CAPSTONE DESIGN COURSE

MIM 1501

Technical Design Report

Abutment Hammering Tool for Dental Implants

Project # W01/S01-2
Design Review

Design Advisor: Prof. Sinan Muftu

Design Team
Robert Birichi, Timothy Goddard, John Jagodnik, John O'Callaghan, Sean Westbrock

March 5, 2001

Dept. of Mechanical, Industrial, and Manufacturing Engineering
College of Engineering, Northeastern University
Boston, MA 02115
Abstract:
The following report analyzes the need for a device to be used in conjunction with the Bicon taper lock abutment system used for prosthetic reconstruction when a tooth has become extensively damaged. In this system, an implant is placed in the jaw and integrated through bone regeneration. The titanium implant acts as a synthetic root, producing a solid foundation for the prosthesis. An abutment is then placed in the implant. The abutment serves as a post for cementing a prosthetic tooth to the implant. The abutment in the Bicon system is affixed to the implant through a taper lock mechanism. The taper lock system, which is comprised of a tapered interference fit for affixing the two components together, prevents rotation or loosening of the prosthesis over time. The effectiveness of the taper lock is highly dependent on the accuracy of the installation force. The current method of application involves placing a handle against the abutment and tapping the handle with a dental mallet. The force used in the current method is inconsistent and often leads to insufficient installation force. To improve the success of this prosthetic fastening system, a new tool design is proposed. The proposed tool design will apply a consistent seating-force, install the abutment in one step, and properly function in different areas of the mouth. The tool design must be compatible with all types of abutments currently produced by Bicon. The proposed tool will also be economical, ergonomical, and durable enough to withstand sterilization and repeated use.

There is a displayed need for this device as indicated by the lack of information found in the literature search concerning single activation dental hammering tools. Because of the established need, the taper lock system was analyzed and a tool design was conceptualized. A single, functional design was chosen from these concepts. The components of the chosen device were analyzed and material and cost analyses were performed. Engineering drawings will define the dimensions of the components for initial prototyping. The final conclusion indicates that this project can be developed to achieve all specifications and requirements prior to the end of the spring quarter 2001.
Table of Contents

Abstract: ......................................................................................................................................................... I

List of Figures.................................................................................................................................................. vi

List of Tables ................................................................................................................................................... vi

Introduction .................................................................................................................................................... 1

Background ..................................................................................................................................................... 1

Problem Statement .......................................................................................................................................... 3

Design Considerations ................................................................................................................................. 4

  Requirements ................................................................................................................................................ 4
    Application of Force .................................................................................................................................. 4
    Functional Specifications .......................................................................................................................... 4
    Industrial Engineering Design .................................................................................................................. 4
  Specifications .............................................................................................................................................. 6

State of the Art Review .................................................................................................................................. 7

  Patent Summary ........................................................................................................................................ 7
    0270187 Electric Mallet ............................................................................................................................. 7
    1772852 Electrically Operated Hand Tool ............................................................................................... 7
    2714918 Power Operated Pick Hammer for Sheet Metal Work ................................................................ 8
    3286558 Angularly Operable Head for Impact Tools ........................................................................... 9
    3921044 Electrical Dental Mallet ............................................................................................................ 9

  Summary of Current Technology .............................................................................................................. 10
    Mechanical .............................................................................................................................................. 11
    Electric ..................................................................................................................................................... 11
    Compressed Air ...................................................................................................................................... 12
    Hydraulic ............................................................................................................................................... 12

Design Review ............................................................................................................................................... 13

Analysis of the Taper Lock System ............................................................................................................. 13

  Experimental Analysis .............................................................................................................................. 13
  Analytical Model: ..................................................................................................................................... 15
    Friction: ................................................................................................................................................... 15
    Stress versus Insertion Length: .............................................................................................................. 16
    Discussion .............................................................................................................................................. 17

Tool Design .................................................................................................................................................... 20

  Electrical .................................................................................................................................................... 21
  Hydraulic ................................................................................................................................................... 21
  Pneumatic ............................................................................................................................................... 21
  Mechanical .............................................................................................................................................. 22

Design Proposals .......................................................................................................................................... 23

  Phase 1: Functionality ............................................................................................................................... 23
  Phase 2: Reliability ................................................................................................................................. 23
Phase 3: Manufacturability and Scale ................................................................. 23
Phase 4: Ease of Use ......................................................................................... 23
Phase 5: Analysis of Desirable and Non-Desirable Attributes ..................... 23
Phase 6: Choice of Design ............................................................................. 23

Analysis of Proposed Designs ........................................................................... 25

Phase 1: Functionality .................................................................................. 25
  Force Redirection ....................................................................................... 25
  Power Source ........................................................................................... 25
Phase 2: Reliability ....................................................................................... 26
  Force Redirection ....................................................................................... 26
  Power Source ........................................................................................... 26
Phase 2 (Supplementary): Defining specifics of general devices: ............... 26
Phase 3: Manufacturability and Scale .............................................................. 27
  Force Redirection ....................................................................................... 27
  Power Source ........................................................................................... 28
Phase 4: Ease of Use ...................................................................................... 28
  Force Redirection ....................................................................................... 28
  Power Source ........................................................................................... 28
Phase 5: Analysis of Desirable and Non-Desirable Attributes .................... 29
  Force Redirection ....................................................................................... 29
  Power Source ........................................................................................... 30
Phase 6: Choice of Design ............................................................................. 30
  Force Redirection ....................................................................................... 30
  Power Source ........................................................................................... 30

Theoretical Analysis of Chosen Design ............................................................ 31

Hinge Pin Analysis ......................................................................................... 32
  Stress in the Pins: ...................................................................................... 33
  Trigger Pin Stress Analysis: ........................................................................ 34
  Hinge Pin Analysis .................................................................................... 34
Transfer Head Spring ..................................................................................... 36
Shaft Spring .................................................................................................. 37
Assembly Issues ............................................................................................ 38

Regulatory ....................................................................................................... 39

Industrial Design ............................................................................................ 39

Economical analysis ......................................................................................... 40

Conclusions .................................................................................................... 41

Analysis of the State of the Art Review .......................................................... 41
Evaluation of the Taper Lock Mechanism: .................................................... 41
  Mathematical modeling ............................................................................ 41
  Experimental testing .................................................................................. 41
Tool Design ..................................................................................................... 41
Industrial Design ............................................................................................ 42
Other Considerations: ..................................................................................... 42
  Regulatory .................................................................................................. 42
  Economics .................................................................................................. 42
Outlook ............................................................................................................ 42

Design Testing and Evaluation: ..................................................................... 42
Concluding Works: ........................................................................................ 43
Summary............................................................................................................................................. 44

APPENDIX A: Experimental Data ........................................................................................................ 45

APPENDIX B: Discussion of the Analytical Model ............................................................................... 46
  Assumptions ..................................................................................................................................... 46
  Model Nomenclature .......................................................................................................................... 47
  Other Concerns ................................................................................................................................. 51
  Impact Stress .................................................................................................................................... 52

APPENDIX C: Derivation of Strain Energy for Implants [ROARK’S] .................................................. 53

APPENDIX D: Engineering Drawings ................................................................................................. 54

References............................................................................................................................................... I
List of Figures
Figure 1: Current Abutment Installation Procedure .................................................................2
Figure 2: Abutment Sizes and Angles .......................................................................................5
Figure 3: Electric Mallet ............................................................................................................7
Figure 4: Electrically Operated Hand Tool ...............................................................................7
Figure 5: Power Operated Pick Hammer for Sheet Metal Work ..............................................8
Figure 6: Power-Driven Hand Unit for Rotary and Reciprocating Tools ..................................8
Figure 7: Angularly Operable Head for Impact Tools .............................................................9
Figure 8: Hand-Held Pneumatic Implement for Removing Dental Scale ................................10
Figure 9: Abutment Delivery System .......................................................................................10
Figure 10: Test Setup ...............................................................................................................14
Figure 11: Experimental Insertion Length vs. Input Energy ......................................................14
Figure 12: Representation of Abutment Stresses ....................................................................16
Figure 13: Maximum Tangential Stress versus Insertion Length ............................................17
Figure 14: Insertion Length versus Input Energy ....................................................................18
Figure 15: Insertion Length versus Input Energy if F+ = 0.25 oz(f) .........................................19
Figure 16: Hydraulic Redirection of Force ................................................................................25
Figure 17: Solenoid ..................................................................................................................25
Figure 18: Magnetic .................................................................................................................26
Figure 19: Cam Action .............................................................................................................26
Figure 20: Pinned Beam Method ............................................................................................27
Figure 21: Button Trigger Mechanism and Basic Trigger Mechanism ..................................28
Figure 22: Proposed Solution ..................................................................................................31
Figure 23: Clearance for the Trigger Lock Mechanism ...........................................................32
Figure 24: Trigger Lock Clearances Dimensions ....................................................................32
Figure 25: The Trigger Pin .......................................................................................................33
Figure 26: The Hinge Pin .........................................................................................................33
Figure 27: Simply Supported Circular Beam in Bending .........................................................34
Figure 28: Transfer Spring Compression ..................................................................................36
Figure 29: Shaft Compression Spring .....................................................................................37
Figure 30: Hand Movement Planes ........................................................................................39
Figure 31: Ergonomic Handle Design ......................................................................................40
Figure A1: Abutment and Implant Segments and Steps .........................................................46

List of Tables
Table 1: Ten-Year Life Study of Bicon Implants ......................................................................3
Table 2: Mechanical and Thermal Properties of Grade 5 Titanium ..........................................6
Table 3: Stress from the Analytical Model ..............................................................................17
Table 4: Retention Force versus Energy ..................................................................................19
Table 5: Elimination Matrix ....................................................................................................24
Table 6: Consumable Lubricants .............................................................................................25
Table 7: Specific Designs .........................................................................................................27
Table 8: Analysis of Concept Attributes ................................................................................30
Table 9: Spring Force and Energy in the Transfer Spring .........................................................37
Table 10: Spring Force and Energy in the Shaft Spring ............................................................38
Table 11: Economic Analysis ................................................................................................40
Table A1: Additional properties of Ti6Al4V ...........................................................................48
Table A2: Results Summary ....................................................................................................49
Table A3: Example of One Iteration of Taper Lock Mathematical Analysis ...........................50
Table A4: Bone Properties ......................................................................................................51
Table A5: Material Impact Properties ......................................................................................52
Introduction

This document will display a need for and outline in detail a preliminary design concerning a dental tool to replace the current method for hammering abutments into implants in the taper-lock system. This document will begin with an outline of the current procedure for installation of a prosthetic tooth and discussion will be given on the problem. A literature search will be performed based on defined functional criteria. The results of a state of the art review will show that no current designs for this purpose exist in patent literature, but that the principles behind striking mechanisms can be applied to this dental application through substantial alterations. With the project need developed, analysis of the taper-lock system will be performed through theoretical analysis and experimental means. Hammering devices for this purpose will be conceptualized and the most efficient and accurate method will be chosen by a rational method. The components of this design will be scrutinized to determine expected weak points and some material and cost considerations. Engineering drawings based on the given analysis of the tool design will be presented in order to begin the prototyping phase. With the goals met for this stage of the project, an outlook will present the remaining tasks needed to successfully complete this project.

Background

The procedure for replacing one tooth or a series of teeth with the taper lock system involves six basic steps [1]:

1. The patient’s jaw is x-rayed and impressions are taken as necessary to determine if the patient is a candidate for this procedure. Tooth extraction (for damaged teeth) and any necessary bone grafts are performed.

2. After recovery time, a titanium implant is placed into the jaw of the patient. This implant serves as the root of the prosthetic tooth. A hole large enough to house the implant is drilled into either the mandible (lower jaw) or maxilla (upper jaw). The surgeon cleans the hole and places the implant into the jaw. The top of the implant typically rests three millimeters below the crest of the bone. A temporary abutment is pressed into the implant, and over the next three to six months, the gums are allowed to heal.

3. During this waiting period, the bone begins to regenerate, and forms around the outside of the implant through osseointegration (integration of bone with the implant). This creates a solid foundation for the prosthesis. Through x-rays and regular check-ups, the clinician determines when the bone integration is sufficient to continue the process.

4. When the implant is sufficiently integrated with the bone, the temporary abutment is removed, exposing the top of the implant. The dentist cleans the surface to remove any saliva or blood, and then inserts the abutment into the implant. The implant and abutment have mating pieces consisting of a tapered hole and a similarly tapered post (typically 3 to 5 millimeters in diameter, depending on bone conditions and area of the mouth). The abutment post is placed loosely into the implant and aligned properly with the surrounding teeth.
5. Once the proper alignment has been achieved, the abutment is locked into the implant. The components are locked together with an interference fit, and the abutment must be seated with sufficient force in order to lock the pieces together. Proper initial alignment is important, since separation of the coupled pieces requires a high amount of force. A handle, which is a chisel with a removable blunt tool at the tip, is placed against the abutment. The head of the handle is tapped with a dental mallet to securely mate the two pieces. The friction between the mating pieces joins the two components securely together and seals the joint from contamination.

6. With the abutment installed, the prosthetic tooth is placed over the abutment and cemented into place. The artificial tooth, with the implant acting as an anchor, will not rotate or loosen over time if the abutment has been installed with sufficient force.

Other methods of securing the abutment to the implant exist. Screws are the most common joining method for alternative procedures. With this method, the connection of the two pieces is initially secure, but over time the screw loses its pre-load if not installed with a precise torque. In these cases, the prosthetic loosens and rotates under stresses in the mouth typically caused by tooth interaction and biting. The taper lock system retains the abutment through frictional forces. Unlike screw preload, taper lock forces do not diminish due to cyclic loading. The major drawback of the taper-lock system is the uncertainty in applying the correct force during abutment installation.
Problem Statement

Because of the mechanics of the taper locking connection in the Bicon dental implant system, an optimal range of force exists to seat the abutment. Insufficient force will cause the prosthetic to loosen over time. Excessive force could cause fracture to the abutment, implant or the patient’s jawbone. Misdirected or excessive force during the seating of the abutment also causes pain to the patient, since the procedure is often performed without anesthesia. Table 1 presents the results of an independent study of the success rate for Bicon implants over the last ten years [2].

<table>
<thead>
<tr>
<th>Year</th>
<th>Failures</th>
<th>Survival Rate</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1889</td>
<td>0.9805</td>
</tr>
<tr>
<td>2</td>
<td>1658</td>
<td>0.9711</td>
</tr>
<tr>
<td>3</td>
<td>1059</td>
<td>0.9688</td>
</tr>
<tr>
<td>4</td>
<td>605</td>
<td>0.9688</td>
</tr>
<tr>
<td>5</td>
<td>435</td>
<td>0.9688</td>
</tr>
<tr>
<td>6</td>
<td>346</td>
<td>0.9688</td>
</tr>
<tr>
<td>7</td>
<td>291</td>
<td>0.9688</td>
</tr>
<tr>
<td>8</td>
<td>132</td>
<td>0.9688</td>
</tr>
<tr>
<td>9</td>
<td>123</td>
<td>0.9688</td>
</tr>
<tr>
<td>10</td>
<td>12</td>
<td>0.9688</td>
</tr>
</tbody>
</table>

As mentioned previously, the current method of installing the abutment to the implant involves tapping a handle that is placed against the abutment with a dental mallet. This is an inaccurate method since the applied force varies among practitioners. Implementing such a method on the molar section of the jaw is also cumbersome and inconsistent.

As a result of the variations and inconsistencies that exist in the current installation procedure, an accurate and stable device is needed to improve the success rate. The proposed device will be economical, ergonomical, and comply with government regulations as needed regarding sterility, materials and safety in medical devices. Further details of the proposed design are described in the following section.
Design Considerations

Requirements
The requirements for the successful solution to this problem statement can be categorized into three sections:

Application of Force
An important consideration in this design is the amount of force to be applied. In order to successfully deliver the installation force, the minimum and maximum forces must first be identified. Some important factors in determining the application force are:

- The maximum loading of the jawbone (maxilla and mandible)
- The maximum impact loading the medical grade titanium (Ti6Al4V) components can withstand before plastic deformation
- The frictional forces involved in installation and removal

These values will aid in developing and evaluating the specified range of forces. The final design will be analyzed statistically to ensure that the applied force is consistent within the desired range.

Functional Specifications
To comply with basic regulations for medical devices, the tool must be designed with certain considerations. First, the tool must have the ability to be sterilized. Dental tools are autoclaved at 40-60 psi and 120°C or sprayed with chemicals to achieve sterilization [3]. Material selection will be important for strength under mechanical and thermal stresses as well as chemical resistance. Stainless steel is the best potential material for this application for it possesses these characteristics and is sterilizable [4]. Possible modes of failure will also be analyzed during the design process to ensure that the tool is effective, safe, and ethical.

Industrial Engineering Design
From an industrial engineering viewpoint, the ergonomics of the design with respect to the patient and the clinician are extremely important. A few of the considerations are as follows:
**Versatility**

The proposed tool is to be used in all sections of the mouth and therefore must be versatile and appropriately shaped for different abutment types. Since abutments come in various sizes and angles (see Figure 2), the ability to fixture the head of the hammering device with different tools will be important. The tool should be versatile enough to install abutments in a minimum of 99.5 percent of mouth structures. Clearance of the human jaw will play an important role to achieve this statistical goal.

![Figure 2. Abutment Sizes and Angles](image)

**Ergonomics**

The handle of the device should be comfortable to the operator. The offset angle of the handle and the center of gravity will be examined to determine the optimum casing for positioning of the tool. Since the amount of control the operator has on the tool relates directly to the effectiveness and accuracy of the device, secure grip and balance in the handle are important aspects of the design. If failure occurs in the tool such as a low battery in an electrical device or o-ring failure in a pneumatic device, the failure should be obvious to the user or the device should cease to operate as a safety precaution.

If the range of applied forces varies dramatically for different abutment sizes, the force will need to be made adjustable. In either the adjustable or non-adjustable case, the power source should be calibrated at regular intervals to ensure consistency.

**Economics**

The economical analysis of this device is broad at this stage of development. The initial demand for the product will be low; therefore, methods of manufacturing will reflect this assumption. Expected market demand, projected cost per tool, and other economical aspects will be further defined as the project progresses.
Specifications

The claims of a clinician involved with the locking taper project define the current specifications of the implant system. These claims include:

- Materials properties of the medical grade titanium used to make the abutment and implant are shown in Table 2.

<table>
<thead>
<tr>
<th>Ti-6Al-4V - Grade 5 - Titanium</th>
<th>550</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.43</td>
<td>113.8</td>
</tr>
<tr>
<td>950</td>
<td>17</td>
</tr>
<tr>
<td>880</td>
<td>0.342</td>
</tr>
<tr>
<td>970</td>
<td>8.6</td>
</tr>
<tr>
<td>44</td>
<td>6.7</td>
</tr>
</tbody>
</table>

- 1.5% failure rate in 600 parts over 4 years due to loosening [5].
- 0.005% failure rate over 4 years due to post shearing at installation [5].
- Abutment deflection of approximately 10 microns into the implant [5].
- 8 oz-force to seat abutment, (less than one Newton), 90 lbs. to remove the abutment (about 390 Newtons) [5].
- The impact force to seat the abutment is equivalent to dropping a one-ounce weight eight inches [1].

Note that the “eight-ounce-inch” force discussed above is actually an energy value. To keep consistent with the convention in the literature, this energy will be used throughout this discussion. The unit “ounce-inches” is not commonly used in practice. One ounce inch actually refers to one-ounce force over a height of eight inches, or 1/16 of a pound force dropped eight inches.

The specifications will be tested analytically and experimentally in order to determine the accuracy of these claims. The acceptable range of force for adequate installation of the abutment into the implant will be established from this analysis. Important data to consider in such an analysis will be the minimum required force to remove the abutment and the typical forces the prosthesis experiences under normal conditions. The above-mentioned specifications will also be used in the analysis and selection of the initial tool design. The required force is an important factor in the design of this tool. This force relates directly to the magnitude of the energy supplied; hence it will be a determining factor in the selection of a power source.

There are several other considerations regarding the power. The power supply should be consistent or regulated. For example, if air power is used to drive the tool, the variability in the pressure supply used should be minimal, or the pressure delivered to the tool should be regulated to retain consistency. If electrical power is used to drive the tool, it should be as consistent as possible. Effects of variable voltage on the impact mechanism should be analyzed prior to exclusion. Regardless of the power supply chosen, the fatigue of moving parts should be minimal over a considerable number of cycles.
State of the Art Review

Patent Summary

The following eight patents were chosen to be most relevant from the 65 patents analyzed.

**0270187 Electric Mallet**

Jan 2, 1883 by J.R. Finney [6]

This patent describes an electro-magnetically actuated mallet. A button completes the circuit of an electro-magnet and uses the property of repulsion of similar poles to drive a hammering piece. A spring returns the hammer to its previous position when the electro-magnet is disengaged.

This device is intended for dentistry and is single actuating, but the design does not suit the requirements in certain aspects. While the applied force is consistent in this device, the device is vertical and cumbersome, reducing its versatility and ergonomics. Also, the design does not provide suitable electric shielding between the hammering piece and the electro-magnet. This causes safety issues since in our application the hammering head is in direct contact with titanium components.

![Figure 3. Electric Mallet](image)

**1772852 Electrically Operated Hand Tool**

Aug. 12, 1930 by L.G. Bates [7]

This is a device that incorporates the hammer and handle into one casing and activates impact with an electric switch.

This is a feasible design for the needs of installing the abutment, but still has issues with the design. The consistency of the applied force is not much better than the current method used. While the design does allow for sterility and safety, the device would still be awkward to use and lacks versatility for installing different parts in various regions of a mouth.

![Figure 4. Electrically Operated Hand Tool](image)
Power Operated Pick Hammer for Sheet Metal Work

Aug. 9, 1955 by E.L. Hopkins [8]

This shows an example of a pneumatic device that could be adapted to the needs of this project. The mechanics of the design is a compressed spring connected to a piston that hits the impact element. It is used to place predictable deformations in sheet metal. The size of this device and the magnitude of the impact force would have to be reduced; however, this type of device shows a valid example of pneumatics driving a hammering head. This design would not work in our circumstances, because it is a reciprocating tool that hammers in parallel with the internal striking mechanism. A reciprocating device would be unsafe since proper seating of the abutment between strokes of the tool would be impossible. This is another device that would not be suitable for some areas of the mouth.

Figure 5. Power Operated Pick Hammer for Sheet Metal Work

Power-Driven Hand Unit for Rotary and Reciprocating Tools

Nov. 4, 1958 by F.P. Wilcox [9]

This patent illustrates a method for turning rotational motion into reciprocating motion. This design essentially uses a lead screw to turn a spur gear. The spur gear is connected to a cam, which transfers the power to the driving member.

While this is an effective method for reciprocating motion, single strike impact using a cam would be unpredictable, since the starting point of the cam is arbitrary after every use. If this design could be converted to a single strike device, it would suit the project needs; however, it does not appear adaptable.

Figure 6. Power-Driven Hand Unit for Rotary and Reciprocating Tools
This design is a right angle force redirection system. The tip consists of a curved path that spans a right angle. A spring rides inside this channel, with cylinders attached to both ends. When the impact tool hits one of the cylinders, the force is redirected perpendicular to the original applied force.

Because the transmitted force has a wide variability (due to force dampening in the spring during transmission), this design is not feasible for the problem given. The ergonomics of such a design are desirable, and the tool is adequately versatile. Sterilization may prove to be an issue with such a device, because the design relies on the spring to transmit the force.

Figure 7. Angularly Operable Head for Impact Tools

This patent summarizes a group of patents found on dental pluggers. These devices are used for performing root canals. A hole is first drilled to remove a contaminated cavity. This hole is then filled with amalgam or gold. The main purpose of the plugger is to condense the metal by packing it into the hole, resulting in a dense tooth filling. Most of these pluggers have some vibratory motion at the tip for compacting means, while a few tips use heat. The exact procedure that this device uses was not accurately disclosed in any of these patents. A few of the patents describe a packing or compressive motion at the tip to dispense the amalgam. This compacting motion is gradual and therefore does not suit the project’s purposes.

Figure 8. Electric Dental Mallet
This device removes dental scale by vibrating and oscillating a head. This is another set of dental tools that vibrated, oscillated, or performed both, but did not provide impact loading. Rotary tools are common in the dental offices, and converting the rotational motion to linear motion might suit the purposes of the design. This particular design performs both oscillation and head rotation.

![Hand-Held Pneumatic Implement for Removing Dental Scale](image)

Figure 8: Hand-Held Pneumatic Implement for Removing Dental Scale

An air powered rotational motor that is secured off-balance (for vibration) powers the device. This does not serve the group's needs. Ergonomical and versatile aspects of this design are favorable; however, this type of device is difficult to use for installing the abutment with sufficient force.

This patent describes one of the only abutment delivery systems found in all of the patent literature. This device is designed for screw retained abutment systems. The tool has the ability to grip the abutment and to tighten the screw using a common torque wrench. This tool is not used in the taper lock system and is only meant as an illustration of what little technology exists for the application of this procedure. From the analysis above and other patent evaluations, no device is found to exist that describes a single actuating hammering tool for dental use. Furthermore, no existing devices can be easily altered to function properly as a dental abutment hammer.

![Abutment Delivery System](image)

Figure 9: Abutment Delivery System

Summary of Current Technology

The Bicon system is unique in its method of installation. Most alternative methods use screw-retained abutments, therefore tooling for the taper lock system has not been developed sufficiently. Through searches of dental tool catalogs, no single actuating dental devices with a stroking motion were found.

Searches of non-dental hammering and single actuating devices were analyzed to determine their adaptability. The following briefly describes the types of items found. They are grouped based on the type of mechanism used, namely mechanical, electrical, air powered, and hydraulic.
**Mechanical**

*Spring-loaded umbrella:* A mechanism where an umbrella is pushed into a spring (located in the handle) and held in place by a catch. Depressing the catch releases the umbrella. The umbrella then hits the end of travel with a certain force. The downfall of this setup is that the spring must be compressed before every use, which could prove cumbersome.

*Gun firing pin:* The firing pin is held back with a spring. The spring is compressed by a catch. Once the catch is released, the firing pin hits the primer of a bullet with enough force to ignite the primer. Some examples of spring forces in guns are 15-28 lbs. (as demonstrated in a Colt or Wolff 1911 gun) [14]. This type of device would have to be scaled down for use in the abutment hammer project. A device with a trigger may also be difficult to position and activate simultaneously.

*Piezoelectric lighter, hammer mechanism:* A small hammer activates this device. When the trigger is pulled back, the hammer compresses a spring, storing energy. When the trigger mechanism slips past a notch in the hammer, the hammer is released. While the trigger in this device would be easy to convert to a button, the consistency of the applied force could diminish over time.

*Automatic punch:* The punch is activated when a lever is pressed, releasing the compressed spring and attached striker. Configuring this type of activation for dentistry would be awkward, and the mechanism has the potential of producing highly variable forces.

*Staple gun:* Squeezing the handle raises the staple hammer and compresses the spring. When the handle is squeezed to the desired distance, the spring releases, driving the staple hammer down. The ergonomics of a large sized manual trigger are not suitable for the dental hammer design.

**Electric**

*Electric lock:* This type of pin, commonly used as a deadbolt lock, is actuated by an electronic solenoid. Fitting a powerful solenoid into a small dental tool may not be feasible. [15]

*Electric stapler:* The stapler is actuated through an electric motor connected to a torque increasing gear train. This design is applicable if it can be effectively scaled down to a usable size. [16]

*Electric nail gun:* This device is activated with a double spring system compressed by two motor driven cogwheels. A cam and lever system pivots to compress the springs. As with previous electrical systems, scaling the design to an acceptable size would be the major concern. [17]
**Compressed Air**

*Pneumatic nail gun:* This device makes use of a piston to drive nails at a high velocity. High-pressure air acts on the rear face of the piston, creating a rapid acceleration. The front of the piston strikes the head of the nail with a nearly perfect elastic collision. The nail then accelerates through an opening, driving it into the desired location. Scaling down the size and transmitted force would be a factor if this type of device is used in the dental hammer project. [18]

*Paintball gun:* This is a system involving a piston inside a cylinder, which is actuated by a pressurized gas supply. For this device the return mechanism is not a spring, but rather a second return chamber. Consistency and size would be important factors if this type of actuation is selected. [19]

*Pneumatic staple gun:* This device consists of a driver actuated by valves, which are connected to a pressurized gas supply. The size and force applied by such a device would not prove practical for the dental hammer design. [20]

**Hydraulic**

*Hydraulic hammer:* A hydraulic hammer is actuated through a piston installed in a fluid filled cylinder. There is a safety concern associated with hydraulic fluid leak from a broken tool. Sterilization may also affect the hydraulic fluid or seals. [21]

Many different types of reciprocating hammer devices were also researched. Analysis of these devices concluded that adapting these drive mechanisms to single actuation would be ineffective. The majority of these types of tools require significant time to attain a nominal speed. Nominal speed is needed to deliver a consistent force.
Design Review

After determining that the current state of the art yielded no sufficient devices that would perform the task of hammering the Bicon dental abutment into its mating implant, work has been performed on the first version of the tool. In order to design the tool for a specific force, the force must be tested. Exceeding the force has the potential of increasing the stresses in the abutment, implant or both and could result in plastic deformation in the components.

Due to issues with proprietary information, the corporate sponsor of this project provided no data on the product. The group was also asked not to reveal excessive information on the nature of the taper lock system from any analysis performed on the test pieces. Because of this, only data provided from the company literature regarding dimensions and materials will be reported. Test results will be presented in such a way not to conflict with proprietary issues. Testing has been performed to determine values for theoretical testing, but only theoretical values can be revealed due to the “black art” of this locking system.

Analysis of the Taper Lock System

Experimental Analysis

The taper lock system was evaluated by experimental testing in order to relate the energy required to seat the abutment to the insertion length. This is to be used to validate the theoretical analysis. Insertion length refers to the distance that the abutment is inserted into the implant during the hammering process. This is directly related to the impact energy given to the abutment and is a factor in the amount of force needed to remove the abutment (retention force). This will be shown in the analytical analysis below.

The first step was to design a fixture for impacting the abutment with a known energy. Free-falling weights are unpredictable because there is variability in where the weight hits with relation to its center of mass. If the weight is off center, it will apply a moment to the piece and could affect seating force. Free-falling weights would produce inconsistent results and therefore will not be used. As previously mentioned, the Bicon procedural manual states that the force required to seat the abutment is equivalent to dropping a one-ounce weight a distance of eight inches [1].

A test fixture was designed in order to both test this statement and aid in relating input force to the retention force and insertion length. For this reason, the “hammer” of the impact fixture is a known weight of approximately one ounce (actual weight is 30.11 grams, or 1.062 oz.). By varying the height of the dropped weight, a graph of impact energy and insertion distance can be obtained. This graph will be compared to the values obtained from the theoretical analysis.

The first step is in the experiment is to vertically cement implants into small cubes using a sheet metal mold. Cement has similar elastic properties of bone and makes it a good model for this experiment. Especially important are the similarities between the Young’s modulus of both materials. The Young’s modulus for bone is 1.2 $\times 10^6$ psi, while the modulus for cement is 2.7 – 3.8 $E6$ psi. Since these materials both are generally brittle in tension and have
a Young's modulus of the same order of magnitude, cement is a valid approximation for a preliminary testing material [22],[23].

The fixture was designed to easily test the eight ounce-inch statement while still allowing other input energies to be tested. The fixture consists of two rods press-fit into a base plate on both sides. (See Figure 10) A sliding carriage fits over the rods and is tolerated in such a way that it can easily slide up and down the rods without twisting excessively in the vertical direction. The principle of the fixture is that by varying the height at which the carriage is dropped, the impact energy can be found. A well fixtured tensile test can relate retention force to the height dropped and a simple measurement can relate the impact energy to insertion distance. While this fixture does have some frictional losses due to contact between the carriage and the guide rods, the losses can be evaluated using the theoretical model or by experimentation. See the Discussion section of the Taper Lock Analysis for details.

The testing to determine insertion length as a function of impact energy for this system was performed. The hammer of the drop test fixture was raised from 0.5 to 22 inches above the abutment in half-inch increments and the insertion length was measured. The numerical results are presented in Appendix B, and the graphical representation is as follows:

```
Figure 10: Test Setup

Figure 11: Experimental Insertion Length vs. Input Energy
```
As can be seen, the data points in the graph above are consistent with the predicted insertion lengths from the analytical model.

**Analytical Model:**
To properly evaluate the taper lock mechanism, a model is needed to describe three important features. The first feature is the relationship between input energy and insertion length. This is important for analyzing the impact energy within the tool. The second function of the model is to estimate the maximum stress within the implant for a given insertion length. The force that the tool applies will have a factor of safety based on the maximum energy that can be applied before the yield stress of titanium is reached. The final function of the model is to evaluate the relationship between abutment retention force and input energy. With this relationship, testing can be conducted on samples to determine the actual energy that a clinician uses in the current installation method. Tensile testing can also be used to compare the current method of installation to the prototype design.

**Friction:**
An in-depth discussion of the assumptions, equations, and methods used in the analytical model are presented in Appendix A. A major concern with this model is the friction between the mating components of the system. The coefficients of dynamic and static friction between two pieces of 6Al4V titanium are not well defined in literature due to environment-based and surface condition variability. This grade of titanium is rarely used as both halves of mating components and therefore the friction factor is not readily available. The closest approximation to the coefficient is that of 6Al4V titanium on 1080 steel. The average coefficient for this type of surface-to-surface contact was found to be approximately 0.35 static and 0.25 dynamic [24].

Another concern is the surface finish of the components. This is critical to the assumptions concerning friction in the theoretical model. While it is difficult to evaluate the surface finish inside the implant, it is observed that the abutment post is either given a high-quality surface finish or coated. This will create a drastic change in the coefficients of friction. An example of this is steel. The coefficients of static and kinetic friction for hard steel on hard steel without lubrication are 0.78 and 0.42, respectively. When a thin coating of Teflon is added to one surface, both coefficients of friction reduce to 0.04 [25].

Bicon has not supplied specific material properties or manufacturing techniques for the taper lock mechanism. The material specifications for the abutment and implant have been collected from publicly available literature on the company’s website [1]. Other relevant data that was not provided includes surface finishes, coating information, and engineering drawings. For this reason, experiments will be conducted to estimate the coefficient of static friction for the abutment and implant. If the coefficient of static friction is low, the coefficient of dynamic friction may be estimated using static to dynamic friction ratios from other materials. If the coefficient static of friction is moderate to high based on the initial Ti6Al4V to 1080 steel approximation, an experiment will be devised to estimate the coefficient of dynamic friction. Further discussion of this experiment will be given in the project outlook section.
The current analytical model assumes the coefficients of static and dynamic friction are 0.13. This value is an approximation based on the assumption that the material behaves closer to Teflon coated titanium on titanium, than titanium on titanium.

This analytical results section is broken into three parts. The first part of the model is independent of friction. In this first section of analysis, the stress in the implant and abutment as a function of insertion length is presented. The laws of elasticity govern the relation and friction is not present in any of these laws. This portion of the analysis can be accepted as the final solution at this stage of the project.

The second part of the model is independent of the coefficient of static friction and only mildly dependent on the coefficient of dynamic friction. The insertion length as a function of input energy falls in this section. The coefficient of dynamic friction is present in calculations of the work to overcome friction during abutment installation. This is only one of the five terms present in the principle of work and energy equation, and does not dramatically change the results for small changes in the dynamic friction coefficient. For all practical purposes, it is impossible for this coefficient to exceed 0.25 [24] (the coefficient of dynamic friction for 6Al4V titanium on 1080 steel) and these results can therefore be accepted as good estimates for insertion length as a function of input energy.

The final part of the model is dependent on the static friction and slightly dependent on the coefficient of dynamic friction. As with second section of testing, the retention force as a function of input energy is somewhat dependent on the work to overcome friction during insertion. The retention force is directly proportional to the coefficient of static friction as shown in the calculations in Appendix B. The retention force is a rough approximation at this stage of the project since it is feasible for coefficient of static friction to vary between 0.35 and 0.04. A variation of this value would have a drastic effect on the calculated retention force.

**Stress versus Insertion Length:**

The maximum radial stress occurs at the interface of the abutment and the implant at the bottom edge of the abutment as represented in Figure 12. The radial stress is in compression for both the abutment and implant. The maximum tangential stress also occurs at the interface of the abutment and implant. The tangential stress is in compression in the abutment and tension in the implant. For this reason, the maximum planar shear stress is located on the bottom edge of the abutment. The results of the calculations are shown below in Table 3. A plot of maximum tangential stress versus insertion length is shown below in Figure 13. All yield stresses are divided by a factor of safety of 1.5 in order to produce working stresses.
Table 3: Stress from the Analytical Model

<table>
<thead>
<tr>
<th>Deflection</th>
<th>Max Normal Stress</th>
<th>Max Shear Stress</th>
<th>Max Radial Stress</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inches</td>
<td>Psi</td>
<td>Psi</td>
<td>Psi</td>
</tr>
<tr>
<td>0.00111</td>
<td>7092</td>
<td>4389</td>
<td>3986</td>
</tr>
<tr>
<td>0.00221</td>
<td>16863</td>
<td>10437</td>
<td>9508</td>
</tr>
<tr>
<td>0.00332</td>
<td>26634</td>
<td>16484</td>
<td>15066</td>
</tr>
<tr>
<td>0.00442</td>
<td>36405</td>
<td>22531</td>
<td>20657</td>
</tr>
<tr>
<td>0.00553</td>
<td>46176</td>
<td>28578</td>
<td>26282</td>
</tr>
<tr>
<td>0.00663</td>
<td>55947</td>
<td>34625</td>
<td>31939</td>
</tr>
<tr>
<td>0.00774</td>
<td>65717</td>
<td>40672</td>
<td>37629</td>
</tr>
<tr>
<td>0.00884</td>
<td>75488</td>
<td>46719</td>
<td>43350</td>
</tr>
<tr>
<td>0.00995</td>
<td>85259</td>
<td>52766</td>
<td>49102</td>
</tr>
</tbody>
</table>

Max Shear Stress: 79750 Psi
Working Shear Stress: 53167 Psi

Figure 13: Maximum Tangential Stress versus Insertion Length

Discussion
When impact energy is applied to the top of the abutment, the abutment moves into the implant a distance that is referred to as the insertion length. Evaluation of this event was performed experimentally using the test fixture described in the Experimental Analysis section above. Figure 14 displays a graph of insertion length versus input energy for both the experimental results and the analytical model.
The average difference between the measured insertion lengths and the theoretical insertion lengths is 0.001 inches. There are two factors that are believed to be the cause of this difference. The first cause is the measurement error. A caliper with a minimum scale of 0.001 inches was used to measure the insertion length, which results in a half-minimum-scale measurement error of ±0.0005-inches. The second cause of error is the friction between the fixture shafts and the falling weight. The interface between the two did not contain any lubrication, which results in a friction based energy loss rate. The goal is to estimate the frictional losses and adjust the input energy to incorporate these losses. Time-lapse photography or testing using free-falling weights could be used to evaluate the current system and to determine a function relating the frictional losses to input energy. As rationalized in the experiment section, free-falling weights are unpredictable; however, this concept can be used to find a crossover value to compensate for energy loss. Using the principle of work and energy, kinetic energy at the bottom of the striking path should equal potential energy at the top minus frictional losses. Through a series of tests, a frictional loss coefficient will be determined such that the following relation holds:

\[
\text{True Input Energy} = m \cdot g \cdot h - F^* \cdot h
\]

Where:

\[F^* = \text{Friction loss coefficient [oz \cdot in]}\]

More information about this test will be presented in the outlook section. Figure 15 displays the results of an analysis approximating \(F^*\) (frictional energy loss) as 0.25 oz\cdotin. Observation concludes that the corrected data corresponds well with the model. This demonstrates the need for further testing since approximating \(F^*\) improves
this error significantly. Without correction, the experimental data matches the theoretical model’s predictions fairly well, particularly the general power-function trend of both plots.

![Deflection vs. Input Energy (Projected)]

Figure 15: Insertion Length versus Input Energy if F+ = 0.25 oz(f)

The theoretical model is also able to predict a retention force for a given input energy. This theoretical result has a six percent error, which is acceptable for this purpose. The retention force can be calculated from the equation below:

$$F_{retention} = P \cdot A \cdot f_{static} = \sigma_s \cdot A_{contact} \cdot f_{static}$$

The results for these calculations are shown below in Table 4. There is currently no experimental data to compare these results to. One reassurance of the validity of the retention force is the previously discussed statement that an 8 oz-in energy produces a 90-lbf retention-force. The theoretical model predicts a retention force of approximately 95.4 pound for an eight oz-in input energy.

<table>
<thead>
<tr>
<th>Input Energy</th>
<th>Retention Force</th>
</tr>
</thead>
<tbody>
<tr>
<td>oz(\cdot)in</td>
<td>lbf</td>
</tr>
<tr>
<td>0.095</td>
<td>9.7</td>
</tr>
<tr>
<td>0.414</td>
<td>21.5</td>
</tr>
<tr>
<td>0.976</td>
<td>33.5</td>
</tr>
<tr>
<td>1.788</td>
<td>45.8</td>
</tr>
<tr>
<td>2.965</td>
<td>58.4</td>
</tr>
<tr>
<td>4.183</td>
<td>71.3</td>
</tr>
<tr>
<td>5.779</td>
<td>84.5</td>
</tr>
<tr>
<td>7.548</td>
<td>98.0</td>
</tr>
<tr>
<td>9.796</td>
<td>111.8</td>
</tr>
<tr>
<td>12.220</td>
<td>125.9</td>
</tr>
<tr>
<td>14.954</td>
<td>140.2</td>
</tr>
</tbody>
</table>
Tool Design
The conceptual design and theory behind the chosen design is given below:

First, general design considerations are discussed. The main objectives of this first design review are:
- To discuss how to improve upon the defined shortcomings of the current system
- To discuss the general concerns that would govern the success of the final design choice.

First, the current method is examined. While the single actuation of the current method is desirable in the final tool design, a major concern is the ergonomics of the current handle and mallet system. The handle is parallel to the applied force, which has the potential of creating inadequate seating, especially in the molar section of both the mandible and the maxilla. Considering the required compactness of the final tool, it is determined that there are a few different methods of redirecting the force perpendicular to the handle.

- Patent 3286558 [9] discussed the “Angularly Operable Head for Impact Tools” that is used for reciprocating tools. The design consists of a spring-loaded curved bar inside a similarly curved path at the head of the tool (Figure 7 above). When one end of the bar is force applied to it, the bar moves along the path to transmit the force. The spring returns the hammering piece to the original position to prepare for subsequent impacts. This design has some variability to the force it applies and was deemed insufficient for the goals of this project. The metal piece will have frictional losses inside the path of movement and depending on how the tool is held could cause a change in the frictional losses. Secondly, the tool is capable of binding at the tip, which is inadequate for an abutment-hammering tool. Finally, the prototype to be designed is focused on a single actuation. The spring and rod system is good for a cyclical motion, but does not have a consistent transient response at the start of cycling. This would produce variation in applied force and thus inadequate for the purpose presented. Altering the design to replace the bar with a semi-rigid bar was discussed, but rejected because of the variability in the spring.

- One method consists of a center-pin-supported beam inside the handle. This method will be acceptable only if the force to seat the abutment is small. In the case of a high force to seat the abutment, the required length of the lever arm will be too long to fit into a reasonable length handle.

- Another approach consisted of the striking mechanism transmitting the force along the length of the handle by some method. The force will then redirect the force perpendicular to the direction of the applied force. The method for redirecting the force is crucial for the success of this design.

- Hydraulics is also a means of redirecting the force with relation to the handle. Pressure in fluids acts in all directions, so a rigid cylindrical (or other shaped) elbow capped at both ends by movable pistons can easily redirect the force.

- Finally, the striking mechanism could be located in the head of the tool, thus requiring no redirection of force. Finding a striking mechanism compact enough to fit inside the head of a dental tool is a challenge for this method of force redirection.
During analysis of the force transitions to the head of the tool, the power sources were discussed. Without detailing the specifics of the design, the power sources can be divided into four groups.

**Electrical**

Electrical designs are limited by the availability of a power source in a typical environment for abutment installation. Electric designs also have the disadvantage of the cord to connect it to the power supply. A battery-powered tool could eliminate the cord connection, but would need to be recharged. The electrical designs are generalized into two main categories.

- Magnet systems exploit the attraction of the opposite poles of a magnet. Somewhere in the handle, a user-engaged button or switch would send current through a stationary electromagnet. This would cause an activation device with an attached magnet of opposite polarity to move, creating an impact force. Almost all of the force redirection methods work for the magnetic redirection method.

- Prior to in-depth analysis, the solenoid appeared to be ideal. A solenoid is a compact electromagnet that engages an impact mechanism for a short amount of time and then returns the striking mechanism to its original position. Solenoids are generally delicate devices but show the potential of being located directly in the head of the tool.

**Hydraulic**

- Hydraulic devices were examined because they apply a uniform pressure over a horizontal surface and have the ability to redirect force based on the physical properties of the fluid. The only loss in this type of device would be the frictional losses between the pistons on either end of the fluid and the wall of the tool handle.

**Pneumatic**

- Air power is examined carefully since many dental tools that require power use the compressors located in the office. These designs consisted of a piston that traveled down the length of the handle in order to store enough energy for the required impact. The o-rings attached to the piston would be chosen to have low friction with the walls of the handle and the piston would return to its original position by means of a compression or tension spring. A valve at the back of the tool would regulate the air pressure in the handle. If the required force was discovered to be relatively low and the required volume of air in the handle calculated to be small, the piston could move freely in the handle to create the impact. If a greater impact is needed, air could be allowed to build up behind the cylinder before a button releases the piston. The final velocity per unit length of the piston is greater in this method than in the previous method and would apply a larger impact load.
Mechanical

- Cam action is used in many dental tools and is powered by transferring linear air pressure to rotary action. This rotational motion can be transferred to linear action through differential gearing and a cam. Such a system would create reciprocating action and a device to limit the cam to one rotation would have to be devised in order to make this power source useful.

- The concept of this design is much like the hammer of a gun and examining the workings of a gun’s trigger mechanism is beneficial when considering the concept as a means of applying force. In trigger mechanisms, the hammer is rotated against the resistance of a spring until a bar comes in contact with a stop on the hammer. This bar is connected to the trigger in such a fashion that when the trigger is pulled, the stop on the hammer slips past the bar and the stored spring energy is used to drive the striking hammer. Automatic guns use the recoil in the explosion of bullets to reset themselves to the locking mechanism. The trigger mechanism in this device would have to be cocked prior to use.

- The compression spring system is much like the trigger mechanism but is arranged in a linear fashion. This device would be a two-action activation as well. First, a plunger at the back of the device (resembling the pull back device on older pinball machines) is pulled back and a mechanism in the handle would lock it in place. With the device loaded, pressing a button would release the catch on the compression spring and activate the tool.
Design Proposals
The elimination process is broken up into phases to accurately decide the optimal design for the project. The brainstorming described above can be considered "Phase 0". The other six phases are defined as follows:

**Phase 1: Functionality**
This step consisted of eliminating any design that did not meet the required functional specification. Some initial research is performed on the power sources and a brief discussion of potential failure modes of each device is given. Functionality and safety are examined for each proposed solution.

**Phase 2: Reliability**
Some designs are eliminated because they are potentially unreliable. Critical components were discussed to determine if variability was likely in the applied force. Designs will become more specific after this phase is complete.

**Phase 3: Manufacturability and Scale**
The manufacturability of the tool was an important characteristic since this device has the potential to attract a moderate to large market demand. The overall size of this device is very important because of the low clearance between both jaws. Elaborate mechanisms are generally discarded in this step and the simpler, more compact methods remain.

**Phase 4: Ease of Use**
In this step, specific failure and potential failures are discussed. The ease of assembly, complexity of the design and the ability to use the tool quickly and easily are evaluated for the remaining design. The goal in this project for phase four is to limit the potential solutions to three designs or less.

**Phase 5: Analysis of Desirable and Non-Desirable Attributes**
In this phase the pros and cons of each of the devices are listed in detail. More detailed drawings are created, and the potential failures at each of the components are analyzed briefly.

**Phase 6: Choice of Design**
The list from phase five is analyzed in this step of the review. First, the positive and negative aspects of each design are listed a pro to con ratio of the analysis in phase five is calculated. Using this as a tool, the final design is chosen. Final examination of the device is performed, and the analytical needs of the device are determined to analyze any potential failures that have been overlooked in the review. Finally, a consensus of the group will confirm the selection of the proposed solution.
The matrix presented in Table 5 shows the elimination process and reasons for elimination by phase. A more detailed analysis is presented in the next section.

<table>
<thead>
<tr>
<th>REDIRECTION</th>
<th>Phase Eliminated</th>
<th>Rationale</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hydraulic</td>
<td>Top Hit 1</td>
<td>Human contact with contaminant</td>
</tr>
<tr>
<td>Patent Concept</td>
<td>Forward Hit 2</td>
<td>Jamming</td>
</tr>
<tr>
<td>See-Saw</td>
<td>Top Hit 3</td>
<td>Long pivot arm</td>
</tr>
<tr>
<td>Power in head</td>
<td>Forward Hit 3</td>
<td>Long pivot arm</td>
</tr>
<tr>
<td>Linear fire to be redirected</td>
<td>Spring 3</td>
<td>Too small clearance to be effective</td>
</tr>
<tr>
<td>Hinge</td>
<td>4</td>
<td>Rotation of tip into tool after engagement</td>
</tr>
<tr>
<td>Hinge and Hammer</td>
<td>5</td>
<td>More cons in this concept</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>POWER</th>
<th>Phase Eliminated</th>
<th>Rationale</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electrical</td>
<td>1</td>
<td>Autoclave</td>
</tr>
<tr>
<td>Magnet</td>
<td>1</td>
<td>Autoclave</td>
</tr>
<tr>
<td>Solenoid</td>
<td>1</td>
<td>Autoclave</td>
</tr>
<tr>
<td>Hydraullic</td>
<td>Cam 2</td>
<td>Human contact with contaminant</td>
</tr>
<tr>
<td>Gun Hammer</td>
<td>3</td>
<td>Variability in cam zero position, complex</td>
</tr>
<tr>
<td>Torsion Hammer, Spin Disc</td>
<td>3</td>
<td>Will not work well with tolerancing</td>
</tr>
<tr>
<td>Button trigger, return in button</td>
<td>3</td>
<td>Will not work well with tolerancing</td>
</tr>
<tr>
<td>Pneumatic</td>
<td>6</td>
<td>More cons in this concept</td>
</tr>
<tr>
<td>Straight Air</td>
<td>4</td>
<td>Valve and position simultaneously</td>
</tr>
<tr>
<td>Pneumatic button</td>
<td>6</td>
<td>More cons in this concept</td>
</tr>
<tr>
<td>Compression Spring</td>
<td>7</td>
<td>Slide is not user friendly</td>
</tr>
<tr>
<td>Locking Slide method</td>
<td>4</td>
<td></td>
</tr>
<tr>
<td>Locking Pinball method</td>
<td>9</td>
<td></td>
</tr>
</tbody>
</table>
Analysis of Proposed Designs

Phase 1: Functionality

Force Redirection

Certain factors are discovered in researching the force redirection methods. First, hydraulic redirection can be eliminated because leaking of the device would create a faulty device. Many hydraulic fluids are not recommended for human consumption, and leaking of the tool during use could be potentially harmful to the patient. Consumable fluids were investigated, a list of a few are in Table 6. These types of fluids, while digestible, could cause the patient to choke if leaking occurred during a procedure. The effects of the autoclavable cycles on the fluids are also an important factor in analyzing this method. Because this method would have many issues that could result in the injury of the patient, this method of force reduction was eliminated from the possible design considerations.

Table 6: Consumable Lubricants

<table>
<thead>
<tr>
<th>Product Name</th>
<th>Component</th>
<th>Temperature</th>
<th>Grade</th>
</tr>
</thead>
<tbody>
<tr>
<td>PQ 40 AA-1</td>
<td>Anderal Whit Oil Based Grease</td>
<td>-10 to 145</td>
<td>H-1</td>
</tr>
<tr>
<td>PQ 40 AA-2</td>
<td>Anderal Clear and White Oil Based Grease</td>
<td>-10 to 145</td>
<td>H-1</td>
</tr>
<tr>
<td>783</td>
<td>Anderal PAO Oil Based Grease</td>
<td>-40 to 149</td>
<td>H-1</td>
</tr>
<tr>
<td>785</td>
<td>Anderal PAO Oil Based Grease</td>
<td>-40 to 149</td>
<td>H-1</td>
</tr>
<tr>
<td>Wafer Grease</td>
<td>Anderal PFPE Oil Based Grease</td>
<td>-40 to 240</td>
<td>H-1</td>
</tr>
<tr>
<td>PQ AA 10</td>
<td>Anderal White Oil</td>
<td>-15 to 180</td>
<td>H-1</td>
</tr>
<tr>
<td>FG 10</td>
<td>Anderal PAO Oil</td>
<td>-50 to 260</td>
<td>H-1</td>
</tr>
<tr>
<td>HTC PG 150</td>
<td>Anderal PAG Oil</td>
<td>-29 to 230</td>
<td>H-1</td>
</tr>
</tbody>
</table>

Note: H-1 grade lubricants are permitted to have incidental contact with foods during processing.

Power Source

Because of human factors and functionality of the device, two categories of power sources were eliminated: electrical and hydraulic.

Hydraulic power has the benefit of applying force and redirecting it. It does, however, have safety issues concerning fluid contact with human tissue. Because of the possible safety risk, this power source is not considered feasible.

Electrical devices are also eliminated in this phase. First, there are potential safety issues with electrical equipment with human tissues. While safety can be built into such a device, the tool will also have issues with autoclavability. Solenoids are very delicate and because of this, the internal wires can easily melt under high temperatures. Specialty solenoids are available that can be autoclaved, but they are expensive and sensitive to vibration. The main failure mode of a solenoid occurs when the current being applied through the device is applied over an extended period of time. The wires in the solenoid coil heat up quickly and melt, shorting out the coil.
Magnetic actuation is also considered as part of the electrical device elimination. While the magnets in the device are less flimsy than solenoids, there are still material concerns at higher temperatures. Because of this, the entire range of electrical devices will be disregarded as adequate designs.

**Phase 2: Reliability**

**Force Redirection**

As discussed above, the redirection method displayed in patent number 3286558 [10] has some major issues that have the potential of causing the tool to bind. Because this device could not guarantee a consistent and reliable force, it was eliminated from the list of possible problem solutions.

**Power Source**

Cam power is eliminated at this stage because of the complexity in making this power source reliable. The main issue with the cam system is the location of the cam when the rotary motion begins. As illustrated in Figure 19, the force could vary if the cam had a full half-rotation or if only a quarter-rotation was obtained. This type of device would be hard to control in a single actuating device. The rotation of the rotary power behind the cam would have to coincide with exactly one rotation of the cam. If it did not line up precisely, the small difference in angle of rotation would build up over time and cause the decay of the applied force. Also, in this device, the hammering head would have to move a considerable amount in order to allow the cam to return to the “home” position. This method is rejected because the reliability is questionable and the other designs are simpler and more reliable.

**Phase 2 (Supplementary): Defining specifics of general devices:**

The remaining phases rely heavily upon the specifics of the design. Each power source and force redirecting mechanism is examined and one to three specific designs were modeled for each. The following are the more defined solutions for the problem.

---

**Figure 18: Magnetic**

**Figure 19: Cam Action**
Table 7: Specific Designs

<table>
<thead>
<tr>
<th>REDIRECTION</th>
<th>POWER</th>
</tr>
</thead>
<tbody>
<tr>
<td>01 Seesaw – upward force redirection</td>
<td>07 Pneumatic – Air valve on to projectile</td>
</tr>
<tr>
<td>02 Seesaw – forward force hit</td>
<td>08 Pneumatic – Air valve on to load, release button projectile</td>
</tr>
<tr>
<td>03 Linear Strike – Hinge</td>
<td>09 Trigger – Torsion hammer spin disc</td>
</tr>
<tr>
<td>04 Linear Strike – Hinge and Hammer</td>
<td>10 Trigger – button trigger return in button</td>
</tr>
<tr>
<td>05 Linear Strike – Round floating piece, non moving</td>
<td>11 Spring – Locking pinball method</td>
</tr>
<tr>
<td>06 Head power – spring in head</td>
<td>12 Spring – Locking slide method</td>
</tr>
</tbody>
</table>

Phase 3: Manufacturability and Scale

Force Redirection

While the pinned beam is a valid method for redirecting force, the size of such a device would need to be large for high forces. To produce a retention force of 90 lbs., the impact force will need to be high. To create such a high force, the power supply would have to transmit a large force or the angle of rotation of the beam would have to be high if the pin is close to the hammering tip. Increasing the power output of the power supply is detrimental to this device. One goal of the redirection method is to create a high impact force from a small amount of input energy from the power supply. The only other option for this design would be to increase the lever arm transmitting the force. This method would be difficult to fit inside a handle since the length of the handle would be long and the diameter of the handle limits the amount of angular deflection that can be achieved. For these reasons, this method will no longer be considered. This method is being eliminated for both the top hit and forward hit concepts since both would involve long pivot arms.

Figure 20: Pinned Beam Method
Size of the head is crucial for this design and should be kept as small as possible due to the clearance between the mandible and maxilla, especially in the molar section. Due to this constraint, the trigger method as presented is also eliminated from consideration. While this type of mechanism is an excellent example of a single actuating hammer, the amount of clearance needed for the rotating hammer is not available in the head of the design.

![Figure 21: Button Trigger Mechanism and Basic Trigger Mechanism](image)

Both methods for trigger activation are eliminated because of the tight tolerance of clearance between human jaws. Unless the hammer mechanism can be altered some way, it will not be considered as feasible for the design.

**Power Source**

The size and manufacturability of the hammering device plays an important role in the power choice. Having the power source in the head of the device is not feasible with the remaining designs, so this option is eliminated from the list of choices. The only potential mechanism for this method is placing a compression spring in the head. This application would be difficult since the action used to compress the spring would be awkward in the head of the device. The loading mechanism would be complex, since placing the pull back close to the head would create a potential pinch point inside the mouth. Because this concept is neither safe nor simple, it will not be considered.

This power source will not be considered in further analysis.

**Phase 4: Ease of Use**

**Force Redirection**

The hinge design without a striking head is not desirable for this device. The device should be closed off from human contact and the force act perpendicular to the handle. If the head is not installed in the tool, the hinge will be the mechanism hitting the abutment. The hinge would strike and most likely bounce back into the tool. The tip of the device could not easily be adapted to the hinge for different sizes and shapes of abutments because the whole striking head would be rotating the attachment into the handle.

**Power Source**

In pneumatic devices, the ease of use for the valve actuating design has several problems. First, the valve would need to be turned on with the tool head in place and aligned. The release valve would also need to be opened at the
conclusion of the procedure to allow the piston to return to its zero position. The button design for the pneumatic device eliminates this error.

In the mechanical devices, the sliding lock is less user-friendly than the pinball lock. A plunger on the end of the handle will allow leverage for pulling back on the compression spring. A mechanical slide along the length of the handle is awkward to use, especially since a sliding piece moving up and down the length of the shaft could cause pinch points.

**Phase 5: Analysis of Desirable and Non-Desirable Attributes**

*Force Redirection*

Based on the above analysis, the hinge mechanism is the optimal choice to redirect the force 90°.
Power Source

This phase will analyze each design’s desirable and undesirable attributes. The two designs that remain are the pneumatic projectile and the locking “pinball” methods. The following table indicates the desirable and non-desirable traits of each:

<table>
<thead>
<tr>
<th></th>
<th>Pneumatic Projectile</th>
<th>Locking Pinball Mechanism</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Desirable</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>• Supplied power</td>
<td></td>
<td>• Less complicated</td>
</tr>
<tr>
<td>• Ease of use</td>
<td></td>
<td>• Less design time</td>
</tr>
<tr>
<td>• One motion, press button</td>
<td></td>
<td>• Versatile trigger</td>
</tr>
<tr>
<td>• Lightweight</td>
<td></td>
<td>• No sealing problems</td>
</tr>
<tr>
<td>• Size is small</td>
<td></td>
<td>• Internal path can be</td>
</tr>
<tr>
<td></td>
<td></td>
<td>anything if assembled</td>
</tr>
<tr>
<td></td>
<td></td>
<td>with half-handles</td>
</tr>
<tr>
<td></td>
<td></td>
<td>• Ease of assembly</td>
</tr>
<tr>
<td></td>
<td></td>
<td>• Off-the-shelf comp.</td>
</tr>
<tr>
<td></td>
<td></td>
<td>• Inexpensive</td>
</tr>
<tr>
<td></td>
<td></td>
<td>• Cordless</td>
</tr>
<tr>
<td><strong>Undesirable</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>• O-ring wear</td>
<td></td>
<td>• 2-motion device</td>
</tr>
<tr>
<td>• Cost/complexity</td>
<td></td>
<td>(load then release)</td>
</tr>
<tr>
<td>• Hoses attached</td>
<td></td>
<td>• Wear in components</td>
</tr>
<tr>
<td>• Placement of the</td>
<td></td>
<td>(springs)</td>
</tr>
<tr>
<td>trigger is defined</td>
<td></td>
<td>• Force to pull plunger</td>
</tr>
<tr>
<td>• Handle must be</td>
<td></td>
<td>• Heavy</td>
</tr>
<tr>
<td>sealed</td>
<td></td>
<td>• Firing ball end</td>
</tr>
<tr>
<td>• Lubrication problems</td>
<td></td>
<td>creates a pinch point</td>
</tr>
<tr>
<td>• Value problems</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Phase 6: Choice of Design

Force Redirection

The hinge mechanism has previously been chosen as the method of redirecting force.

Power Source

From the analysis in phase 5, the pro-to-con ratio is 6/7 for the pneumatic device and is 9/5 for the mechanical. While this does not indicate the level of severity of the cons for each design, it can also be seen that failure in the pneumatic system has a higher potential of injuring the patient. Lubrication would also be a problem, and o-rings would wear quickly due to the long linear path it would need to travel per each activation.

As in any design, the chosen design has a few critical components that would be a major source of a potential failure in the design. Theoretical analysis is performed to design around these potential failure modes.
Theoretical Analysis of Chosen Design

The proposed design is shown in below:

![Proposed Solution]

**Figure 22: Proposed Solution**

The following is an analysis of parts of the assembly that could cause failure if not analyzed properly before implementation. Dimensioned drawings detailing the components of the assembly shown in the figure above are given in Appendix D.

Use of this device consists of two steps. In the first step, energy is stored in the compression spring by pulling back on the plunger. The latch is designed to hold the plunger in the loaded position until the button is pressed. To use the device, a clinician would load the device and then line the head of the tool with the loose and aligned abutment. After lining the tool up at the proper angle, a push of the button will release the button. The plunger will be forced forward and strike the hinge. The hinge will redirect the force 90°, and the energy will be transferred and consumed in pressing the abutment and implant together.
Hinge Pin Analysis

The trigger mechanism is constrained within a small area of the handle. For this reason, clearance of the trigger lock is an important feature of this design. Figure 23 displays the trigger lock in both the loaded and released positions. The distance D is the original distance between the inside of the handle and the top of the trigger lock. The distance C represents the length of the trigger lock that is in contact with the ring. This is also the distance that the trigger lock will need to displace in order to release the shaft. When the trigger is released, the distance J represents the final clearance between the top point of the trigger lock and the inside of the handle. If J is greater than zero the design will work in theory, however a minimum clearance of .015 inches is needed to allow for machining tolerances.

Figure 23: Clearance for the Trigger Lock Mechanism

Figure 24 displays an isolated view of the trigger lock in both the locked and unlocked positions.

Figure 24: Trigger Lock Clearances Dimensions

The following trigonometric analysis can be used to find X, θ and Z:
\[ X = Y - C = 0.090 \text{ inches} \]
\[ \theta = \arcsin(C/T) = 23.6 \text{ degrees} \]
\[ Z = (S \cdot \sin(\theta)) + H = 0.099 \text{ inches} \]

With \( Z \) now known, the clearance height \( J \) can be computed as follows:

\[ J = D - (Z - H) = 0.026 \text{ inches} \]

Since \( J \) is greater than 0.015 inches, clearance will not prove to be a problem if the machining is completed within the specified tolerances.

**Stress in the Pins:**

One of the major concerns with the abutment hammer design is the stresses in the two pins. Both the trigger pin and the hinge pin are subject to loads that will produce bending stresses. The trigger pin, shown below in Figure 26, will experience a static force. The small diameter of this pin creates a strength concern that warrants a stress analysis.

![Figure 25: The Trigger Pin](image)

The hinge pin, depicted below in Figure 26, is subject to an impact force. While the diameter of this pin is three times greater than the diameter of the trigger pin, the potential for high stresses resulting from impact loading creates a need for this stress analysis.

![Figure 26: The Hinge Pin](image)
Figure 27 presents the general results for a simply supported circular beam subject to a uniform load. These results include: maximum moments, shear force and normal stress from bending. These equations, as well as the shear and moment diagrams, hold true for both the trigger and hinge pins.

**Shear and Moment Diagram**
- Simply supported beam model

**Reaction Forces and Uniform Load**
\[
\begin{align*}
w &= \frac{F}{L} \\
Ra &= \frac{F}{2} \\
M_{max} &= \frac{F \cdot L}{2} - \frac{F \cdot L}{8} = \frac{3}{8}FL \\
\end{align*}
\]

**Maximum Shear and Moment**
\[
\begin{align*}
V_{max} &= Ra = \frac{F}{2} \\
M_{max} &= Ra \cdot x - w \cdot x^2 \\
\end{align*}
\]

**Bending Stress**
\[
\sigma_{max} = \frac{M_{max}}{I} = \frac{3/8 \cdot F \cdot L \cdot (D/2)}{\pi \cdot D^4/64} = \frac{12 \cdot F \cdot L}{\pi \cdot D^3}
\]

Figure 27: Simply Supported Circular Beam in Bending

**Trigger Pin Stress Analysis:**
The hinge pin is subject to a static load of 3.75 pounds. This load is a result of the force stored in the shaft spring when the tool is in the loaded position (as explained in the spring calculation section). This force can be directly input to equation 1 of Figure 27 to determine the maximum bending stress in the trigger pin. With the knowledge that the pin diameter is 0.040 inches and the free length of the pin is 0.25 inches, the maximum normal stress is computed to be 56,700 psi. The yield stress of 6Al4V titanium in tension is 127,633 psi. With a factor of safety of 2, the working stress becomes 63,816 psi. The working stress is more than 7,000 psi greater than the actual stress; therefore, the hinge pin will function adequately. It should also be noted that a safety pin is added to prevent the shaft from being pulled back beyond the required distance. If this safety were not added, it would be possible for the dentist to pull the shaft beyond the loaded position. The clinician would release the shaft and an impact loading would be produced on the trigger lock. This could result in failure of the trigger pin.

**Hinge Pin Analysis**
The analysis of the hinge pin is similar to the trigger pin analysis, however, an impact force will need to be determined. The diameter of this pin is 0.125 inches, while its length is only 0.25 inches. Due to the small diameter
to length ratio, shear stress from bending will also be examined. To estimate the impact force, the principle of work and energy will be employed.

\[ E = F' \cdot X \] (assumes that the force is constant throughout the distance X)

Where:
- \( E \) = Input Energy = 9.55 ounce inches
- \( F' \) = Impact Force
- \( X \) = Distance that \( F' \) act through = 0.00478 inches (See Figure 27)

From this equation, the impact force is calculated to be 128.9 pounds. Impact force is an abstract concept that warrants explanation. For purposes of this analysis, it is convenient to define an impact force rather than impact stresses. Impact stresses are computed as the product of the static stress and an impact factor \((S' = S \cdot n)\). Because the static stress and the impact stress have the same cross sectional areas, these stresses can be converted directly to forces. The conclusion drawn from this analysis shows that the impact force is equivalent to the product of the impact stress and contact area.

The physical concept of an impact force is also important. Impact force refers to the static force that would produce an equivalent stress to that seen in the actual dynamic process. In this case, a 1-ounce weight dropped 9.55 inches will produce the same stresses in the hinge pin as a 128.9-pound weight placed statically on the pin.

Equation 1 from Figure 27 can be used to find the maximum normal stress in the hinge pin. The following analysis will also compute the maximum shear stress form bending.

\[
\sigma_{\text{max}} = \frac{12 \cdot F' \cdot L}{\pi \cdot D^3} = \frac{12 \cdot 128.9 \text{ lbs} \cdot 0.25 \text{ in}}{\pi \cdot (0.125 \text{ in})^3}
\]

\[
\sigma_{\text{max}} = 63,022 \text{ psi}
\]

\[
V_{\text{max}} = F'/2 = 64.4 \text{ lbs.}
\]

\[
\tau_{\text{max}} = \frac{4 \cdot V}{I \cdot t} = \frac{4 \cdot V}{\pi \cdot r^2}
\]

\[
\tau_{\text{max}} = 7,002 \text{ psi}
\]
The yield stress of 6Al4V titanium in tension is 127,633 psi. With a factor of safety of 2, the working stress becomes 63,816 psi. The maximum normal stress is approximately 800 psi below the working stress so bending will not cause failure in the hinge pin. The ultimate shear stress for titanium is 79,750 psi. With a factor of safety of 2 the working shear stress becomes 39,860 psi. The working shear stress is well above the actual shear stress. This analysis shows that the hinge pin will be able to withstand this impact load.

**Transfer Head Spring**

The abutment-hammering tool contains two compression springs. The first compression spring, namely the transfer spring, is located in the head of the tool. The primary function of this spring is to return the hammer to its original position after abutment installation has occurred. The free length of this spring is 0.38 inches and the maximum load that this spring can withstand is 2.4 pounds. The spring can be analyzed in two stages (as shown below in Figure 28). In the first stage the spring has been compressed from its original free length (A) to its nominal length. The tool is now in the rest position where the operator has not depressed the trigger. In the second stage, the transfer shaft is in the operating position. The trigger is depressed and the abutment is now installed. An additional compression length (C) is now added to the spring. For purposes of this analysis the deflection will be assumed to be 0.007 inches, as this is what the model predicts for abutment insertion length.

![Figure 28: Transfer Spring Compression](image)

The energy stored in the spring as well as the spring force at the nominal and final displacements can now be found. The spring is constrained by the following two equations [27]:

\[
F_{spring} = k \cdot \delta \\
PE_{spring} = \frac{1}{2} \cdot k \cdot \delta^2
\]

Applying the following two equations for spring force and potential energy stored in a spring as well as the spring constant (k), Table 9 can be computed to show the force and energy at both the nominal and final positions.
Table 9: Spring Force and Energy in the Transfer Spring

<table>
<thead>
<tr>
<th>Nominal Position</th>
<th>Final Position</th>
</tr>
</thead>
<tbody>
<tr>
<td>E (oz-in)</td>
<td>K (lbs./in)</td>
</tr>
<tr>
<td>1.35</td>
<td>10</td>
</tr>
</tbody>
</table>

From this information, it is clear that in addition to the energy needed to seat the abutment, an additional 1.5 ounce-inches of energy will be needed to compress the transfer spring. It should also be noted that the force at the final stage is only 57 percent of the maximum force that the spring can withstand.

**Shaft Spring**

The shaft spring is shown below in Figure 29.

The length of the spring prior to the input force is referred to as original length (A). The length of the spring when the trigger is loaded is referred to as the loaded length (B). In stage 0 the spring is at the original length. In stage 1 the user pulls the shaft back to compress the spring to the loaded length. Stage 2 consists of the steady state condition after the trigger has been released and the shaft has contacted the transfer head. Stage 2 is identical to stage 1.
Applying the same equations that were used in the transfer spring analysis, the required deflection for a given energy storage can be determined. The associated spring force can also be determined for a given energy storage. The results of these calculations are shown below in Table 10.

<table>
<thead>
<tr>
<th>E (oz-in)</th>
<th>K (lbs/in)</th>
<th>δ (in)</th>
<th>F (lbs)</th>
</tr>
</thead>
<tbody>
<tr>
<td>12.4</td>
<td>12.0</td>
<td>0.36</td>
<td>4.3</td>
</tr>
<tr>
<td>11.8</td>
<td>12.0</td>
<td>0.35</td>
<td>4.2</td>
</tr>
<tr>
<td>11.1</td>
<td>12.0</td>
<td>0.34</td>
<td>4.1</td>
</tr>
<tr>
<td>10.5</td>
<td>12.0</td>
<td>0.33</td>
<td>4.0</td>
</tr>
<tr>
<td>9.83</td>
<td>12.0</td>
<td>0.32</td>
<td>3.8</td>
</tr>
<tr>
<td>9.23</td>
<td>12.0</td>
<td>0.31</td>
<td>3.7</td>
</tr>
<tr>
<td>8.64</td>
<td>12.0</td>
<td>0.30</td>
<td>3.6</td>
</tr>
<tr>
<td>8.07</td>
<td>12.0</td>
<td>0.29</td>
<td>3.5</td>
</tr>
<tr>
<td>7.53</td>
<td>12.0</td>
<td>0.28</td>
<td>3.4</td>
</tr>
<tr>
<td>7.00</td>
<td>12.0</td>
<td>0.27</td>
<td>3.2</td>
</tr>
<tr>
<td>6.49</td>
<td>12.0</td>
<td>0.26</td>
<td>3.1</td>
</tr>
</tbody>
</table>

While this table presents the results for a series of input energies, it is important to choose the appropriate input energy and design the tool for that deflection. It is assumed that the linear bearing has no frictional losses and the only two energy losses within the tool are stressing of the hinge pin and compression of the transfer spring. With these assumptions, the input energy is simply the sum of the required energy for installing the abutment and the energy stored in the transfer spring. These energies add up to a total input energy of 9.55 ounce inches. With this input energy the actual displacement and spring force are 0.315 inches and 3.75 pounds, respectively.

**Assembly Issues**

Because this design is powered by mechanical means, there are no sealing requirements for the design. Because of this, the handle will be designed as half handles for ease of assembly. Manufacturing of these handles will have a tight tolerance for all the pins and bolts in order to mate the pieces together properly. The outer diameter will be contoured as designated by industrial design (discussed below). The inner diameter will be milled to create such features as countersinks for pins and milled paths for bearings and other tool components. Once assembled, the half handles will be placed together and screwed in place with four screws. These screws will be set inside the handles to increase handling comfort during use.
Regulatory
From research into the government regulations needed to market this design, it was determined that little regulatory work needs to be performed. As with any device, there is a procedure for good manufacturing practice [27] that assures that the design is safe to be used in surgical or clinical procedures. This document is in place to assure that the tool design is safe and has been tested sufficiently to assure no injury will occur to the patient as a result of tool failure. This device is classified as a class 1 device, meaning that it is an accessory to noninvasive surgery [28]. There is also no external power connected to the device, which eliminates many regulations from applying to this design. Because of the class and mechanical nature of the design, this device needs no pre-market 501k-approval. 501k approval requires a review of the design by the FDA for safety during use, and usually applies to tools for invasive procedures or devices that have the potential to injure a patient if safety are not in place.

Industrial Design
The following human factors will be taken into consideration when designing the handle for the abutment hammer:

- The first plane allows palmar flexion or when performed in opposite direction dorsiflexion.
- The second plane consists of ulnar or radial deviation.

Due to the configuration of the wrist joint, the hand can move in two planes of movement. These planes are demonstrated in Figure 30:

One of the main goals in the design of a handle is to maintain a straight wrist. This prevents bending and bunching of the tendons in the carpel tunnel, which mainly occurs in palmar flexion or ulnar deviation. Design of the dental hammer without considering any human factors would force the wrist into ulnar deviation. The key rule in handle design is to prevent or minimize ulnar deviation. Ideally, the handle should have a slight bend to avoid this, but by designing the angle of the striking head to be 90°, this bend has been compensated for. The angling also allows the tool to get to the hard-to-reach areas at the back of the mouth.

Another goal in handle design is to avoid tissue compression. The dental abutment hammer is not held for considerable lengths of time and therefore will not greatly affect the ergonomics of our handle design. To minimize tissue compression, contours will be added to the handle. The handle will be designed to have a large enough contact surface to distribute the force over a larger area of the hand. The handle will also be designed to direct it to less-sensitive areas, such as the tough tissue between thumb and the index finger. [29]
It is also desirable to avoid repetitive finger action. As a rule, frequent use of the index finger should be avoided and thumb-operated controls should be used. To satisfy this, the trigger button will be located in the area where the thumb will rest. Figure 31 shows a conceptual ergonomic design.

Due to time constraints and policies of the Division of Institutional Compliance of Northeastern University concerning external surveys, the dentist survey of dental handles will not be performed. In place of this analysis, analysis of the handle designs of current dental tools will be examined to determine the optimal design. This data will compliment the ergonomical data presented above.

**Economical analysis**

Based on the above design and the parts reflected in the engineering drawings (SEE Appendix D), a preliminary cost analysis produced the following results [30][35]:

<table>
<thead>
<tr>
<th>Part</th>
<th>Part No.</th>
<th>Supplier</th>
<th>Supplier Part No.</th>
<th>Material</th>
<th>Cost (stock)</th>
<th>Initial Total Cost</th>
</tr>
</thead>
<tbody>
<tr>
<td>Housing, Left</td>
<td>PBM-101L</td>
<td>McMaster-Carr</td>
<td>9083K21</td>
<td>SS 316</td>
<td>$67.74/3 ft.</td>
<td>$22.40</td>
</tr>
<tr>
<td>Housing, Right</td>
<td>PBM-101R</td>
<td>McMaster-Carr</td>
<td>9083K21</td>
<td>SS 316</td>
<td>$67.74/3 ft.</td>
<td>$22.40</td>
</tr>
<tr>
<td>Shaft</td>
<td>PBM-105</td>
<td>McMaster-Carr</td>
<td>8636K13</td>
<td>SS 316</td>
<td>$17.56/6 ft.</td>
<td>$4.42</td>
</tr>
<tr>
<td>Ball Handle</td>
<td>PBM-107</td>
<td>MSC</td>
<td>6671248</td>
<td>Brass*</td>
<td>$6.69</td>
<td>$6.69</td>
</tr>
<tr>
<td>Linear Bearing</td>
<td>PBM-109</td>
<td>McMaster-Carr</td>
<td>6483K11</td>
<td>SS</td>
<td>$22.04 ea</td>
<td>$22.04</td>
</tr>
<tr>
<td>Retention Ring</td>
<td>PBM-115</td>
<td>McMaster-Carr</td>
<td>6483K22</td>
<td>SS 316</td>
<td>$33.53/ft.</td>
<td>$2.60</td>
</tr>
<tr>
<td>Hinge Pin</td>
<td>PBM-123</td>
<td>McMaster-Carr</td>
<td>69055K111</td>
<td>TIBA4AV</td>
<td>$22.50/ft.</td>
<td>$1.23</td>
</tr>
<tr>
<td>Push Button</td>
<td>PBM-205</td>
<td>McMaster-Carr</td>
<td>6483K22</td>
<td>SS 316</td>
<td>$2.80</td>
<td>$2.80</td>
</tr>
<tr>
<td>Trigger Lock</td>
<td>PBM-201</td>
<td>McMaster-Carr</td>
<td>9083K21</td>
<td>SS 316</td>
<td>$0.84</td>
<td>$0.84</td>
</tr>
<tr>
<td>Trigger Pin</td>
<td>PBM-207</td>
<td>McMaster-Carr</td>
<td>69055K111</td>
<td>TIBA4AV</td>
<td>$0.84</td>
<td>$0.84</td>
</tr>
<tr>
<td>Transfer Head</td>
<td>PBM-121</td>
<td>McMaster-Carr</td>
<td>9083K21</td>
<td>SS 316</td>
<td>$2.00</td>
<td>$2.00</td>
</tr>
<tr>
<td>Hammer</td>
<td>PBM-125</td>
<td>McMaster-Carr</td>
<td>6483K16</td>
<td>SS 316</td>
<td>$40.48/6 ft.</td>
<td>$0.25</td>
</tr>
<tr>
<td>Torsion Springs (2)</td>
<td>PBM-209</td>
<td>N/A</td>
<td>TED</td>
<td>SS</td>
<td>$4.00</td>
<td>$4.00</td>
</tr>
<tr>
<td>Shaft Spring</td>
<td>PBM-111</td>
<td>Century Spring</td>
<td>E15-60</td>
<td>SS</td>
<td>TED</td>
<td>TED</td>
</tr>
<tr>
<td>Transfer Spring</td>
<td>PBM-112</td>
<td>Century Spring</td>
<td>0C-86</td>
<td>SS</td>
<td>TED</td>
<td>TED</td>
</tr>
<tr>
<td>Ring Set Screws (2)</td>
<td>PBM-113</td>
<td>McMaster-Carr</td>
<td>9231A146</td>
<td>SS 316</td>
<td>$22.85/100</td>
<td>$0.45</td>
</tr>
<tr>
<td>Set Screw, Head</td>
<td>PBM-102</td>
<td>McMaster-Carr</td>
<td>92313A000</td>
<td>SS 316</td>
<td>$100.00/100</td>
<td>$1.00</td>
</tr>
<tr>
<td>Bearing Screws (4)</td>
<td>PBM-117</td>
<td>McMaster-Carr</td>
<td>91735A148</td>
<td>SS 316</td>
<td>$17.17/100</td>
<td>$0.69</td>
</tr>
<tr>
<td>Handle Screws (4)</td>
<td>PBM-104</td>
<td>McMaster-Carr</td>
<td>91735A148</td>
<td>SS 316</td>
<td>$17.17/100</td>
<td>$0.69</td>
</tr>
<tr>
<td>Washers (4)</td>
<td>PBM-119</td>
<td>McMaster-Carr</td>
<td>90107A007</td>
<td>SS 316</td>
<td>$2.54/100</td>
<td>$0.10</td>
</tr>
<tr>
<td>Lock Washers (4)</td>
<td>PBM-127</td>
<td>McMaster-Carr</td>
<td>92147A420</td>
<td>SS 316</td>
<td>$2.01/100</td>
<td>$0.08</td>
</tr>
<tr>
<td>Whole Assy</td>
<td>PBM-100</td>
<td>N/A</td>
<td>TED</td>
<td>SS</td>
<td>363.21</td>
<td>$100.12</td>
</tr>
</tbody>
</table>

***Does not include labor

Based on this analysis, this tool is more than economical to produce. Most components can be bought as off-the-shelf components, which reduces the cost of the tool. A cost of $363.21 is estimated at this time, but the cost of the tool excluding labor should be $100.00. This cost would be considerably reduced if stock were bought in larger quantities.
Conclusions

Analysis of the State of the Art Review

From the literature researched, no available device was found that would effectively solve the problem described. Many impact devices were evaluated and can be categorized by power source into four groups: mechanical, pneumatic, electric, and hydraulic. Many single actuation devices were researched to determine the effectiveness of general hammering tools. In searching for specific dental tools and a broad scope of hammering tools, no devices were found that could be feasibly adapted for the dental industry and provide a consistent hammering force. Because of the established need for this tool and the inability to find an existing solution, design of this innovative project is necessary to solve the abutment-hammering problem.

Evaluation of the Taper Lock Mechanism:
The evaluation of the taper lock mechanism was divided into mathematical modeling and experimental testing. Literature searches produced no valuable models for analysis of locking tapers.

Mathematical modeling

A model for evaluating the stresses in the implant and abutment post has been developed. Further experiments to determine the friction factors are needed before this model is completed.

Experimental testing

Testing was conducted on samples in order to validate the mathematical models. Insertion length was measured as a function of known input energies. This is related directly to the insertion lengths in the mathematical analysis with some calculable error due to friction in the test setup. The values obtained from this experiment coincide with the values obtained from the models, and thereby validate the theoretical analysis.

Tool Design:
The tool design is highly dependent on the force requirements for the hammering mechanism. Analysis of the tool consisted of two parts: the force redirection mechanism and the power source. Initial research into the available types of power in a dental office was completed. This search revealed that compressed air with a line pressure of approximately 40 psi and 110 volt standard electricity is available. Initial research of air, electric, hydraulic, and mechanical powered devices was conducted to reveal the design of different hammering mechanisms. Once the desired forces were specified, an in-depth evaluation of power sources and hammering mechanisms was completed and divided into four types: mechanical, electrical, pneumatic, and hydraulic. Different types of force redirection methods were also examined. By a seven-phase analysis method, proposed solutions were eliminated until the optimal design was chosen. Phase zero of the analysis involved brainstorming. The remaining six phases involved narrowing the potential choices down to one choice. With the given solution, theoretical analysis of the main components was performed and the first revisions of engineering drawings were drafted.
Industrial Design

The ergonomic design of the handle will be designed to conform to the natural anatomy of the hand. Factors to reduce stresses and occurrences of repetitive motion issues (such as carpal tunnel syndrome) are considered in this design. The button was placed on the handle in order to allow for thumb actuation of the device. This will reduce the stresses that could result in carpal tunnel syndrome. The handle will also be contoured in order to increase grip and transfer the holding position of the tool to an ergonomically comfortable position. Optimal design of the handle involves a multi-angled handle, but this cannot be used because of machining issues and mechanics of the chosen design. The industrial design of the handle will result in efficiency of use and reduction of stresses in the hand and wrist of the user.

Other Considerations:

Regulatory

Consultation by personnel at the FDA and document searches reveals that this design does not need to follow any regulations regarding pre-market approval. The only regulation that needs to be followed is the good manufacturing procedure outlined by the FDA[27][33].

Economics

Economical analysis revealed that the cost of the prototype is inexpensive since it consists of mostly off-the-shelf components. Cost of raw stock is approximated to be about $370, while the actual stock used for the tool is around $100 dollars [30][31]. In a manufacturing environment, labor would be added to the cost of production, but raw stock would be purchased in bulk and therefore be less expensive.

Outlook

From this point forward, the initial prototype will be fabricated, revised and tested. Many of the components of the tool are small off-the-shelf components. Custom components include the handle, button mechanism and other small components. These will be made and the prototype will be assembled. Ergonomics of the handle will be further examined and any necessary features will be designed to incorporate additional ergonomic features. Issues with the assembly will be noted and adjustments will be made as necessary. The following testing will be performed to validate the design and determine flaws in the device:

Design Testing and Evaluation:

In depth testing of the prototype will be performed in order to validate the quality, consistency and accuracy of the tool. The following list demonstrates the types of design evaluation that will be performed:

- Prototyping will be performed at both the component and assembly levels.
- The spring distance for the target force will be tested and compared to the calculations.
- Accuracy of the spring to create a consistent force will be tested by a repetitive test to analyze this feature.
• The trigger assembly will be prototyped to show that the button requires a minimal force to unlock the compressed spring and that the button mechanism is sufficient to hold the spring in place.

• Pull back force of the plunger will be examined to ensure that the required force to set the plunger is reasonable. This will allow the clinician to set the plunger with ease.

• The coefficient of friction between the abutment and the implant will be determined experimentally in order to reduce the error in the theoretical models. Some samples will be placed in cement to simulate osseointegration as before. Once dry, the abutments will be seated with the prototyped tool. The same number of samples will also be installed into the cement cubes with the current mallet and handle method. Both test groups will be tensile tested to determine the average retention force and the standard deviation of each sample group.

• The device will be autoclaved and re-tested to ensure that no detrimental thermal or corrosive effects are seen in the device. The unit will be disassembled and the components examined for physical changes.

After testing, redesign cycles will be performed until the design is considered adequate. Once this is done, the final prototype will be ready for presentation. Statistical analysis will be present to display the accuracy, repeatability and reliability of the device.

**Concluding Works:**

After the final design has been tested and validated, the final touches will be applied to the design process. Cost analysis for the design will be conducted for the final product. A product documentation package, which includes a final bill of materials, will be generated. A procedural document package will also be generated. This package will include a user manual, instruction sheet, instructional video, or any combination of these three resources.
Summary

Taper lock abutments present a significant advantage over screw-retained abutments. The taper lock mechanism is more effective for prosthetic retention, provided the installation force is correctly applied. The current mallet and handle installation procedure creates variable forces, resulting in diminished success rates. This presents a need to improve the installation process through development of an abutment-hammering tool. Specifications and requirements can be determined through evaluation of the current installation procedure, taper lock mechanics and bone configurations. An analysis of the state of the art reveals that no product has been developed to fulfill these specifications and requirements. Further research shows that the mechanics of striking devices can be applied to this abutment-hammering tool; however, this will require innovative design work. Theoretical analysis was performed to determine the stresses in the implant and abutment. Experiments were performed on samples that proved the validity of the models. With the physical analysis of the taper lock system established, the tool was designed and the major components analyzed. Ergonomics of the handle design, regulatory requirements and economics were analyzed for the chosen design. Engineering drawings were drafted as a final step for reviewing, and the prototyping and testing phase of the project can begin on schedule. The current process plan indicates that the prototype can be tested, further developed and validated by the end of the spring quarter 2001.
APPENDIX A: Experimental Data

The following is the data collected from the insertion distance experiment:

<table>
<thead>
<tr>
<th>Drop L</th>
<th>Original H</th>
<th>Corrected After Impact</th>
<th>Insertion Distance</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.0</td>
<td>1.0</td>
<td>0.486</td>
<td>1.01</td>
</tr>
<tr>
<td>2.0</td>
<td>1.0</td>
<td>0.986</td>
<td>1.02</td>
</tr>
<tr>
<td>2.5</td>
<td>1.0</td>
<td>1.486</td>
<td>1.02</td>
</tr>
<tr>
<td>3.0</td>
<td>1.0</td>
<td>1.986</td>
<td>1.01</td>
</tr>
<tr>
<td>3.5</td>
<td>1.0</td>
<td>2.486</td>
<td>1.01</td>
</tr>
<tr>
<td>4.0</td>
<td>1.0</td>
<td>2.986</td>
<td>1.00</td>
</tr>
<tr>
<td>4.5</td>
<td>1.0</td>
<td>3.486</td>
<td>1.00</td>
</tr>
<tr>
<td>5.0</td>
<td>1.0</td>
<td>3.986</td>
<td>1.00</td>
</tr>
<tr>
<td>5.5</td>
<td>1.0</td>
<td>4.486</td>
<td>1.00</td>
</tr>
<tr>
<td>6.0</td>
<td>1.0</td>
<td>4.986</td>
<td>1.00</td>
</tr>
<tr>
<td>6.5</td>
<td>1.0</td>
<td>5.486</td>
<td>1.00</td>
</tr>
<tr>
<td>7.0</td>
<td>1.0</td>
<td>5.986</td>
<td>1.00</td>
</tr>
<tr>
<td>7.5</td>
<td>1.0</td>
<td>6.486</td>
<td>1.00</td>
</tr>
<tr>
<td>8.0</td>
<td>1.0</td>
<td>6.986</td>
<td>1.00</td>
</tr>
<tr>
<td>8.5</td>
<td>1.0</td>
<td>7.486</td>
<td>1.00</td>
</tr>
<tr>
<td>9.0</td>
<td>1.0</td>
<td>7.986</td>
<td>1.00</td>
</tr>
<tr>
<td>9.5</td>
<td>1.0</td>
<td>8.486</td>
<td>1.00</td>
</tr>
<tr>
<td>10.0</td>
<td>1.0</td>
<td>8.986</td>
<td>1.00</td>
</tr>
<tr>
<td>10.5</td>
<td>1.0</td>
<td>9.486</td>
<td>1.00</td>
</tr>
<tr>
<td>11.0</td>
<td>1.0</td>
<td>9.986</td>
<td>1.00</td>
</tr>
<tr>
<td>11.5</td>
<td>1.0</td>
<td>10.486</td>
<td>1.00</td>
</tr>
<tr>
<td>12.0</td>
<td>1.0</td>
<td>10.986</td>
<td>1.00</td>
</tr>
<tr>
<td>12.5</td>
<td>1.0</td>
<td>11.486</td>
<td>1.00</td>
</tr>
<tr>
<td>13.0</td>
<td>1.0</td>
<td>11.986</td>
<td>1.00</td>
</tr>
<tr>
<td>13.5</td>
<td>1.0</td>
<td>12.486</td>
<td>1.00</td>
</tr>
<tr>
<td>14.0</td>
<td>1.0</td>
<td>12.986</td>
<td>1.00</td>
</tr>
<tr>
<td>14.5</td>
<td>1.0</td>
<td>13.486</td>
<td>1.00</td>
</tr>
<tr>
<td>15.0</td>
<td>1.0</td>
<td>13.986</td>
<td>1.00</td>
</tr>
<tr>
<td>15.5</td>
<td>1.0</td>
<td>14.486</td>
<td>1.00</td>
</tr>
<tr>
<td>16.0</td>
<td>1.0</td>
<td>14.986</td>
<td>1.00</td>
</tr>
<tr>
<td>16.5</td>
<td>1.0</td>
<td>15.486</td>
<td>1.00</td>
</tr>
<tr>
<td>17.0</td>
<td>1.0</td>
<td>15.986</td>
<td>1.00</td>
</tr>
<tr>
<td>17.5</td>
<td>1.0</td>
<td>16.486</td>
<td>1.00</td>
</tr>
<tr>
<td>18.0</td>
<td>1.0</td>
<td>16.986</td>
<td>1.00</td>
</tr>
<tr>
<td>18.5</td>
<td>1.0</td>
<td>17.486</td>
<td>1.00</td>
</tr>
<tr>
<td>19.0</td>
<td>1.0</td>
<td>17.986</td>
<td>1.00</td>
</tr>
<tr>
<td>19.5</td>
<td>1.0</td>
<td>18.486</td>
<td>1.00</td>
</tr>
<tr>
<td>20.0</td>
<td>1.0</td>
<td>18.986</td>
<td>1.00</td>
</tr>
<tr>
<td>20.5</td>
<td>1.0</td>
<td>19.486</td>
<td>1.00</td>
</tr>
<tr>
<td>21.0</td>
<td>1.0</td>
<td>19.986</td>
<td>1.00</td>
</tr>
<tr>
<td>21.5</td>
<td>1.0</td>
<td>20.486</td>
<td>1.00</td>
</tr>
<tr>
<td>22.0</td>
<td>1.0</td>
<td>20.986</td>
<td>1.00</td>
</tr>
</tbody>
</table>
APPENDIX B: Discussion of the Analytical Model

An analytical model was developed to explain the stresses, insertion lengths and the retention forces within the implant and abutment. Figure AP1 displays the implant and abutment in segmented form. The basic concept is to separate the abutment and implant into small segments (Ai’s and li’s from Figure 1X). These segments can then be treated as independent disks. The abutment can be positioned so that segment A1 is stressing implant segment I7. At the same time A2 is stressing I6, A3 is stressing I5, etc. On the next step A1 is stressing I8, A2 is stressing I7, etc. Each of these disks can then be treated as independent pressure vessels in a press fit configuration. Discussion of this procedure will be presented later in this appendix.

![Figure AP1: Abutment and Implant Segments and Steps](image)

**Assumptions**

Several assumptions are made in the process of examining this model. The first assumption is that each segment can be treated as an independent piece. An example of this can be seen in step 1 of Figure AP1. Abutment segment A1 is stressing implant segment I7. This model predicts that I8 does not experience any stresses from the forces applied to I7. This is clearly not the case in the real situation. I8 would undergo a bending stress in an attempt to support I7. This is the worst assumption that is made in this analysis. The assumption will clearly cause an over estimation of the stresses. While this over estimation of stresses will be noticeable, it is not expected to create a major concern for this model’s validity. Furthermore this will create an over estimation of the stresses, which will serve to increase the factor of safety. If the stresses were under estimated, there would be significant concerns that the stresses may exceed the elastic limit as a result of this error.

The next assumption is that the bone does not affect the implant. For this we assume that the bone is soft enough to expand freely rather than creating a support for on the outside surface of the implant. This assumption is adequate, since the young’s modulus of bone is nearly sixteen times less than the young’s modulus of 6Al4V titanium. This assumption is similar to a steel bar rapped in foam. The foam is so soft that it cannot create a significant resistance to
a force. The steel bar absorbs nearly all of the force. The bone will be analyzed later to determine if plasticity in the bone is a potential failure mode.

The final assumption is that representing the tapered post of the abutment and the tapered hole of the implant is a viable approximation. So long as the number of segments is large, the error from this approximation will be extremely small.

Model Nomenclature:

\( L_{\text{seg}} \): Length of the segments, a property of the model definition

\( a \): original inside radius of the implant segment

\( d_i \): original inside diameter of the implant segment

\( b \): original outer diameter of the implant segment

\( d_o \): original outer diameter of the implant segment

\( u_s \): decrease in abutment segment radius

\( u_b \): increase in the implant segment inner radius

\( \Delta \): diametric interference

Equations for Individual Segments

\[
\Delta = 2 \left[ u_s \right] + 2 \left[ u_h \right]
\]

\[
\varepsilon_{\text{t, abutment}} = \frac{1}{E} \left( \sigma_t \cdot \mu \cdot \sigma_r \right)
\]

\[
\varepsilon_{\text{r, abutment}} = \frac{2 \cdot u_s}{D_{\text{original}}}
\]

\[
\sigma_{r, \text{max}} = -\sigma - \frac{E \cdot A}{4 \cdot a}
\]

\[
\sigma_{r, \text{max}} = \frac{E \cdot A}{4 \cdot a} (1 + (a b)^2)
\]

\[
F_{\text{retention}} = \sigma_r \cdot \pi \cdot d_i \cdot L_{\text{segment}} \cdot f_{\text{static}}
\]

\[
U_{\text{abutment}} = \frac{1}{2} (\varepsilon_{\text{t, abutment}}^2 + \varepsilon_{\text{r, abutment}}^2) \cdot E \cdot V_{\text{abutment}}
\]

\[
V_{\text{abutment}} = \pi \cdot \left[ \frac{D_{\text{original}}^2}{2} \right] \cdot L_{\text{seg}}
\]

\[
U_{\text{implant}} = -\frac{2 \cdot \pi \cdot x \cdot h \cdot a^4 \cdot p^2}{E \cdot (b^2 - a^2)^2} \left[ (a + u_h)^2 - \frac{b^4}{(a + u_h)^2} \right]
\]

\[
\sigma_{r, \text{max ave}} = \frac{\sigma_{r, \text{max t}} + \sigma_{r, \text{max t+1}}}{2}
\]

\[
W_{\text{sliding}} = \sigma_{r, \text{max ave}} \cdot \pi \cdot d_i \cdot L_{\text{segment}} \cdot f_{\text{static}} \cdot L_{\text{segment}}
\]
The above equations are all used to compute values for the individual segments [33][34][35]. Equation 1 shows the average radial stress or radial pressure. This is then used in equation 2 to compute the work required to overcome sliding friction. This work is defined as the product of pressure, the area it acts on, the friction factor and the distance it acts through. The average pressure takes into account the radial stress that is already present form the previous step and averages it in with the new maximum pressure. The friction factor and the full surface area of the segment then multiply this. The final step is to multiply by the length that this force acts (L segment).

Combinations of the forces, work and energy can now be examined. The equations listed below are used to compute the total energy needed to complete the step. It is important to note two things. The change in diameter of the implant segment is always defined as the original diameter minus the final diameter. For this reason, the energy needed to complete the second step automatically incorporates the energy needed to complete step 1. The work to overcome sliding friction is non-conservative. For this reason it is important to add up all of these works from the previous steps in order to predict the energy.

\[ U_{\text{total}} = \sum_{j=1}^{n} U_{\text{abutment}} + \sum_{i=1}^{n} U_{\text{implant}} \]

\[ W_{\text{sliding friction}} = W_{\text{sliding friction total}} + \sum_{i=1}^{n} W_{\text{sliding friction total i-1}} \]

\[ F_{\text{retention total}} = \sum_{i=1}^{n} F_{\text{retention total i}} \]

\[ E_{in} = U_{\text{total}} + W_{\text{sliding friction total}} \]

The model uses a function to determine the outer diameter of the implant segment since the outside of the implant is not a pure cylinder. Due to a privacy agreement between Northeastern University and Bicon, the actual angles and lengths for the implant and abutment cannot be displayed. The properties of 6Al4V titanium and the results of this model for a segment length of 0.001105 inches and a total of 144 segments are shown below. An example step calculation is also displayed below.

### Table A 6: Additional properties of Ti6Al4V

<table>
<thead>
<tr>
<th>Material Properties of 6Al4V</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>E</td>
<td>16505000</td>
<td>psi</td>
</tr>
<tr>
<td>Poissons Ratio</td>
<td>0.342</td>
<td>unit less</td>
</tr>
<tr>
<td>Friction Factor (Static)</td>
<td>0.12</td>
<td>unit less</td>
</tr>
<tr>
<td>Titanium Yield Stress (Tension)</td>
<td>127633</td>
<td>psi</td>
</tr>
<tr>
<td>Titanium Yield Stress ( Compression)</td>
<td>3</td>
<td>psi</td>
</tr>
<tr>
<td>Max Shear Stress</td>
<td>79750</td>
<td>psi</td>
</tr>
<tr>
<td>Friction Factor (Kinetic)</td>
<td>0.12</td>
<td>unit less</td>
</tr>
<tr>
<td>Titanium Working Normal Stress</td>
<td>85089</td>
<td>unit less</td>
</tr>
</tbody>
</table>
Table A 7: Results Summary

<table>
<thead>
<tr>
<th>Deflection</th>
<th>Input Energy</th>
<th>Retention</th>
<th>Max Normal Stress</th>
<th>Max Shear Stress</th>
<th>Max Radial Stress</th>
</tr>
</thead>
<tbody>
<tr>
<td>inches</td>
<td>oz*in</td>
<td>lbs.</td>
<td>psi</td>
<td>psi</td>
<td>psi</td>
</tr>
<tr>
<td>0.001106</td>
<td>0.11</td>
<td>9.31</td>
<td>7092</td>
<td>4389</td>
<td>3986</td>
</tr>
<tr>
<td>0.002211</td>
<td>0.510</td>
<td>22.47</td>
<td>16863</td>
<td>10437</td>
<td>9508</td>
</tr>
<tr>
<td>0.003317</td>
<td>1.250</td>
<td>36.01</td>
<td>26634</td>
<td>16484</td>
<td>15066</td>
</tr>
<tr>
<td>0.004422</td>
<td>2.338</td>
<td>49.94</td>
<td>36405</td>
<td>22531</td>
<td>20657</td>
</tr>
<tr>
<td>0.005528</td>
<td>3.783</td>
<td>64.26</td>
<td>46176</td>
<td>28578</td>
<td>26282</td>
</tr>
<tr>
<td>0.006633</td>
<td>5.597</td>
<td>78.98</td>
<td>55947</td>
<td>34625</td>
<td>31939</td>
</tr>
<tr>
<td>0.007739</td>
<td>7.788</td>
<td>94.08</td>
<td>65717</td>
<td>40672</td>
<td>37629</td>
</tr>
<tr>
<td>0.008844</td>
<td>10.366</td>
<td>109.59</td>
<td>75488</td>
<td>46719</td>
<td>43350</td>
</tr>
<tr>
<td>0.009950</td>
<td>13.342</td>
<td>125.49</td>
<td>85259</td>
<td>52766</td>
<td>49102</td>
</tr>
<tr>
<td>0.011055</td>
<td>16.725</td>
<td>141.79</td>
<td>95030</td>
<td>58813</td>
<td>54884</td>
</tr>
<tr>
<td>0.012161</td>
<td>20.526</td>
<td>158.50</td>
<td>104801</td>
<td>64860</td>
<td>60696</td>
</tr>
</tbody>
</table>
Table A & Example of One Iteration of Taper Lock Mathematical Analysis

<table>
<thead>
<tr>
<th>Iteration</th>
<th>Angle 1</th>
<th>Angle 2</th>
<th>Angle 3</th>
<th>Angle 4</th>
<th>Angle 5</th>
<th>Angle 6</th>
<th>Angle 7</th>
<th>Angle 8</th>
<th>Angle 9</th>
<th>Angle 10</th>
<th>Angle 11</th>
<th>Angle 12</th>
<th>Angle 13</th>
<th>Angle 14</th>
<th>Angle 15</th>
</tr>
</thead>
<tbody>
<tr>
<td>No.</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>3</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>4</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>5</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>6</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>7</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>8</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Notes:
- Table A contains data for one iteration of Taper Lock Mathematical Analysis.
- Each row represents a different iteration, with columns for various angles.

- Calculations and results are detailed in the body of the table.
Other Concerns:
While the major stress concern is on the abutment and implant, the jawbone should also be analyzed to ensure safety. As previously stated, the jawbone is assumed to provide no resistance to the titanium implant. With this understood, the top segment of the implant is allowed to expand freely. The following equation can be used to find the change in radius of the outside of the implant ($\Delta b$):

$$\Delta b = (\sigma_t) \cdot \left( \frac{b}{E} \right) \cdot \left( \frac{2 \pi a^2}{b^2 - a^2} \right)$$

Where:

- $b$: original outer implant radius
- $a$: original inner implant radius

While Bicon will not allow the disclosure of the values for $a$ and $b$, if the yield stress of 6Al4V titanium is inserted for $\sigma_t$, then the maximum possible $\Delta b$ is computed to be 0.0012546 inches.

The jawbone can now be treated as a thick walled pressure vessel with an outside radius ($b'$) equal to 0.25 inches. The change in internal radius ($\Delta a'$) will be the same as the change in the outer radius of the abutment ($\Delta b$). These dimensions combined with the following table of material properties for bone will allow for evaluation of the stresses in the bone.

### Table A 9: Bone Properties

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
<th>Units</th>
<th>Bone Type Tested to Obtain this Property</th>
<th>Species the Bone Came From</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\mu$ (radial)</td>
<td>0.19</td>
<td>unit less</td>
<td>Skull Bone</td>
<td>Human</td>
</tr>
<tr>
<td>$E_{rad}$ and tangential</td>
<td>1,100,000</td>
<td>psi</td>
<td>Cortical Bone</td>
<td>Human</td>
</tr>
<tr>
<td>$\sigma_t$ max compression</td>
<td>16,000</td>
<td>psi</td>
<td>Cortical Bone</td>
<td>Human</td>
</tr>
<tr>
<td>$\sigma_t$ max in tension</td>
<td>13,068</td>
<td>psi</td>
<td>Tibia and Cortical</td>
<td>Bovine</td>
</tr>
</tbody>
</table>

$$\Delta a' = p a' \frac{1}{E} \left( \frac{a'^2 + b'^2}{a'^2 - b'^2} - \nu \right) = 0.001254588 \text{in}$$

$$\sigma_r = p = \frac{\Delta a' E}{a'} \left[ \frac{1}{b'^2 + a'^2 / \left( b'^2 - a'^2 \right) + \nu} \right] = 11668 \text{psi}$$

$$\sigma_t = p \left( \frac{b'^2 + a'^2}{b'^2 - a'^2} \right) = 12637 \text{psi}$$
Since the actual radial and tangential stresses are lower than the maximum radial and tangential stresses, the bone is not a failure point. Plastic deformation will occur in the abutment and implant well before plastic deformation occurs in the jawbone. [22]

**Impact Stress**

The impact energy delivered to the jawbone and abutment must be less the maximum impact that 6Al4V titanium, the jawbone and 316 stainless steel. The following table shows the maximum impact energy that each of the three materials can withstand. It is clear that all three of these materials are able to absorb 8 oz-in of energy without shattering.

<table>
<thead>
<tr>
<th>Material</th>
<th>Charpy Impact Strength</th>
<th>Units</th>
<th>Charpy Impact Strength</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>6AL4V Titanium</td>
<td>13</td>
<td>ft*lbs</td>
<td>2496</td>
<td>oz*in</td>
</tr>
<tr>
<td>316 Stainless Steel</td>
<td>77</td>
<td>ft*lbs</td>
<td>14784</td>
<td>oz*in</td>
</tr>
<tr>
<td>Human Bone</td>
<td>0.2</td>
<td>ft*lbs</td>
<td>38.4</td>
<td>oz*in</td>
</tr>
</tbody>
</table>

*Table A 10: Material Impact Properties*
APPENDIX C: Derivation of Strain Energy for Implants

\[ \sigma = E \varepsilon \]
\[ U = \int \int \int (\sigma_r \varepsilon_r + \sigma_\theta \varepsilon_\theta) dV \]
\[ \Rightarrow U = \int \int \int (\varepsilon_r^2 + \varepsilon_\theta^2) dV \quad \text{or} \quad \frac{1}{E} \int \int \int (\sigma_r^2 + \sigma_\theta^2) dV \]
\[ \sigma_r = \frac{a^2 p}{b^2 - a^2} \left( 1 - \frac{b^2}{r^2} \right) \quad (\sigma_r \text{ is a function of } r) \]
\[ \sigma_\theta = \frac{a^2 p}{b^2 - a^2} \left( 1 + \frac{b^2}{r^2} \right) \quad (\sigma_\theta \text{ is a function of } r) \]
\[ \Rightarrow U = \frac{1}{E} \int \int \int \left( \frac{a^2 p}{b^2 - a^2} \right)^2 \left[ \left( 1 - \frac{b^2}{r^2} \right)^2 + \left( 1 + \frac{b^2}{r^2} \right)^2 \right] dV \]
\[ \therefore U = \frac{1}{E} \left( \frac{a^2 p}{b^2 - a^2} \right)^2 \int \int \int \left( 1 + \frac{b^4}{r^4} \right) dV \]

\( h \): constant

\[ U = \frac{1}{E} \left( \frac{a^2 p}{b^2 - a^2} \right)^2 \int \int \int \left( r + \frac{b^4}{r^3} \right) dr \]

\[ U_{r, \text{outer}}^{r, \text{outer}} = \left( \frac{4 \pi h}{E} \right) \left( \frac{a^4 p^2}{(b^2 - a^2)^2} \right) \left( \frac{r^2 + b^4 r^{-2}}{2} - \frac{b^4}{2} \right) \]

at \( r_{\text{outer}}, r = \frac{b^2}{2} \)

at \( r_{\text{inner}}, r = a + \mu \star h \)

\[ \left[ \left( \frac{2 \star \mu \star h}{E} \right) \left( \frac{a^4 p^2}{(b^2 - a^2)^2} \right) \left( \frac{b^2}{2} - \frac{b^4}{2 b^2} \right) \right] = 0, \text{ for } r_{\text{outer}} \]

\[ U = 0 - \left[ \left( \frac{2 \star \mu \star h}{E} \right) \left( \frac{a^4 p^2}{(b^2 - a^2)^2} \right) \right] \left[ \left( a + \mu \star h \right)^2 - \frac{b^4}{(a + \mu \star h)^2} \right] \]
APPENDIX D: Engineering Drawings

The following engineering drawings are representative of the final drawings for the components of the tool and the final assembly:
THE INFORMATION CONTAINED IN THIS DRAWING IS THE SOLE PROPERTY OF NORTHEASTERN UNIVERSITY. ANY REPRODUCTION IN PART OR WHOLE WITHOUT THE WRITTEN PERMISSION OF NORTHEASTERN UNIVERSITY IS PROHIBITED.

SECTION A-A
SCALE 1: 1.5

<table>
<thead>
<tr>
<th>PMBA-100</th>
<th>PMBA-100</th>
</tr>
</thead>
<tbody>
<tr>
<td>FINISH: --</td>
<td>FINISH: --</td>
</tr>
<tr>
<td>APPLICATION: DO NOT SCALE DRAWING</td>
<td>APPLICATION: DO NOT SCALE DRAWING</td>
</tr>
</tbody>
</table>

NORTHEASTERN UNIVERSITY
ABUTMENT HAMMERING TOOL
FOR DENTAL IMPLANTS

ABUTMENT HAMMER ASSEMBLY

PMBA-101

DWG. NO. A

FULL SCALE
CAD: F.E., WASHINGTON, D.C.

REV. 1 OF 1

TOLERANCES ARE:
- DIMENSIONS ARE IN INCHES
- TOLERANCES ARE
- FRACTIONS OCTAL ANGLES
  .XXX+ .030 
  .XXX .002

MATERIAL
PMBA-100

DRAWN
IFG 03-MAR-01

CHECKED
JL 03-MAR-01

APPROVED
ABUTMENT HAMMER ASSEMBLY

FOR DENTAL IMPLANTS

ABUTMENT HAMMER ASSEMBLY

PMBA-101

DWG. NO. A

FULL SCALE
CAD: F.E., WASHINGTON, D.C.

REV. 1 OF 1

TOLERANCES ARE:
- DIMENSIONS ARE IN INCHES
- TOLERANCES ARE
- FRACTIONS OCTAL ANGLES
  .XXX+ .030 
  .XXX .002

MATERIAL
PMBA-100

DRAWN
IFG 03-MAR-01

CHECKED
JL 03-MAR-01

APPROVED
ABUTMENT HAMMER ASSEMBLY

FOR DENTAL IMPLANTS

ABUTMENT HAMMER ASSEMBLY

PMBA-101

DWG. NO. A

FULL SCALE
CAD: F.E., WASHINGTON, D.C.
NO. 33 DRILL THRU
CB. .175ψ .15 FROM OPOSITE SIDE
2 PLACES AS SHOWN

DETAIL A
SCALE 2:1

R.22 THRU

φ .100ψ .100

6-32 TAP ψ .30
2 PLACES EQ. PL. ON A .880 BC.

PBMA-100

DIMENSIONS ARE IN INCHES
TOLERANCES ARE:

FRACTIONS DECIMALS ANGLES
+ .XXX+ .01 +0.5°
XXX+ .002

UNLESS OTHERWISE SPECIFIED

CAD-GENERATED DRAWING,
DO NOT MANUALLY UPDATE

NORTHEASTERN UNIVERSITY
ABUTMENT HAMMERING TOOL
FOR DENTAL IMPLANTS

HOUSING, LEFT

316 STAINLESS

DO NOT SCALE DRAWING

PBM-101L

REV. 1

SHEET 1 OF 1

THE INFORMATION CONTAINED IN THIS DRAWING IS THE SOLE PROPERTY OF NORTHEASTERN UNIVERSITY. ANY REPRODUCTION IN PART OR WHOLE WITHOUT THE WRITTEN PERMISSION OF NORTHEASTERN UNIVERSITY IS PROHIBITED.
The information contained in this drawing is the sole property of Northeastern University. Any reproduction in part or whole without the written permission of Northeastern University is prohibited.

D C B A

(1/).107 \(\pm\) .200
NO. 43 DRILL \(\pm\) .20
4-40 TAP \(\pm\) .15

\(\Phi\) .089 \(\pm\) .200
NO. 43 DRILL \(\pm\) .20
4-40 TAP \(\pm\) .15
2 PLACES AS SHOWN

---

MIRROR IMAGE OF DOC PBM101L FOR ALL NON-DISPLAYED DIM.
**SHAFT**

**PBM-105**

**DIMENSIONS**
- No. 36 drill thru 6-32 tap thru
- Ø0.250
- 10.90
- 5.500

**TOLERANCES**
Fractions decimals: angles + .002, - .002

**MATERIAL**
316 Stainless

**NORTHEASTERN UNIVERSITY ABUTMENT HAMMERING TOOL FOR DENTAL IMPLANTS**

<table>
<thead>
<tr>
<th>PART NO.</th>
<th>PART OR IDENTIFYING NO.</th>
<th>NOMENCLATURE</th>
<th>MATERIAL</th>
<th>QTY</th>
</tr>
</thead>
<tbody>
<tr>
<td>PBMA-100</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**DRAWING**
- TFG 03-MAR-01

**CHECKED**
- JJJ 03-MAR-01

**APPROVED**
- REPRINTED

**DRAWING DESCRIPTION**
- Shaft

**SCALE**
- 1:12

**DATE**
- 03-MAR-01

**NOTE**
- Do not scale drawing
NOTES:

1. MSC PART NUMBER: MSC06871016

NO. 7 DRILL $\phi 0.40$

1/4-20 TAP $\phi 0.25$

$\phi 1.375 \pm 0.01$ SPHERE
The information contained in this drawing is the sole property of Northeastern University. Any reproduction in part or whole without the written permission of Northeastern University is prohibited.

The drawing is of a linear bearing for dental implants. The dimensions and tolerances are specified in inches. The parts list includes:

<table>
<thead>
<tr>
<th>PART NO.</th>
<th>PART OR IDENTIFYING NO.</th>
<th>NUMBER OR DESCRIPTION</th>
<th>MATERIAL</th>
<th>INV. ROD</th>
</tr>
</thead>
<tbody>
<tr>
<td>PBMA-109</td>
<td>SST</td>
<td>1.25</td>
<td>STAINLESS STEEL</td>
<td>03-MAR-01</td>
</tr>
</tbody>
</table>

The linear bearing is designed to fit within an 0.880 BC. hole. The tolerated dimensions are specified for various parts of the bearing.
NO. 36 DRILL THRU
6-32 TAP THRU
2 PL. AS SHOWN

.335

.480

.250 THRU

.200

.125

22° ±1°
THE INFORMATION CONTAINED IN THIS DRAWING IS THE SOLE PROPERTY OF NORTHEASTERN UNIVERSITY. ANY REPRODUCTION IN PART OR WHOLE WITHOUT THE WRITTEN PERMISSION OF NORTHEASTERN UNIVERSITY IS PROHIBITED.

NOMENCLATURE
I = IDENTIFYING NO.

PARTS LIST

<table>
<thead>
<tr>
<th>ITEM NO.</th>
<th>DESCRIPTION</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>PBMA-125</td>
</tr>
<tr>
<td>2</td>
<td>HAMMER 1</td>
</tr>
<tr>
<td>3</td>
<td>C</td>
</tr>
<tr>
<td>4</td>
<td>A</td>
</tr>
<tr>
<td>5</td>
<td>B</td>
</tr>
<tr>
<td>6</td>
<td>D</td>
</tr>
</tbody>
</table>

MATERIAL
316 STAINLESS

FINISH
--

APPLICATION
DO NOT SCALE DRAWINGS

SCALE 2:1

DIMENSIONS ARE IN INCHES
TOLERANCES ARE:
FRACTIONS DECIMALS ANGLES
XXX .01 ±0.5°
XXX .002°

NOT SCALE DRAWING

PMMA-100

AUTO ENG

SCALE 2:1

CAD GENERATED DRAWING, DO NOT MANUALLY UPDATE

DFG 03-MAR-01

CHECKED

JJJ 03-MAR-01

NORTHEASTERN UNIVERSITY
ABUTMENT HAMMERING TOOL
FOR DENTAL IMPLANTS

HAMMER

DRAWN

TFG 03-MAR-01

APPROVALS

DATE

DRAW

CHECK

REVISIONS

DESCRIPTION

APPROVED

REV

DRAW

NO. DWG. NO.

PBM-125

1

APPLICAT

1

4-40 TAP ↓ .20

NO. 43 DRILL ↓ .30

R.01 TYP

.10

.60

φ .18

φ .480
NO. 47 DRILL $0.20  
3-48 TAP $0.15 

Φ $0.600 

R $0.10 

.100 

.200 

PBMA-100 

316 STAINLESS
NORTHEASTERN UNIVERSITY
ABUTMENT HAMMERING TOOL
FOR DENTAL IMPLANTS

TRIGGER PIN

MFGTHG

PBM-127

REV. 1

PART NO.

PART OR
IDENTIFYING NO.

MATERIAL

SPE

UNLESS OTHERWISE SPECIFIED
DIMENSIONS ARE IN INCHES
TOLERANCES ARE:

FRACTIONS DECIMALS ANGLES

+ .XX + .01 + 0.5°

.

.040

45° X .005
2 PL. AS SHOWN

.40

316 STAINLESS

FULL SCALE CAD FILE: "WASHER.SLDPMI"
References

    http://www.bicon.com/documentation/PDF_Final_00/Bicon_Procedure.pdf


