16/2023	Songhee drv8825 Stepper Motor Driver	Amazon	Manoel Aguirre Lara	\$14.99
5/16/2023	NEMA 17 Stepper Motor	Amazon	Manoel Aguirre Lara	\$22.96
5/17/2023	PTFE Standard Tubing - All Sizes - Fractional (in) / 1/4" ID X 5/16" OD	Fluorostore	Michael Gigoux	\$56.25
5/17/2023	PTFE Standard Tubing - All Sizes - Fractional (in) / 3/16" ID X 1/4" OD	Fluorostore	Michael Gigoux	\$35.75
5/17/2023	Fluorostore - Sales Tax & Shipping Fees	Fluorostore	Michael Gigoux	\$15.99
5/21/2023	NEMA 23 Stepper Motor	Nanotec	Manoel Aguirre Lara	\$145.19
5/23/2023	Galvanized Steel/HDPE Lining Push-Pull Casing	Sava Cables	Michael Gigoux	\$299.00
5/23/2023	Plated Steel Bulkhead Terminal	Sava Cables	Michael Gigoux	\$148.00

5/24/2023	Single-Turn Spiral Internal Retaining Rings for 5/8" ID, 1060-1090 Spring Steel, packs of 25	McMaster	Michael Gigoux	\$10.97
5/24/2023	Single-Turn Spiral Internal Retaining Rings for 3/4" ID, 1060-1090 Spring Steel, packs of 25	McMaster	Michael Gigoux	\$11.52
5/24/2023	McMaster Carr - Sales Tax & Shipping Fees	McMaster	Michael Gigoux	\$8.55
5/24/2023	Nut for 1/8" Tube OD 37 Degree Flared Fitting for Stainless Steel Tubing	McMaster	Michael Gigoux	\$22.20
5/24/2023	Nut for 1/4" Tube OD 37 Degree Flared Fitting for Stainless Steel Tubing	McMaster	Michael Gigoux	\$13.08
5/24/2023	McMaster Carr - Sales Tax & Shipping Fees	McMaster	Michael Gigoux	\$10.08
5/25/2023	Sleeve For 1/8" Tube Od 37 Degree Flared Fitting For Stainless Steel Tubing	McMaster	Michael Gigoux	\$14.24
5/25/2023	McMaster Carr - Sales Tax & Shipping Fees	McMaster	Michael Gigoux	\$7.91
5/26/2023	PTFE AWG Spaghetti Tubing - AWG 8 Standard Wall	Fluorostore	Michael Gigoux	\$16.00
5/26/2023	Fluorostore - Sales Tax & Shipping Fees	Fluorostore	Michael Gigoux	\$15.99
5/27/2023	Micro Traders 4pcs M10 10mm Brake Lever Adjusting Regulator Screws for Mountain Road Bike MTB Bikes Cable	Amazon	Matthew Cordova	\$8.09

	Adjusters Bolts Brake Parts Aluminum Alloy Black M10x26X10mm			
5/27/2023	Micro Traders 4pcs M7 7mm Brake Lever Adjusting Regulator Screws for Mountain Road Bike MTB Bikes Cable Adjusters Bolts Brake Parts Aluminum Alloy Black M7x26.6x14.6mm	Amazon	Matthew Cordova	\$6.89
5/27/2023	Amazon Sales Tax	Amazon	Matthew Cordova	\$1.16
5/27/2023	CYCLEZ Premium 6mm Bicycle Brake Caliper Barrel Adjusters, Steel, Silver Barrel Adjuster Bolt with Locknut, 10 Pack	Amazon	Matthew Cordova	\$10.49
5/27/2023	Amazon Sales Tax	Amazon	Matthew Cordova	\$0.81
5/27/2023	10PCS Cycles Brake Part Lever Bike Rear Derailleur Brake Cable Line Thumb Shifter Screw Brake Lever Adjusting Barrel(RC122-M5)	Amazon	Matthew Cordova	\$10.30
5/27/2023	Juscycling Index Adjusting Barrel M5 Downtube Barrel Adjuster - 6 pcs/Bag	Amazon	Matthew Cordova	\$8.49
5/27/2023	Amazon Sales Tax	Amazon	Matthew Cordova	\$1.46
5/30/2023	Universal-Thread Push-To-Connect Tube Fitting For Air And Water,	McMaster	Michael Gigoux	\$13.76

	Stainless Steel, Straight, For 1/4" Od, 1/8 Pipe Size			
5/30/2023	McMaster Carr - Sales Tax & Shipping Fees	McMaster	Michael Gigoux	\$7.88
5/30/2023	Wire Rope Stops-Not For Lifting For Steel Rope, Copper, For 1/16" Rope Diameter, packs of 50	McMaster	Matthew Cordova	\$9.06
5/30/2023	McMaster Carr - Sales Tax & Shipping Fees	McMaster	Matthew Cordova	\$8.05
5/30/2023	Wood for Test Stand	Home Depot	Manoel Aguirre Lara	\$27.35
5/31/2023	18-8 Stainless Steel Flat-Tip Set Screws, 8-32 Thread, 1/4" Long, packs of 25	McMaster	Matthew Cordova	\$3.30
5/31/2023	Wire Rope Compression Sleeve-For Lifting For Steel Rope, Copper, For 1/16" Rope Diameter, packs of 50	McMaster	Matthew Cordova	\$9.48
5/31/2023	Multipurpose 6061 Aluminum, 7/16" Thick, 6" x 6"	McMaster	Matthew Cordova	\$16.92
5/31/2023	McMaster Carr - Sales Tax & Shipping Fees	McMaster	Matthew Cordova	\$9.94
6/1/2023	Universal-Thread Push-To-Connect Tube Fitting For Air And Water, Stainless Steel, Straight, For 1/4" Od, 1/4 Pipe Size	McMaster	Michael Gigoux	\$14.35
6/1/2023	McMaster Carr - Sales Tax & Shipping Fees	McMaster	Michael Gigoux	\$7.92

6/1/2023	Self-Locking External Retaining Rings For 1/2" Od, Black-Phosphate 1060-1090 Spring Steel	McMaster	Michael Gigoux	\$8.41
6/1/2023	Fixed-Tip Retaining Ring Plier With Cushion Grip For External Rings, 0.038" Diameter Straight Tip	McMaster	Michael Gigoux	\$26.73
6/1/2023	McMaster Carr - Sales Tax & Shipping Fees	McMaster	Michael Gigoux	\$16.88
6/1/2023	Aluminum Loop Clamp, 1/4" Id, 13/16" Long, packs of 5	McMaster	Matthew Cordova	\$14.94
6/1/2023	Multipurpose 6061 Aluminum 90 Degree Angle With Square Edge, 1/16" Thickness,.5" Outside Height, 1' Long	McMaster	Matthew Cordova	\$2.40
6/1/2023	Zinc-Plated Steel Corner Bracket, 7/8" x 7/8" x 5/8"	McMaster	Matthew Cordova	\$9.30
6/1/2023	Mcmaster Carr - Sales Tax & Shipping Fees	McMaster	Matthew Cordova	\$9.41
6/6/2023	HiLetgo 5pcs DRV8825 Stepper Motor Driver Module for 3D Printer RepRap 4 RAMPS1.4 StepStick	Amazon	Matthew Cordova	\$14.49
6/6/2023	Amazon Sales Tax	Amazon	Matthew Cordova	\$1.12
6/6/2023	Stainless Steel Cable Tie, Solid Locking, Standard, 8" Long	McMaster	Matthew Cordova	\$7.90

6/6/2023	Stainless Steel Cable Tie, Solid Locking, Standard, 10" Long	McMaster	Matthew Cordova	\$18.10
6/6/2023	Mcmaster Carr - Sales Tax & Shipping Fees	McMaster	Matthew Cordova	\$8.08
6/6/2023	24V Power Supply signcomplex	Amazon	Manoel Aguirre Lara	\$19.99
6/6/2023	24V Power Supply alitove	Amazon	Manoel Aguirre Lara	\$9.59
6/8/2023	McMaster - Chemical-Resistant Low-Temperature Fluorosilicone O-Ring, 1/16 Fractional Width, Dash Number 014, Packs of 5	McMaster	Bora Gursel	\$6.25
6/8/2023	McMaster - Chemical-Resistant Low-Temperature Fluorosilicone O-Ring, 1/16 Fractional Width, Dash Number 018, Packs of 5	McMaster	Bora Gursel	\$8.50
6/8/2023	McMaster Carr - Sales Tax & Shipping	McMaster	Bora Gursel	\$8.11
6/8/2023	McMaster - Belleville Spring Lock Washer	McMaster	Bora Gursel	\$11.72
6/8/2023	McMaster - 316 Stainless Steel Socket Head Screw, 4-40 Thread Size	McMaster	Bora Gursel	\$5.75
6/8/2023	Multipurpose 6061 Aluminum, 1-1/2" Diameter, 1 Foot Long	McMaster	Bora Gursel	\$20.67

6/8/2023	McMaster - Chemical-Resistant Low-Temperature Fluorosilicone O-Ring, 1/16 Fractional Width, Dash Number 016	McMaster	Bora Gursel	\$7.12
6/8/2023	McMaster Carr - Sales Tax & Shipping	McMaster	Bora Gursel	\$11.83
6/8/2023	DRV8825 Stepper Motor Driver Carrier, High Current (Header Pins Soldered)	Pololu	Matthew Cordova	\$32.18
6/8/2023	Pololu - Sales Tax & Shipping and Handling	Pololu	Matthew Cordova	\$26.90
6/8/2023	DIGITEN 1/4 Quick Connect 0.3-10L/min Water Hall Effect Flow Sensor Meter	Amazon	Anannaya Vij	\$9.49