

# **Redesign of Alpine Ski Bindings to Prevent Inadvertent Release**

A Major Qualifying Report

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**Degree of Bachelor of Science**

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## Abstract

Inadvertent release is one of the major causes of serious injuries to alpine skiers. The cause of inadvertent release is the repeated flex and counter-flex of the ski and binding, commonly called chatter. Due to current ski binding's high mass and low natural frequency, they are ill equipped to handle this phenomenon. The results of this project were the design and prototyping of a binding conceived to prevent inadvertent release by increasing the natural frequency of the binding, based on a patent held by Prof. Christopher A. Brown. This project also realized the creation of a vibration-generating machine, which was used to test current ski bindings under a vibrating load.

## Acknowledgments

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# 1 Introduction

## 1.1 Objective

The primary objective for this project is to reduce inadvertent release in alpine ski bindings, caused by vibrations encountered in the ski binding and ski. The vibrations are caused by a phenomenon known as chatter, a repeated cycle of flex and counterflex in the ski. Chatter causes dynamic loading and movement of the binding pieces. To reduce the effects of this phenomenon, a prototype of an improved alpine ski binding was designed and tested based on a patent filed by the project advisor, Professor Christopher A. Brown.

The secondary objective for this project is to provide data pertaining to the relationship between chatter and inadvertent release. Through testing of both conventional alpine ski bindings and the created improved prototype, this project aims to increase the industry's understanding of the correlation between chatter experienced in skis and bindings, and its role in inadvertent release.

## 1.2 Rationale

Winter mountain sports are extremely popular worldwide. In the United States alone, the annual attendance rate for alpine skiing and snowboarding resorts averages 57.5 million people. Additionally, the United States spends \$7 billion annually on these winter sports (Lagran, 2015). The safety of alpine skiing has been enhanced recently due to improved binding, boot, and ski designs. The overall injury rate over the past 34 years for alpine skiing has decreased to approximately 1.9 per thousand skier visits due to the new skiing technology available on the market (Johnson et al., 2009). However, skiing still has the potential to be dangerous. The leading cause of serious alpine ski injuries is uncontrollable falls whereas the leading cause of death for skiers is impact with another person or a static object, usually caused by falling. This mode of injury accounts for 0.563 deaths per million participant visits (Lagran, 2015).

The ski-binding-boot (SBB) system is responsible for properly retaining the skier on the skis. In addition, the SBB system is designed to release the boot from the binding when

certain conditions are met in order to prevent injury (Shealy et al., 2005). This system has two main methods of failure. The first failure method is inadvertent release, which is when the boot is prematurely ejected from the ski binding. This can lead to falls that can cause severe injury or even death. The second method of failure for the SBB system is termed a miss. This is when the binding fails to release the boot during periods where the loads encountered in the system exceed those that should cause a release. This can cause severe lower extremity injuries (Shealy et al., 2005).

### **1.3 State of the Art**

#### **1.3.1 Inadvertent Release**

Modern research on ski bindings focuses on two areas: retention and release concerns. Retention concerns focus on the retention of the boot within the binding. These stem from performance considerations, as the binding must transfer control loads from the boot to the ski to maintain control of the skis. In order to prevent inadvertent release, a common practice in professional skiing is to increase the release settings of the binding, which increases the force required to induce release. This is commonly accepted as the only way to prevent inadvertent release (Brown & Madura, 2014).

On the other hand, release concerns deal with the proper release of the boot from the binding when loads in the system exceed the limits determined by the release settings. Proper release aids with the avoidance of injuries sustained as a result of skiing. Anterior Cruciate Ligament (ACL) injuries have specifically been targeted as an area of concern, along with lower extremity equipment related (LEER) injuries. The prevention of these injuries has largely concentrated on multi-directional release in both lateral and vertical directions, in accordance with twisting and bending moments about the tibial shaft, which can cause injuries to the lower leg. Lower release settings and the point of application of the force on the ski are also important factors in avoiding LEER injuries, as bindings then require less impulse force to release (Brown & Madura, 2014).

In summation, higher binding release settings increase the likelihood of lower leg and knee injuries while decreasing the possibility of inadvertent release. This is where the goal of preventing inadvertent release comes into direct conflict with the prevention of injuries

resulting from failed or delayed release. However, research suggests that while lower release settings have decreased the occurrence of lower leg injuries, they have not been effective in preventing ACL injuries. One study looked at approximately 17,000 injuries in order to investigate the relationship between release settings and injuries (Shealy, 2005). The study showed that the trade-offs between injuries due to failure to release and injuries due to inadvertent release have minimal difference. There are, however, other methods that can be considered to improve retention without compromising release settings. Since the model for release settings and injuries is based on load and the model for retention is based on work done, the two could be related to find a common threshold and obtain a balanced release setting.

Another study specifically explored knee injuries sustained during alpine skiing, noting that while the general trend for alpine skiing injuries shows a decline over the past few decades, knee injuries are still relatively common. This study stated that the knee is the most frequently injured part of the body during alpine skiing, with knee injuries making up as much as 33% of the total injuries in adult skiers, and recommended technical improvements to the equipment design to avoid these injuries (Senner et al., 2013). With respect to the binding, the recommendations include the minimization of the effects of external forces, which could be accomplished by adjusting the design of the sliding elements and bearings accordingly. In addition, adding more degrees of freedom to the mechanism would enable multi-directional release including rotating and vertical release. The performance of the binding in dynamic conditions must also be improved in order to prevent inadvertent release, and finally, higher adjustability allows for skiers with various techniques to safely operate the ski binding.

### **1.3.2 Binding Design**

When recreational skiing began to increase in popularity, new designs needed to be created in order to improve the safety and reliability of both the ski and binding. The most prominent of these is the binding itself, as it determines whether to hold the boot in place while in control, or to release during a fall to prevent injury. Patent research was used to study the design intent of previous iterations of safety bindings.



The first patent that was studied was a binding that uses an electronic binding release system (Hull, 1983). This binding system utilizes a dynamometer to measure the stresses within the binding. When the stress focal point is calculated to be approximately at the lower extremities of the skier, the binding releases. Equations within the system are formulated using inertial, damping, and yieldable stiffness measurements.

Another patent described a cableless safety ski binding that holds the heel of the boot in place with adjustable release pressure (Hans, 1967). This system is similar to ski binding designs utilized today. The binding works by locking to the boot using a spring mechanism when the boot is forced in place. The binding then releases when a certain force is met. The release force can be manually adjusted to suit the weight and ski style of the user.

The final patent that was studied was a heel-binding piece that incorporated a secondary heel release mechanism along with a vector decoupler to reduce inadvertent release (Howell, 2015). The two release points incorporate a lateral pivot release and a vertical lever release. With the addition of the secondary release method, the binding is able to release the boot from the ski under more circumstances to combat injury during a fall. The vertical decoupler and lateral cam system combats benign forces that are neither vertical nor lateral to prevent the binding from releasing inadvertently.

#### 1.4 Approach

This project designed and prototyped improved ski binding toe and heel pieces. Spring-loaded extensions within these pieces increase the responsiveness of the binding to displacement caused by vibration. This allows the bindings to adapt to the displacement and return to their normal positions, which helps to prevent inadvertent release.

Current research has limited information regarding the causation of inadvertent release by the vibration of skis and bindings. As previously mentioned, this is commonly caused by repeated cycles of flexion and extension in the ski during a turn, which is referred to as chatter. Through testing of both the current bindings on the market as well as the prototype binding, this project aims to provide new information on the relationship between chatter

and inadvertent release. Testing was accomplished by the design and construction of a vibration machine.

Finally, this project uses Axiomatic Design to design the improved alpine ski binding, which serves to reduce inadvertent release. Collectively exhaustive and mutually exclusive (CEME) functional requirements (FRs) and associated design parameters (DPs) were created to describe and quantify all aspects of the design.

## **1.5 Methods**

### **1.5.1 Model Conventional Binding Using CAD Software**

Using 3D modeling software, a solid model of a conventional ski and binding was developed in order to gain an understanding of how the system worked through visual representation. Autodesk Inventor was used as the CAD software for this project, as it has the ability to conduct modal, finite element analysis (FEA), and material fatigue simulations. These simulations provide important information on the design intent for our improved binding.

### **1.5.2 Design and Manufacture Vibration Generating Equipment**

In order to determine how vibrations affect the release mechanism of alpine ski bindings, a machine capable of generating vibrations in the bindings was designed and manufactured. The machine was used to generate a sinusoidal vibration in the ski, through means of displacing a testing sole affixed into the binding of the ski. The machine was constructed with a sturdy frame, which was secured to a ground surface in order to control the vibrations.

### **1.5.3 Test Conventional Ski Binding with Vibration Test Machine to Induce Inadvertent Release**

To test conventional bindings using the vibration machine, the ski and binding were oscillated at increasing frequencies. During vibration, the frequency and amplitude of the vibrations were recorded via a high-speed camera system to determine the frequency and displacement at the point of release. The data obtained from testing of conventional

bindings was saved for use as a baseline to compare with test data obtained from the prototype ski binding.

#### **1.5.4 Design of Improved Ski Binding Using CAD Software**

After using Axiomatic Design to create a functional decomposition for our improved binding, a CAD model was constructed to help visualize the geometry. The use of 3D modeling in the design of our ski binding allowed our team to perform various geometrical tweaks and simulations on the binding, in order to improve its performance.

To determine the efficacy of the improved binding design, we conducted modal, FEA, and material fatigue simulations to further improve the design of the prototype. The simulations implemented on the improved binding provided important measurements that would otherwise be difficult to measure through physical testing.

The modal, FEA, and material fatigue simulation testing on the improved binding allowed us to compare the results of the conventional ski and binding setup with our improved design. This provided more conclusive evidence of an improvement from the standard binding to our improved design. In addition, these simulations also alleviated bias or measurement error encountered in physical testing equipment. Data from these simulations gave insight on our physical testing readings to allow for better testing procedures.

#### **1.5.5 Create Functional Prototype of Improved Binding**

The group used the design mentioned in the above section to create a prototype. This prototype was created using fused deposition modeling (FDM), and form of 3D printing, or rapid prototyping. The manufacturing process works by extruding layers of molten ABS plastic on top of each other, until the final geometry of the prototype is realized. This technology was used in order to both minimize costs of production, as well as reduce lead-time on highly custom parts.

#### **1.5.6 Test Improved Binding Prototype**

Using the vibration testing machine designed by the group, the prototype was to be put through a series of tests in order to ensure that the binding retains correct release setting

values per ASTM F939-12. This is an important test to complete, because if the binding does not hold up to these standards, it would not be viable for use, regardless of whether or not it improves the safety of skiing through limiting the chance of inadvertent release. After testing the prototype for its compliance with ASTM F939-12, it was to be tested to determine if it was possible to induce inadvertent release in the binding, using the constructed vibration machine.

Unfortunately, testing could not be performed on the prototyped binding. Due to the strength limitations of models created using FDM production methods, the prototype did not have the requisite strength to be able to undergo standard and vibrational release testing. Models created using FDM exhibit only compressive strength along their Z axis (the axis perpendicular to the plane the layers are printed on). Due to the complex geometry of the binding pieces and the function of the binding, it was not possible to limit loads to only one axial direction in the binding. This is due to the bindings requirement for both lateral and vertical release, which require loads along a minimum of two orthogonal axes.

## 2 Design Process

### 2.1 Axiomatic Design

Created by Nam Suh, axiomatic design is a method of design that makes use of two axioms that form the fundamentals of all good designs. This design method consists of two axioms that will be further described below. The first axiom is to maximize independence while the second axiom is to minimize information.

The first axiom dictates that all functional parameters (FR) of the design should be "mutually exclusive" and "collectively exhaustive." Customer needs (CN) are transformed into FRs to fulfill each specific need, and each FR has its own design parameter (DP). A DP is a corporeal part of the design that satisfies a specific FR. This axiom enables the designer to identify any coupling between the FR and DPs. In a design that is completely decoupled, only one DP satisfies each FR. This ensures that varying any one FR has no

effect on the others, and that none of the DPs needs to execute more than one function simultaneously.

The second axiom focuses on simplifying the design problem by removing any physical information from the FRs. This allows for a simpler design process with room for modifications. According to Suh, the simplest solution will have the highest chance of success; therefore it is always the most fitting choice (Suh, 2001).

One of the primary advantages of using this approach is the elimination of nonessential or superfluous iterations, which in turn minimizes the time invested during the design process. A truly independent design matrix will also allow for modifications of the design later in the machining process.

The task of forming FRs and DPs for this project was made easy by using the Acclaro program, which arranged the decomposition in a matrix form, allowing any coupling to easily be identified.

## **2.2 Vibration Test Machine**

### **2.2.1 Purpose**

The chief goal of this project is to obtain a prototype binding design that can offset the effect of chatter while maintaining the normal function of a conventional binding. In order to accomplish this, chatter conditions in conventional bindings must be simulated and analyzed. Hence, a vibration generating machine was needed to carry out testing and analysis of inadvertent release of conventional ski bindings due to chatter.

### **2.2.2 Constraints**

The following constraining factors were taken into consideration while developing the design matrix:

- Budget: Prototype must be within allotted budget provided by the ME department (budget shared with binding prototype)
- Material: Only Aluminum may be used for machining
- Construction: CNC Machining must be used

- Functioning: Aside from AC motor, device must be completely mechanical.
- According to measurements taken from several bindings, at least 15mm displacement must be induced to cause inadvertent release of binding

### 2.2.3 Design Matrix

The individual components of the functional decomposition for the vibration ski-testing machine are illustrated in Figure 1 below. This decomposition will be further expanded upon in the following section of the report.

#	[FR] Functional Requirements	[DP] Design Parameters	FR Measurement
0	Vibrate ski to induce inadvertent release	Linear vibration generator	
1	Generate mechanical power for use by system	AC induction motor	
1.1	Transfer rotary motion from motor to machine	Input shaft linking motor and machine	
1.1.1	Maintain axial alignment	Machined spacing blocks between shaft bearings and mounting plate	
1.1.2	Prevent shaft from moving linearly	Set screws between shaft and bearing	
1.1.3	Reduce rotary friction of shaft	Mounted steel ball bearings on shaft	
1.2	Prevent vibrational load from being transmitted to motor shaft	Flex coupling between motor shaft and machine input shaft	
2	Allow for variable frequency of oscillation	VFD controller to adjust power frequency provided to motor	
3	Convert rotary motion to linear motion	Slider crank mechanism	
3.1	Reduce friction on vertical axis	Linear ball bearing to allow smooth axial motion	
3.2	Maintain linear bearing vertical positioning	Rigid frame between linear bearing and mounting plate	
3.3	Restrict motion to vertical axis	Static interface between linear bearing and rigid frame	
3.4	Reduce friction between vertical linkage and shaft	Steel ball bearings on shaft	
4	Provide common attachment point for bearing and motor location	Mounting plate with tapped holes to locate components	
5	Provide secure attachment to ground	Mounting plate bolted into floor	
6	Prevent vibration from being transferred to ground	Polyurethane bushing between mounting plate and floor	

Figure 1: Functional Decomposition for the Vibration Ski-Testing Machine



## 2.2.4 Functional Requirements and Design Parameters

### 2.2.4.1 *FR0 and DP0*

The primary goal of this design was to reproduce chatter conditions in a controlled environment. Therefore, the zero level decomposition focused on vibrating the ski with the view of meeting the main objective of this project; to understand the mechanism of inadvertent release due to chatter (FR0).

The overall goal described under the above functional requirement, was satisfied by a system which met each of the individual higher and lower level FRs as shown in Figure 1 above. This system is the linear vibration generator system (DP0).

### 2.2.4.2 *FR1 and DP1*

A power source was needed to drive the mechanical system (FR1). This is divided into two children. According to FR1.1, the rotary motion produced by the power source must then be transferred to the system. This is further divided into three children FRs (FR1.1.1, FR1.1.2, and FR1.1.3). First, the components of the system needed to be precisely aligned. The first consideration was to maintain axial alignment. In addition, linear movement of the shaft had to be precluded to ensure axial alignment. The presence of rotary friction at the shaft was also bound to cause alignment issues, and hence it needed to be minimized. FR1.2 focused on preventing undesirable forces from interfering with system functioning by isolating the vibrational load from the motor shaft.

The power source required by FR1 was provided in the form of an AC induction motor (DP1). A shaft functioned as the drivetrain to transmit motion between the motor and rotary drivetrain (DP1.1). This DP has three children that focus on the alignment of the shaft. Axial alignment is maintained using spacing blocks between shaft bearings and mounting plate (DP1.1.1). Linear movement of the shaft is avoided using setscrews between the shaft and bearing (DP1.1.2). The third child, DP1.1.3, minimizes rotational friction using steel ball bearings that are mounted onto the rotary shaft. Finally, DP1.2 isolates the vibrational



load from the motor shaft using a flex coupling between the motor shaft and machine input shaft. This coupling protects the motor by allowing a small amount of torsional movement.

#### **2.2.4.3 FR2 and DP2**

In order to obtain a comprehensive data set on the amplitude and pinpoint the natural frequency, a range of variable frequency oscillations were required (FR2). This was accomplished using a VFD controller that could input variable power frequencies to the motor (DP2).

#### **2.2.4.4 FR3 and DP3**

The top level (FR3) of this functional requirement concerns the conversion of rotary motion from the power source to linear motion. This is broken down into four children FRs (FR3.1, FR3.2, FR3.3, and FR3.4). The first child FR states that friction must be minimized in the vertical axis (FR3.1). Next, the vertical positioning of the linear bearing must be maintained in order to minimize vertical friction (FR3.2). The motion must be restricted to the vertical axis to ensure the vibration is unidirectional and no power is wasted through undesirable motion (FR3.3). The last child FR states that the friction between the vertical linkage and shaft must be minimized (FR3.4).

These functional requirements are satisfied using a slider crank mechanism that converts the rotary motion to linear motion using the following components (DP3). A linear ball bearing allows smooth axial motion (DP3.1), while a rigid frame between the linear bearing and mounting plate maintains the vertical position of the linear bearing (DP3.2). Motion is restricted along the vertical axis using a static interface between the linear bearing and the rigid frame (DP3.3). Lastly, steel ball bearings mounted on the shaft are used to minimize friction between the vertical linkage and shaft (DP3.4).

#### **2.2.4.5 FR4 and DP4**

It is necessary to have a common attachment point for the bearing and motor in order to ensure precise alignment between the motor and the machine shaft (FR4).

This requirement is accomplished by attaching the entire machine to a common mounting plate with tapped holes that are strategically placed for the individual components (DP4).

#### **2.2.4.6 FR5 and DP5**

For safety purposes, a secure attachment to the ground is necessary in order to anchor the system (FR5). In addition to providing safety, this also provides vibrational damping for the machine.

This function requirement is accomplished by bolting the aluminum mounting plate to the floor that provides a secure ground attachment (DP5).

#### **2.2.4.7 FR6 and DP6**

The last functional requirement (FR6) states that the vibration machine must be isolated from the ground such that a minimal amount of vibration is transmitted or lost to the floor.

The design parameter that allowed this FR to be possible (DP6) mentioned that polyurethane bushings are placed at each of the holes between the mounting plate and the floor in order to provide the necessary vibrational damping.

### **2.2.5 Finite Element Analysis of Vibration Machine Assembly**

Finite element analysis, or FEA, was employed in order to determine whether or not the designed vibration machine would stand up to the extreme stresses of testing. FEA calculates stress within a given CAD model by breaking the model down into tiny segments, called elements. It then calculates the stress for each segment, and collates these together to determine stress at all points within the model. In order to perform FEA, an applied load must be calculated. Calculation of the maximum downwards force on the ski, due to its maximum acceleration during the vibration cycle, yielded a load of 886.1 pounds of force. This load was rounded to 1000 pounds, which ensured that any anomalies encountered during real-world testing would not affect the performance of the machine. As such, all loads used in the FEA environment were set to 1000 pounds. The FEA results for the entire assembly are detailed below, in Figure 4 and Figure 3 below. For FEA results of individual components, please refer to Appendix 3: FEA of Machined Parts.

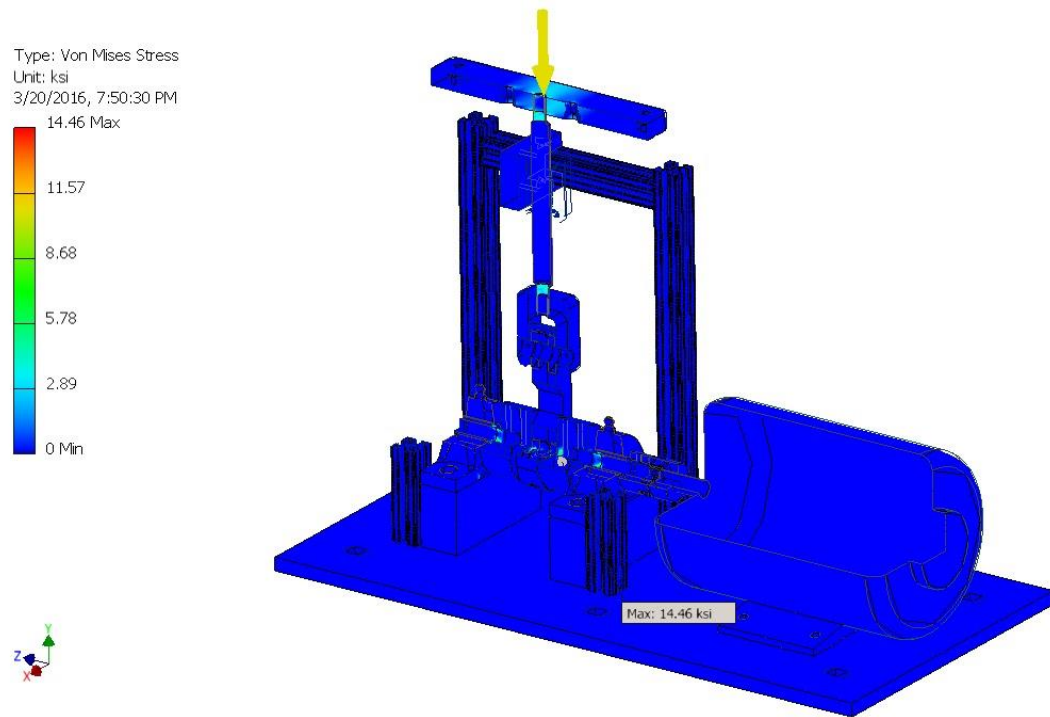


Figure 4: FEA Showing Von Mises Stress for the Vibration Testing Machine

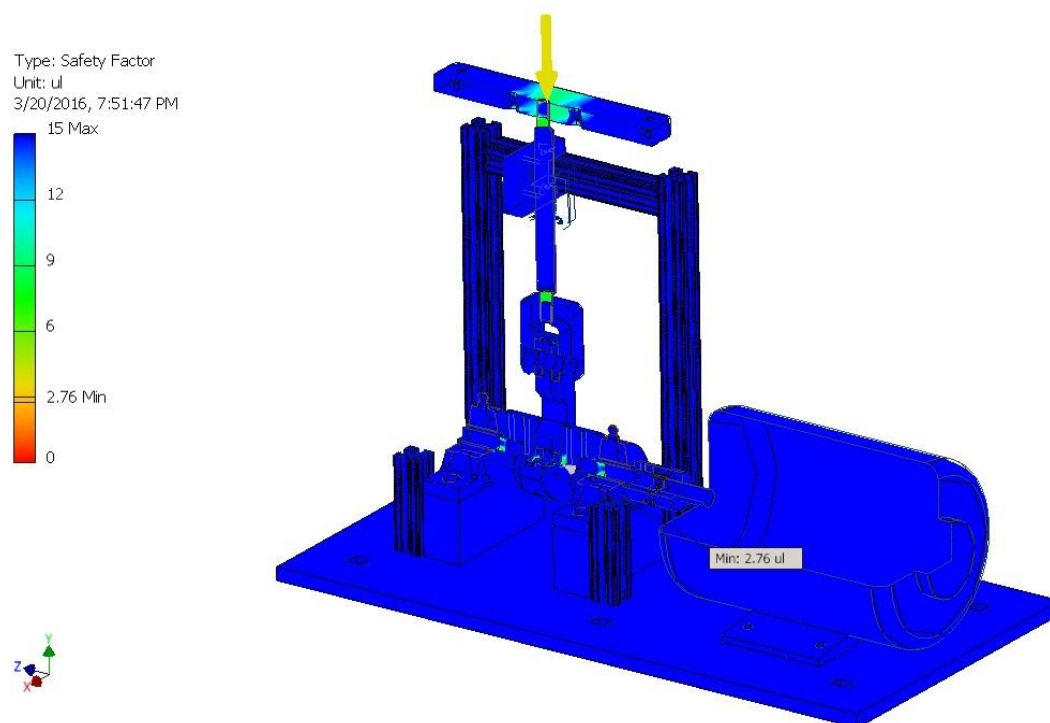


Figure 3: FEA Showing Safety Factors for the Vibration Testing Machine

The FEA results for the vibration machine assembly were tabulated for both the maximum stress encountered in the model, as well as the minimum safety factor. As denoted in Figure 4 and Figure 3, the maximum effected Von Mises stress in the machine was calculated at 14.46ksi, which gave a minimum safety factor of 2.76 for the assembly, indicating the design was viable from a durability standpoint. The FEA results also indicated that design would be compliant with the calculated fatigue limit at  $10^7$  cycles. If the machine were to run at a continuous 40Hz, this would grant us 69.44 hours of available testing time, ensuring that the machine would be viable for extended testing.

## **2.3 Redesigned Ski Binding**

### **2.3.1 Purpose**

The main purpose of the redesigned binding is to increase the frequency in order to counteract the effects of chatter. It is important that the binding needed to be redesigned without altering its normal functions.

### **2.3.2 Constraints**

The following constraining factors were taken into consideration while developing the design matrix:

- Budget: Prototype must be constructed within allotted budget provided by the ME department (budget shared with vibration machine)
- Must retain normal binding functioning in both static and dynamic conditions
- The binding must be able to meet the normal release setting calculated for a conventional binding, as set forth in ASTM F939.
- Natural frequency of prototyped binding must be greater than that of conventional binding

### **2.3.3 Design Matrix**

Figure 5 below shows the functional decomposition for the redesigned ski binding.

#	[FR] Functional Requirements	[DP] Design Parameters
0	Reduce inadvertent release due to chatter	Maintain binding contact with boot during ski oscillation
1	Increase natural frequency of the binding	Reduced weight motional components of binding
2	Compensate for displacement of heel piece due to flexure	A spring loaded member between boot and heel piece
2.1	Allow heel piece to maintain contact with boot	Mechanism response faster than motion of boot
2.2	Allow boot to move bilaterally relative to ski	Mechanism at half load during normal circumstances to allow for bilateral movement
3	Compensate for displacement of toe piece due to flexure	A spring loaded member between boot and toe piece
3.1	Allow toe piece to maintain contact with boot	Mechanism response faster than motion of boot
3.2	Allow boot to move backwards relative to ski	Mechanism at near full load during normal circumstances to allow for unidirectional movement
4	Transmit loads between heel piece and boot	Interface(s) to transmit loads between boot and heel piece
4.1	Transmit moment about x axis	Pin to transmit moment about x axis
4.2	Transmit moment about y axis	Pin to transmit moment about y axis
4.3	Transmit moment about z axis	Pin to transmit moment about z axis
5	Transmit loads between toe piece and boot	Interface(s) to transmit loads between boot and toe piece
5.1	Transmit moment about x axis	Pin to transmit moment about x axis
5.2	Transmit moment about y axis	Pin to transmit moment about y axis
5.3	Transmit moment about z axis	Pin to transmit moment about z axis

Figure 5: Functional Decomposition for the Redesigned Ski Binding

The following figure, Figure 6, displays the corresponding design matrix for the redesigned ski binding.

<div> <div>FR0: Reduce inadvertent release due to chatter</div> <div>FR1: Increase natural frequency of the binding</div> <div>FR2: Compensate for displacement of heel piece due to flexure <div>FR2.1: Allow heel piece to maintain contact with boot</div> <div>FR2.2: Allow boot to move bilaterally relative to ski</div> </div> <div>FR3: Compensate for displacement of toe piece due to flexure <div>FR3.1: Allow toe piece to maintain contact with boot</div> <div>FR3.2: Allow boot to move backwards relative to ski</div> </div> <div>FR4: Transmit loads between heel piece and boot <div>FR4.1: Transmit moment about x axis</div> <div>FR4.2: Transmit moment about y axis</div> <div>FR4.3: Transmit moment about z axis</div> </div> <div>FR5: Transmit loads between toe piece and boot <div>FR5.1: Transmit moment about x axis</div> <div>FR5.2: Transmit moment about y axis</div> <div>FR5.3: Transmit moment about z axis</div> </div> </div>	<div> <div>DP0: Maintain binding contact with boot during ski oscillation</div> <div>DP1: Reduced weight motional components of binding</div> <div>DP2: A spring loaded member between boot and heel piece <div>DP2.1: Mechanism response faster than motion of boot</div> <div>DP2.2: Mechanism at half load during normal circumstances to allow for bilateral movement</div> </div> <div>DP3: A spring loaded member between boot and toe piece <div>DP3.1: Mechanism response faster than motion of boot</div> <div>DP3.2: Mechanism at half near full load during normal circumstances to allow for unidirectional movement</div> </div> <div>DP4: Interface(s) to transmit loads between boot and heel piece <div>DP4.1: Pin to transmit moment about x axis</div> <div>DP4.2: Pin to transmit moment about y axis</div> <div>DP4.3: Pin to transmit moment about z axis</div> </div> <div>DP5: Interface(s) to transmit loads between boot and toe piece <div>DP5.1: Pin to transmit moment about x axis</div> <div>DP5.2: Pin to transmit moment about y axis</div> <div>DP5.3: Pin to transmit moment about z axis</div> </div> </div>
---	---

Figure 6: Design Matrix for the Redesigned Ski Binding

### 2.3.4 Functional Requirements and Design Parameters

#### 2.3.4.1 *FR0 and DP0*

The zero level functional requirement (FR0) is that the primary purpose of this design is the same as the overarching goal of the project; to reduce inadvertent release due to chatter. The corresponding design parameter (DP0) is that the main goal of this design is to be satisfied by a modified ski binding which has the ability to maintain contact with the boot during ski flex and counter flex.

#### 2.3.4.2 *FR1 and DP1*

The first functional requirement on the next level (FR1) is that the natural frequency of the binding must be increased in order to decrease the response time of the binding. The design parameter that comes along with this (DP1) is that the natural frequency of the redesigned binding must be increased. This is accomplished by reducing the weight of the motional components of the binding (DP1).

#### 2.3.4.3 *FR2 and DP2*

As mentioned previously, chatter causes flexure in the ski that results in displacement of the binding pieces. This leads to the next parent function requirement (FR2) that focuses on compensating for this displacement in the heelpiece. This can be accomplished under two conditions described by the two children functional requirements (FR2.1 and FR2.2). The first child FR states that the heelpiece must always maintain contact with the boot. The second child FR states that the boot must be able to move bilaterally relative to the ski.

The design parameter (DP2) that corresponds with the parent FR is that contact between the boot and heelpiece will be maintained by a spring-loaded member. In order to follow the process of axiomatic design, the two children FRs have corresponding DPs (DP2.1 and DP2.2). The first child DP is that the response of the mechanism needs to be faster than the motion of the boot during flexure. The second child DP states that the mechanism will be at half load during normal circumstances to allow for bilateral movement, thus preserving normal binding function and release.

#### 2.3.4.4 *FR3 and DP3*

The next functional requirement (FR3) states that the displacement of the toe piece during flexure must be compensated in addition to the displacement of the heelpiece. As in FR2,



this can be carried out in two components, or child FRs (FR3.1 and FR3.2). The first child FR states that the toe piece must always maintain contact with the boot. The second states that the boot must be able to move backwards relative to the ski.

The parent design parameter for this level (DP3) states that contact between the boot and the toe piece needs to be maintained using another spring-loaded member. The first child design parameter (DP3.1) states that the mechanism response needs to be faster than motion of boot during flexure. In addition, the second child (DP3.2) says that the mechanism is to be near full load during normal circumstances to allow for unidirectional movement.

#### **2.3.4.5 FR4 and DP4**

This level of FRs (FR4) takes into consideration the transmission of loads between heelpiece and boot. This is very significant in order to maintain the normal functionality of the ski binding. This is broken down into three lower level FRs, or children FRs (FR4.1, FR4.2, and FR4.3), which state that moment must be effectively transmitted about the x, y, and z-axes, respectively.

The parent design parameter (DP4) deals mainly with the transmission of loads between the binding and the boot. The loads between the boot and heelpiece are transmitted using the following children DPs (DP4.1, DP4.2, and DP4.3). These state that a pin will be used to transmit moments about the x, y, and z-axes, respectively.

#### **2.3.4.6 FR5 and DP5**

The last function requirement parent (FR5) states that maintaining effective transmission of loads between the toe piece and the boot is essential to the normal function of the ski. The children of this FR (FR5.1, FR5.2, and FR5.3) state that the moment must be effectively transmitted about the x, y, and z-axes, respectively.

The parent design parameter (DP5) states that the boot and toe piece has a similar interface as the boot and heelpiece. The children DPs (DP5.1, DP5.2, and DP5.3) state that a pin is used to transmit the moment about the x, y, and z-axes, respectively.



### 2.3.5 Finite Element Analysis of Redesigned Binding

Similar to the vibration machine, FEA was also performed for both the heelpiece and toe piece assemblies of the redesigned ski binding. To determine the loads that the binding would experience, the release values standards set forth in ASTM F939-12 were followed to determine the appropriate release limits for the binding. This resulted in a maximum vertical release torque of 52.463ft-lb, and a maximum lateral release torque of 221.232ft-lb. These torques translated into a lateral release force of 52.256 pounds-force and a vertical release force of 277.615 pounds-force.

#### 2.3.5.1 Heel Cup

As the heelpiece provides both the vertical release as well as the reaction force for lateral release at the toe, both loads were applied to the heel cup. The FEA results for the heelpiece of the binding are shown below, in Figure 7 and Figure 8. The results for individual components of the heelpiece can be found in Appendix 3: FEA of Machined Parts.

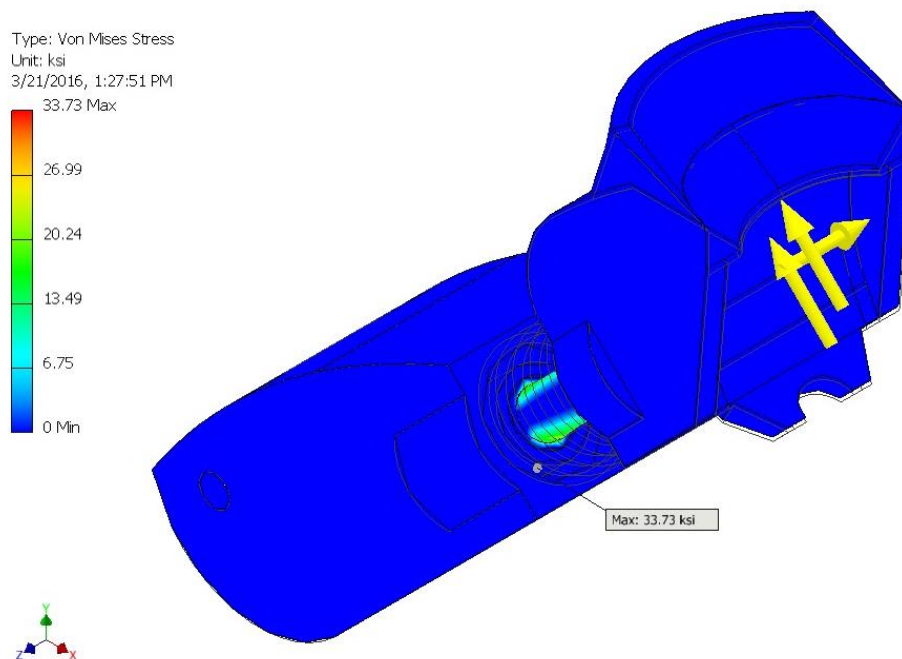
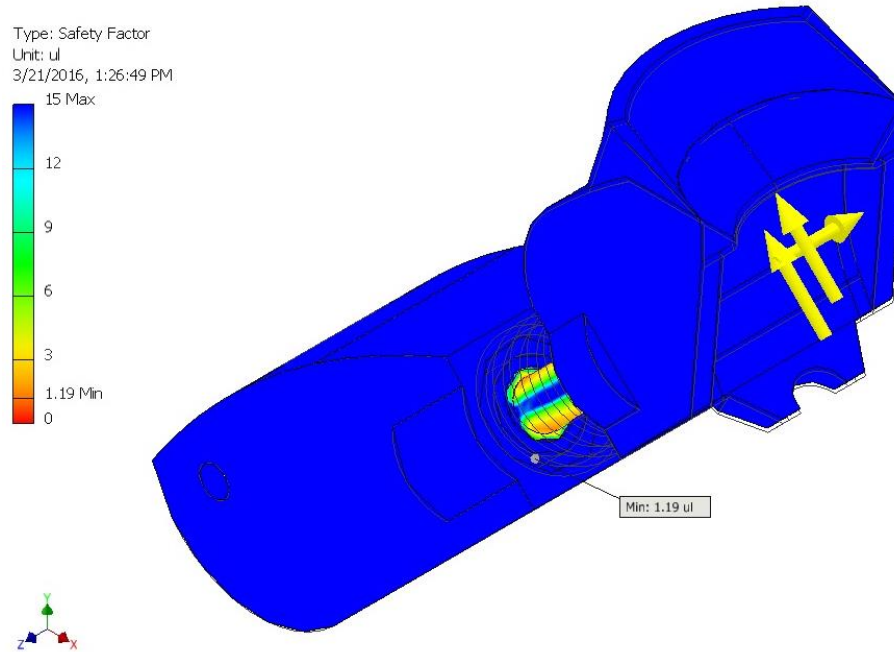


Figure 7: FEA showing Von Mises Stress of the Heelpiece



**Figure 8: FEA Showing the Safety Factors for the Heelpiece**

As shown in the FEA results, the maximum Von Mises stress in the heel piece assembly was 33.73ksi, giving a minimum safety factor of 1.19. Although this safety factor was lower than desired, it occurred in an ‘off the shelf’ brass bushing, which was rated for more load than would be applied to it in our design, so this result was discounted. The lowest safety factor encountered in any of the custom design components was 2.74, which indicated the design would be successful.

#### **2.3.5.2 Toe Cup**

Unlike the heelpiece, the toe piece only provides lateral release, so the only load applied to the toe piece in the FEA simulations was the lateral release force. The FEA results for the toe piece of the binding are shown below, in Figure 9 and Figure 10. The results for individual components of the toe piece can be found in Appendix 3: FEA of Machined Parts.

Type: Von Mises Stress  
Unit: ksi  
3/21/2016, 1:43:06 PM  
24.52 Max

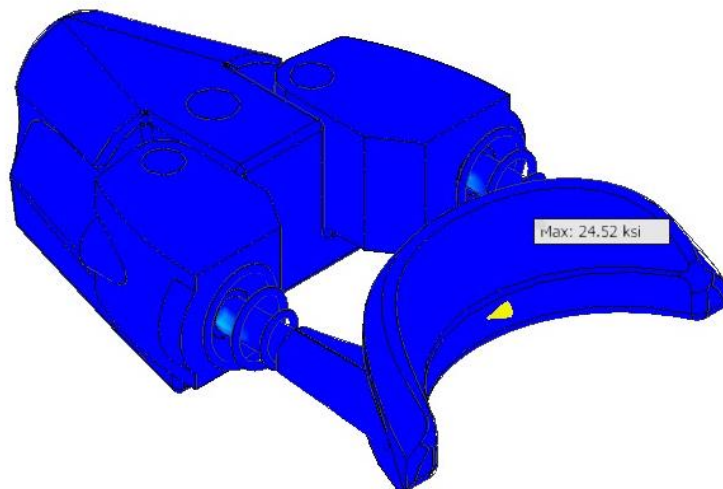
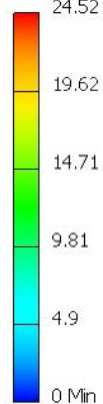


Figure 9: FEA showing Von Mises Stress for the Toe Piece

Type: Safety Factor  
Unit: ul  
3/21/2016, 1:42:40 PM  
15 Max

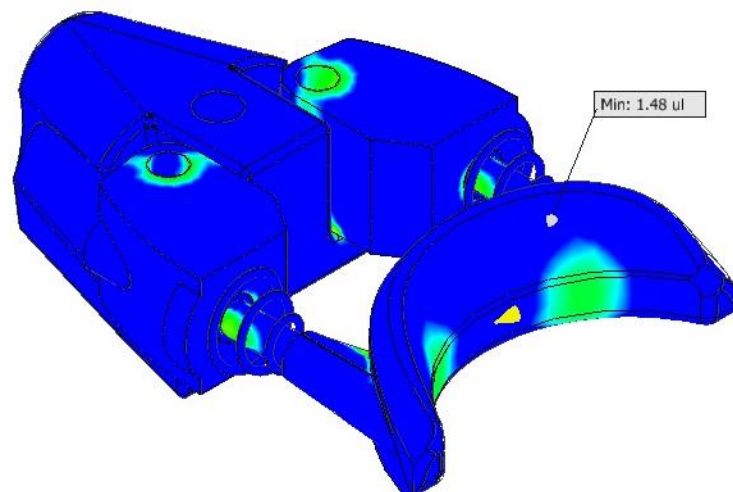
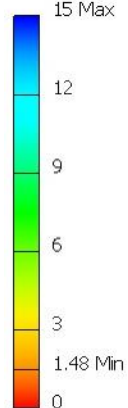


Figure 10: FEA Showing the Safety Factors for the Toe Piece

The maximum Von Mises stress encountered in the toe piece was 24.52ksi, which corresponded with a minimum safety factor of 1.48. This was deemed acceptable for the design, as the loads applied were the maximum allowable per the release limits calculated using ASTM F939-12.

### 2.3.6 Modal Analysis of Redesigned Ski Binding

In addition to FEA, modal analysis was performed to calculate the natural frequency of binding heel and toe pieces. Due to the complex geometry of the system, modal analysis results produced within CAD environments were unstable. To remedy this, the natural frequencies of the bindings were calculated by hand, simplifying the assemblies down to basic spring-mass systems. Using the equation for natural frequency, given as:

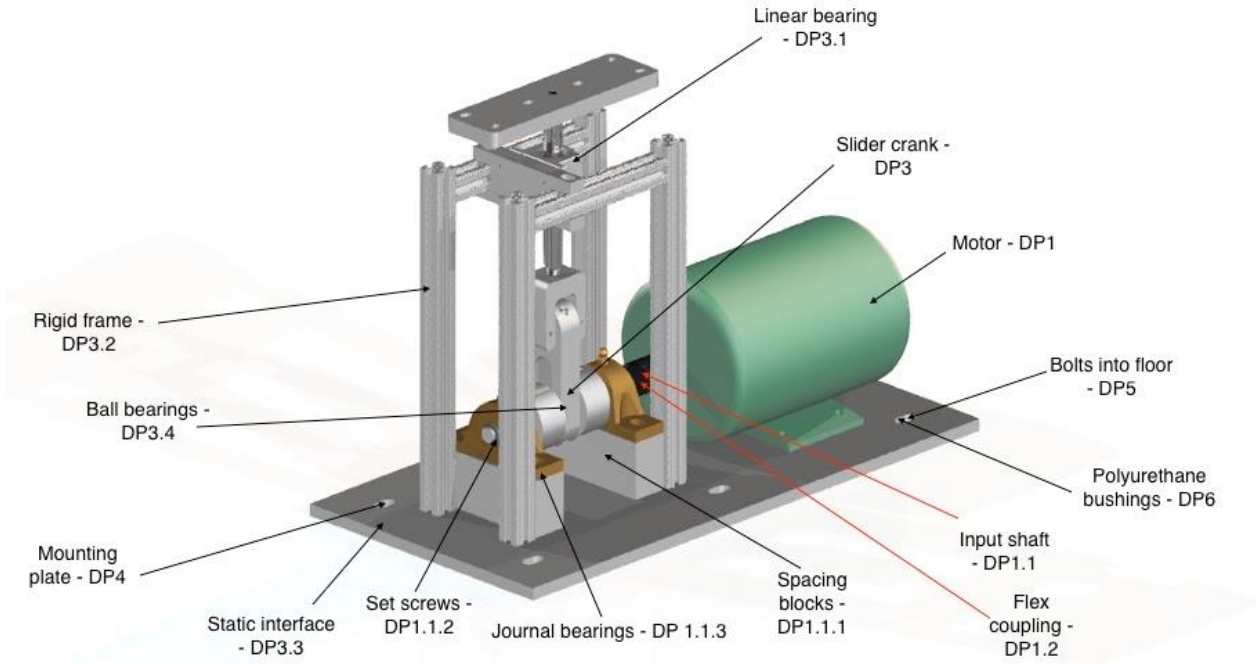
$$f_n = \frac{1}{2} * \pi * \sqrt{\frac{k}{m}}$$

the fundamental frequency for the toe piece was calculated as 64.838Hz, and the fundamental frequency for the heel piece was calculated at 60.083Hz. As the natural frequency for the bindings exceeded the expected resonance for mode 1 oscillation in the ski, which was calculated as 33.61Hz using a CAD modal analysis environment, the binding response would be conducive to the elimination of inadvertent release.

## 3 Physical Implementation

### 3.1 Vibration Test Machine

Once the final functional decomposition and design matrix were prepared, we were able to practically implement them using CAD. Figure 11 shows the completed design with each component represented by its DP. The drawings of each individual component of the machine can be found in Appendix 1, Technical Drawings of Machined Parts.



**Figure 11: Vibration Test Machine Final Design**

### 3.2 Redesigned Ski Binding

The CAD design for the redesigned ski binding was obtained in a similar fashion. As seen in the complete heelpiece and toe piece renderings (Figure 12 and Figure 13, respectively), each part of the design consists of a DP that satisfies the corresponding FR. All levels of DP2 and DP4 are applied to the heelpiece, whereas DP3 and DP5 are implemented in the toe piece. DP1 is a part of the cups of both pieces. The drawings of the individual components of both the heel and toe piece can be found in Appendix 1, Alpine Ski Binding.

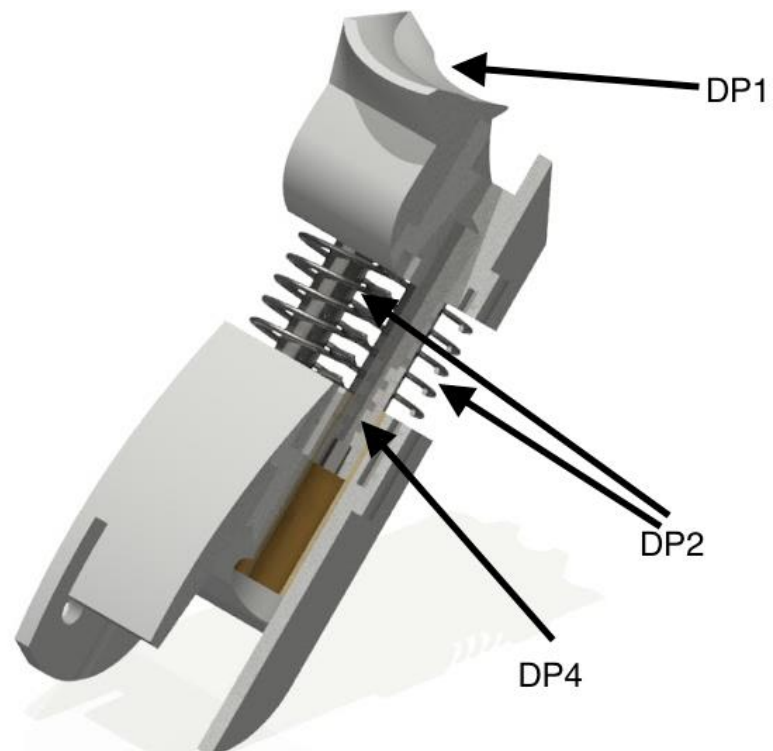


Figure 12: Heelpiece Final Design

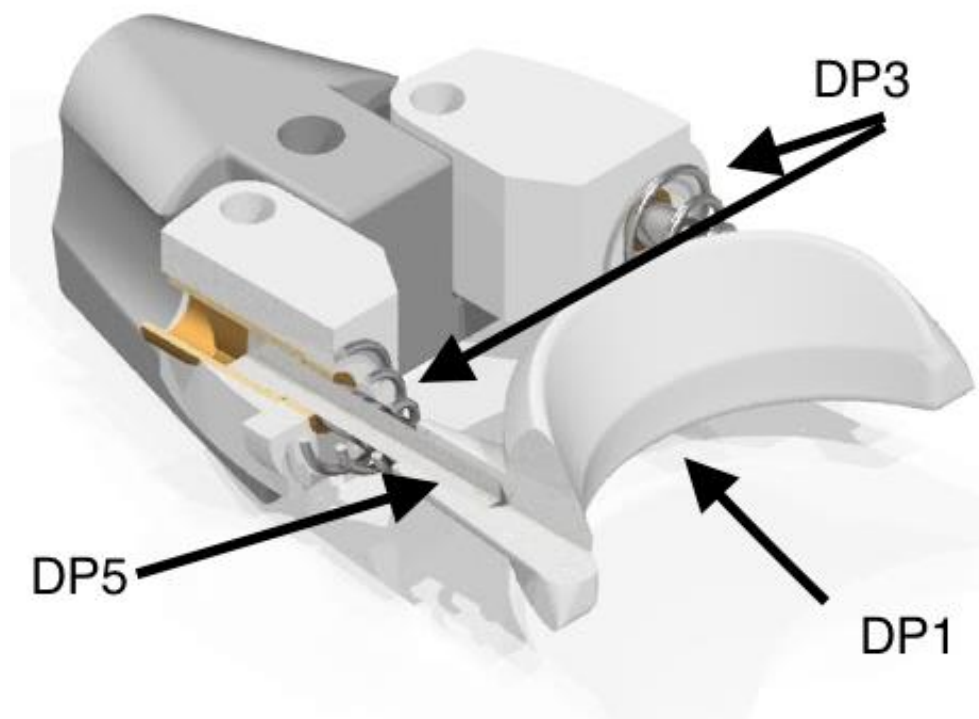


Figure 13: Toe Piece Final Design

## 4 Manufacturing

### 4.1 Vibration Ski Testing

#### 4.1.1 CAD Software (Rendering)

The CAD rendering of the complete vibration machine assembly is shown below in Figure 14.

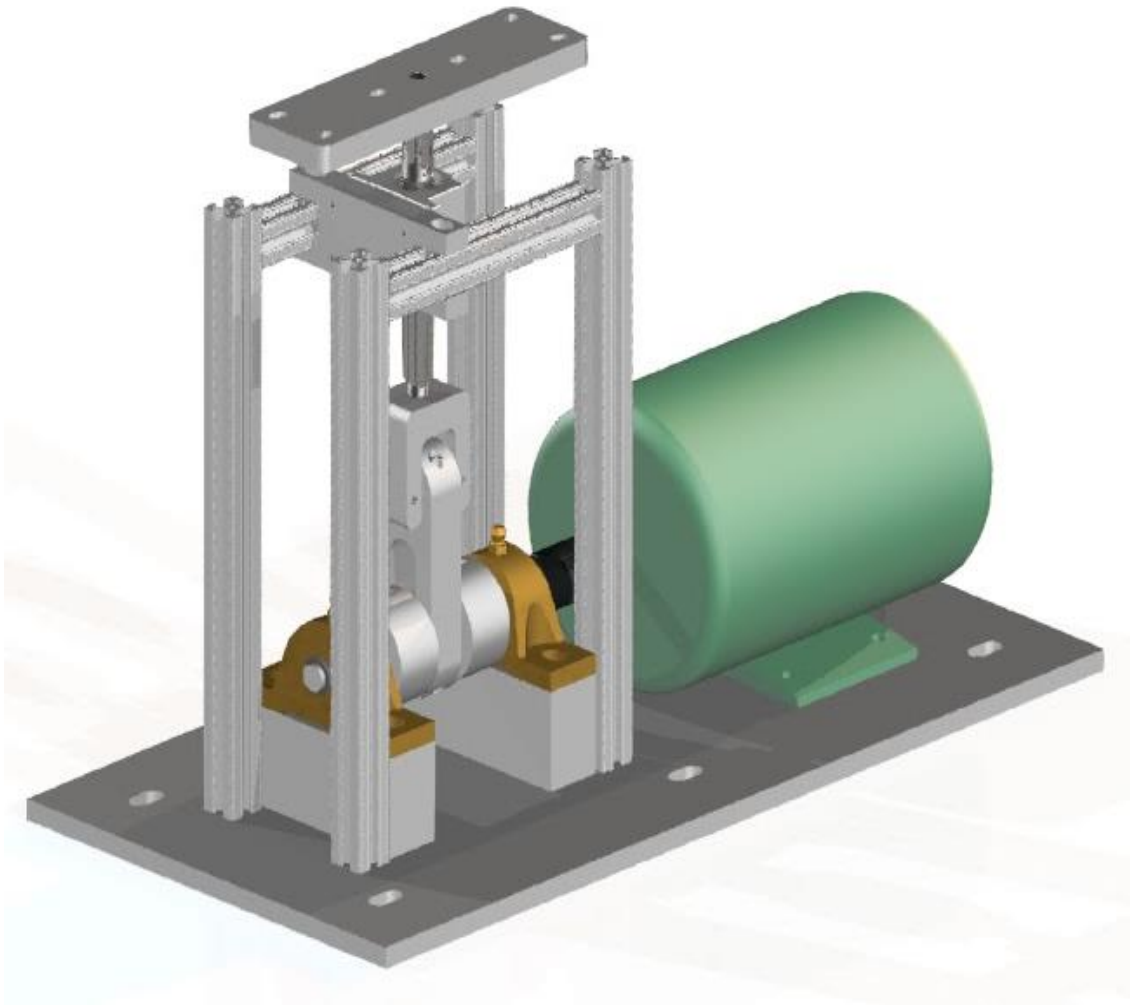


Figure 14: Completed Design of Vibration Testing Machine

#### 4.1.2 CNC Machining (Mini Mill & VM-2)

To machine the custom parts designed for the vibration machine, standard 3-axis CNC milling was implemented. Using the solid models for the parts, CAM software was used to create the necessary toolpaths for the milling operations. All toolpaths used standard tooling available in the Washburn Shops facilities. Tooling used for specific parts is

detailed in Appendix 2, Table of Tooling Used. All machining, except for the mounting plate, was done using the HAAS Mini Mills. All square stock was fixtured in the standard machining vice installed in the mill. Round stock parts were secured using the vice in conjunction with a V-block. Due to the large size of the mounting plate, machining for this part was conducted in the VM-2 vertical machining center. Machine straps and spacer blocks were used to fixture the large aluminum plate onto the table of the machine tool.

#### **4.1.3 Unguarded Machinery**

Unguarded machining, as well as hand tools were used to manufacture the shafts used in the rotary linkage, as well as to construct the extruded aluminum frame. These parts were cut to length using a vertical band saw, and sharp edges of the parts were broken using a deburring wheel. The faces of the aluminum extrusions were smoothed using a powered belt sander. The center hole of the aluminum extrusion was then threaded using a manual tapping station.

#### **4.1.4 Final Assembly**

In the first iteration of the connector rod, the circular bearings were placed between the connector rod and connector rod end clamps, which were then secured together using socket head cap screws. Due to machining errors, the end clamps did not sit flush with the face of the connector rod, causing the bearings to be misaligned. In the second iteration, the connector rod was re-machined as one part, eliminating the possibility of misalignment, and the ball bearings were inserted into their seats into it using the arbor press. The press was also used to push the corresponding rotary shafts through the connector rod, the journal bearings, and the crank mechanism. Once this subassembly was complete, the rest of the machine was assembled around it as shown in the CAD rendering. Figure 15 below shows the manufactured design of the vibration tester machine.



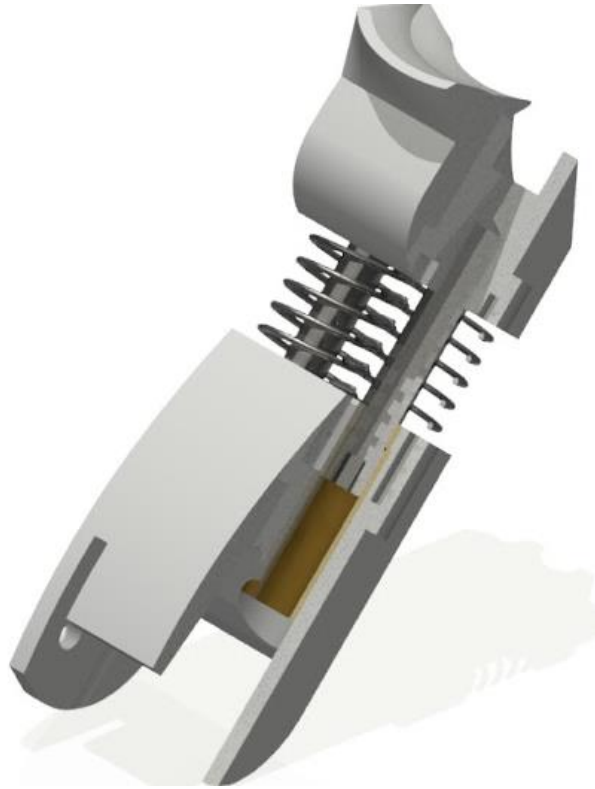


**Figure 15: Completed Vibration Testing Machine**

## **4.2 Ski Binding Production**

### **4.2.1 CAD Software (Rendering)**

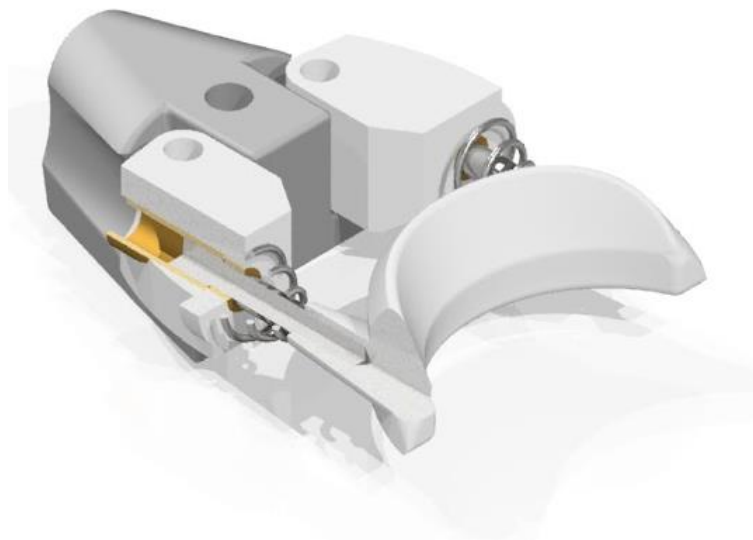
The CAD rendering for the heelpiece is shown below in Figure 16.



**Figure 16: Redesigned Heelpiece**

It is comprised of the pivot and the heel cup that are held together by a pair of pins. Each of the pins is encompassed by a coil spring.

The toe piece was separately rendered as shown below in Figure 17.



**Figure 17: Redesigned Toe Piece**

It consists of two spring bases on either side and the toe cup. The dark grey part depicts the original binding that was be modified to add the spring bases and toe cup. Each of the bases is connected to the cup by a pin, which is encompassed by a conical spring.

#### **4.2.2 3D Printing & CNC Machining (SST-1200 and ST-10)**

The two spring bases, heel cup, pivot and toe cup were produced using FDM rapid prototyping. ABS or acrylonitrile butadiene styrene plastic was used as the FDM material. The Dimension SST-1200 FDM printer located in Higgins Laboratories was used to produce the printed parts. The pins were machined in a HAAS ST-10 lathe using aluminum stock. Standard lathe tooling available in the Washburn Shops facility was used during the lathing operations, including a 55-degree lathing insert, and a cutoff tool. In total, four pins were machined. Two pins were produced for the toe piece, each with a stepped diameter of 5/16 and 3/8 inch. Similarly, two pins were produced for use in the heelpiece, each with a stepped diameter of 7/16 and 1/2 inch. The ends of the pins were then threaded for the appropriate thread size. To prepare for assembly, the holes in the 3D printed toe cup and heel cup were also threaded to complement the threading of the pins.

#### **4.2.3 Final Assembly**

Two coil springs and four brass bushings were ordered for the final assembly of the heelpiece. The holes in the pivot were lined with epoxy resin and the bushings were pressed into them using the arbor press. This held the bushings securely in place, and allowed the

pins to be pressed through them. The bushings were sized such that one bushing per pin would encompass the small diameter of the pin, and one bushing would encompass the larger diameter. The coil springs were inserted around the pins, and the threaded ends of pins were then driven into their respective threads in the heel cup.

The assembly process for the toe piece was nearly identical to that of the heelpiece, except conical springs were used in place of standard coils. Brass bushings were pressed into the holes in each of the spring bases, with epoxy to keep them in place, and the pins were pushed through the bushings. The conical springs were placed over the reverse side of the pins, which were then drive into the threaded holes in the toe cup.

Figure 18 below shows a photograph of the final redesigning binding prototype. This figure shows the heelpiece and toe piece installed on the ski that was used for testing.



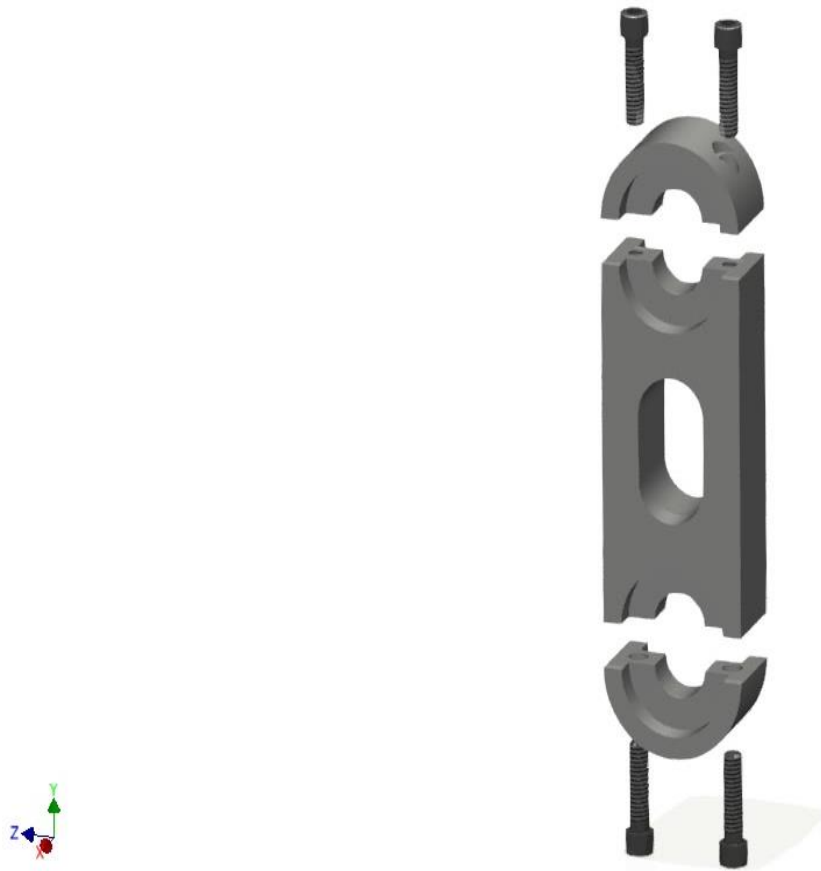
Figure 18: Redesigned Ski Binding

### 4.3 Component Iteration

#### 4.3.1 Linkage

Initially the vertical linkage of the vibration machine was designed using three separate pieces, where each end was fastened together to hold the bearings in their pocket. However, when these ends were cut, they were not perfectly flush with the middle link, causing the linkage to be misaligned with the axle running through the linear bearing. This created friction in the yoke and vertical shaft, making the forces too great for the motor to operate effectively. To fix this issue, the linkage was redesigned to be a single piece of aluminum with the bearings that are lightly press fit into pockets in the linkage. This allowed the

linkage to stay aligned to the vertical axle, and lessened the load on the motor. Figure 19 and Figure 20 show the first iteration and the final iteration, respectively.



**Figure 19: First Linkage Iteration**





Figure 20: Final Linkage Iteration

#### 4.3.2 Yoke

After machining the yoke, we discovered the threading for the vertical shaft was not aligned axially with the yoke, causing excessive friction and stress on the connecting pin of the linkage and yoke. For these reasons, the yoke was re-machined. The second production of this part yielded an acceptable alignment, allow for smooth operation of the vertical shaft.

#### 4.3.3 Linear Bearing

The original linear bearing used for our vertical shaft allowed for 2-degrees of axial misalignment total, in the event the shaft was not vertically aligned. However, this allowed the vertical shaft to rock back and forth within the linear bearing. When the machine was at maximum run speed, the rocking in the bearing created impact forces on the inner workings of the bearing, causing it to fail. Our solution was then to use a fixed, lubricated ceramic bearing to keep the vertical shaft from rocking and creating a smoother overall operating motion.

## 5 Testing of Final Design and Results

### 5.1 Testing Procedures

#### 5.1.1 Safety Precautions

In order to ensure that the use of the vibration testing machine was conducted safely, the team followed several important steps. The measures that were followed during testing are described in detail in the following sections below.

##### *5.1.1.1 Securing the Machine to Ground Plane*

With the help of the facilities department at WPI, holes were drilled into the solid concrete floor at the testing location in Kaven Hall 202, and threaded metal inserts were installed in them. The aluminum mounting plate attached to the base of the machine was then bolted to the floor through these holes, using high strength Grade 8 steel bolts. Polyurethane rubber dampers were placed between the plate and the floor to further to prevent vibrations generated by the machine from being transferred into the floor. These steps helped reduced the effects of the vibration on the components of the machine by providing a makeshift damper for the vibrations, and also significantly reduced the possibility of the machine breaking free by providing a secure attachment to a stationary ground plane.

##### *5.1.1.2 Nylon Paracord Rope*

To ensure the ski remained constrained in the event of release from the vibration machine, paracord rope was used to secure the ski to a stationary object separate from the machine itself. The tip and the tail of the ski were secured to the framework of a heavy-duty impact tester located directly behind the vibration machine mounting location. In case of binding release, the rope would prevent the ski from flying out of control by pulling the ski upwards and absorbing some of its energy. The rope was tied so that it was loose enough to allow the ski to vibrate freely but tight enough to pull it upwards in case of inadvertent release. The rope also prevented the ski from moving an exorbitant distance away from the vibration machine laterally, to ensure the safety of any lab users within the proximity of the machine.

##### *5.1.1.3 “B Netting”*

Once the machine was fully assembled and set up, the testing area was surrounded by “B Netting”, a safety net commonly used during alpine ski races as a safety precaution. The



netting would act as a failsafe to prevent the ski from flying out of control in the event of inadvertent release. This would also protect the team from projectile components thrown out by the vibration machine in the event of machine failure.

#### **5.1.1.4 *Standard Safety Precautions***

During the testing process, the general safety procedures for a power-driven machine were observed. All team members wore safety goggles at all times. While the machine was operating, all members stood to the sides, rather than directly in front of the vibrating machine, to prevent injury in the event components were ejected from the machine. In addition, the machine was operated from a safe distance with the frequency drive placed at a distance to the side of the apparatus.

#### **5.1.2 *Equipment Setup***

The slow motion videos showing the effects of chatter during normal skiing suggested that mode 1 vibrations could be a significant factor in this phenomenon. By mounting the ski at the center and vibrating it, the machine would be able to induce mode 1 vibration in the shape of a half sine curve, as shown in Figure 21 below. Using the high-speed video camera system, we examined the behavior of the ski in ultra slow motion, thus enabling us to evaluate the effect of mode 1 vibration on the ski bindings by taking measurements of displacement and frequency in the ski binding.

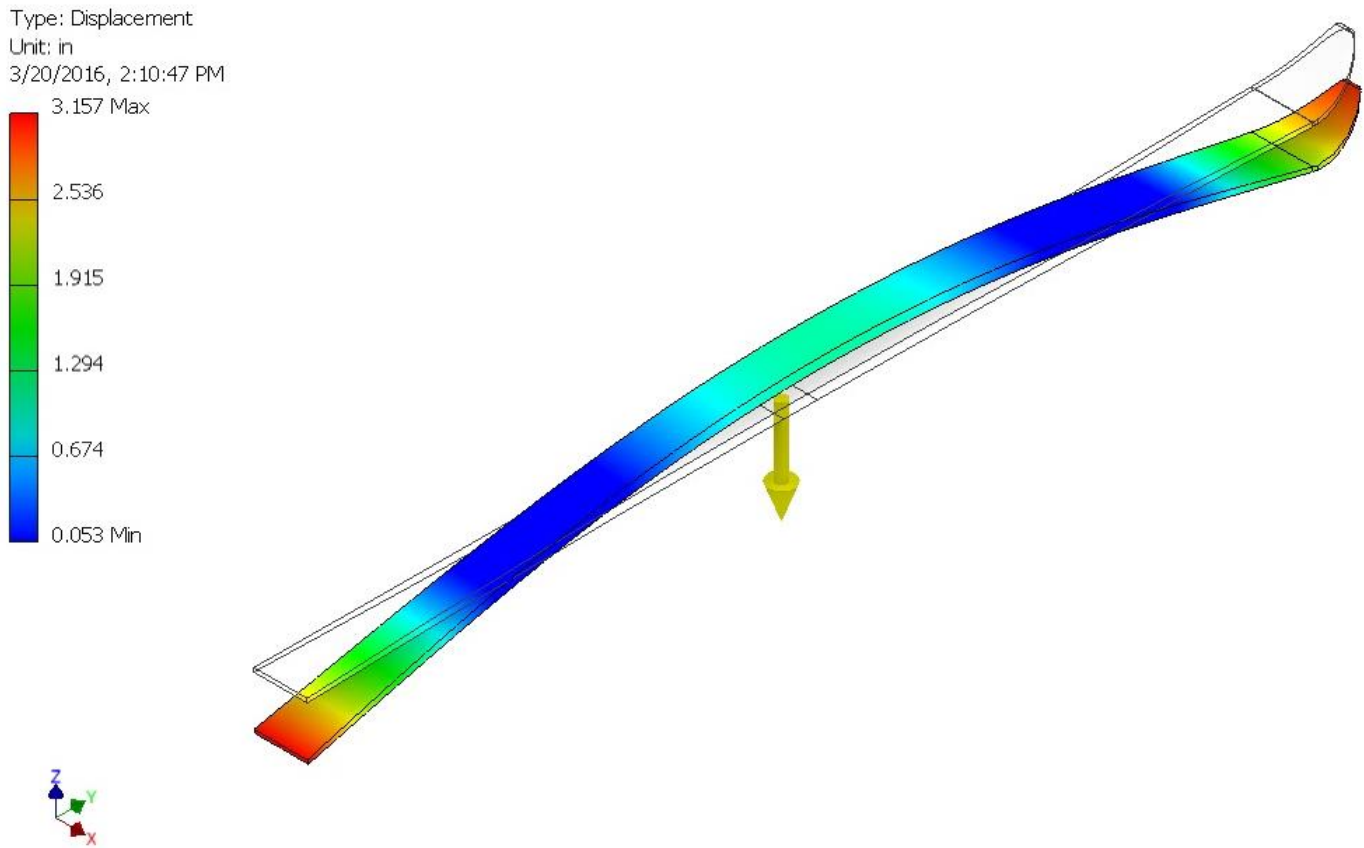


Figure 21: Vibration Mode 1 on Ski

The controller, which was to be used to vary the frequency of the motor, was placed outside the ski netting to the side of the machine. The high-speed camera was placed facing the machine, and a video light mounted on a tripod was used for illumination. During the full load test, a scale was placed directly behind the mounted ski parallel to the machine and perpendicular to the line of sight of the camera so that reliable displacement measurements could be obtained from the slow-motion videos.

### 5.1.3 Testing Process

#### 5.1.3.1 No Load

Initially, the machine was run without the test sole adapter attached to the vertical linkage; in order evaluate the operation of the machine with no load applied. The motor was used to drive the machine, starting at a frequency of 5Hz and gradually increasing to 40Hz. The high-speed camera was used to take video of the apparatus, and the corresponding actual

machine operating frequencies were calculated from these videos at regular intervals. As expected, these frequencies were lower than the controller frequency, but within a reasonable margin or error. The maximum frequency reached by the machine was 36.7Hz at a controller frequency of 40Hz. The no-load expected frequency of the machine at a control frequency of 40Hz is 38.33Hz, giving a total error of 4.26%. This process was repeated after modifications were made to the machine.

#### *5.1.3.2 Dry Run*

After the no load experiments, the test sole adapter and test sole were attached to the vertical linkage. The machine was operated at increasing frequencies again, and the actual machine operating frequencies were calculated from slow-motion videos. Once again, the machine frequencies were found to be lower than the input frequency from the controller, but the deficit remained within a reasonable margin. The maximum machine frequency was largely unchanged from the no load run. This was repeated after modifications were made to the machine.

#### *5.1.3.3 Full Load*

After safe operation of the machine was confirmed through several dry runs, a ski and binding were mounted on the test sole adapter. Rossignol 9X Pro skis and Salomon Model 900S bindings were used for this purpose, as this combination of ski and binding was modified in the construction of the binding prototype. Operating the machine at full load led to a significant drop-off in the machine operating frequency as compared to the controller frequency. This was in part due to the additional weight of the ski, which was approximately 12lbs. The uneven distribution of weight between the tip and the tail of the ski also caused a force normal to the vertical axis to be applied to the linear bearing, creating excessive friction, and making it difficult for the motor to drive the machine. This was counteracted by securing the tip of the ski while the machine was operating, thus eliminating the normal force. Following this adjustment, videos of the vibrating ski were taken for controller frequencies between 20 and 40 Hz, which spanned the expected natural frequency of the ski of 33.61Hz. The videos were then used to find the natural frequency of the ski, and the displacement at which it occurred. The maximum machine frequency was found to be 17.71 Hz at a controller frequency of 32 Hz. The following section further expands on the data collected from these videos.

#### 5.1.4 Results of Evaluation

##### 5.1.4.1 Stills From High Speed Camera

Figure 22 and Figure 23 below are a series of still photographs taken from the high-speed camera program used to gather the data for the project. The first two stills show two parts of a slow motion video of the ski being shaken by the vibration test machine. The top picture shows the maximum deflection of the ski while the bottom picture shows the minimum deflection.

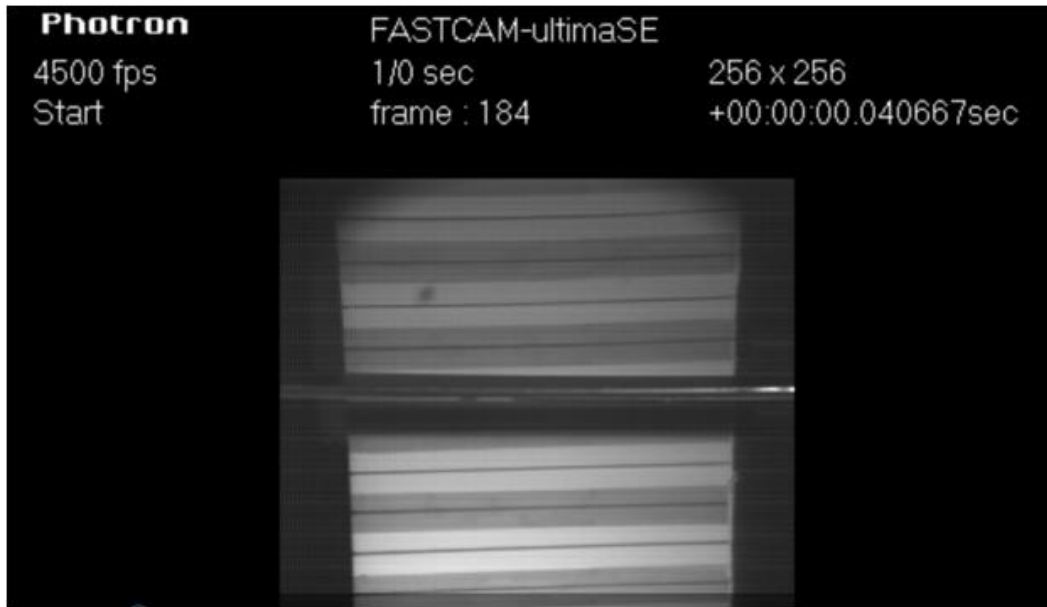


Figure 22: Maximum Deflection of Ski



Figure 23: Minimum Deflection of Ski

#### 5.1.4.2 Displacement Tables and Plots

The data collected during the full load testing is represented below in tabular form (Table 1), as well as graphical form (Figure 24 and Figure 25). The experiment was carried out for a total of 21 runs, with frequencies ranging from 20Hz to 40Hz. Both the frequency and displacement results showed a great deal of fluctuation as seen in the graphs, however, a peak of 17.71 Hz can be seen in the Experimental Frequency vs. Controller Frequency graph. This was found to be the natural frequency of the ski, at a corresponding displacement of 2.25 inches. The fluctuation can be attributed to various factors. Chief among these was the hold-down force securing the tip of the ski, which was not constant for all of the runs. In addition, the discrepancies between controller and experimental frequencies also caused inconstancy in the results.

**Table 1: Data Collected from Frequency Testing**

Run	Controller Frequency (Hz)	Experimental Frequency (Hz)	Displacement (in)
1	20	11.42	2
2	21	12.47	2.1
3	22	10.98	1.9
4	23	10.56	2.3
5	24	14.71	2.3
6	25	10.32	1.65
7	26	12.23	1.75
8	27	11.78	1.7
9	28	12.03	1.775
10	29	10.98	1.7
11	30	11.6	2.25
12	31	12.86	2.25
13	32	17.71	2.25
14	33	11.34	2
15	34	12.3	2.1
16	35	12.47	2.15
17	36	11.65	2.5
18	37	11.25	2.25
19	38	12.09	2.5
20	39	12.5	2.625
21	40	11.45	2.25

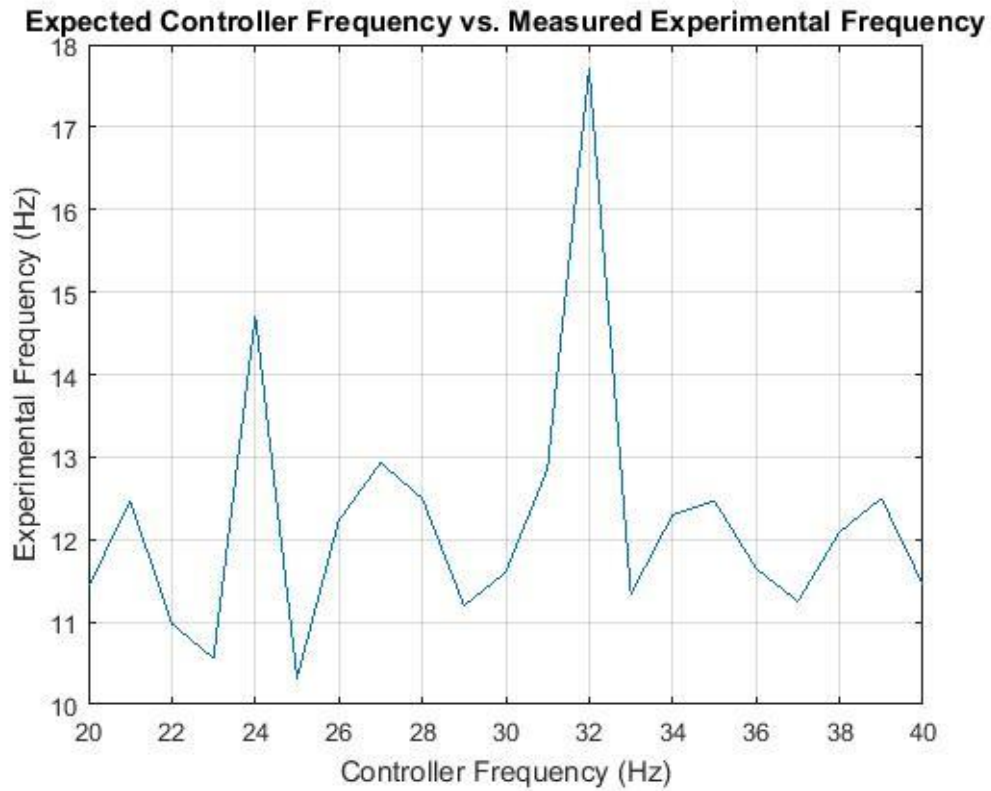


Figure 24: Expected Controller Frequency vs. Measured Experimental Frequency

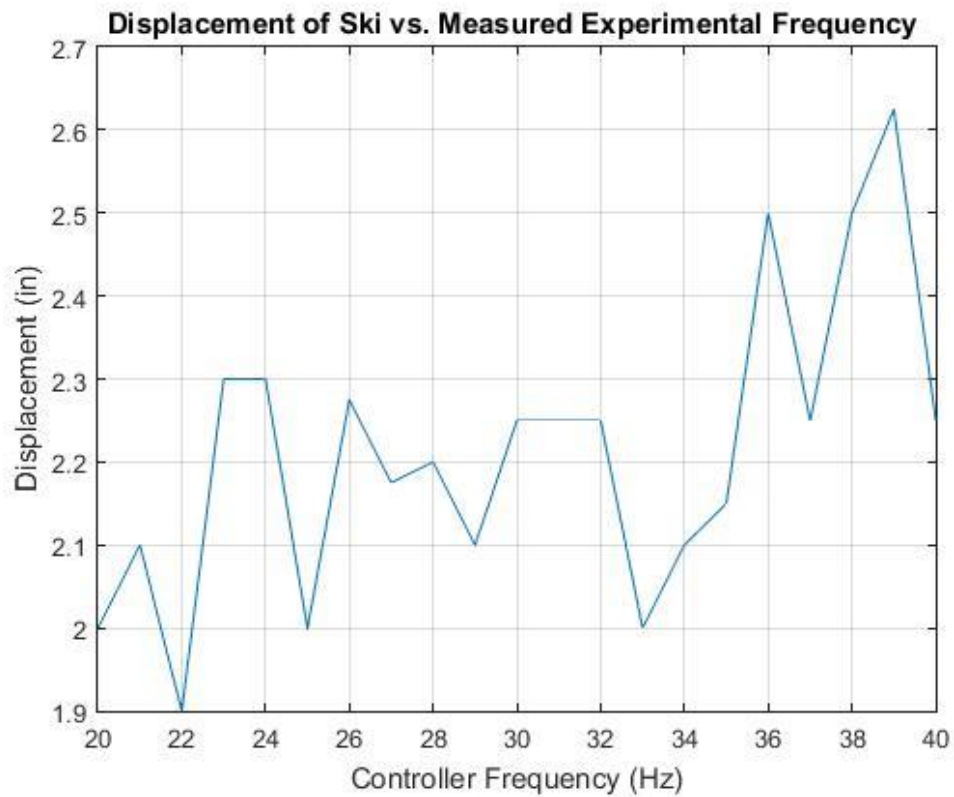


Figure 25: Displacement of Ski vs. Measured Experimental Frequency

## 6 Discussion

### 6.1 Evaluation of the Design Process

By using axiomatic design, the team was able to minimize the number of redundant iterations needed to optimize the design for both the vibration machine and the binding. Aside from minor changes (outlined in Section 4.3, Component Iterations), only one major iteration was needed. In hindsight, some of the issues that arose during testing or assembly could have been avoided by using axiomatic design principles more effectively to identify all of these potential issues during the design process. According to axiom number one, the FRs must be completely independent of one another. As a result, the DPs used to fulfill these FRs must also be completely independent. This should allow the designer to identify and counteract any coupling beforehand. During the formulation of the design matrix for the vibration machine, the team was not able to identify the coupling between the shaft and the connector rod due to the fact that the connector rod was a combination of three distinct pieces. Therefore, it should have been divided into three separate FRs with distinct DPs. However, we were still able to make use of axiomatic design principles by revisiting the original design matrix whenever this problem arose, which allowed us to make modifications easily without affecting the functioning of the design in any unexpected way. As described above, this issue was countered by redesigning the connector rod as one distinct piece to complement the design matrix.

### 6.2 Accomplishments

#### 6.2.1 Created a Ski Binding Testing Machine

We developed an operational ski vibration machine using a cam system that would raise and lower a test sole adapter 0.9 inches vertically at approximately 36Hz. With a ski attached, the system could reach approximately 17Hz with a max displacement of about 2.625 inches at the tip of a ski. The vibration machine is powered by a 1HP three phase AC motor that is controlled by a 1HP variable frequency/phase controller. This allowed us to test the shaker at any frequency between 0.1Hz and 200Hz, providing a wide range of data available to collect.

### **6.2.2 Developed a Representational Model of Rapid Response Ski Binding Patent**

Keeping in line with our key objective, we were able to use Prof. Brown's Patent as a model to design a ski binding which possesses the potential to eliminate inadvertent release due to chatter. The drawings and physical information included in the patent formed the basis for the concept of using springs to offset displacement due to chatter, and pins to transmit loads to retain normal binding function. By testing the CAD simulation for the design, we confirmed that the natural frequency increased significantly in comparison to the original conventional binding. Using 3D printed ABS plastic, aluminum pins, and coil springs, we manufactured a working representational model that can demonstrate the changed function of the binding.

## **6.3 Critical Assessment of Design Method**

### **6.3.1 Satisfaction of the Objective**

The objectives we created at the beginning of this project were not fully met due to unforeseen issues with the vibration testing machine, as well as material strength constraints of the created binding. Our primary objective was to reduce inadvertent release in alpine ski bindings caused by vibrations encountered in the ski binding and ski. We were able to create a representational model of the binding patent filed by Professor Brown. Due to the material of the binding being 3D printed ABS plastic, it was incapable of withstand testing forces. This prevented us from being able to test the effectiveness of the binding in a vibrational situation.

Our secondary objective for this project was to provide data pertaining to the relationship between chatter and inadvertent release. Unfortunately our vibration testing machine was not powerful enough to recreate the vibration necessary to prove the relationship in our testing. We were able to create a table and graphs showing the displacement and frequency of a conventional ski and binding ranging from 20Hz to 40Hz on our vibration testing machine. The highest frequency of the ski recorded in that testing was 17.71Hz with the highest displacement being 2.625 inches.

### **6.3.2 Results and Satisfaction of the Constraints**

This project satisfied many of the constraints that were set in the beginning stages of the design. The first constraint was to create a prototype that must cost no more than the



allotted budget. This constraint was not met, as the budget needed to be expanded in order to buy the aluminum stock to create the second iteration of the linkage and also to get the redesigned ski binding printed. Our second constraint states parts machined for the vibration machine must be made of aluminum. This constraint exists because it would not be possible to machine the parts if a stronger material was used based on the machinery that was available to use in Washburn Shops. The final design of the vibration machine consisted of only aluminum parts in accordance to this constraint. Another constraint for this project was that the parts for the vibration testing machine must be created using CNC machining. We met this constraint as the HAAS Mini Mill and ST-10 lathe were used to machine the components. There was a functioning constraint that stated that besides the motor, the machine must be purely mechanical. The final design implements only mechanical components, with the exception of the AC motor used to drive the machine.

### **6.3.3 Impact of Solution**

When this project is successfully developed and a working binding prototype is produced, there will be meaningful global, economic, and societal impacts. Alpine skiing is a popular sport with many people participating in both leisure skiing and also competitive downhill skiing. Current bindings do not take into account displacement that is needed in the toe piece in order to hold the boot within the binding during chatter, or the flexion of the ski while turning. Inadvertent release affects many people who participate in this sport because of the fact that the boot is able to release from the binding. This binding would have societal impacts because it would help to make the sport safer by preventing severe injury as result of inadvertent release.

This ski binding also has the potential to have an economic impact if it is proven to work effectively. With the proper support of both professional and amateur skiers, the binding has the potential to be a commercial success.

### **6.3.4 Future Work and Recommendations**

#### **6.3.4.1 Creation of Current Binding Model with Simulations**

A 3D conventional binding model along with the modal simulations, FEA and fatigue simulations should be created. These simulations can provide extensive data showing areas

of vulnerability that current bindings have in both design and choice of material, aiding in improved design decisions and testing procedures. The simulations could allow for savings in cost, time, and material, by allowing the ability to see failures in conventional bindings without physically destroying specimens.

#### ***6.3.4.2 Operating the Vibration Machine at High Frequency***

As discussed, the vibration machine that was created during this project was only able to reach a maximum frequency of 17.71Hz with the ski, whereas the frequency required for reproducing chatter conditions is closer to 33Hz. While the motor used in this design was rated at a nominal power frequency of 60Hz, and this frequency was approached without load on the machine, it was insufficient when a ski was mounted to the test sole. We believe that a significantly higher frequency could be reached if more power was provided to drive the machine. Furthermore, a considerable amount of power was lost due to friction between the machine parts. This was in part due to alignment issues. These could be remedied by using a more rigid frame, and ensuring it has a more precise alignment. Reducing the length of the connector rod and the overall frame height would also help eliminate friction from the system.

#### ***6.3.4.3 Achieving Inadvertent Release in a Conventional Binding***

As previously stated, the vibration machine induced mode 1 vibrations in the ski, but was unable to recreate chatter conditions. Hence, future work should take mode 2 oscillations into consideration, which induce an S-shaped curve in the ski and may play a critical role in the mechanism of inadvertent release due to chatter. An image of mode 2 vibration simulated in the CAD model of the ski can be found in Figure 26.

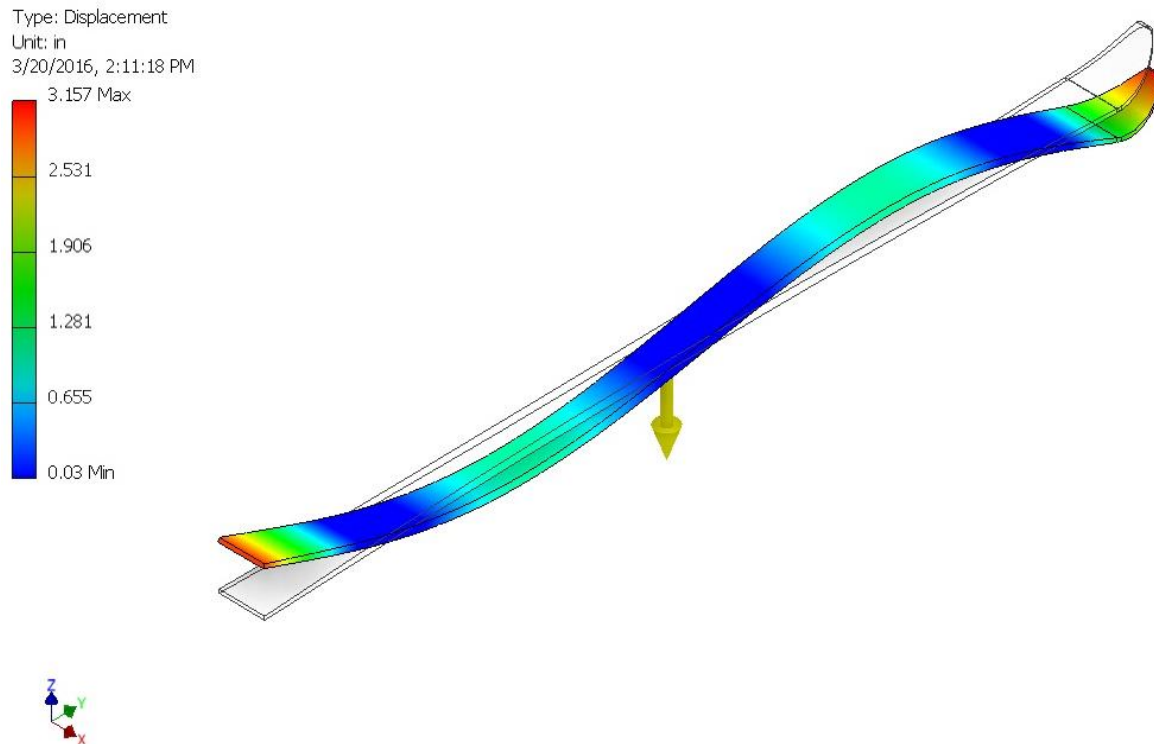


Figure 26: Vibration Mode 2 on Ski

An S-shaped oscillation could be induced by clamping the ski at the tip and tail rather than the center, and then vibrating asynchronously. This modification, coupled with the use of more power to drive the machine, could potentially reproduce chatter more accurately and induce inadvertent release.

#### 6.3.4.4 Testing the Redesigned Binding

Once chatter conditions are satisfactorily simulated, the next step is to obtain proof of concept for the redesigned binding. The binding should be vibrated at the frequency at which inadvertent release occurred for the conventional binding to prove that it can offset the effect of chatter and retain the boot.

#### 6.3.4.5 Developing a Functioning Prototype

Since the redesigned binding model created for this project would not be able to withstand testing, a more effective prototype is needed. By creating a prototype with stronger materials, such as injection-molded plastic, the binding could successfully be tested at the frequency at which inadvertent release occurs. CNC machining could also be used to machine a fully functional prototype out of aluminum.

## 6.4 Summary of What Was Learned

This project was a highly challenging and informative experience for the team. Progress was slow at the beginning as we tried to understand the principles of axiomatic design. We learned that when the principles were not properly followed, the design process could become disorganized and inefficient. As our understanding of axiomatic design grew, however, the process became and more strategic, and we were able to reach a complete design in a short period of time which enabled us to remain on schedule.

The team also gained valuable machining experience over the course of this project. While all the team members had basic machining background, the parts required for the vibration machine were more difficult to machine. The machining was accomplished with some guidance from the lab monitors at Washburn shops.

## 6.5 Concluding Remarks

### 6.5.1.1 Major Design Accomplishments

- Created a Ski Binding Testing Machine
- Collected data on displacement and frequency of vibration of conventional skis
- Developed a representational model of the Rapid Response Ski Binding Patent

### 6.5.1.2 Critical Assessment on Effectiveness of Design Method

- Early problems with the design resulted in the need for another iteration of connector rod
- The axiomatic design matrix allowed easy modification of the design for the resolution of this issue
- Friction was a greater issue than expected, and decreased the efficiency of the machine

### 6.5.1.3 Concluding Issues Remaining

- There were issues of alignment between the vertical shaft and linear bearing
- The linear bearing mounting frame was misaligned and allowed too much vibration
- There was a ¼ inch gap between coupler yokes, adding to required drive power
- The test sole adapter threaded hole for vertical shaft was misaligned
- Motor power was insufficient to overcome friction points in the system
- Strength of 3D printed ABS plastic binding was insufficient for testing conditions

## 7 References

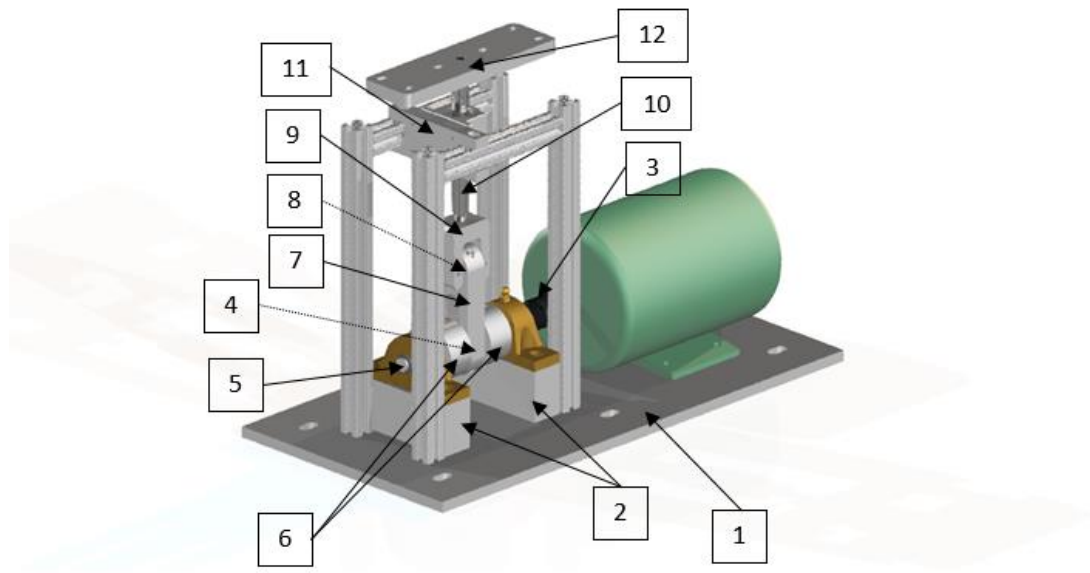
- ASTM F939. (2012). Standard Practice for Selection of Release Torque Values for Alpine Ski Bindings. ASTM Standards. Pennsylvania, United States of America: American Society for Testing and Materials.
- Brown, C., and Madura, J. 'Rapid Response Ski Binding'. 2014: n. pag. Print.
- Hans, P. 'Heel downholder for cableless safety ski bindings'. 1967: n. pag. Print
- Howell, R. 'Alpine ski binding heel unit'. 2015: n. pag. Print
- Hull, M. 'Method For Programmed Release In Ski Bindings'. 1983: n. pag. Print.
- Johnson, R. J., Ettlinger, C. F., & Shealy, J. E. (2009). Update on injury trends in alpine skiing. *J ASTM Int*, 5, 11-22.
- Lagran, M. (2015). An Introduction to Snow Sports Injuries and Safety. Retrieved: October 28, 2015, from: Ski Injury
- Senner, V., Michel, F. I., Lehner, S., & Brügger, O. (2013). Technical possibilities for optimising the ski-binding-boot functional unit to reduce knee injuries in recreational alpine skiing. *Sports Engineering*, 16(4), 211-228. doi:10.1007/s12283-013-0138-7
- Shealy, J. E., Ettlinger, C. F., & Johnson, R. J. (2005). Using signal detection theory as a model to evaluate release/retention criteria in alpine skiing. In *Journal of ASTM International* (Vol. 2, No. 7, pp. 1-12). ASTM International.
- Suh, N. (2001). *Axiomatic Design: Advances and Applications*. Oxford University Press.

## Appendix 1: Technical Drawings of Machined Parts

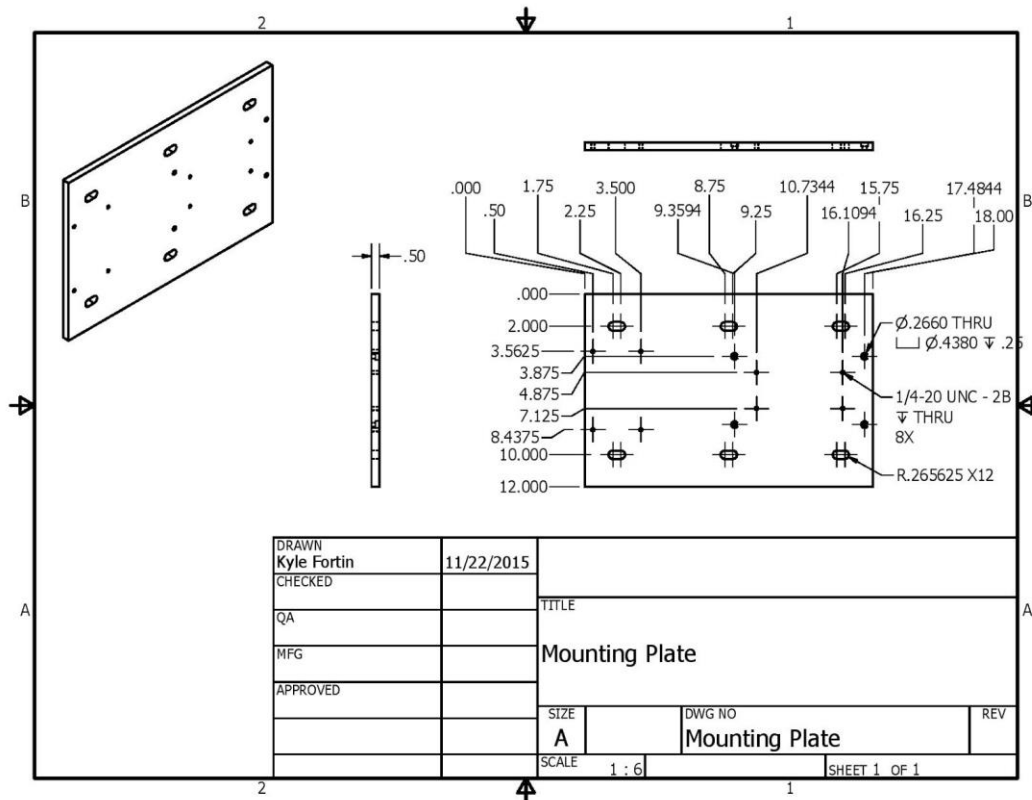
This appendix contains technical drawings of all parts that were machined at WPI for the completion of this MQP.

### Vibration Machine

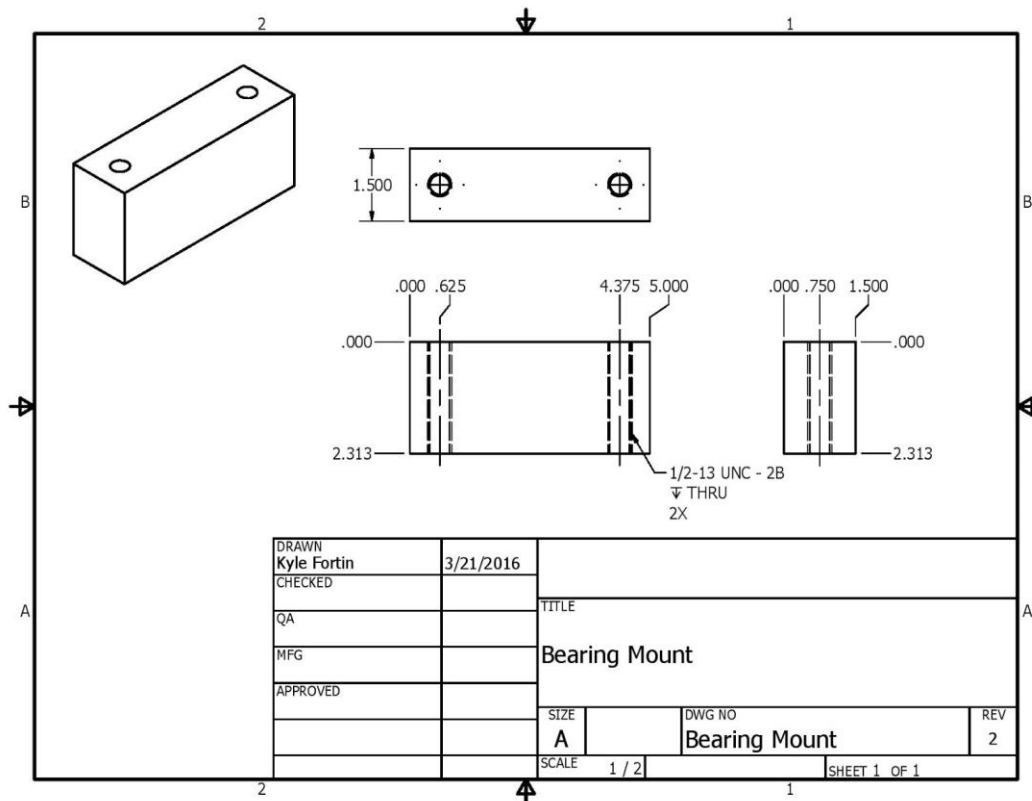
#### Parts Diagram



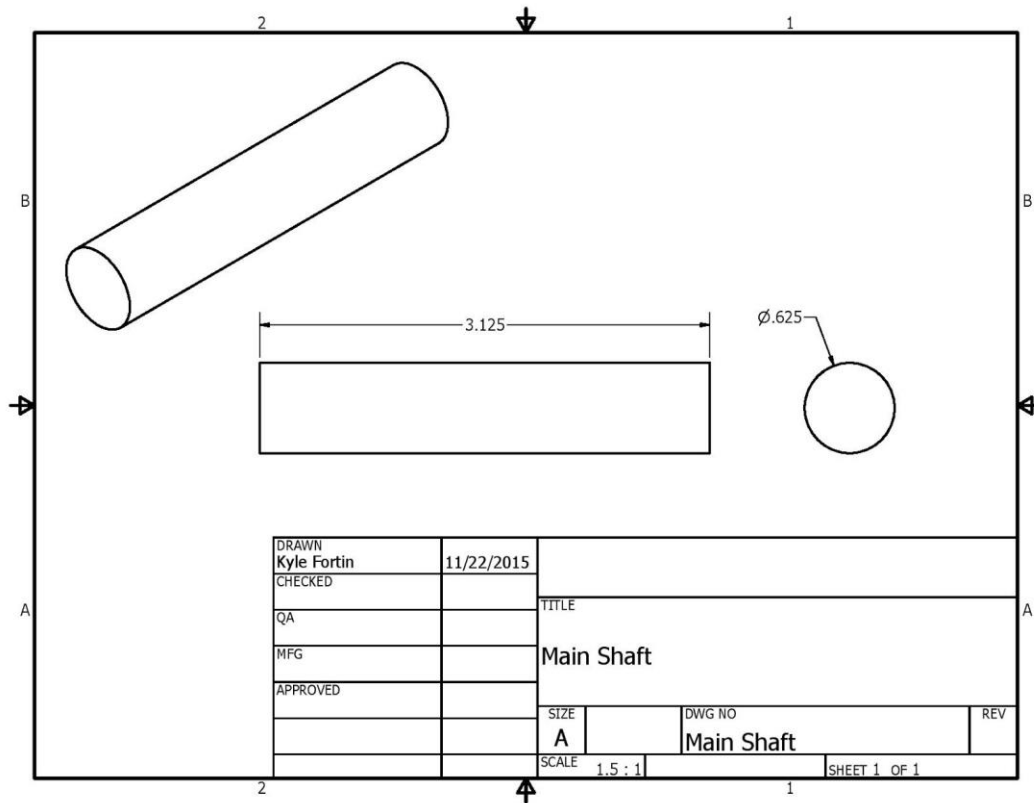
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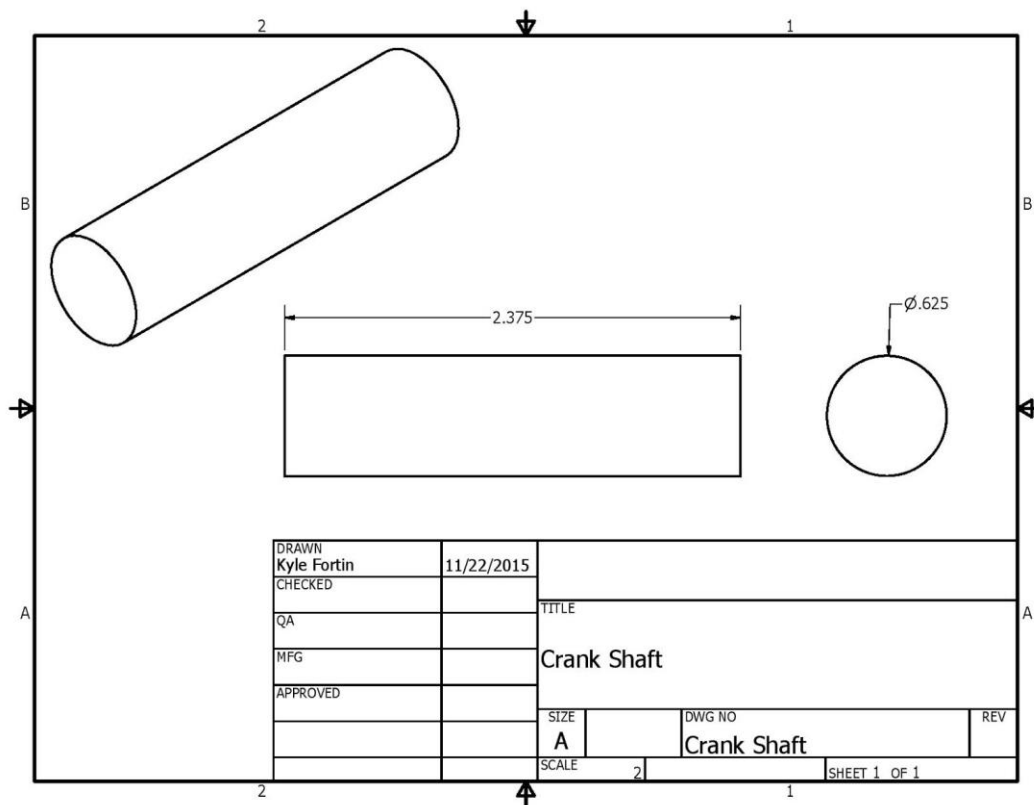
Bearing Mount (2)



Main Rotary Shaft (3)

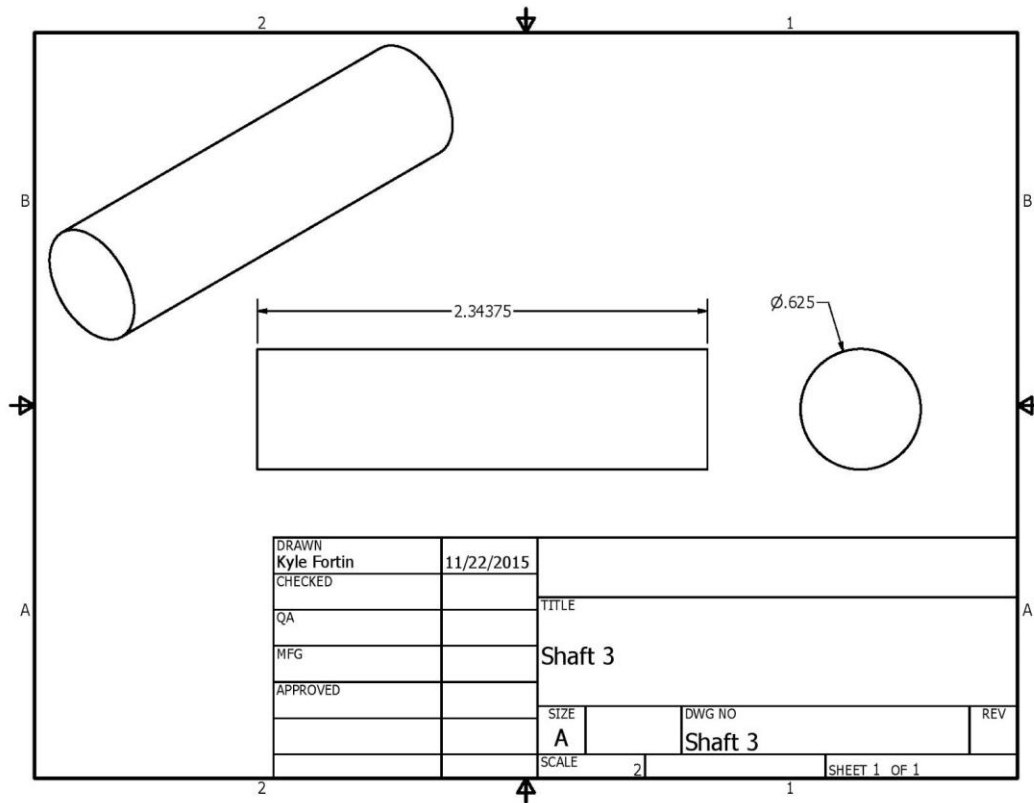


Crank Shaft (4)

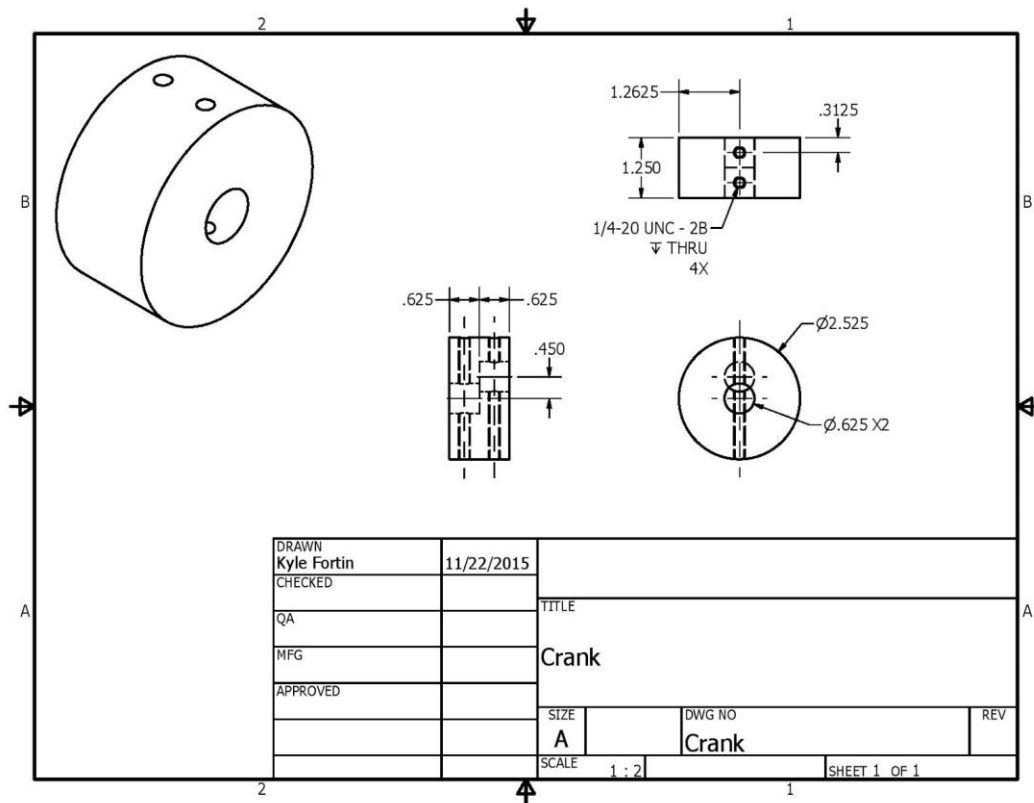




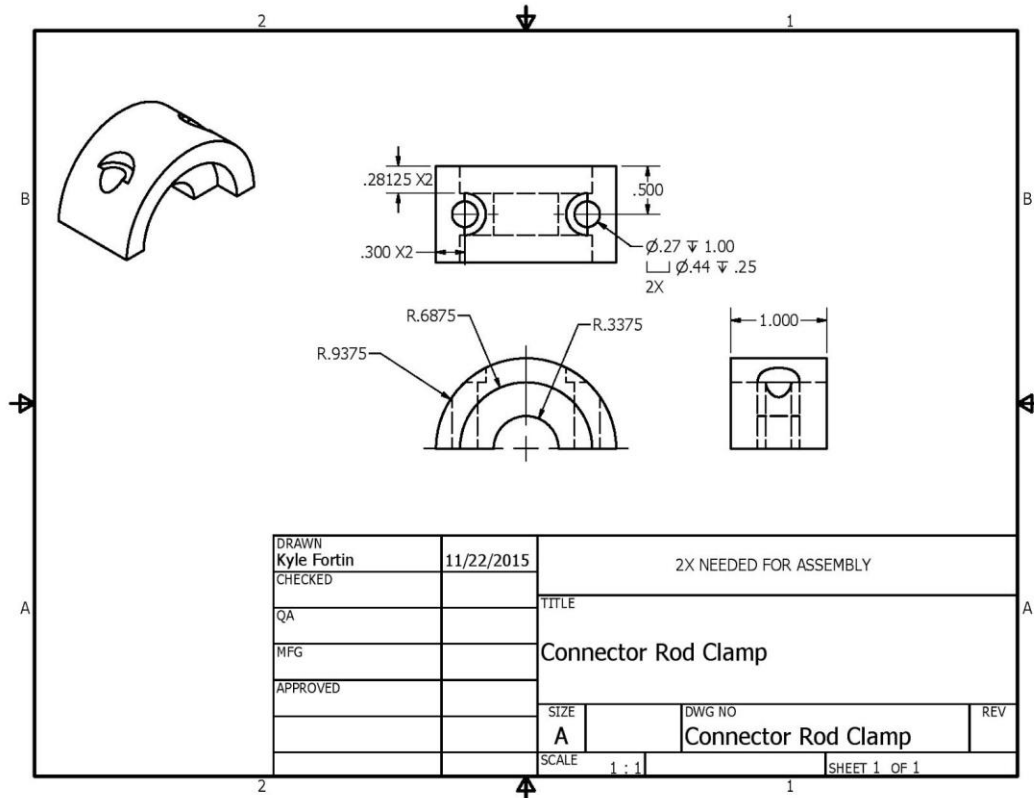
Tail Shaft (5)



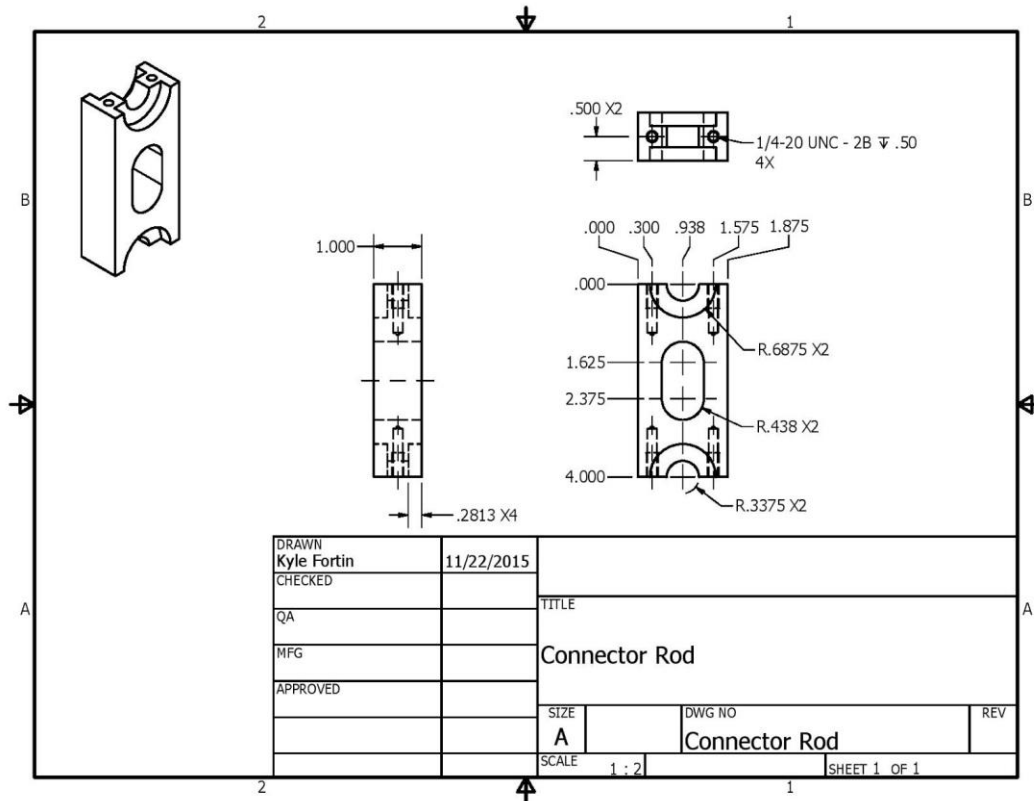
Crank (6)



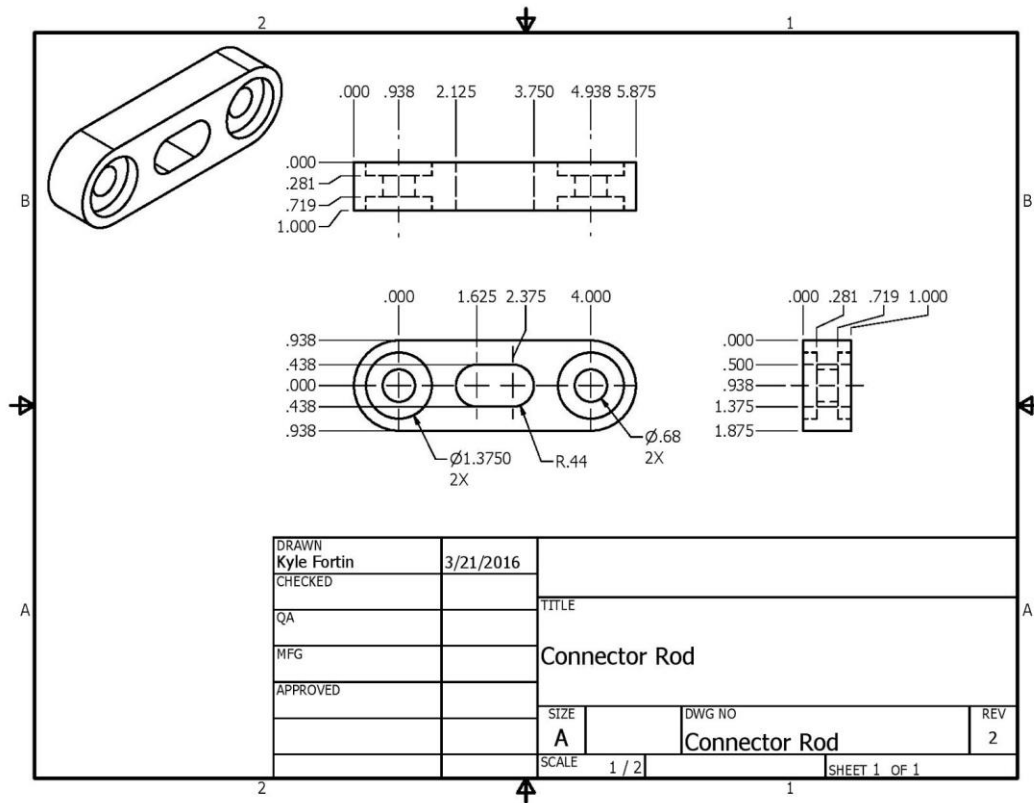
### Connector Rod End Clamp [Replaced by Connector Rod (Rev 2)]



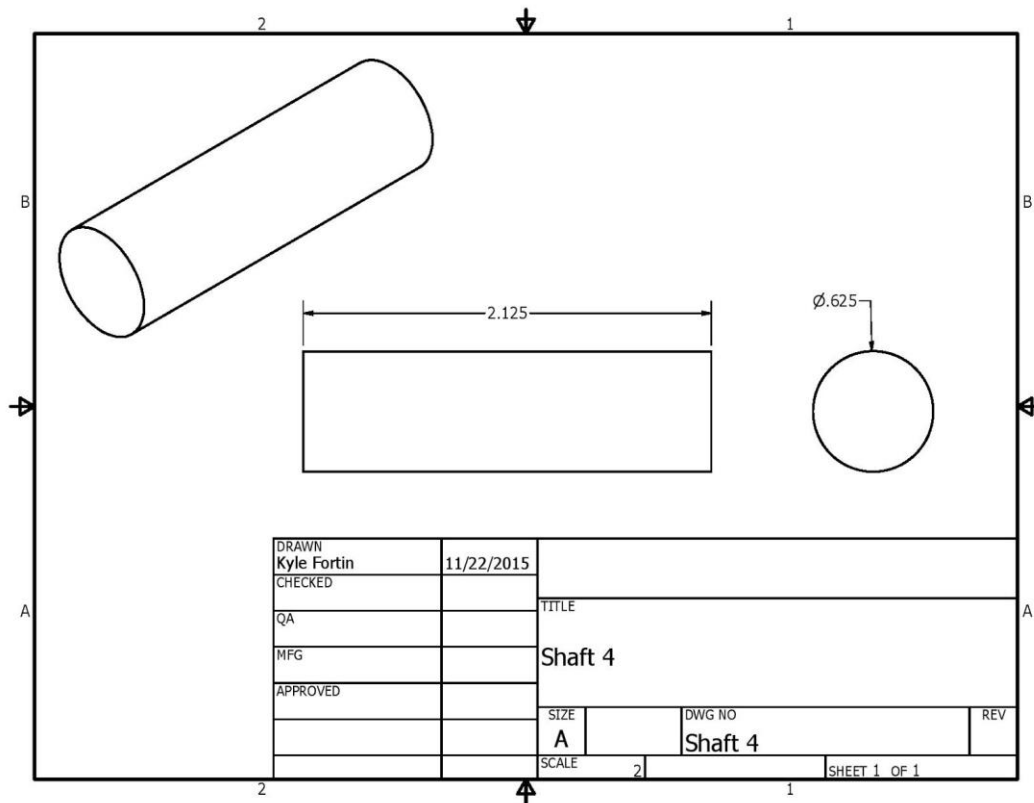
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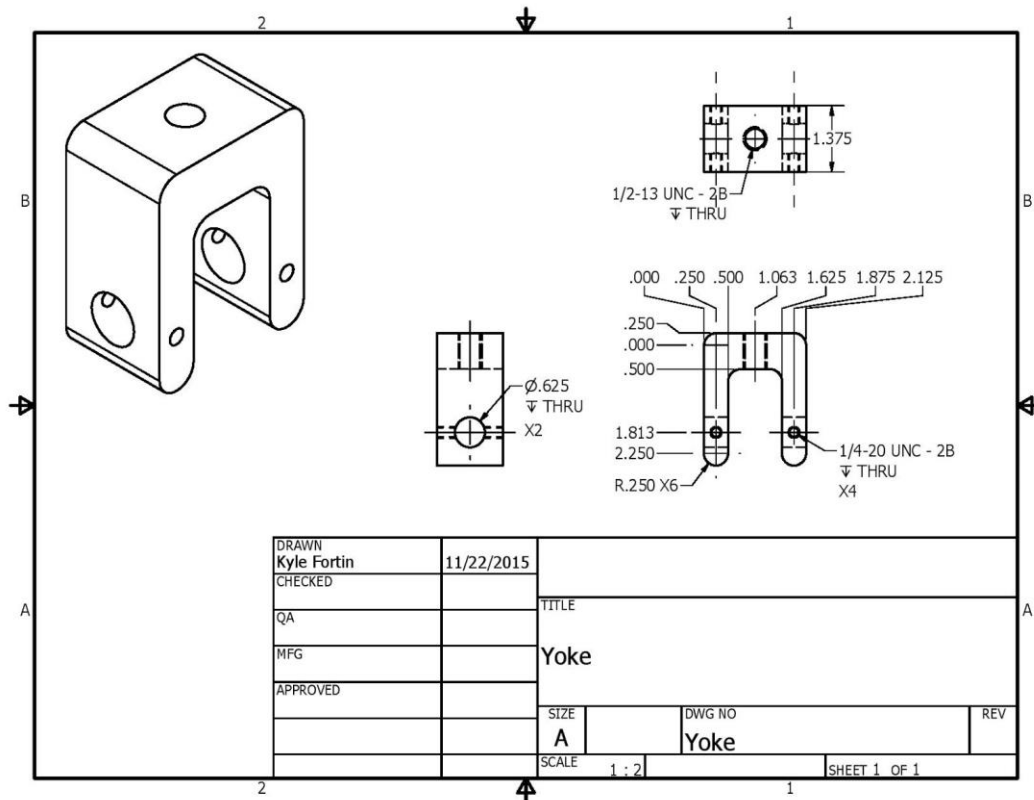
Connector Rod (Rev 2) (7)



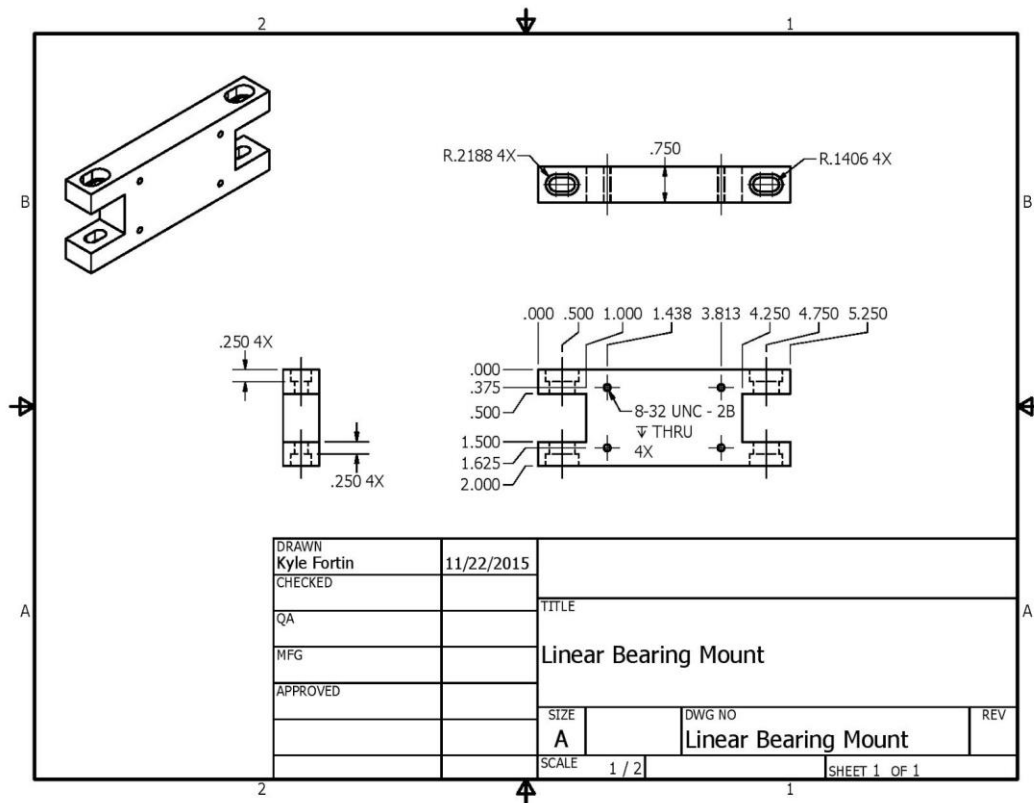
Upper Linkage Shaft (8)



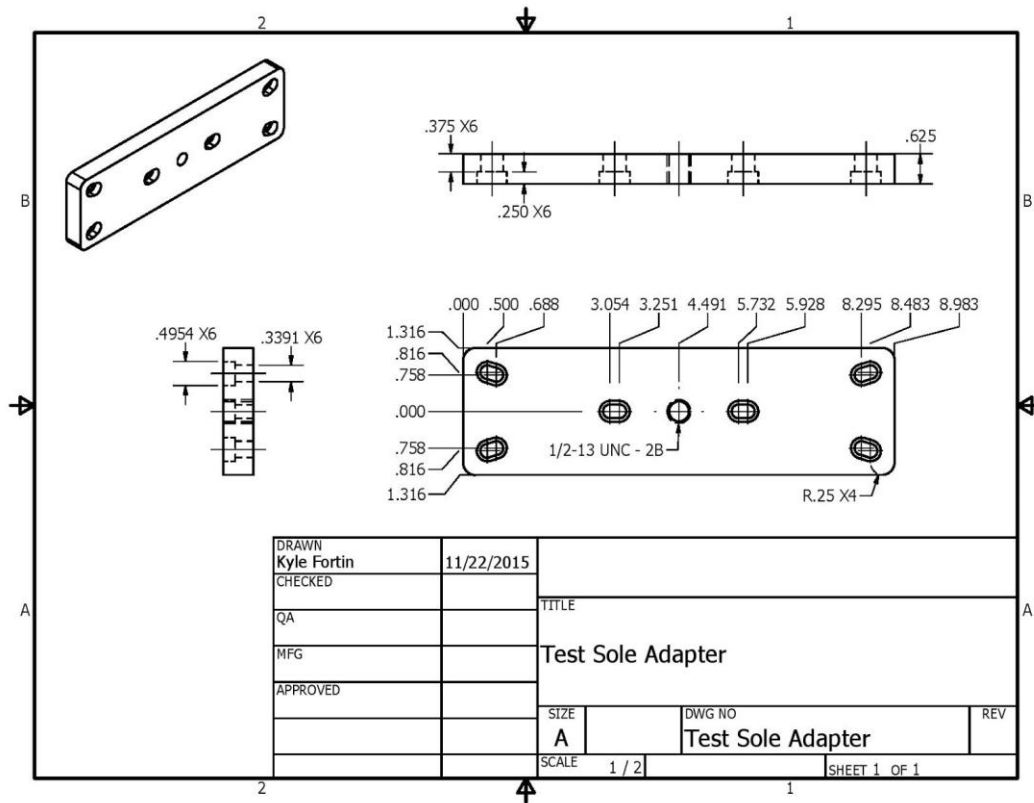
Yoke (9)



Linear Bearing Mount (11)

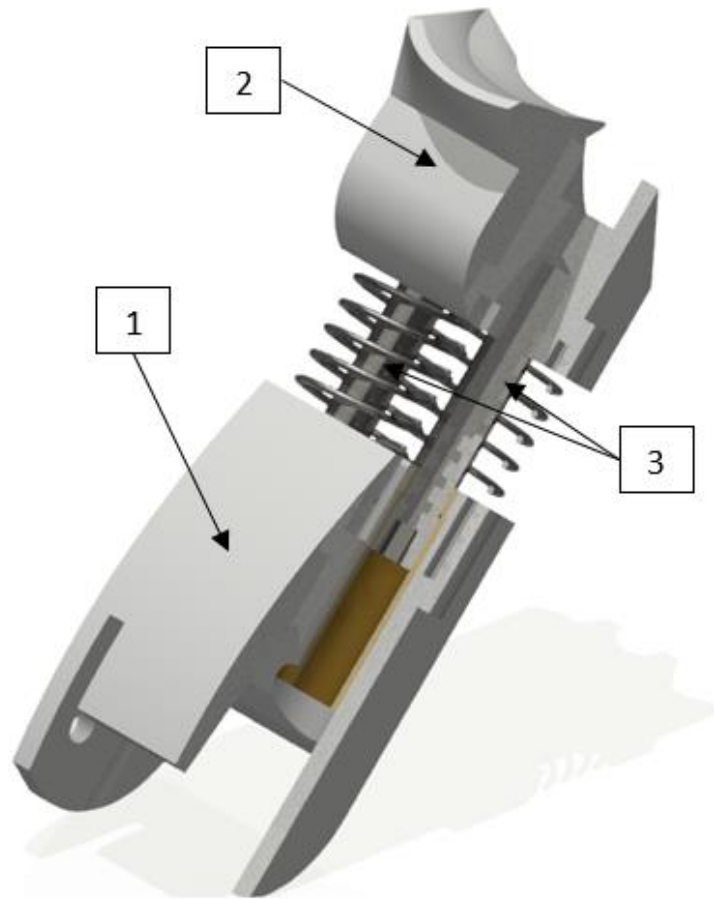


## Test Sole Adapter (12)

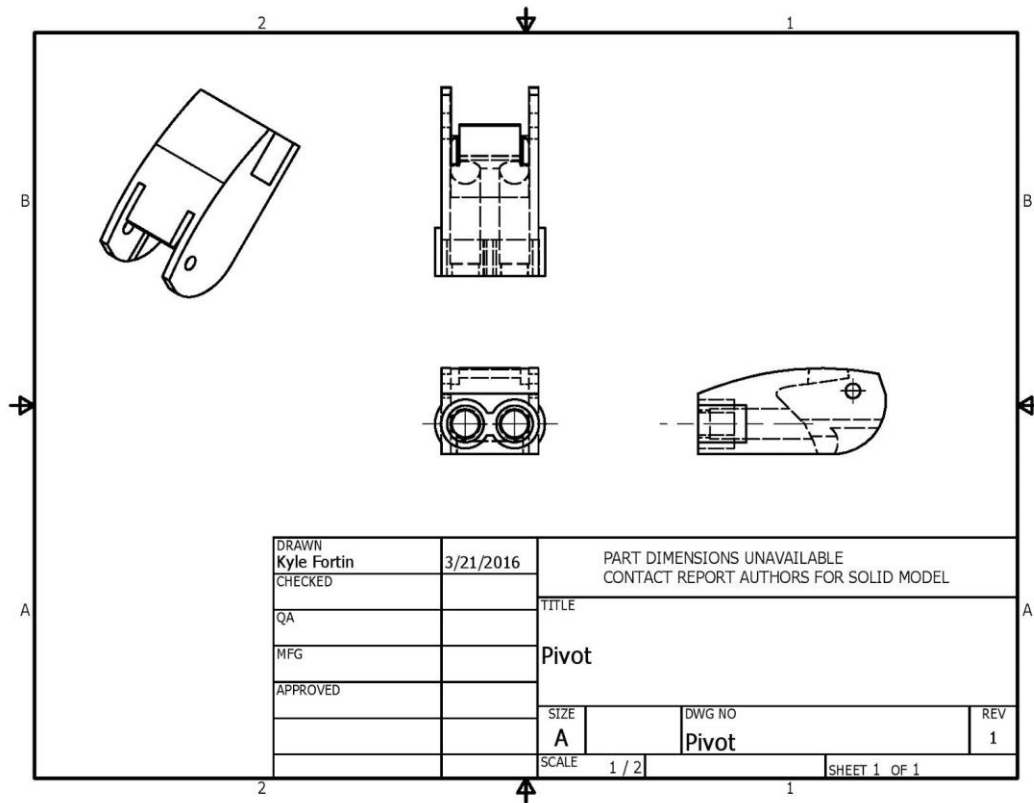


## Alpine Ski Binding – Heel Piece

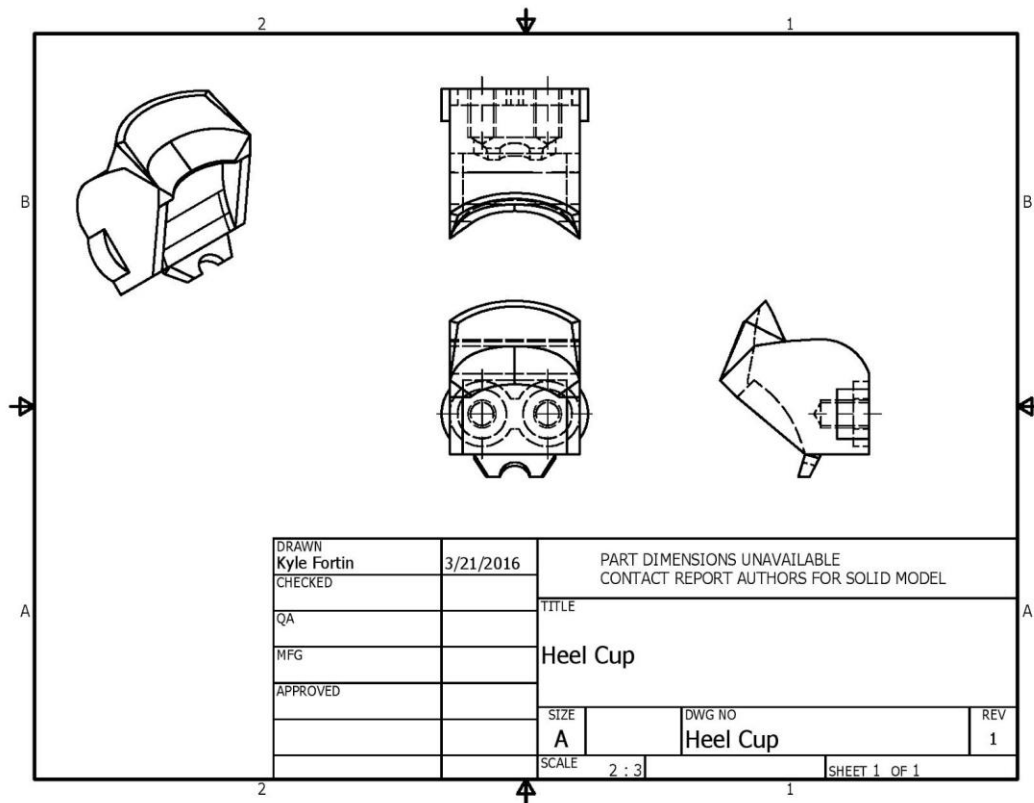
Part Diagram



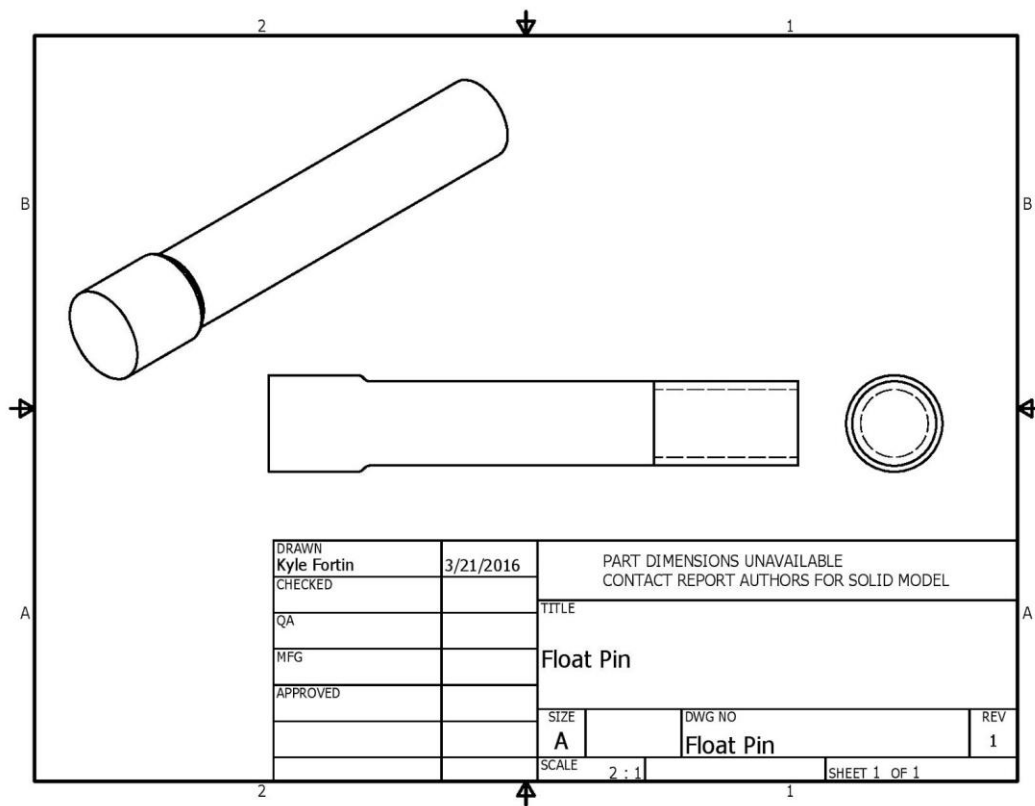
Pivot (1)



Heel Cup (2)



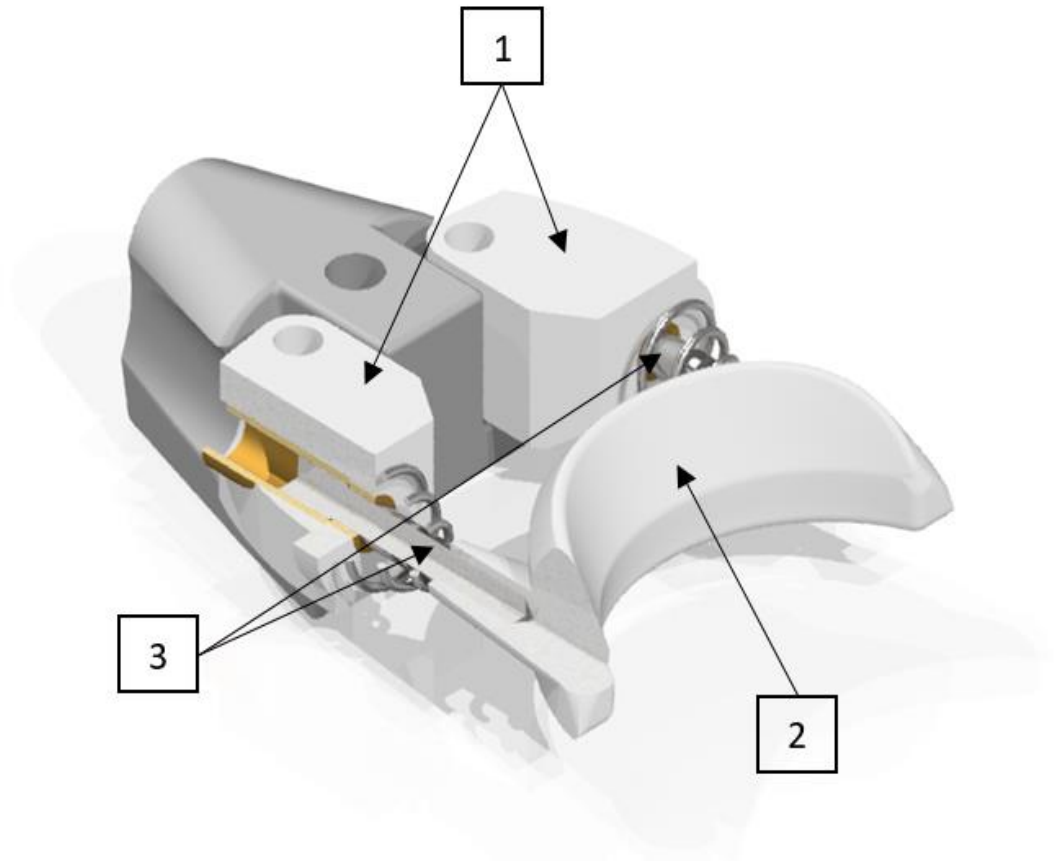
Float Pin (3)



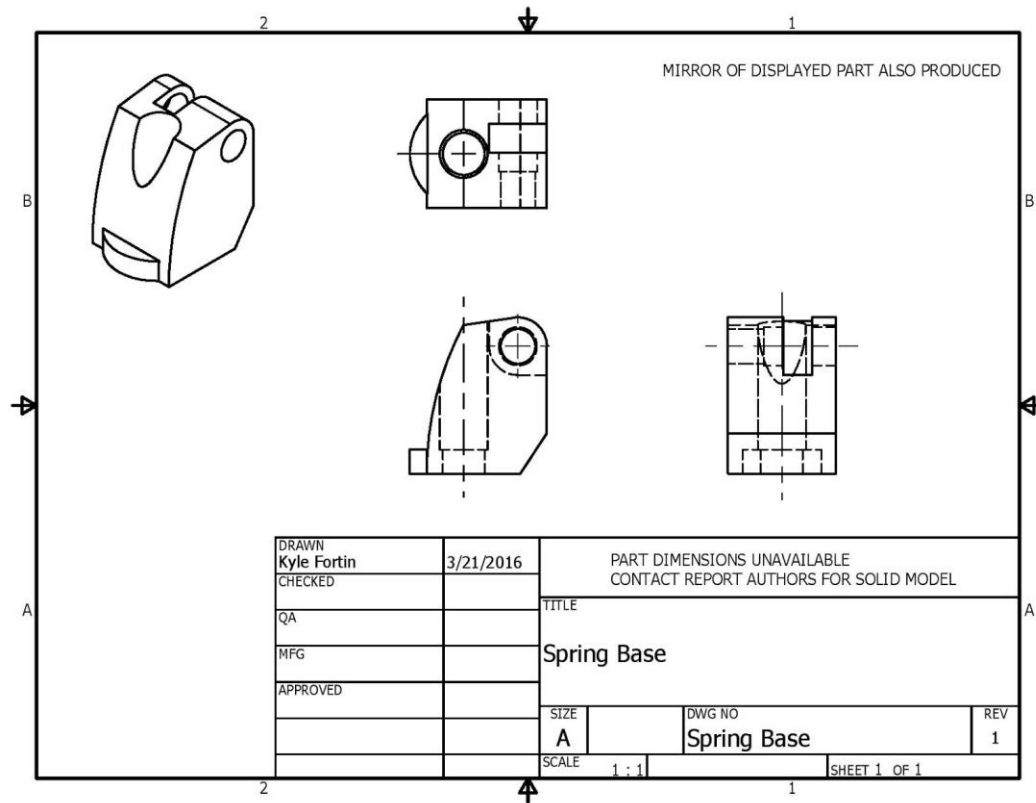


## Alpine Ski Binding – Toe Piece

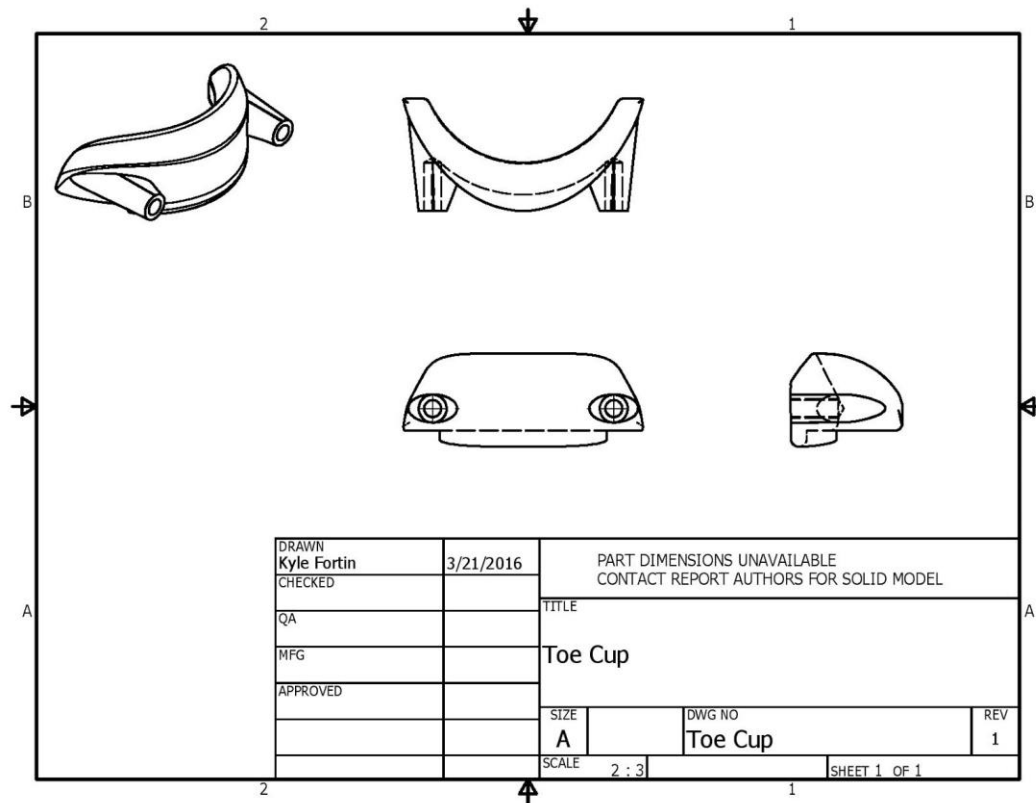
### Parts Diagram



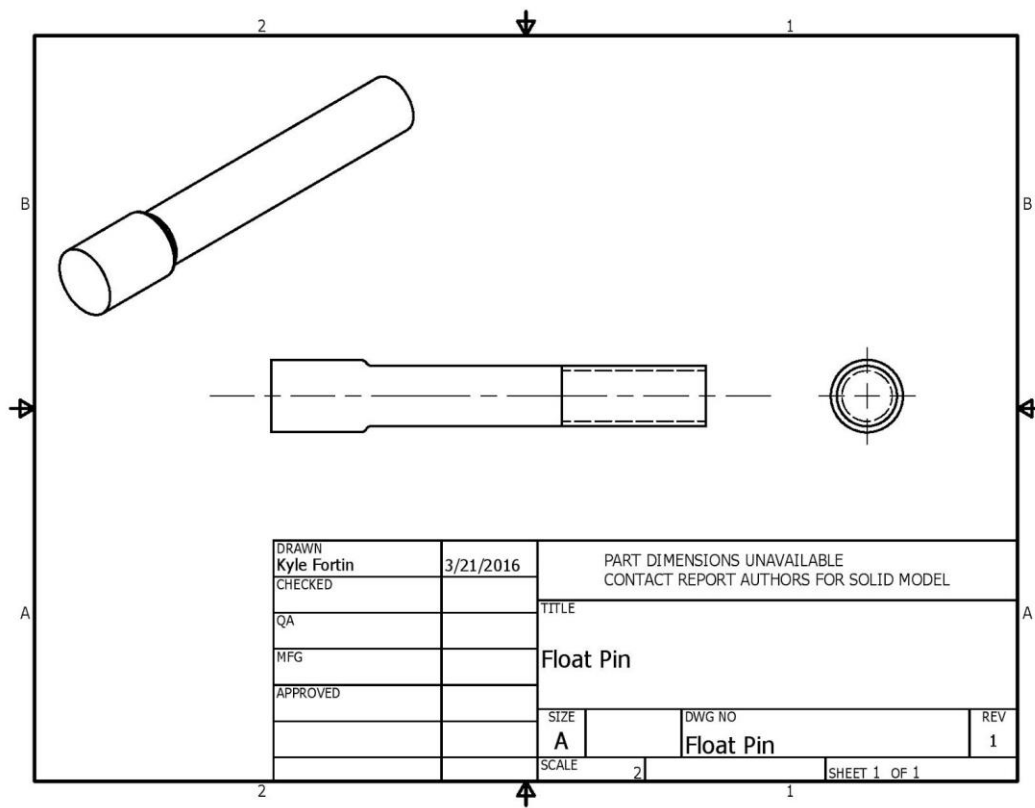
## Spring Base (1)



## Toe Cup (2)



Float Pin (3)





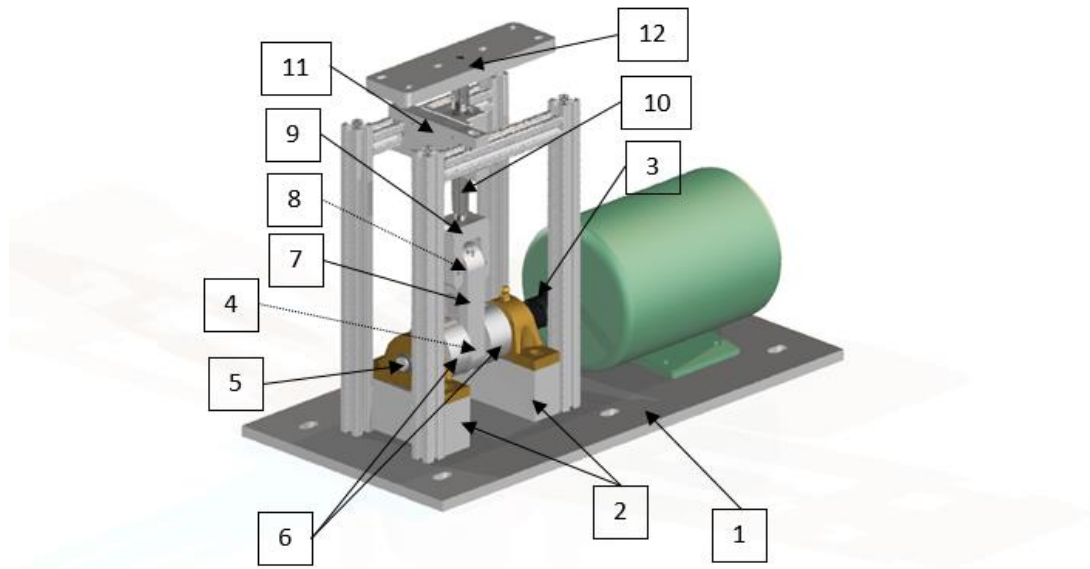
## Appendix 3: FEA of Machined Parts

Finite element analysis was used to evaluate the strength of all custom parts that were to be built for this project. The following images depict the results of the analysis. FEA analysis results were recorded both in terms of stress experienced throughout the parts, as well as the associated safety factor of the parts. The maximum Von-Mises Stress and the minimum safety factor are indicated on each result.

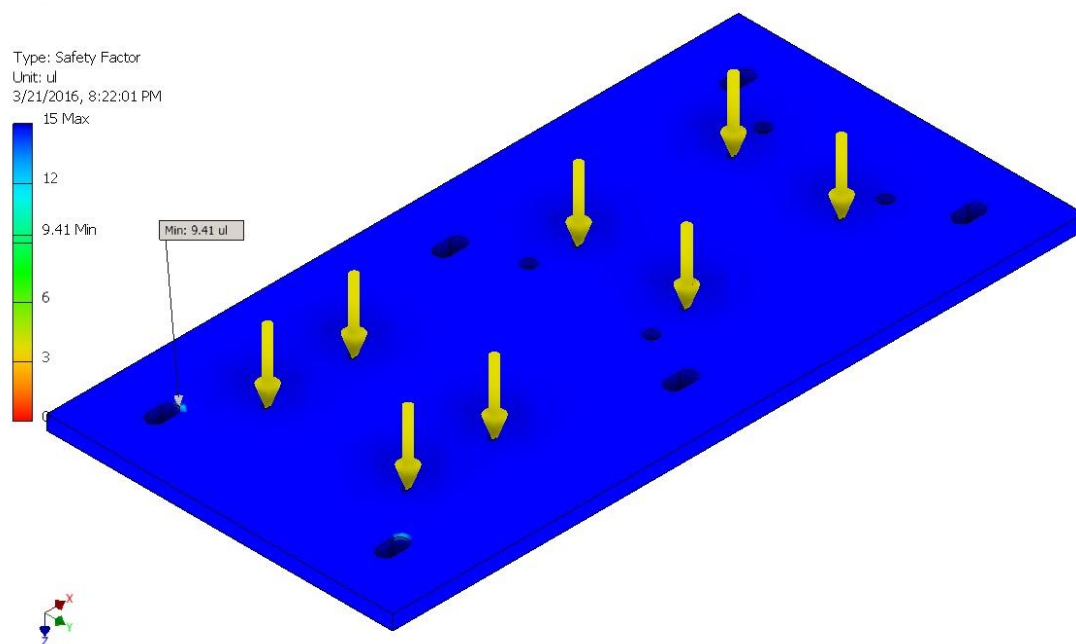
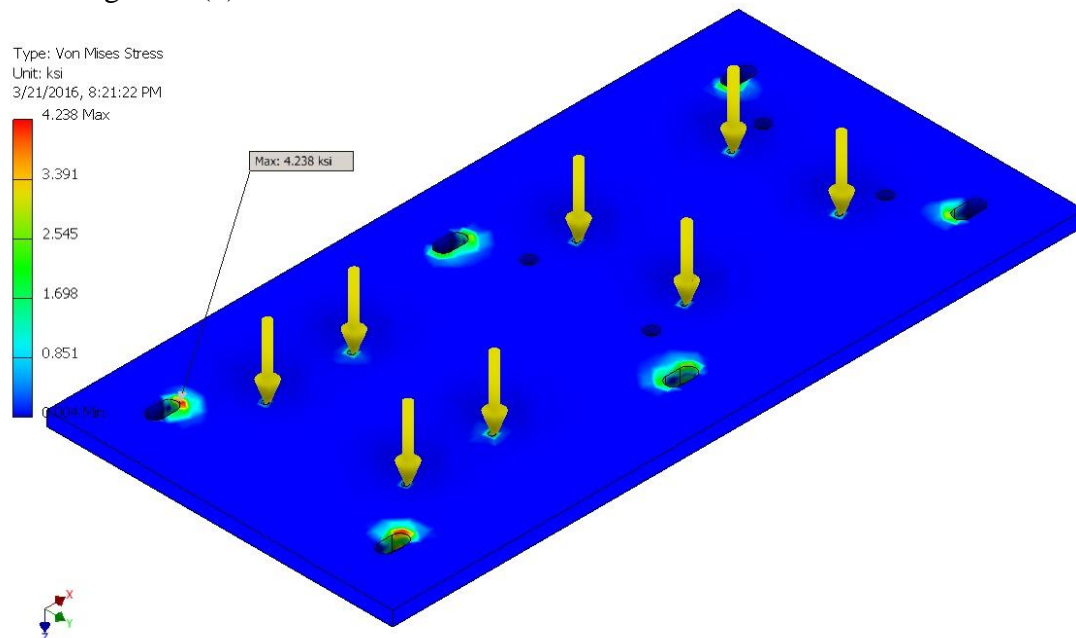
### Vibration Machine

Calculation of the maximum downwards force on the ski, due to its maximum acceleration during the vibration cycle, yielded a load of 886.1 pounds of force. This load was rounded to 1000 pounds, which ensured that any anomalies encountered during real-world testing would not affect the performance of the machine. As such, all loads used in the FEA were set to 1000 pounds. The minimum calculated safety factor for any individual component was 3.68, indicating that the design would be successful. Analysis for the assembly as a whole is discussed in Section 2.2.5.

### Parts Diagram

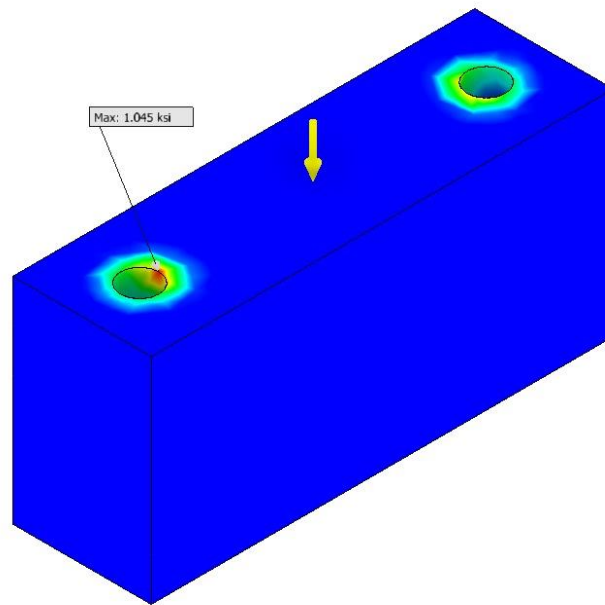
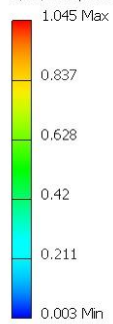


## Mounting Plate (1)

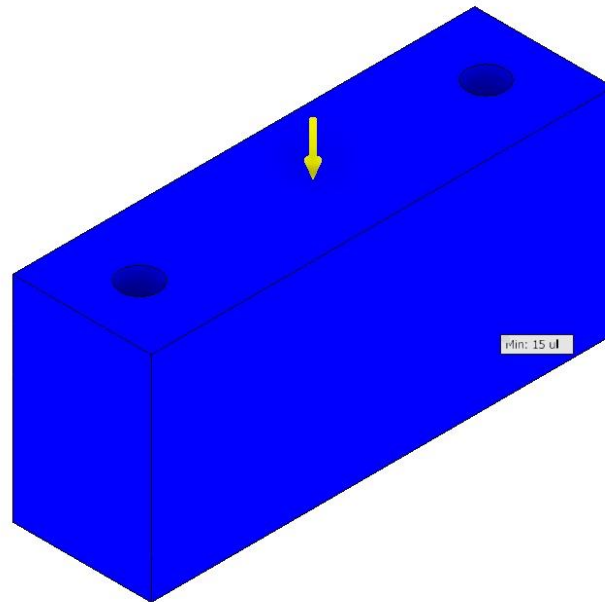
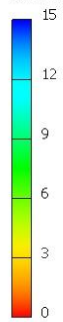


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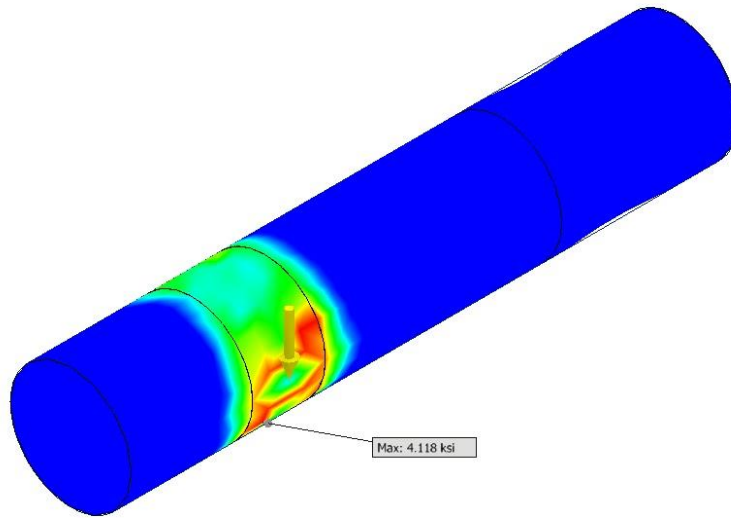
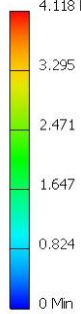


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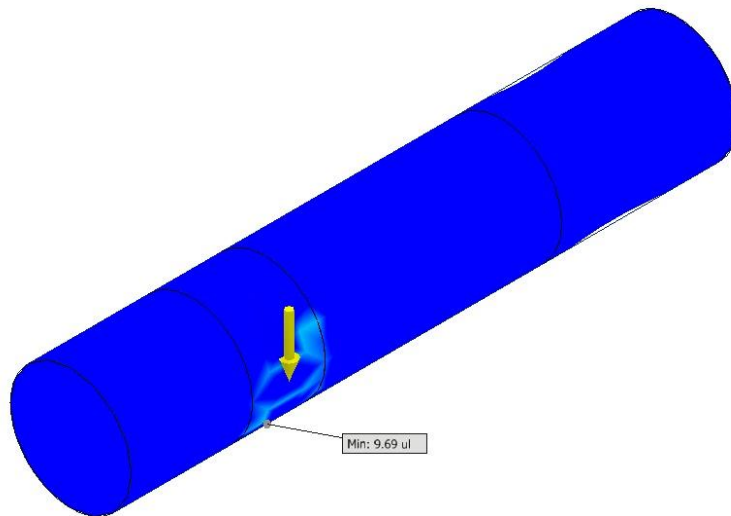
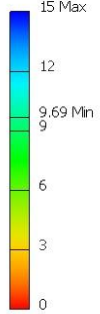


### Main Rotary Shaft (3)

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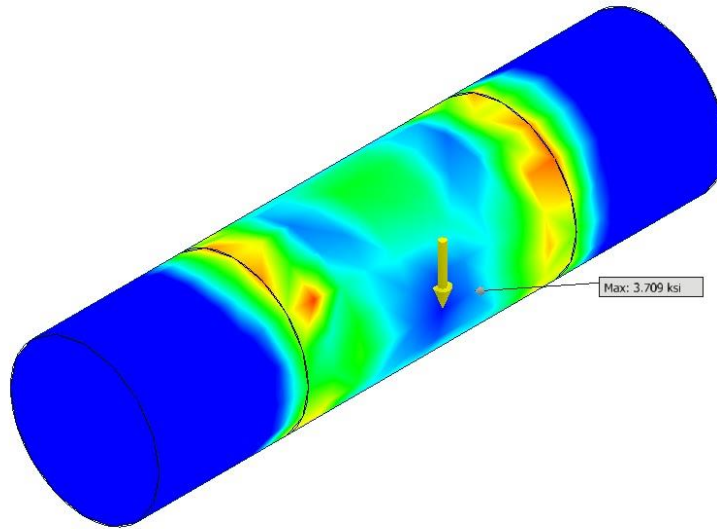
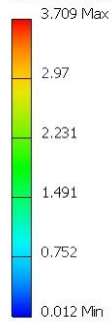
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15 Max



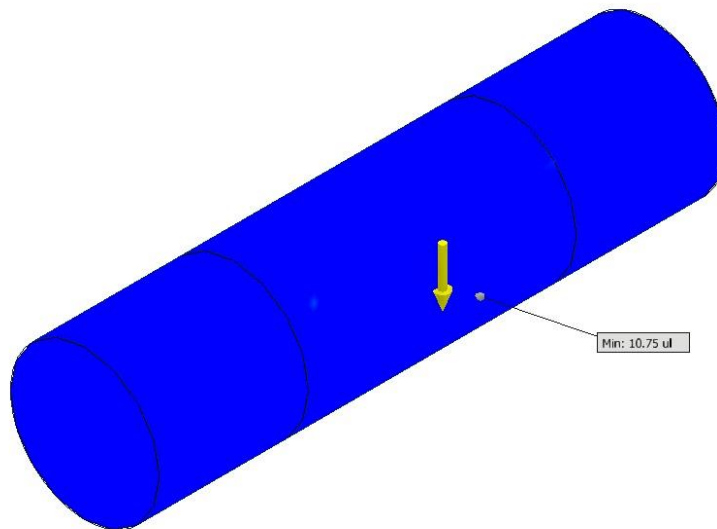
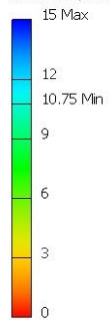


## Crank Shaft (4)

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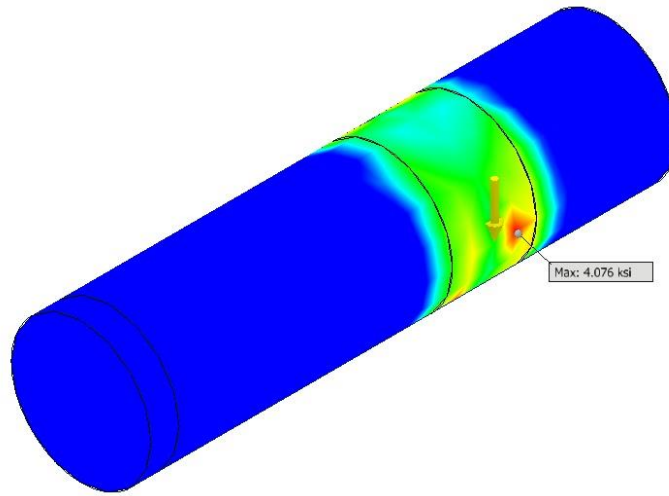
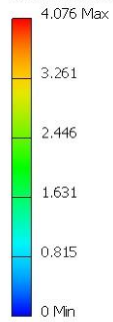


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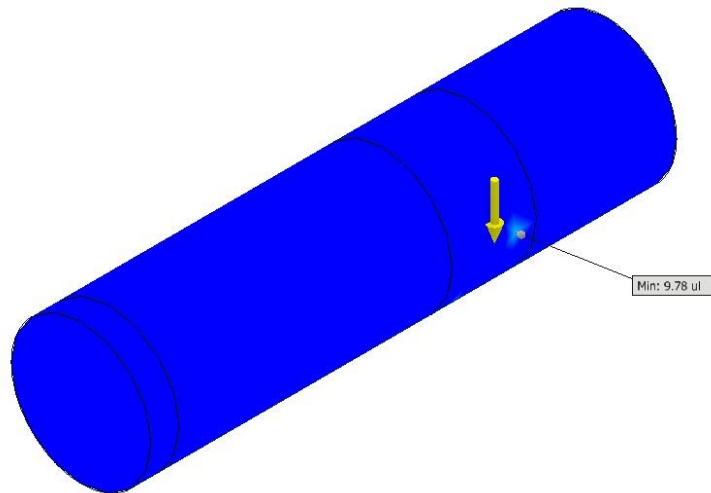
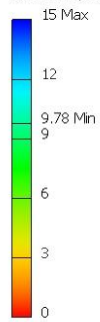


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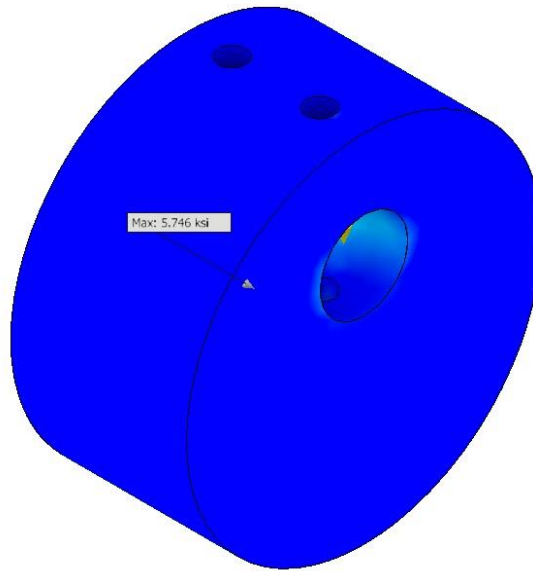
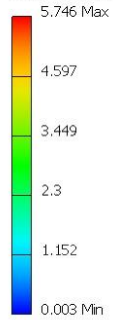


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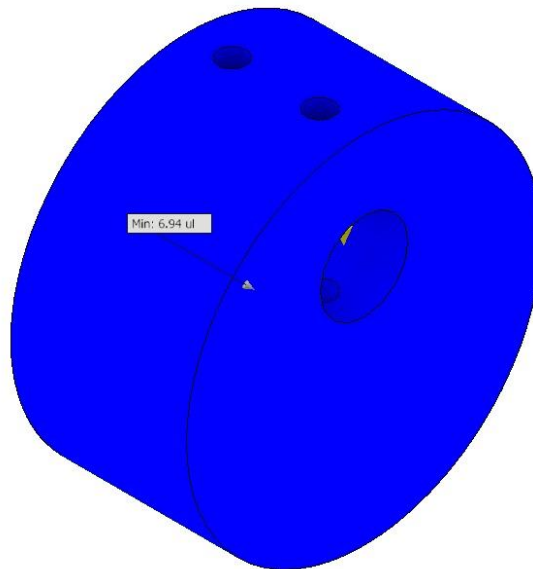
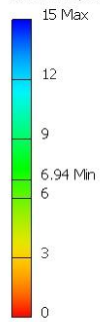


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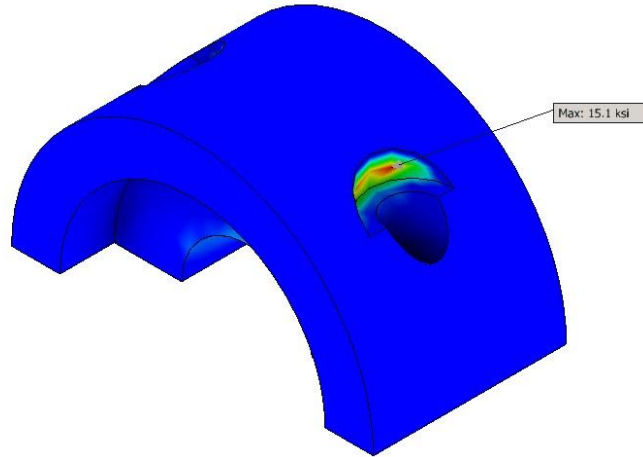
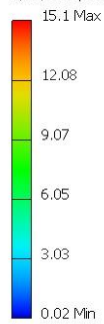


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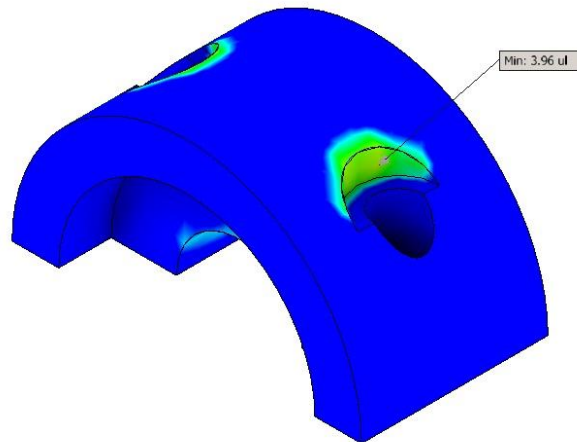
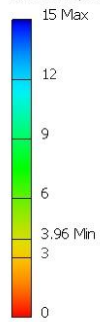


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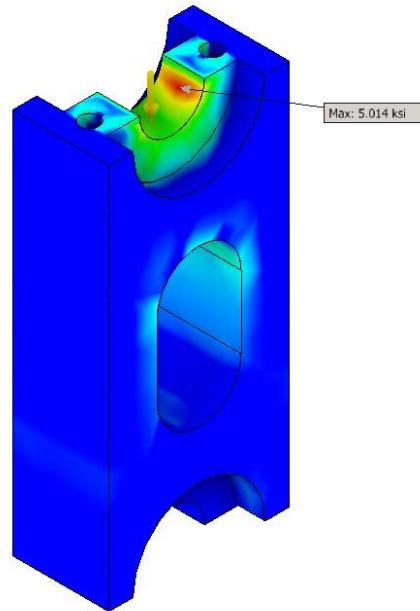
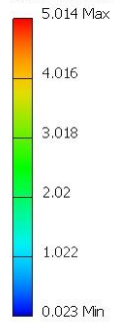


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Unit: ul  
3/20/2016, 8:17:09 PM

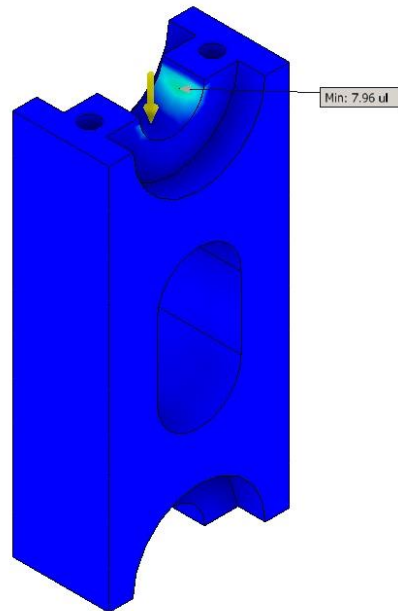
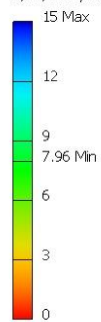


## Connector Rod (Rev 1) [Replaced with Connector Rod (Rev 2)]

Type: Von Mises Stress  
Unit: ksi  
3/20/2016, 8:19:16 PM

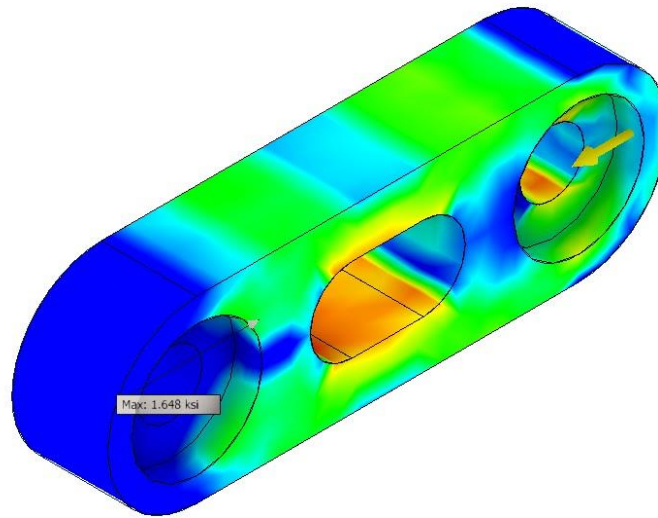
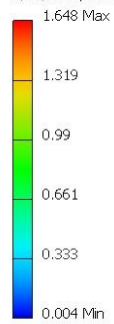


Type: Safety Factor  
Unit: ul  
3/20/2016, 8:19:41 PM

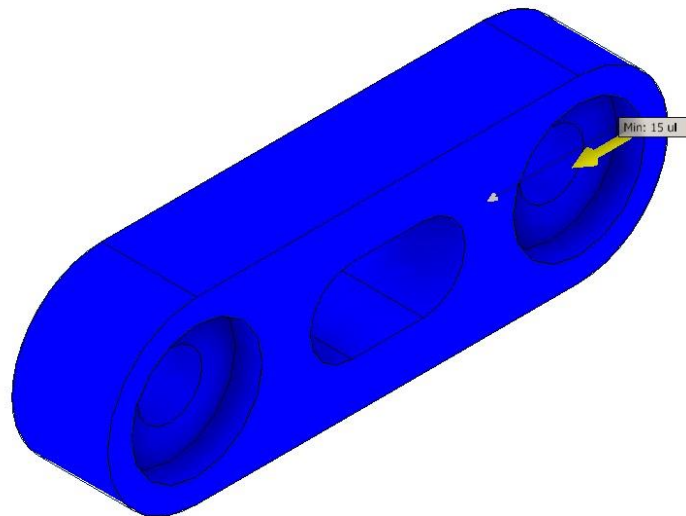
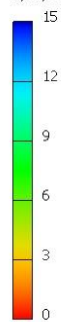


## Connector Rod (Rev 2) (7)

Type: Von Mises Stress  
Unit: ksi  
3/21/2016, 8:36:41 PM

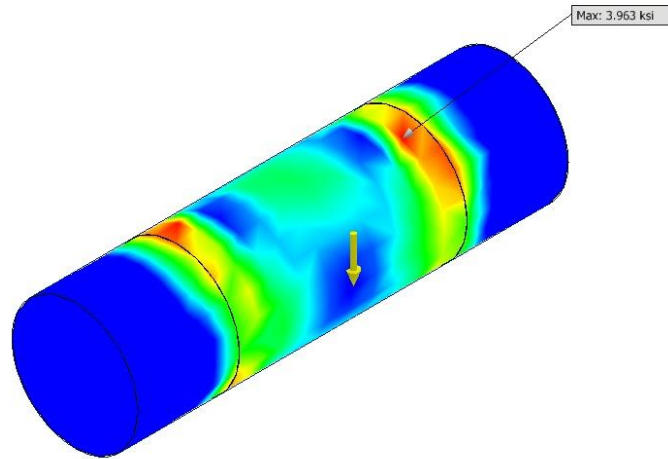
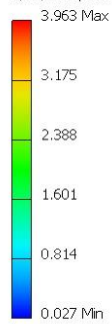


Type: Safety Factor  
Unit: ul  
3/21/2016, 8:36:53 PM

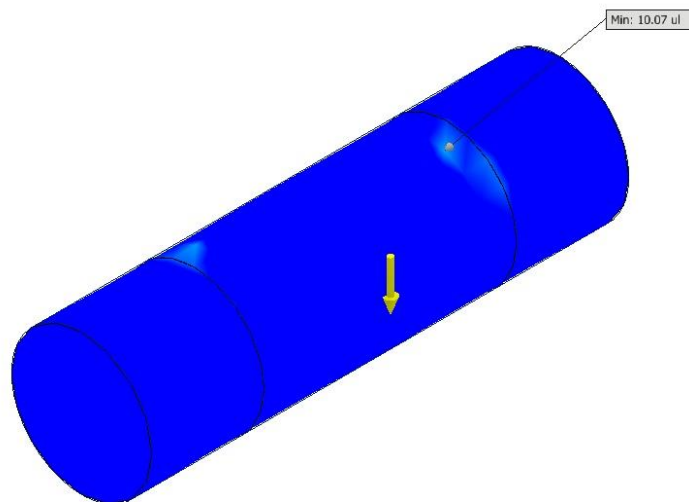
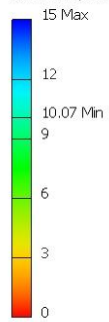


## Upper Linkage Shaft (8)

Type: Von Mises Stress  
Unit: ksi  
3/20/2016, 8:12:53 PM

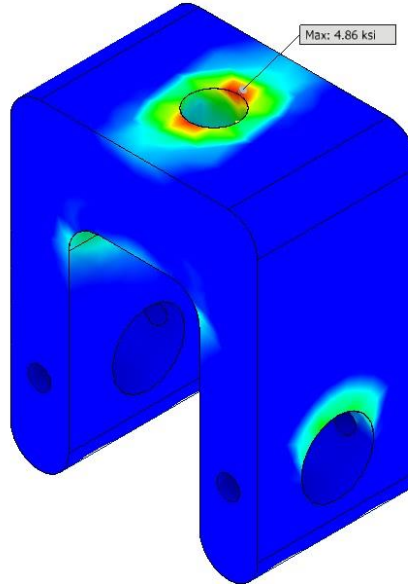
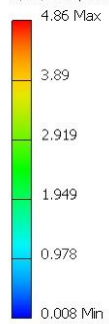


Type: Safety Factor  
Unit: ul  
3/20/2016, 8:13:45 PM

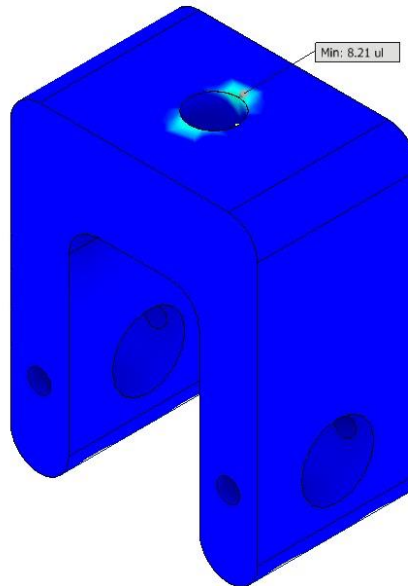
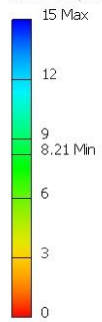


## Yoke (9)

Type: Von Mises Stress  
Unit: ksi  
3/20/2016, 8:08:28 PM



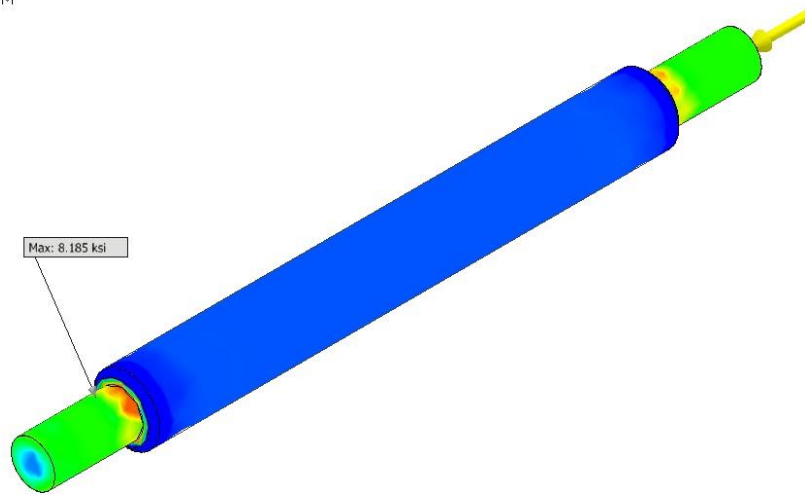
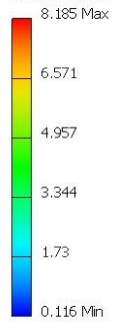
Type: Safety Factor  
Unit: ul  
3/20/2016, 8:08:54 PM



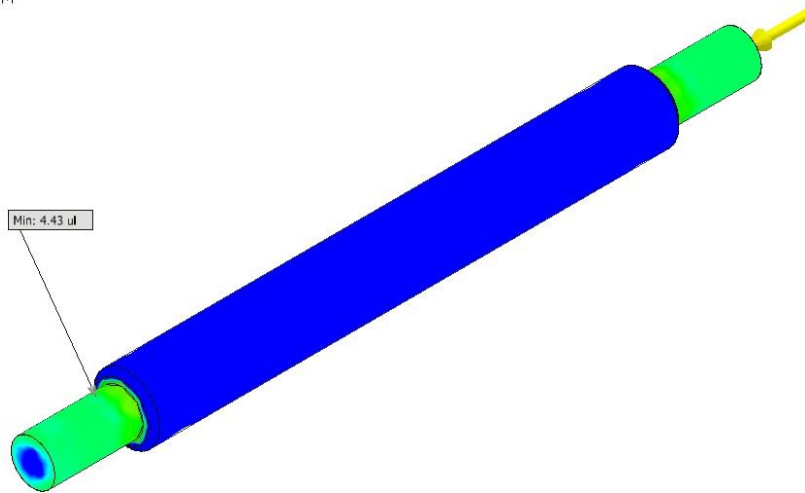
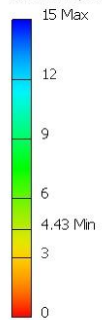


## Vertical Linear Shaft (10)

Type: Von Mises Stress  
Unit: ksi  
3/20/2016, 8:00:43 PM

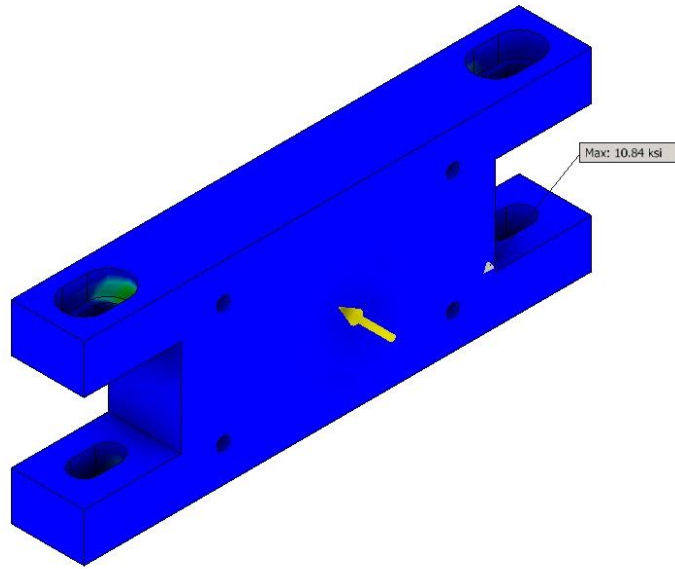
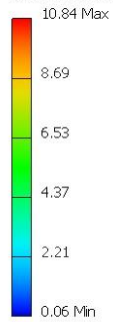


Type: Safety Factor  
Unit: ul  
3/20/2016, 8:01:13 PM

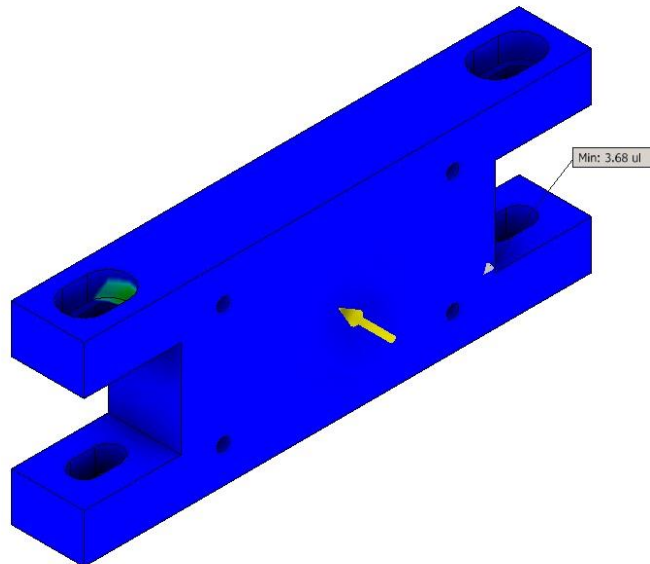
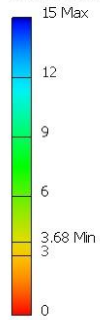


## Linear Bearing Mount (11)

Type: Von Mises Stress  
Unit: ksi  
3/20/2016, 8:07:00 PM

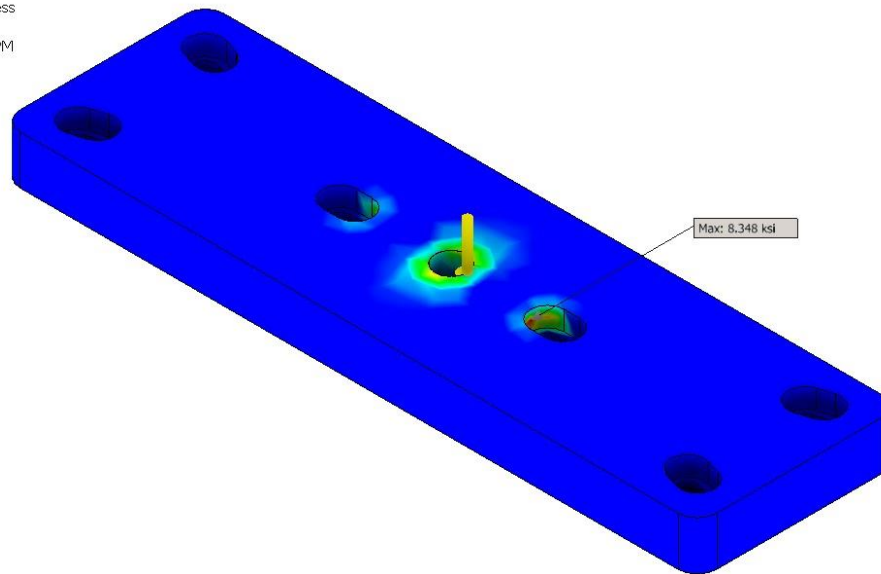
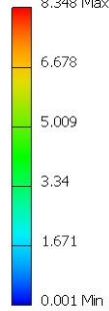


Type: Safety Factor  
Unit: ul  
3/20/2016, 8:07:22 PM

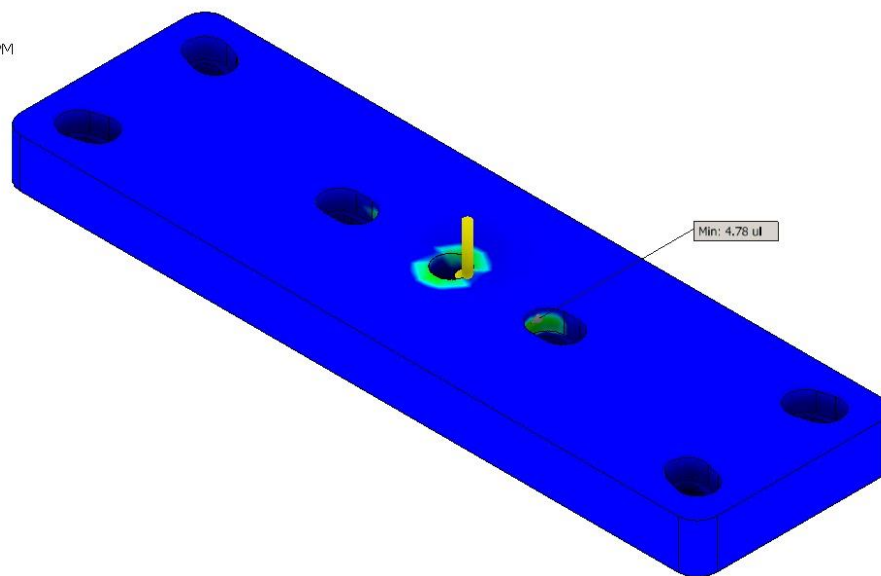
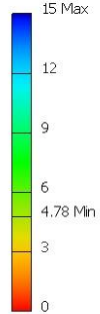


## Test Sole Adapter (12)

Type: Von Mises Stress  
Unit: ksi  
3/20/2016, 7:55:04 PM  
8.348 Max



Type: Safety Factor  
Unit: ul  
3/20/2016, 7:56:06 PM  
15 Max

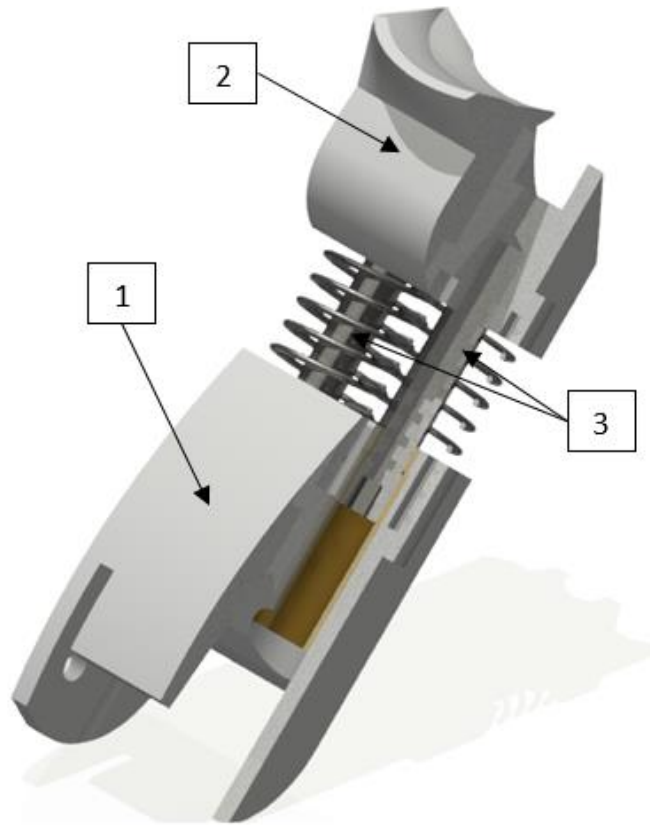


## Alpine Ski Binding

To analyze the stress experienced by the heel piece of the binding, the release limits for the binding were first calculated using the methods laid out in ASTM F939-12. This resulted in a maximum vertical release torque of 52.463ft-lb, and a maximum lateral release torque of 221.232ft-lb. These translated to a lateral release force of 52.256 pounds-force and a vertical release force of 277.615 pounds-force. These loads were used in the FEA for all binding pieces (the heel piece experience both lateral and vertical release forces, while the toe piece experiences only lateral release forces). As these loads were the maximum allowable per ASTM standards, the numbers were not rounded for ease of calculation. The FEA for the toe and heel assemblies can be found in Section 2.3.5.

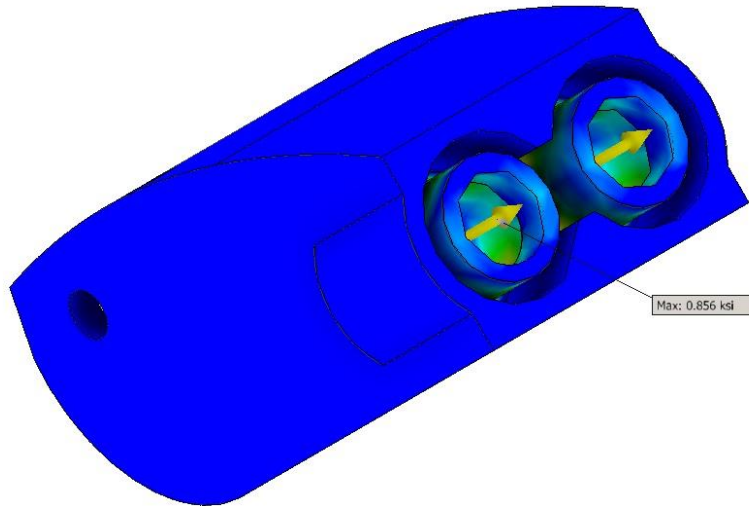
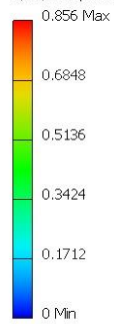
### Heel Piece

#### Parts Diagram

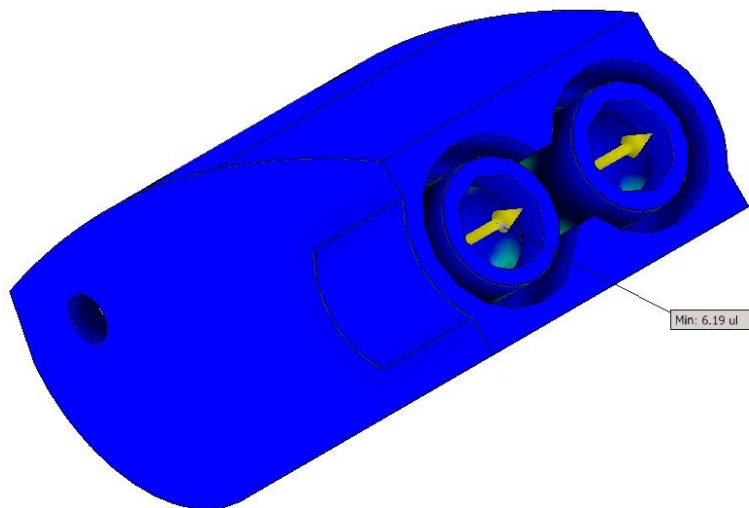
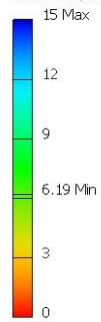


## Pivot (1)

Type: Von Mises Stress  
Unit: ksi  
3/21/2016, 1:34:12 PM

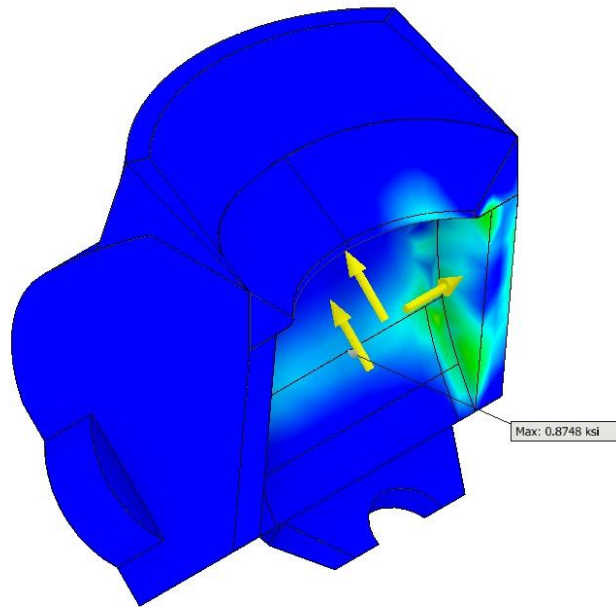
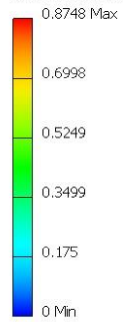


Type: Safety Factor  
Unit: ul  
3/21/2016, 1:34:46 PM

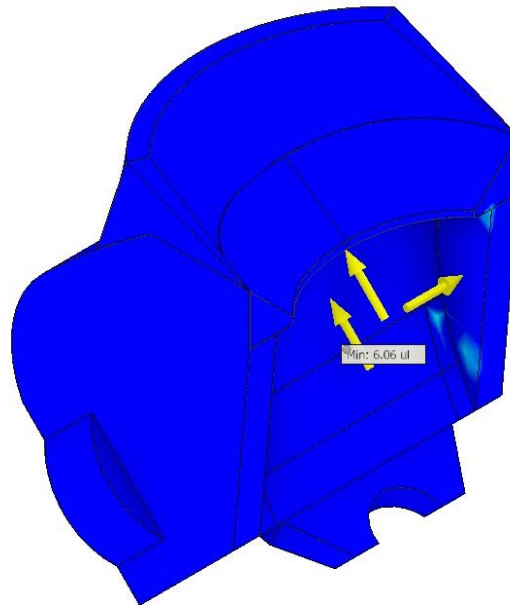
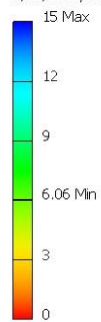


## Heel Cup (2)

Type: Von Mises Stress  
Unit: ksi  
3/21/2016, 1:29:38 PM

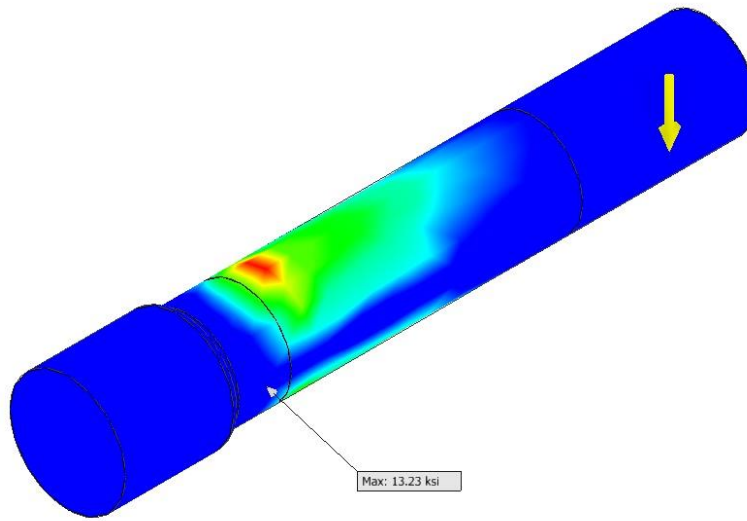
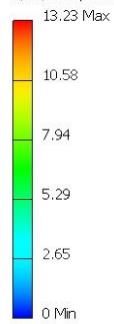


Type: Safety Factor  
Unit: ul  
3/21/2016, 1:30:04 PM

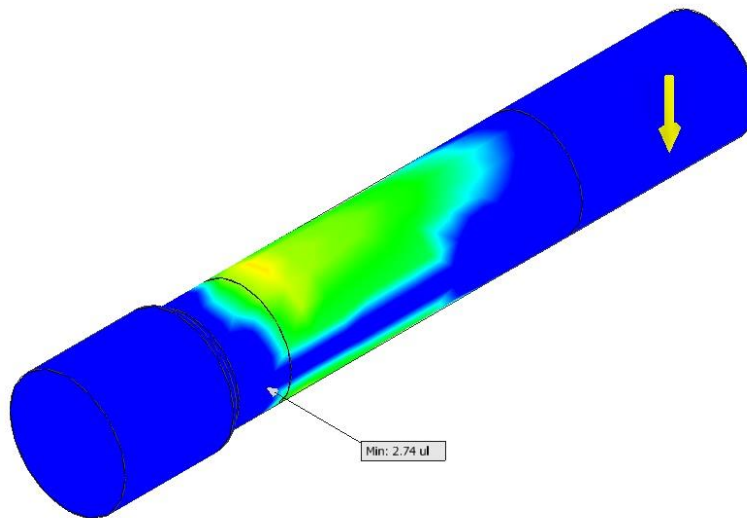
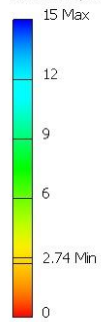


## Float Pin (3)

Type: Von Mises Stress  
Unit: ksi  
3/21/2016, 1:39:00 PM

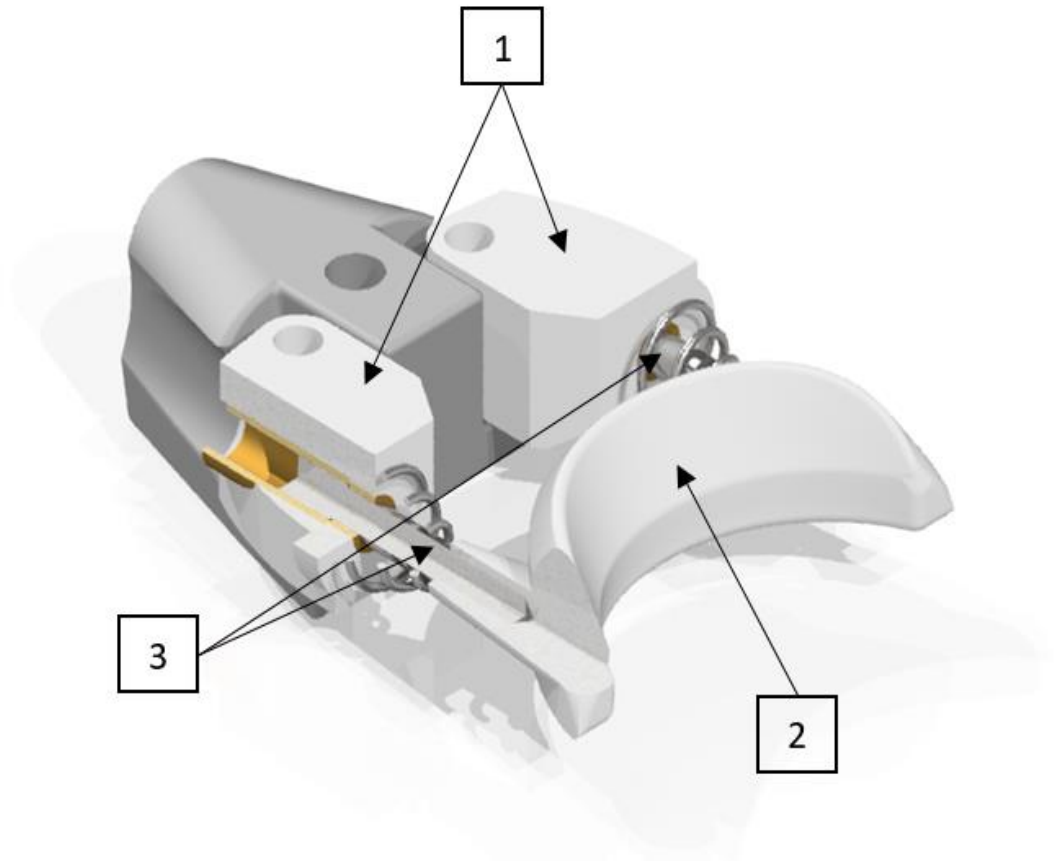


Type: Safety Factor  
Unit: ul  
3/21/2016, 1:39:18 PM



**Toe Piece**

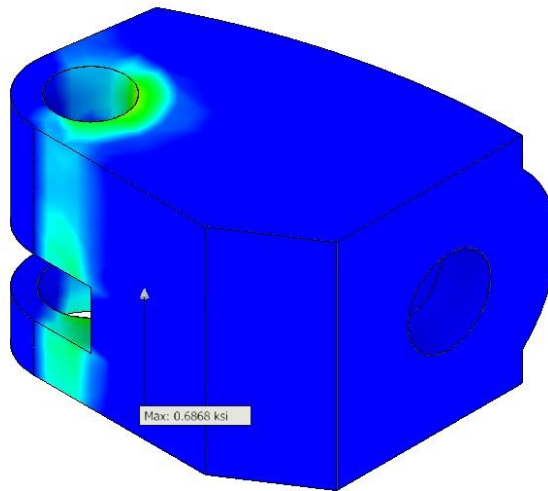
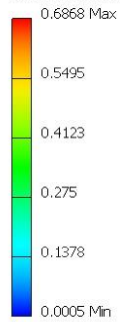
## Parts Diagram



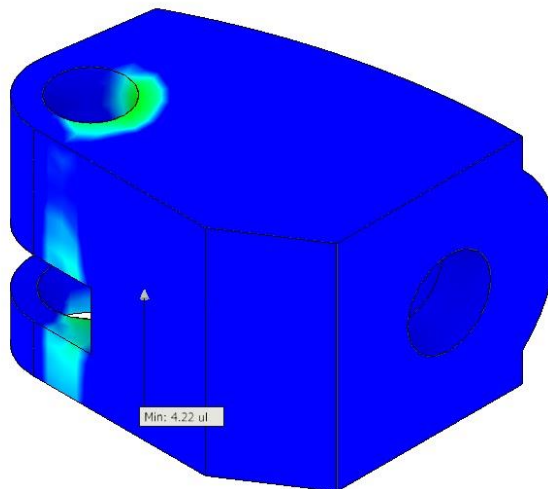
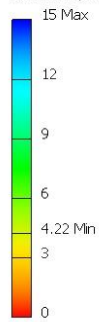


## Spring Base (1)

Type: Von Mises Stress  
Unit: ksi  
3/21/2016, 1:44:36 PM

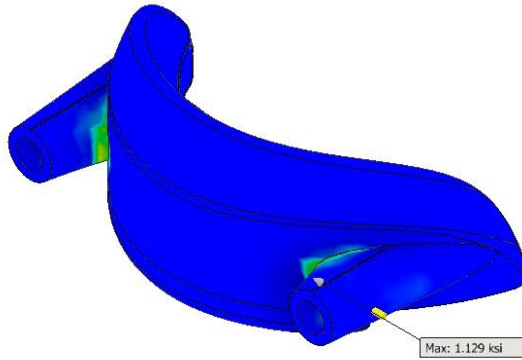
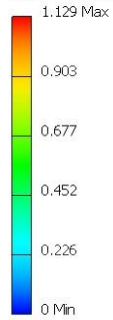


Type: Safety Factor  
Unit: ul  
3/21/2016, 1:44:57 PM

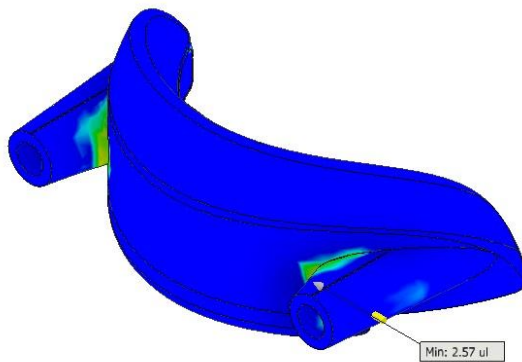
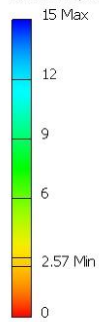


## Toe Cup (2)

Type: Von Mises Stress  
Unit: ksi  
3/21/2016, 1:46:43 PM

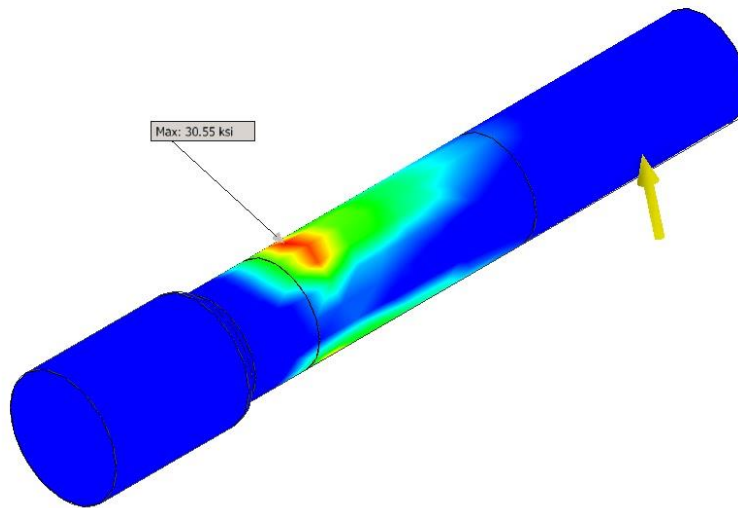
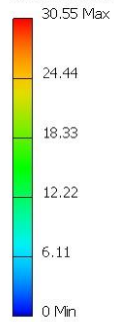


Type: Safety Factor  
Unit: ul  
3/21/2016, 1:47:19 PM



## Float Pin (3)

Type: Von Mises Stress  
Unit: ksi  
3/21/2016, 1:49:42 PM



Type: Safety Factor  
Unit: ul  
3/21/2016, 1:49:59 PM

