Development of a Strain Energy Storage Mechanism Using Tension Elements to Enhance Golf Club Performance

Marc A. Whitezell
Brigham Young University - Provo

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DEVELOPMENT OF A STRAIN ENERGY STORAGE MECHANISM
USING TENSION ELEMENTS TO ENHANCE GOLF
CLUB PERFORMANCE

by
Marc A. Whitezell

A thesis submitted to the faculty of
Brigham Young University
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GRADUATE COMMITTEE APPROVAL

of a thesis submitted by

Marc A. Whitezell

This thesis has been read by each member of the following graduate committee and by majority vote has been found to be satisfactory.

_________________________________  ____________________________________
Date                             Spencer P. Magleby, Chair
_________________________________  ____________________________________
Date                             Larry L. Howell
_________________________________  ____________________________________
Date                             Jonathan D. Blotter
As chair of the candidate’s graduate committee, I have read the thesis of Marc A. Whitezell in its final form and have found that (1) its format, citations, and bibliographical style are consistent and acceptable and fulfill university and department style requirements; (2) its illustrative materials including figures, tables, and charts are in place; and (3) the final manuscript is satisfactory to the graduate committee and is ready for submission to the university library.

Date

Spencer P. Magleby
Chair, Graduate Committee

Accepted for the Department

Matthew R. Jones
Graduate Coordinator

Accepted for the College

Alan R. Parkinson
Dean, Ira A. Fulton College of Engineering and Technology
The development of current golf club designs has followed an evolutionary process starting with the original wooden heads of a hundred years ago, to the thin-walled, hollow body titanium heads of today. Current designs utilize what has become known as the trampoline effect to increase the efficiency of the ball-club impact, which has a number of limiting factors that restrict clubhead performance. These limitations provided the motivation for this research to explore new mechanisms by which the efficiency of the ball club impact could be increased. In particular this research focuses on the development of compliant mechanisms to increase club performance.

The results of this research, from concept development to initial prototype plans, are included in this study. A discussion of past and current research in the area of golf club
design is presented. A new list of performance metrics for golf clubs and a number of new golf club concepts is also presented. This is followed by a static and dynamic analysis of the most promising golf club configuration. The study is concluded with a concept validation analysis and a presentation of possible prototype configurations for a new golf club design.
I would like to acknowledge a number of different people for their support in helping me complete this research and thesis.

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Chapter 1: Introduction

1.1 Motivation

Many devices are designed to operate under conditions of high impact loading. In some
cases, such as armor plating, these devices are designed to absorb and dissipate the
energy of the impact. In other situations, these devices are designed to return as much
energy as possible to the impacting object. It is the latter case that provides the
motivation for this research.

During a collision, kinetic energy is transferred between two or more impacting objects.
If the kinetic energy of the system (which includes the impacting objects) is conserved,
the impact is said to be elastic. In all other cases, when the kinetic energy is not
conserved, the impact is defined as inelastic. Since actual impacts include energy losses
due to friction, heat, etc., they are inelastic. In an effort to quantify the efficiency of an
impact, a metric called the Coefficient of Restitution (COR) is used. The COR is simply
the ratio of the post impact velocity to the pre impact velocity of the impacting objects.
Thus, the two extreme values for the COR of an impact would be one if the impact were
completely elastic, and zero if the impact were completely inelastic.
It is common in many situations, especially in sports, to create impacts that exhibit high levels of COR. These impacts usually involve two objects; a ball, and some sort of impact device such as a bat or club. In most cases the key characteristics of the ball are fixed to maintain the integrity of the sport. As a result, there has been much effort to modify the impact devices to increase the COR of the impact with the ball. Since the COR of an impact is essentially a measurement of the efficiency of the collision, it is desirable to limit the deformation of the impacted object, or ball.

Commonly, the objects that are impacted by such devices are inelastic. Since the COR of an impact is essentially a measurement of the efficiency of the collision, it is desirable to limit the deformation of the impacted object. Current designs utilize what has become known as the “trampoline effect” to limit this deformation, which is created by making the impacting face of the device more compliant, allowing it to flex like a trampoline when impacting an object. This allows the more elastic face of the device to deform, storing more of the impact energy as strain energy in the deformation of the face, while limiting the inelastic deformation of the impacting object. This thesis will focus on golf clubs, but the processes used to generate high COR devices for golf clubs could be used for increasing the COR exhibited by any device.

Although current golf club designs have worked well, they have a number of limiting factors that restrict their performance. The first limiting factor has been the durability of these designs. In an attempt to increase the COR exhibited by golf clubs, engineers have sought after configurations that would increase the trampoline effect and maximize the
efficiency of the system. The methods used in the past to increase the trampoline effect have been to make the impacting face larger and thinner, giving the face more compliance. Current designs have become limited to the availability of materials that will allow the face to become larger and thinner, while still providing the necessary strength to withstand the impact forces experienced when the club impacts a ball.

The next limiting factor is the area of the face over which a high COR is exhibited. This limitation can be directly correlated to the trampoline effect that is used to propel the ball. Since the face behaves like a trampoline, only a hit at the very center of the face provides the maximum COR, while hitting an object off-center dramatically lowers the COR.

These inherent limitations connected with the current trampoline effect designs have motivated the exploration of new and innovative ways to increase the COR of an impact. While many design approaches could be considered, some of the most promising appear to be those that utilize mechanisms. A mechanism is defined as a component of a machine consisting of two or more bodies arranged so that the motion of one compels the motion of the other (Wilson, 2003). A mechanism behind the impacting face of the club could be utilized to provide compliance, relieving the dependency on the face itself to provide all the compliance. A particular group of mechanisms, called compliant mechanisms, would allow for the most promising designs. A compliant mechanism is a mechanism that gains all or at least some of its mobility from the deflection of flexible members rather than from movable joints only (Howell, 2001). Since compliant mechanisms do not contain any joints, they add no friction to the system, which will
work to maximize the efficiency of the impact. While current designs utilize compliance (deflection of the impact face) to increase the COR of an impact, it is likely that there exists more optimal compliant mechanism configurations for increasing the COR of an impact.

1.2 Objective

The development of current golf club designs has followed an evolutionary process starting with the original wooden heads of a hundred years ago, to the thin, hollow body titanium heads of today. Current designs all utilize what has become known as the trampoline effect to increase the efficiency of the ball-club impact, and there has been no success to date in generating a new mechanism by which golf club performance could be enhanced. It is the objective of this research to develop a completely new strain energy storage mechanism, outside of the face, to enhance golf club performance.

1.3 Contribution

The primary contribution of this thesis will be to provide a new mechanism by which golf club performance can be enhanced. The development of this new mechanism will allow researchers to overcome the limitations of current designs by relieving the dependence on the face to act as the sole strain energy storage mechanism, making it possible to increase both the durability of high COR designs and the size of the high COR zone. This will be highly beneficial to many different products, especially to the area of sports equipment,
where the performance of devices such as tennis rackets and golf clubs has become limited by the traditional configurations utilized to increase their COR.

### 1.4 Outline

The results of this research will be presented following the outline listed below.

- **Ch. 2: Literature Review**
  
  Previous work in the area of golf club design will be presented. In particular, a previously established design process for developing compliant mechanisms possessing high coefficients of restitution will be presented.

- **Ch. 3: Research Approach**
  
  The approach used to develop the new mechanism will be presented and outlined in this section.

- **Ch. 4: Metric Definition**
  
  Since the new configurations developed during the research were vastly different from current club designs, a new set of metrics was required to fully evaluate and compare the new designs.

- **Ch. 5: Concept Generation**
  
  The concepts generated as part of this research will be presented in this section.

- **Ch. 6: Concept Evaluation/Selection**
  
  Using the pre-established set of metrics, the field of concepts was narrowed down to those with the most promise, and the results of this process will be presented are presented in this chapter.

- **Ch. 7: Static Analysis**
The results obtained from the initial static tests used to validate the feasibility of the most promising concepts, as determined in Chapter 6, are presented in this chapter.

- **Ch. 8: Dynamic Analysis**
  Based on the results of the static analysis, a number of dynamic analyses were performed and the results of which are included in this section.

- **Ch. 9: Concept Validation**
  A final, more advanced dynamic model was used to validate the feasibility of the design and provide support for pursuing detailed development and experimental validation of the design.

- **Ch. 10: Prototype Plans**
  In order to show the feasibility of producing the design, initial prototype plans will be presented.

- **Ch. 10: Conclusion**
Chapter 2: Background and Literature Review

Research in the area of enhancing golf club head performance has exploded in recent years. Current research efforts are focused on improving two different areas of club head performance: increasing the launch velocity of the golf ball, and increasing the forgiveness of the club. The following chapter will include a discussion of these two areas, along with a discussion of the methods used to predict club performance.

2.1 Increasing Golf Ball Launch Velocity

Increasing golf ball launch velocity is of prime importance since it is directly correlated to increasing the distance the ball will travel. Golf clubs allow players to achieve much higher ball velocities than would otherwise be possible. This is accomplished by two separate means: increasing the radius of rotation, and creating an impact with the ball.

2.1.1 Increasing Radius of Rotation

By using a club, a player effectively increases the radius of rotation between the torso of their body and the ball. For a constant angular velocity (created by swinging the arms) the tangential velocity will increase as the radius of rotation is increased. This allows players to achieve much higher ball velocities than would otherwise be possible by simply throwing the ball. Since the length of a club becomes limited by the loss of
control experienced by the player as the club gets longer, there is a limit to how long the radius of rotation can be extended. Once the optimum club length has been determined for a player, the only other way to increase ball velocity is to increase the efficiency of the impact between the ball and club (Michal, 2001).

2.1.2 Creating an Efficient Impact

By increasing the efficiency of the impact between a golf ball and club, the launch velocity of the ball can be increased. Using the conservation of momentum, this phenomenon can be understood. In simple terms, the conservation of momentum states that the momentum of two objects that collide must be the same before and after impact (see equation (2.1) below for a golf ball/club momentum equation)

\[ MV_1 + mu_1 = MV_2 + mu_2 \]  

(2.1)

Where:

\begin{align*}
  M &= \text{mass of club} \\
  m &= \text{mass of ball} \\
  V &= \text{velocity of club} \\
  u &= \text{velocity of ball} \\
  1 &= \text{pre impact} \\
  2 &= \text{post impact}
\end{align*}

Since the velocity of the ball before impact is zero \((u_1 = 0)\), Equation (2.1) can be rewritten and solved for \(u_2\) as follows:

\[ u_2 = \frac{M}{m} (V_1 - V_2) \]  

(2.2)
Equations (2.1) and (2.2) assume that the impact is perfectly elastic, that is, that there are no energy losses (due to friction, heat, etc.) between the ball and club at impact. In reality, the ball/club impact is not perfectly elastic due to the inelastic response of the ball as it undergoes compression and recovery of its shape. In an effort to quantify these losses, a parameter called the Coefficient of Restitution (COR) has been defined as follows:

\[
\text{COR} = \frac{u_2 - V_2}{V_1} \quad (2.3)
\]

The COR is simply a ratio of the relative velocities of two colliding objects after and before impact and represents the efficiency of a collision. A COR of one indicates a perfectly elastic collision (all kinetic energy is conserved), while a COR of zero indicates a perfectly plastic collision (all kinetic energy is lost and the club and ball end up stuck together after impact). Solving the COR equation (2.3) for \( V_2 \) and substituting the solution for \( V_2 \) into Equation 2.2, the velocity of the ball after impact can be rewritten as:

\[
u_2 = \frac{M V_1 (1 + \text{COR})}{M + m} \quad (2.4)\]

With this solution for the launch velocity of the ball, equation (2.4), it can be illustrated how creating an impact between the club and ball actually multiplies the ball velocity. Assuming the club head to golf ball mass ratio is 4.3, and the COR of the collision is 0.83, it can be seen that the launch velocity of the ball is 1.48 times the club head speed.
before impact. Thus, by creating an impact between the club and the ball, one is able to impart a launch velocity almost 1.5 times faster than the speed of the club.

As can be seen from equation (2.4), the only variable that can be modified to increase the launch velocity of the ball is the COR of the impact. Since the COR is a measure of the efficiency of an impact, much effort has been placed on making the ball/club impact more efficient. As was explained earlier, the major loss mechanism at impact is the ball as it undergoes compression and then recovery of its shape. Michal and Novak (2001) performed an analysis on a golf ball to determine its behavior as it is compressed and then recovers its shape. Figure 2.1 below shows the force displacement curves attained by Michal and Novak for various loadings/unloadings of a golf ball.

Figure 2.1 Load displacement hystereses at various displacements for a wound golf ball (Michal, 2001).
As can be observed in Figure 2.1, the force displacement curve for unloading falls below the loading curve. This indicates that the ball recovers only a fraction of the total strain energy that was stored in compression as it regains its shape. Michal and Novak then performed a numerical integration of the force displacement curves and compared them with the maximum load reached by each curve (see Figure 2.2).

![Figure 2.2 Percent recovered energy for a single load cycle as a function of peak load (Michal, 2001).](image)

As can be seen from Figure 2.2, the amount of strain energy recovered by a golf ball is directly correlated to the maximum load experienced by the ball. The conclusion of these findings is that in order to maximize the efficiency of the ball club impact (maximize the COR), the maximum load experienced by the ball must be minimized. Following is a summary of the different areas of research that have been explored to minimize the maximum force on the ball and increase the COR of the ball/club impact.
2.1.2.1 Face Flexibility

Work in this area includes an investigation into the relationship between the impact face flexibility and its effect on the COR of an impact involving this face and another object (also known as the “trampoline effect”). This research concluded that increasing the flexibility of the impact face can increase the COR of the collision by about 12% over a very rigid face. This research also concluded that the COR is maximized when the half period of the impact face is matched with that of the ball (Cochran, 1999).

2.1.2.2 Materials

Previous work has focused primarily on increasing the trampoline effect exhibited by the devices. In order to increase the COR of these devices, the impacting face has been made thinner and larger, prompting research in the area of material science to find new materials that can withstand the tremendous forces experienced in the face as it becomes thinner. Currently, titanium alloys such as TI-6AL-4V and stainless steel alloys such as 413 and 17Cr-4N are being utilized in golf club designs for their high ratio of yield strength to modulus of elasticity, which helps to increase the COR and durability of the clubs by making the club face more compliant (Ogando, 2002).

2.1.2.3 Impedance Matching

Work has also been performed in the area of impedance matching of the impacting and impacted objects. It has been found that the COR between two objects can be maximized in the case that the frequencies at the minimal value of their respective mechanical impedances coincide with each other (Yamaguchi, 1999).
2.1.2.4 New Mechanism Development

A mechanism is defined as a component of a machine consisting of two or more bodies arranged so that the motion of one compels the motion of the other (Wilson, 2003). Since current club head designs have become limited by the availability of materials that allow them to become larger and thinner, mechanisms may provide the added flexibility needed without sacrificing durability.

Compliant Mechanisms

A compliant mechanism is a mechanism that gains all or at least some of its mobility from the deflection of flexible members rather than from movable joints only (Howell, 2001). Since compliant mechanisms do not contain any joints, they add no friction to the system, which will work to maximize the efficiency of the impact. While current golf club designs utilize compliance (deflection of the impact face) to increase the COR of an impact, it is likely that there exists more optimal compliant mechanism configurations for increasing the COR of the club-ball impact.

COR Mechanism Development Research

There has been some initial research into the development of concepts for compliant mechanisms possessing high coefficients of restitution (Woolley, 2003). Woolley outlined the framework for a process to create compliant mechanisms that maximize the COR of an impact. The first step outlined in this processes included defining the mechanism using lumped element models. The
next step involved using closed form static models and static finite element models to provide initial geometries of the mechanisms. The final step involved using dynamic finite-element simulations to refine the configurations and determine their viability for final design. This thesis will attempt to build upon and improve the process proposed by Woolley.

2.2 Increasing Golf Club Forgiveness

Golf club forgiveness refers to a club’s ability to hit long straight shots, regardless of how it was swung. In an effort to increase the forgiveness of golf clubs, three different techniques have traditionally been used. These include perimeter weighting the club, lowering the center of gravity of the club, and increasing the size of the sweet spot (area of the face over which a high COR impact will occur). The following sections will discuss these techniques and how they are used to increase a club’s forgiveness.

2.2.1 Perimeter Weighting

Perimeter weighting involves moving as much of the mass of the club as possible to the outer edges of the club. This works to increase the moment of inertia of the club, helping it to resist rotation from moments created by off-center shots.

2.2.2 Lowering the Center of Gravity

Center of gravity is defined as the average location of the weight of an object (Hibbeler, 1998). Lowering the center of gravity of a golf club increases a player’s ability to hit the
ball up into the air. Since this is a common problem most golfers share, a low center of gravity is desired for most clubs.

2.2.3 Increasing the Size of the High COR Zone

In an effort to increase the area of the impacting face over which a high COR is exhibited, previous research has focused on modifying the thickness of the impacting face. TaylorMade Golf Company has developed a driver using this very technology (Ogando, 2002). An “inverted cone” is machined into the reverse side of the faceplate, varying the thickness of the face from about 2.5 mm at the center of the cone to about 4.0 mm at the top. This cone shape helps to more evenly distribute the impacting forces over a larger area of the face, resulting in a larger area high COR zone on the face. In fact, this increased the size of the high COR zone from the size of a tee-head to the size of a quarter. Refer to Figure 2.3 for an illustration of this technology.

Figure 2.3 A driver that uses the inverted cone to expand the area of the impacting face over which a high COR is exhibited. A view of the back side of the club face, showing the inverted cone, can be seen on the right side of the figure.
2.3 Methods for Predicting Golf Club Performance

Methods that can be used to accurately predict the performance of a particular club design allow engineers the ability to refine and optimize designs before they are built. This section will explain the different methods designers are currently using to predict club performance.

2.3.1 Traditional Approach

Since current golf club designs are the result of years of evolutionary development, involving many small variations and improvements over previous configurations, predicting club performance is largely based upon the performance of previous versions of the designs. Using computer simulations, engineers are able to make small modifications to current designs and predict the performance of the new derivative configurations with relative accuracy.

2.3.2 Non-Traditional Approaches

Given that current golf club design focused on making incremental improvements to traditional designs, there has been little investigation into processes that could be used to predict the performance of completely new designs. Of the literature that exists, much of it focuses on the development of lumped mass models to predict performance. This section will explore applications lumped mass models, along with a new approach developed to predict the performance of compliant mechanisms possessing high coefficients of restitution.
2.3.2.1 Lumped Mass Models

Simplifying complex design problems into smaller, less complex pieces is a technique used commonly in engineering. In an effort to simplify the analysis of the club/ball impact, research efforts have focused on modeling the club and ball as lumped mass models.

Club Models

Current club designs utilize a thin plate as the impacting surface with the ball. This plate acts like a spring since it deflects upon impact with the ball. This club configuration is modeled using a two-mass, one linear spring model (see Figure 2.4).

As can be seen in the figure above, the club is separated into two masses: the moving mass of the face, and the mass of the remaining club. The two masses are attached by a linear spring, which accurately models small deflections of a flat plate. It should be noted that no literature currently exists on modeling club configurations that have non-linear face deflections.
Ball Models

The deflection of a golf ball is more complex than that of the club. As a result of this complexity, there exist a number of different lumped-mass ball models.

One approach to modeling a golf ball was proposed by Ujihashi (1994). This model consists of a linear spring in series with a parallel configuration of second linear spring and a dashpot (see Figure 2.5).

![Figure 2.5 Ujihashi lumped mass model](image)

Lieberman and Johnson (1994) developed a ball model that uses non-linear springs to model the deflection of the clubface. This model was also developed through experimental research. It consists of a non-linear spring in parallel with another non-linear spring and linear dashpot that are in series (see Figure 2.6)
A simpler, less complex ball model was developed by Goldsmith (1960). This model uses a non-linear spring in parallel with a non-linear dashpot to model the ball (see Figure 2.7).

Using the lumped mass models, the behavior of the dynamic system (created by combining the club and ball models) can be determined. Woolley (2003) used the Ujihashi ball model, in combination with the club model presented earlier, to predict the COR behavior of different club configurations. Since the United States Golf Association
has fixed the mass and stiffness characteristics of the ball, the only variables left to be modified are those of the club. Below is a plot showing the effect of varying clubface mass and flexibility on the COR of the impact between the club and the ball (see Figure 2.8).

This figure shows that there exist optimum combinations of clubface stiffness and mass that result in high COR impacts. The results of this plot can be used as a starting point for the initial design of a high COR club.
2.3.3 Computer Simulations

Using the results from the lumped-mass models as a reference point, the performance of the club configuration can be further predicted using computer simulations. Computer software programs such as Ansys and Abaqus can be used to model the actual geometric design of the club and predict its behavior upon impact with a ball.

2.4 Summary

Existing literature contains much information on enhancing the performance of club designs. While it is understood how a club’s performance can be improved, little research has been conducted to find new mechanisms, outside of the traditional thin-faced hallow-body designs, by which a high COR impact may be achieved. Also, there has been little effort to develop club simulations outside of those to predict the performance traditional linear bending configurations. These deficiencies provide the focus of this research: To investigate new mechanisms by which the COR of the ball club impact can be increased, and to develop new tools to simulate and predict their performance.
Chapter 3: Research Approach

Since previous golf club designs have followed an evolutionary development path, there has been little research focused on developing entirely new mechanisms to increase golf club head performance.

One of the few resources, upon which much of the approach of this research is based, is the process outlined by Woolley (2003). Woolley’s research outlines a process for the development of concepts for compliant mechanisms possessing high coefficients of restitution. As was explained in the previous chapter, Woolley’s process consists of three major stages:

1. Defining/modeling the mechanism using lumped-mass models.

2. Using closed form static models and static finite element models to provide initial geometries for the mechanisms.

3. Analyzing the mechanisms using dynamic finite-element simulations to refine the configurations and determine their viability for design.

While much was added to this process, it provided the basic framework upon which much of the analysis techniques used in this research were based. This chapter will outline the approach followed by this research to develop a new mechanism by which golf club performance could be increased.
3.1 Metric Generation

The first step in the developmental process involved the generation of a suitable set of metrics to evaluate and compare the new concept designs. A key element of the metric generation phase was to fully understand the functional specifications that were to be met or exceeded by new designs. Since many of the new concepts would differ greatly from current club designs, previous metric sets used to evaluate club designs had to be expanded to allow a complete evaluation of the new designs. The creation of a new set of metrics provided a starting point for the concept generation phase, by establishing targets for optimum performance characteristics for the new designs.

3.2 Concept Generation

With a sufficient set of metrics defined to provide target performance specifications for new concepts, the concept generation process could be initiated. To ensure that all concept possibilities were exhausted, an organizational system was created to structure the concept generation process. The organizational system was created by decomposing the golf club functionally into two different categories. The first category focused on where the energy of the impact would be stored in the club. The second category dealt with how the energy of the impact would be stored. After the organizational system was established, the concept generation process was initiated. Each category was individually targeted to generate a number of different concepts. A more detailed description of the concept generation process will be presented in Chapter 4.
3.3 Concept Selection

The next phase in the design process involved narrowing down the concepts generated in the previous phase to those that showed the most promise. The first step in the selection process involved the creation a concept screening matrix. Since detailed quantitative comparisons were difficult to achieve at this point in the design process, a coarse comparative screening technique was used to narrow down the concepts to those with the most potential.

3.4 Static Analysis

Using simplified models of the most promising concepts, static analysis techniques were used to quantitatively evaluate the different configurations and also to obtain initial geometries for the designs. Using the results of the static analysis, the best performing concept was selected for future study.

3.5 Dynamic Analysis

The next step in the development process involved using dynamic analysis techniques to predict the performance of the selected design. Using the results obtained from the static analysis of the new design, lumped mass models were generated and used to predict the dynamic behavior of the system. The lumped mass analysis provided an estimation of the design space for the particular configuration under investigation. With an estimation of the optimum design parameters, simplified models of the club were then simulated on
a computer using dynamic analysis software in an attempt to verify and identify the most promising configuration parameters.

### 3.6 Concept Validation

In earlier stages, more simplified dynamic models were used to allow more efficient parametric analysis of alternative design configurations. In an effort to achieve more accurate predictions of the concept’s performance, a more advanced model was created. The results from this more advanced model were used to better predict the design’s performance and finalize its geometry. It was at this point that it was decided if the design warranted further study, in particular prototyping.

### 3.7 Prototype Designs

Using the club geometry developed during previous stages, basic designs for a functional prototype of the high COR mechanism were produced.

### 3.8 Summary

Following a unique development process, largely based on Woolley’s process for developing compliant mechanisms that exhibit high levels of COR, plans for a completely new mechanism by which golf club performance can be increased were produced.
Chapter 4: Metric Generation

Since many of the new concepts generated during this research would differ greatly from current club designs, previous metric sets used to evaluate club designs had to be expanded to allow a complete evaluation of the new designs. This chapter outlines the process used to generate this new set of metrics.

4.1 Previous Metrics

The first step in generating a new set of metrics involved gaining an understanding of those used in previous research. Since there exist little information on processes used for designing completely new club designs, much of this metric research was based on that of Woolley (2003). Woolley established a basic set of metrics to evaluate new mechanisms that exhibit high levels of COR (refer to Table 4.1 below for a list of these metrics).
As illustrated in Table 4.1, Woolley identified three basic metrics: Weight, Durability, and COR. Club weight is essentially a fixed quantity at around 200 grams. While this weight is variable, 200 grams is the accepted standard across the golf industry, allowing club designers little or no freedom to create clubs that deviate greatly from this weight.

The next metric defined by Woolley was durability of the configuration. Woolley explained that the durability of a specific design is determined using targets for the maximum stresses experienced by the configuration under an impact loading. Since each design uses different configurations to provide compliance, the critical stresses vary according to the design and the configuration it utilizes (i.e. tension, compression, bending, etc.). The third metric used by Woolley was COR. Since the purpose of the design process was to produce mechanisms exhibiting high levels of COR, this metric would provide a direct measure of how the design performed.

### 4.2 Expanded Metrics

After reviewing the previous metrics, it was determined that a more expanded set of metrics would be required to evaluate and compare the many new and unique designs that
would be generated. Two different sets of metrics were generated: Expanded COR
Metrics, and Shot Straightness Metrics.

**4.2.1 Expanded COR Metrics**

The first set of metrics, Expanded COR Metrics, were generated to allow a more detailed
evaluation of the COR characteristics for each of the designs (see Table 4.2 for a list of
these new metrics).

**Table 4.2 An expanded list of metrics to be used in the concept evaluation and comparison process.**

<table>
<thead>
<tr>
<th>Expanded COR Metrics</th>
<th>Abbreviation</th>
<th>Description</th>
<th>Target Value</th>
<th>Acceptable Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Peak COR</td>
<td>COR_peak</td>
<td>Maximum COR produced by the club.</td>
<td>1</td>
<td>0.83</td>
<td>%</td>
</tr>
<tr>
<td>Active Face Area</td>
<td>AFA</td>
<td>Includes all portions of the club face that exhibit a COR greater than the specified COR_{min} (a.k.a. the “sweet spot”).</td>
<td></td>
<td></td>
<td>mm^2</td>
</tr>
<tr>
<td>Total Face Area</td>
<td></td>
<td>Area of the entire club face.</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Average COR</td>
<td>COR_{avg}</td>
<td>Average COR over the Active Face</td>
<td>1</td>
<td>0.83</td>
<td>%</td>
</tr>
<tr>
<td>Minimum COR</td>
<td>COR_{min}</td>
<td>A minimum COR value. This will specify the boundary of the Active Face</td>
<td>0.83</td>
<td>0.83</td>
<td>%</td>
</tr>
<tr>
<td>Active Face Ratio</td>
<td>AFR</td>
<td>This is the ratio of the Active Face Area to the Total Face Area</td>
<td>1</td>
<td>0.2</td>
<td>%</td>
</tr>
<tr>
<td>Max COR Area</td>
<td>MCA</td>
<td>This is the theoretical limit for the COR area under a distribution curve that is greater than COR_{min}. It is a box whose base is the line created by COR_{min}, and whose height is COR_{peak} - COR_{min}.</td>
<td></td>
<td></td>
<td>mm^2</td>
</tr>
<tr>
<td>Real COR Area</td>
<td>RCA</td>
<td>This is the actual area under the COR distribution curve that is located above the COR_{min}.</td>
<td></td>
<td></td>
<td>mm^2</td>
</tr>
<tr>
<td>COR Distribution Rating</td>
<td>CDR</td>
<td>This rating shows how the COR is distributed over the active face area. A rating of 1 means the dist. is completely flat, and a lesser rating indicates that the COR distribution is curved.</td>
<td>1</td>
<td></td>
<td>%</td>
</tr>
<tr>
<td>Club Rating</td>
<td>(COR_{avg} / AFR)</td>
<td>This rating can vary from 0 to 1. A rating of 1 indicating a design with a COR of 1 and an AFA that covers the entire face.</td>
<td>1</td>
<td>0.166</td>
<td>%</td>
</tr>
</tbody>
</table>

This new set of metrics was developed to address two key design issues, including the
maximum COR and the size of the high COR area each configuration utilized (as were
discussed in Chapter 2). These new metrics made it possible to not only compare the
max COR produced by a configuration (as was used in the previous phase of the research
to compare concepts, but also to compare and evaluate the shape and size of the high-COR area (Active Face Area) and the distribution of the COR over this area. The following is a figure to aid in the understanding of the Expanded COR Metrics (see Figure 4.1).

---

**4.2.2 Shot Straightness Metrics**

In an effort to more fully quantify the forgiveness of the designs, a new set of metrics called Shot Straightness Metrics were developed. These metrics were designed to evaluate the straightness of off-center shots, with straightness being defined as the proximity of the ball’s flight with respect to the target (refer to Table 4.3 for a list of the Shot Straightness Metrics).
Table 4.3 Shot straightness metrics.

<table>
<thead>
<tr>
<th>Description</th>
<th>Target Value</th>
<th>Acceptable Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Exit Angle</td>
<td>Angle between Input and Exit Ball Velocity Vectors</td>
<td>0</td>
<td>5</td>
</tr>
<tr>
<td>Critical Angle</td>
<td>Max Exit Angle that will allow a &quot;straight&quot; shot</td>
<td>0</td>
<td>5</td>
</tr>
<tr>
<td>Exit Angle Distribution</td>
<td>Distribution of Exit Angles across the face of the club</td>
<td>0</td>
<td>??</td>
</tr>
<tr>
<td>Distribution Area</td>
<td>Area under Exit Angle Distribution curve. The smaller this area is, the more likely one is to hit a straight shot</td>
<td>0</td>
<td>??</td>
</tr>
</tbody>
</table>

For an illustration of each of these metrics, refer to the following figures (see Figures 4.2 & 4.3).

It should be noted that the ball velocity vectors used to calculate the exit angle are the horizontal velocity vectors (parallel with the hitting surface). The horizontal velocity vectors allow a direct measurement of the side to side deviation of the ball’s flight from its intended path.
4.3 Summary

In order to better evaluate the many new and unique club configurations that would be generated by this research, it was necessary to expand the current list of club metrics. This new set of expanded metrics included two different groups of metrics: Expanded COR metrics and Shot Straightness metrics. With these new sets of metrics defined, work was begun on the concept generation process.
Chapter 5: Concept Generation

With a more thorough set of metrics defined, work began on developing concepts that would perform well when evaluated by these new measurements. This chapter will document the process used to develop the concepts, and will also provide a description of each of the concepts.

5.1 Organizational System

To aid in the process of generating concepts, the problem of increasing the COR of golf clubs had to be decomposed into simpler sub problems. The golf club was decomposed functionally into two different categories. The first category focused on where the energy of the impact would be stored in the club. The second category dealt with how the energy of the impact would be stored. Following is a description of each of these categories along with their corresponding sub-categories.

5.1.1 Location of Energy Storage

After a careful examination of the different possible club configurations, the location of where the impact energy could be stored was divided into two different categories: Deformable Face, and Floating Face configurations.
5.1.1.1 Deformable Face

The energy of an impact can be stored in a number of different areas within the club. Current designs use the face to store the energy of the impact with the ball. When the ball impacts the face, it deforms, storing the energy of the impact. This type of an energy storage configuration was classified as a Deformable Face configuration.

5.1.1.2 Floating Face

Another possible way to store the energy of the impact is to store the energy using a mechanism behind the face. By incorporating a mechanism behind the face of the club, the design will no longer be dependent on the face to provide all the energy storage of the impact. Since the face will float freely on the supporting mechanism, these configurations were classified as Floating Face configurations.

5.1.2 Method of Energy Storage:

Since the COR of a club is essentially a measure of the efficiency of the transfer of energy during impact, it was logical to decompose the golf club functionally into the different energy storage methods that could be used to store the energy of an impact. Five distinct categories of energy storage methods were selected. These categories included Axial, Bending, Torsional, Hybrid, and Other.
5.1.2.1 Axial

The first category is called Axial energy storage. Axial energy can be stored in two different forms: tension, and compression. Of the four categories, Axial is the most efficient since it uses the entire cross section of the material to store strain energy.

5.1.2.2 Bending

The next strain energy storage method was classified as Bending. Bending is the method used by current club configurations to store the energy of impact. Since bending does not use the entire cross section of the material (some of the material is in compression while the other is in tension, while some is not stressed at all), it is the least efficient way to store strain energy.

5.1.2.3 Torsional

Energy can also be stored in the form of torsion. While torsion is a more efficient form of energy storage than bending, it is not as efficient as axial energy storage.

5.1.2.4 Hybrid

As the name implies, Hybrid concepts will be those that incorporate two or more of the energy storage methods. While, for example, most concepts will not be perfectly Bending or Axial, Hybrid concepts will include only those that are deliberately designed to utilize two or more forms of energy storage.
5.1.2.5 Other

This category includes any other form of energy storage that does not fall into any of the categories above. For example, energy could be stored in the form of compressed gas within the body of the club.

5.2 Concept Descriptions

The next step in the developmental processes involved generating concepts within each of the organizational categories described earlier. Following is a description of each of these concepts, organized according to the system defined above.

5.2.1 Deformable Face, Bending Concepts

![Split Tube Hinge Diagram]

**Description:** Split-tubes will be placed around the perimeter of the face, allowing the ends of the face to pivot freely. The split-tubes will also store energy in the form of torsion.

Figure 5.1 Split Tube Hinge.
5.2.2 Floating Face Concepts

5.2.2.1 Axial Concepts

Tension Spoke

Description: Upon impact, the tension rods will elongate and store the majority of the energy in tension. It should be noted that this configuration will provide a nonlinear force to the ball—the force will increase as the face is deflected.

Figure 5.4 Tension Spoke.
**Tension Spoke Hoop**

**Description:** The tension rods will elongate, and the hoop will compress upon impact. This configuration will allow energy storage in the forms of tension and compression. This configuration will also provide a nonlinear force to the ball – the force will increase as the face is deflected.

![Figure 5.5 Tension Spoke Hoop.](image)

**Axial Cylinder**

**Description:** Like the tension rod design, all impact energy will be stored in tension in the cylinder. The cylinder may provide more stability than the rods. This design would also require a long cylinder (> 0.3 m) to function properly.

![Figure 5.6 Axial Cylinder.](image)

**Compression Spoke Hoop**

**Description:** Upon impact the compression rods will compress, elongating the hoop. This will allow energy storage in compression and tension. This will provide a nonlinear force to the ball – the force will initially be high, and then lessen as the face is deflected.

![Figure 5.7 Compression Spoke Hoop.](image)
5.2.2.2 Bending Concepts

**Tension Rod**

Description: All impact energy will be stored in the elongation of the tension rods. In order to function properly, this configuration requires the rods to be long (>0.3 meters).

![Tension Rod](image)

Figure 5.8 Tension Rod mechanism.

**Folded Tension Rod**

Description: All impact energy will be stored in the elongation of the tension rods. By folding the rods, a longer rod can be fit into a smaller area. Since a long rod is required to store the energy of the impact, this may be a way to fit it within an acceptable volume.

![Folded Tension Rod](image)

Figure 5.9 Folded Tension Rod mechanism.

**Single-Side Parallel Motion**

Description: The two beams attached to the floating face will act as a four-bar mechanism, ensuring parallel motion of the face. There will be some lateral translation of the face as it is deflected.

![Single-Side Parallel Motion](image)

Figure 5.10 Single-Side Parallel Motion mechanism.
**Double-Side Parallel Motion**

**Description:** Using this system of beams, the face will move only in the horizontal direction with no lateral deflection.

![Diagram of Double-Side Parallel Motion](image)

**Figure 5.11 Double-Side Parallel Motion mechanism.**

**Initially Curved Beam**

**Description:** All compliance will come from the bending of the initially curved beam behind the floating face.

![Diagram of Initially Curved Beam](image)

**Figure 5.12 Initially Curved Beam mechanism.**

**Orthoplanar**

**Description:** Two or more orthoplanar spring in series will restrict the motion of the floating face to parallel motion. All impact energy will be stored in the bending of the orthoplanar springs.

![Diagram of Orthoplanar mechanism](image)

**Figure 5.13 Orthoplanar mechanism.**
5.2.2.3 Torsional Concepts

**Torsional Parallel Motion**

*Description:* The split-tubes will act as energy storing joints.

![Torsional Parallel Motion](image)

Figure 5.14 Torsional Parallel Motion mechanism.

**Split-Tube Floating Face**

*Description:* The split-tubes will act as energy storing joints.

![Split-Tube Floating Face](image)

Figure 5.15 Split-Tube Floating Face mechanism.

5.2.2.4 Hybrid Concepts

**Axial Torsional Parallel Motion**

*Description:* Energy will be stored in the split-tubes and the tension rod. This will allow energy storage in the form of both torsion and tension.

![Axial Torsional Parallel Motion](image)

Figure 5.16 Axial Torsional Parallel Motion mechanism.
5.2.2.5 Other Concepts

<table>
<thead>
<tr>
<th>Gas Filled Membrane</th>
</tr>
</thead>
<tbody>
<tr>
<td>Floating Face</td>
</tr>
<tr>
<td>Gas Filled Body</td>
</tr>
<tr>
<td>Flexible Club Body</td>
</tr>
</tbody>
</table>

**Description:** Upon impact, the face will compress the body of the club. Energy will be stored both in tension/bending in the body of the club, and in the compressed gas.

---

**Figure 5.17:** Gas Filled Membrane mechanism.

5.3 Summary

In order to begin the concept generation processes with a structured effort, an organizational system was created to help classify the different types of concepts that would be generated. The golf club was decomposed into two main functional categories: location of energy storage and method of energy storage. Within each of these categories a number of sub-categories were enumerated to help classify the concepts with more detail. After the organizational system was established, the concept generation process was initiated. Each category was individually targeted to generate a number of different concepts. After completing the concept generation process, work began on the concept selection process, which will be described in the following chapter.
Chapter 6: Concept Selection

Upon completion of the concept generation process, it became necessary to narrow the number of concepts down to the most promising designs. In an effort to achieve this goal, the concept selection process was initiated. This chapter documents the procedures followed to select the most promising concepts for future study.

6.1 Concept Selection

The first step in concept selection process was to conduct a concept screening. Since detailed quantitative comparisons would be difficult to obtain at this stage in the design process, a coarse comparative concept screening process was ideal to select concepts for further study.

The first step in the concept screening processes involved the creation of a screening matrix. After referring to the metrics described in the Chapter 4 “Concept Generation,” five general selection criteria were derived to be used in the screening matrix. These more general selection criteria allowed each concept to be compared qualitatively with the current club configuration. A sixth selection criterion called “Chance of making viable design” was created to help in selecting a design that could be realized. After the completion of the screening matrix, each of the concepts was rated, and these scores were
summed to determine the most promising concepts. Refer to the following page for the
completed concept screening matrix (see Table 6.1).

Upon completion of the screening matrix, it became evident that there were a number of
promising concepts. In order to differentiate the concepts, they were classified by color
according to the order in which they would receive attention. Following is an
explanation of each category, along with a description of the concepts within them.

### 6.1.1 High Priority Concepts

The highest scoring concepts were highlighted with green. These concepts included the
Tension Spoke, Tension Spoke Hoop, and Compression Spoke Hoop designs. This
outcome was expected since these designs utilized the highly efficient Axial energy
storage method in a package that could be feasibly fit into a golf club. Refer to the
following figure for a better illustration of these configurations.

![A three dimensional sketch of the Tension Spoke Hoop design.](image)

Figure 6.1 A three dimensional sketch of the Tension Spoke Hoop design.
### Table 6.4: Concept screening matrix.

**Color Key**
- First Priority for Modeling
- Second Priority for Modeling
- Non priority, hold for future reference
- Not realizable

<table>
<thead>
<tr>
<th>Size of High COR Zone</th>
<th>Bending</th>
<th>Axial</th>
<th>Torison</th>
<th>Hybrid</th>
<th>Deformable Face</th>
<th>Other</th>
</tr>
</thead>
<tbody>
<tr>
<td>Single-Side Parallel Motion</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Double-Side Parallel Motion</td>
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<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Tension Rod</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Cylinder</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Folded Cylinder</td>
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<td>1</td>
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<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Tension Spoke Hoop</td>
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<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Compression Spoke Hoop</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Split-Tube Folding Face</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Torisonal Parallel Motion</td>
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<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Axial Torsion Parallel Motion</td>
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<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Split-Tube Hinge</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Tapered-Wedge Hinge</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Folded Face Hinge</td>
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<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>Gas-Filled Membrane</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
</tbody>
</table>

**Face Stability**

| Specific Volume Efficiency | 0 | 0 | 1 | 1 | 1 | 1 | 1 | 1 | 0 | 0 | 0 | 0 | 0 | 0 |

| Weight | 0 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 0 | 0 | 0 | 0 | 0 | 0 |

| Moment of Inertia | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 | 1 |

**Chance of making viable design**

- Probably (1), Maybe (0), Probably not (-1), Major Obstacles (-2)

| Chance of making viable design | 0 | 0 | 1 | 1 | 1 | 1 | 1 | 1 | 0 | 0 | 1 | 1 | 2 | 2 | -1 |

**Total**

| 0 | 1 | 2 | 1 | 1 | 1 | 1 | 1 | 3 | 4 | 1 | 4 | 0 | 0 | 1 | 1 | 2 | 2 | -1 |
The major limitation for the Tension designs was the floating face. Finding a material light and strong enough with which to construct the design was critical to its success. Like the Tension designs, the Compression designs were limited by the floating face, but unlike the Tension designs buckling was also an issue since the spokes would be under compression upon impact with the ball.

6.1.2 Second Priority Concepts

Yellow concepts were those that did not score as high as the green, but that still scored high enough to receive second priority in the analysis. While the concepts in this category did not score as high, many of them were much more realizable than those in the green category. The major limiting factor of these designs was also be the floating face.

6.1.3 Non-Priority Concepts

The white concepts were those that were kept in the concept library, but they were not a priority to investigate in the analysis processes that followed.

6.1.4 Non-realizable Concepts

The red concepts were those that were deemed “non realizable” due to the major obstacles that had to be overcome to achieve a successful design. These concepts were eliminated from the concept library.
6.2 Summary

The first step in the concept selection process involved creating a screening matrix that was used to narrow down the concept pool to the most promising concept configurations. The screening matrix allowed each concept to be scored according to a set of general selection criteria, and then these scores were summed. The concepts with the highest scores were selected for future study. These designs included the Tension Spoke designs, which will be the focus of discussion in the chapters that follow.

The next step in the development processes included a more detailed analysis of the selected concepts. This analysis began with a static analysis, the results of which will be presented in the following chapter.
Chapter 7: Static Analysis

After generating a number of different club concepts, the Tension Spoke club designs were selected for further study. The Tension Spoke designs included two variations: the Tension Spoke design which relied on the rods to provide all compliance, and the Tension Spoke Hoop design which relies on the rods and an exterior hoop to provide compliance. In order to obtain the initial geometry of the designs, a static analysis was performed. The relative simplicity of a static analysis (as compared with a dynamic analysis) allowed a number of different geometries to be quickly analyzed. As a consequence of the simplicity of the static analysis, the results were highly dependant on the assumptions used to derive them.

7.1 Results

Four different initial geometries were generated using a number of different assumptions. This section will first discuss these assumptions, and then the results of the analysis will be presented.
7.1.1 Assumptions

The following sections include a description of the assumptions used to make the initial geometries.

7.1.1.1 Maximum Load Experienced by the Ball

Since this load is highly dependant on the compliance of the face (or the mechanism behind it) the maximum ball load at impact can vary considerably. Using data from Woolley’s research as a reference, a range of max loads was generated. The minimum load was set at 9 kN, while the max load was set at 13 kN (Woolley, 2003).

7.1.1.2 Constant Spring Constant (k)

In the first phase of research, a range of optimum spring constants for the face was generated. The range that resulted in the highest COR varied from 2000000 – 5000000 N/m. Since it was not known how well these values would predict the performance dynamically, this range was used to establish the maximum and minimum values for the static analysis.

Another important assumption involving the spring constant was that it was constant over the full deflection of the face. It is known that the Tension Spoke designs will exhibit a non-linear force-deflection curve due to the characteristics of the geometry. Since this non-linear behavior is not known, the assumption of a linear force-deflection curve was used to predict static deflection in the analysis.
7.1.1.3 No Bending

Since the majority of static deflection of the face will occur as a result of the elongation of the tension rods, as opposed to bending of the rods, it was assumed in the static analysis that the rods did not bend.

7.1.1.4 Allowable Yield Stress

As a result of the assumption that the rods are in pure tension (no bending), the maximum allowable yield stress in the rods had to be determined. In pure tension, the entire cross-section of the rod will experience the same stress level and as a result, the maximum allowable stress in the rod was set to a value 10% below the yield stress of the material used to make the rods.

7.1.1.5 Material

It was determined that the entire design would be made using titanium. This decision was based on titanium’s high strength to Young’s modulus ratio (allowing more compliance) and the fact the TaylorMade will be more easily able to construct the prototype if it is constructed of titanium. Following are the material properties of titanium that were used in the analysis:

- Yield stress: $1 \times 10^9$ N/m
- Allowable yield stress: $9 \times 10^8$ N/m
- Young’s modulus: $1.16 \times 10^{11}$ N/m
### 7.1.2 Initial Geometry

Using the assumptions above, four different geometries were generated using different combinations of maximum loads and face spring constants (see Table 6.1).

<table>
<thead>
<tr>
<th>Table 7.1 Four different combinations of max load and spring constants that were used to generate initial geometries.</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Max Load</strong> (N)</td>
</tr>
<tr>
<td>Geometry 1</td>
</tr>
<tr>
<td>Geometry 2</td>
</tr>
<tr>
<td>Geometry 3</td>
</tr>
<tr>
<td>Geometry 4</td>
</tr>
</tbody>
</table>

Following is a description of the geometry required to accommodate each of the conditions displayed in Table 6.1.

#### 7.1.2.1 Geometry 1

The following describes the details of Geometry 1.

**Tension Spoke**

It was found that the rods in for this set of conditions would have to be at least 3.7 cm to withstand the impact and provide the proper amount of compliance.

**Tension Spoke Hoop**

Compliance from the hoop would make smaller designs possible (< 3.7 cm radius). As the geometry gets smaller, more compliance would be required from
the hoop. The length of the spokes could essentially be reduced to zero by
transferring all the storage of strain energy to bending in the surrounding hoops.

7.2.1.2 Geometry 2

The following describes the details of Geometry 2.

**Tension Spoke**

This combination of low max load and a high spring constant resulted in the
smallest possible configuration (using the assumptions listed above). The length
of the rods for this design needed to be at least 1.5 cm.

**Tension Spoke Hoop**

For this set of conditions, it may not be necessary to use the hoop for more
compliance.

7.2.1.3 Geometry 3

The following describes the details of Geometry 3.

**Tension Spoke**

This max load and spring constant combination yield no possible geometries that
had rod 5 cm or shorter. 5 cm was determined to be the maximum rod length
since geometries any larger than this would be very hard to use a golf club.
Tension Spoke Hoop

All of these designs would require some compliance from the outer hoop.

7.1.2.4 Geometry 4

The following describes the details of Geometry 4.

Tension Spoke

This load and stiffness combination requires the rods to be at least 2.1 cm long.

Tension Spoke Hoop

As in the geometries described above, the hoop could be used to reduce the required length of the spokes.

7.2 Discussion of Results

Using the assumptions described earlier, a number of viable configurations were generated. If the Tension Spoke design is desired, the static analysis predicted that the length of the rods may vary within the range of 1.5 – 5 cm (if Geometry 3 is excluded). Since the static analysis was used to see the “best case” and “worst case” scenarios, it is expected that the optimum design will lie within this range. If the Tension Spoke design results in configurations too large to incorporate into a golf club, the Tension Spoke Hoop design can be used to reduce the required rod length. Figure 6.1 displays a possible Tension Spoke Hoop design.
Figure 7.1 A possible Tension Spoke Hoop configuration. A bendable beam is used to provide the “hoop compliance.” The view is straight on (through the club face).

This design (the Tension Spoke Hoop design) would create the necessary compliance to achieve the maximum COR. It should be noted that this design utilizes some bending (in the bendable beam) to store additional strain energy, but maximizing the tension in the rods minimizes this bending.

7.3 Summary

The static analysis confirmed that there exist a number of viable configurations for the Tension Spoke designs. The geometries obtained from this analysis provided a starting point for more detailed dynamic analyses. The next chapter will discuss the dynamic analysis of the Tension Spoke designs.
Chapter 8: Dynamic Analysis

After generating a number of different club concepts, the Tension Spoke club design was selected for further study. The first step in the analysis processes involved a static analysis of the design. The results from the static analysis indicated that there were a number of viable configurations of the design. The next step in the process involved using the static model to predict the behavior of the Tension Spoke design.

8.1 Static Behavior Prediction

In an effort to predict the behavior of the Tension Spoke design, static analysis techniques were used to develop a number of different force deflection curves. This involved comparing the force applied at the face to the resulting deflection of the face. After the force deflection curves were generated, a curve fitting process was used to determine the force deflection equations that could be used to predict the behavior of the Tension Spoke design. These equations were then used in the next step of the analysis, the dynamic analysis of the lumped mass model, to predict the dynamic behavior of the tension spoke design.
The static force deflection analysis was first performed assuming that the spokes stored the energy of the impact in the form of pure tension (with no bending). Using the following equation for the deflection ($\delta$) of an axially loaded beam,

$$\delta = \frac{(F \cdot L)}{(A \cdot E)}$$

(8.1)

Where:

- $F$ = Force applied in axial direction
- $L$ = Length of beam
- $A$ = Cross sectional area of beam.
- $E$ = Modulus of elasticity of beam.

Force deflection curves for various Tension Spoke configurations were generated (see Figure 8.1). It is important to note that the cross sectional area of the beam was calculated from the maximum stress allowed in the beam. Since the entire cross section of the beam would experience the same stress level (since the beam is in pure tension), the maximum allowable stress was established to be 90% of the yield stress of titanium.
As can be seen in Figure 8.1, the force deflection curves for this design are highly nonlinear. Using these curves, a mathematical equation was developed that could adequately predict the nonlinear behavior of the pure tension designs. It was found that the following equation:

\[ F = k x^3 \]  

(8.2)

would predict the behavior of the pure tension design. The following figure shows a plot of the pure tension force deflection curve with the matching curve predicted by equation (8.2).
In order to verify the behavior predicted by the pure tension model, Abaqus was used to generate a number of different force deflection curves. The spokes were modeled with a rectangular cross section since this shape would be easily reproducible in a physical prototype.

After numerous iterations, it was discovered that the spokes had to be made sufficiently thin before they would adequately model the pure tension assumption. This was an obvious phenomenon since making the spokes thinner would reduce the amount of bending used to store strain energy upon impact. In an effort to quantify non-dimensionally how thin the spokes should be, a non-dimensional parameter called the spoke length to thickness ratio was generated. It was found that a spoke length to
thickness ratio of 525 or greater would result in designs that closely approached the behavior of the pure tension designs.

Since the entire Tension Spoke configuration (which includes hundreds of spokes) would be too time consuming and unpractical to model in Abaqus, only a portion of the configuration was analyzed. The model was reduced to an infinitely rigid center plate (to model the floating face) attached to two spokes, one on each side of the plate. The force applied to this new configuration was reduced to equal the equivalent load two spokes would experience in the full configuration (the total force was divided by the total number of spokes the design called for, and then multiplied by two—the number of spokes in this model). The results of this model, compared to those obtained from the theoretical pure tension models are displayed in Figure 8.3 below.

![Curve Fit Comparison](image)

**Figure 8.3** Plot showing curve fit accuracy to Abaqus output (please note that this is a zoomed plot to better show the curve fits).
Since there was still some error between the behavior predicted by Abaqus and the pure tension model (due to the inability to eliminate all bending from the system), a new curve was fitted to the Abaqus force deflection curve. The Abaqus curve was used because it more accurately represents the true physical behavior of the Tension Spoke design. The new equation, also nonlinear, was determined to be:

\[ F = k x^{2.9438} \]  

(8.3)

When compared with the actual force deflection curve predicted by Abaqus, this equation predicts its behavior within an average of a 1.5% error (see Figure 8.3).

8.2 Lumped Mass Models

Using the characteristic force deflection equation determined from the static analysis of the Tension Spoke design, lumped mass models were generated and used to predict the dynamic behavior of the system. Two different lumped mass models were used in this analysis in an effort to observe the effects the different models would have on the dynamic predictions of the club/ball impact. This section will describe the two lumped mass models and will also present the dynamic behavior predictions obtained from each model.
8.2.1 Ujihashi Model

The club was modeled using two masses joined by a nonlinear spring defined by Equation 8.3. The first mass, called the face mass, was used to represent the total moving mass (including the face mass with the contributing mechanism mass). The second mass, called the club mass, represented the remaining (non-moving) mass of the club. Since the total mass of most common metalwood drivers is 200 grams, the sum of the two club masses was set to equal 200 grams. The ball was modeled using the Ujihashi ball model. This model used a linear spring in series with another linear spring and a damper that are then attached to a mass. The entire ball-club lumped mass system, including the specified stiffness and damping values for the ball model, is illustrated in Figure 8.4.

![Figure 8.4 Lumped mass model for the ball/club system. For the Ujihashi ball model, $K_1=4.9e6$, $K_2=4.7e6$, $M_{\text{ball}}=45.1g$, $C=250\ \text{N/(m/s)}$. A total club mass (which includes $M_{\text{club}}$ and $M_{\text{face}}$) of 200g was used, and $K$ is a non-linear spring with $a=2.9438$.](image-url)
With the lumped mass model properly defined, the differential equations of motion for the ball/club system were defined. The next step in the process involved writing the equations of motion in state-space form so a computer could solve them numerically using a specified set of initial conditions. The club (including the face mass and club mass) were given an initial velocity of 48.3 m/s, while the ball was given and initial velocity of 0 m/s. A built-in Runge-Kutta formula was then used in Matlab to solve the equations for the max ball velocity produced by the impact for a given range of face stiffnesses and masses.

8.2.2 Results

Using conservation of momentum, the maximum ball velocity data was used to calculate the COR of the collision using the following equation:

\[
COR = \frac{u_2(M + m)}{MV_1} \tag{8.4}
\]

Where:

\[M = \text{Total club mass}\]
\[m = \text{Ball mass}\]
\[V_1 = \text{Initial club velocity}\]
\[u_2 = \text{Maximum ball velocity}\]

See Figure 8.5 for a plot of face stiffness and mass versus COR.
Figure 8.5 Using the results obtained from the lumped mass model, the above COR plot was generated for the Tension Spoke design.

Figure 8.5 shows that the range of mass and stiffness over which the configuration exhibits a high COR (COR > .83) is relatively large. Comparing the COR plot for the Tension Spoke design with the COR plot for the typical linear face design (see Figure 8.6), it can be seen that the optimum moving mass for the system increases from ~20g (linear face value) to ~33g for the nonlinear face design. This is a very promising phenomenon since it will allow the mass necessary to create the heavier “floating face” designs.
Another interesting behavior of the Tension Spoke design that can be observed from the COR plot is that the optimum design (one that exhibits a COR > .83) is relatively insensitive to stiffness above a certain value. Since the stiffness of the Tension Spoke design is directly correlated to the length of its spokes, the stiffness values in the COR plot were replaced by their corresponding spoke lengths (see Figure 8.7).

Figure 8.6  A comparison of the COR plots for linear and Tension Spoke designs. The dark purple/burgundy regions are those of highest COR (.84-.88). Notice how the high COR region moves to the right (increasing moving mass) for the Tension Spoke design.
Figure 8.7 shows that the optimum spoke length for the Tension Spoke design should lie within the range of 24mm to 31mm long. These spoke lengths are very promising in that they will allow the Tension Spoke mechanism to fit within traditional oversized club head dimensions.

### 8.2.3 Velocity Analysis

Since the actual club head speed will vary considerably (due to the different abilities of the golfers who swing the club), an analysis was performed to observe the effect of changing the initial velocity of the club. Using the same procedure as that described above, the ball/club model was analyzed at three additional initial club velocities: 100
mph, 90 mph, and 80 mph. The results of these analyses can be observed in the figure below.

![Figure 8.8 Plots showing the effect of decreasing club speed on optimal COR club design. The purple regions are those of highest COR and represent the range of .854-.856.](image)
It can be seen that as the club speed decreases, the optimal stiffness at which a high COR is exhibited increases. This result was expected since at lower club speeds the ball will deform less, requiring less compliance from the club to optimize the efficiency of the impact.

8.2.4 Other Lumped Mass Model Results

It was anticipated that there should exist some rate of face energy absorption upon impact with the ball that will provide optimal impact efficiency. In an effort to explore this idea, a number of different curves were analyzed using the closed form solution obtained from the lumped mass model. For a discussion of these results, please refer to Appendix I.

8.3 Abaqus Model

Since the lumped mass model is only capable of providing an estimation of the behavior of a dynamic system, an FEA computer model was used to more realistically simulate the club/ball impact. Using the results obtained from the lumped mass model as a starting point for the initial geometry of the Tension Spoke club design, a number of simulations were run to verify the results obtained from the lumped mass model. The following section will explain the procedure used to model the Tension Spoke design and will also present the results obtained from the computer simulations.
8.3.1 Procedure

To simplify the computer model and reduce the computation required to obtain results, the Tension Spoke design was modeled using two beams attached to an infinitely rigid plate (see Figure 8.9).

![Figure 8.9 Tension Spoke design as modeled and meshed in Abaqus. The center square portion is the infinitely rigid plate, while the attached rectangles are the two beams.](image)

An infinitely rigid plate was used to model the floating face since it did not provide any compliance upon impact with the ball.

The total number of spokes required for each design was determined from the maximum forces that the ball would exert on the face of the club. This force was determined by examining the accelerations of the ball in the lumped mass model. With the maximum force, the total required cross sectional area of the spokes was determined according to the maximum allowable stress that could be reached in the spokes. Since the spokes would be in almost pure tension when impacted by the ball, the maximum stress level was set to be 90% of the yield stress of titanium. Because the spokes had to be so thin (in
order to limit the amount of strain energy stored in bending), many of the configurations would require hundreds of spokes to achieve the necessary cross sectional area. In order to represent the behavior of hundreds of thin spokes, the two spokes in the model maintained their original geometry, but their material properties were modified. The modulus of elasticity and the density of the two spokes were multiplied so as to equal the equivalent stiffness and weight of the complete Tension Spoke design (which includes hundreds of thin spokes).

To model the spokes as being attached to a club, the end nodes of the spokes were given a non rotational boundary condition and they were also given a density such that the entire club mass (including the Tension Spoke mechanism and the supporting club) would equal 200g.

While moving mass was easily defined in the lumped mass model, it became more difficult to define the moving mass of the Abaqus model. The moving mass of the design would include the entire center plate mass, with some portion of the spoke mass also contributing. Since the spokes in the design were very thin, they behaved almost as if they were pinned-pinned. This allowed the calculation of the equivalent moving mass of the spokes at the midspan using the kinetic energy of the spokes. It was calculated that the equivalent moving mass of the spokes would be a third of their total mass. Thus the moving mass of each design was defined as the mass of the center plate (which could be adjusted by modifying its density) plus one third of the mass of the spokes.
Before the Tension Spoke model could be analyzed by Abaqus it had to be meshed. One mesh element was used for the entire model and it was the S4R type in Abaqus. The number of elements used for the spokes in the different configurations was kept constant at 296.

Using the meshed model explained above and a proprietary ball model created by TaylorMade-adidas Golf, the COR of the ball/club impact for a number of different club configurations was then calculated according to current USGA standards.

### 8.3.2 Results

Results were first obtained for the optimum configuration (one that exhibits the highest possible COR) as was determined from the lumped mass analysis. This configuration had spoke lengths of 29.8 mm and a moving mass of approximately 34g. Using this configuration as a starting point, the surrounding design space was explored to find the areas where the highest COR was exhibited according to Abaqus. A total of 24 points were obtained throughout the design space, and then Matlab was used to interpolate a design space surface between these points (refer to Figure 8.10 for a plot of the design space generated from the Abaqus output).
It can be observed in the figure above that the Tension Spoke design is capable of achieving high COR values (> 0.83). One important detail that must not be overlooked is the amount of moving mass for each configuration that is available for construction of the floating face. This value was calculated for each configuration (spoke length and moving mass combination) by subtracting the moving mass of the spokes from the total moving mass. Assuming that the floating face would require at least 8g of mass to be constructed, a new “feasible” design space surface was created (see Figure 8.11 below).
As can be seen in Figure 8.11, the feasible design space as was determined by the Abaqus output shows the capability of producing configurations that exhibit COR values in the area of 0.83.

8.3.3 Swing Speed vs. COR Analysis

The Abaqus model was also used to predict the COR of the Tension Spoke Design for different swing speeds. The COR for a number of Tension Spoke configurations was calculated at three different swing speeds: 90, 109, and 120mph. These results were then compared with a typical deformable face driver and the results are plotted below (see Figure 8.12).
Figure 8.12 COR comparisons at different swing speeds for the Tension Spoke design and a typical deformable face driver.

It should be noted that the Tension Spoke configurations were designed to have the same COR as the typical deformable face driver at a swing speed of 109mph to allow a better comparison (except for the red line which was included to show the effect of changing the moving mass on COR).

A number of interesting trends can be seen in this plot. The first of which is that the length of the spokes determines how sensitive the design will be to changes in swing speed. As the length of the spokes gets longer, the COR of the design becomes less...
sensitive to swing speed. Another interesting trend is that as moving mass is reduced, the COR of the configuration increases while there is little effect on the design’s sensitivity to swing speed (the slope of the line remains constant as mass is changed).

Comparing the Tension Spoke configurations with the typical deformable face driver, it can be seen that the COR of the Tension Spoke designs is far less sensitive to changes in swing speed than is the typical deformable face driver. This means that the Tension Spoke designs will allow higher COR impacts at faster swing speeds when compared to the typical driver. These results are very promising, indicating that players with faster swing speeds will be able to hit the ball further with USGA conforming golf clubs.

### 8.3.4 Discussion of Results

The COR results obtained from Abaqus were different than those obtained from the lumped mass analysis. There are various explanations for why these two outputs did not correlate and these will be explained in the following sections.

#### 8.3.4.1 Moving Mass

The amount of moving mass in the lumped mass model is easily defined, while that of the Abaqus model is more difficult to define. As was explained earlier, the moving mass of the spokes was calculated from theory using principles of kinetic energy. In reality, as Abaqus attempts to portray, the spokes may contribute more, or less to the moving mass of the design. This uncertainty of the exact moving mass in the Abaqus model could be one possible source of error between the two results.
8.3.4.2 Ujihashi Lumped Mass Ball Model

As was explained earlier in this paper, the lumped mass Ujihashi ball model was used to model the golf ball. This model is a one-degree of freedom model of a golf ball and as a result, some error in its behavior is expected.

The Ujihashi model was generated from data of a ball that was fired at a solid steel target. Figure 8.13 includes two plots of the performance characteristics of the ball model compared with the actual experimental ball behavior as reported by Ujihashi (Ujihashi, 1994).

![Figure 8.13: The curves of interest are the “Experiment” and “One-Degree Model” curves. Notice the differences in the curves at high forces and deformations (Ujihashi, 1994).](image)

As can be seen in this figure, at high force and deformation levels, there is considerable error between the ball behavior predicted by the Ujihashi model and that obtained from experiment. To add to the possible uncertainty, the Tension Spoke design behaves highly nonlinearly when impacted by the ball, unlike the rigid steel plate Ujihashi used to obtain the above data. This could be a significant source of error, in that the Ujihashi model may become highly inaccurate when used with a nonlinearly deforming club. In
conclusion, there is no documentation reporting the accuracy of the Ujihashi ball model when impacting nonlinearly deforming surfaces, and thus there is no assurance that the model is accurate in this situation.

After reviewing the above assumptions used in the lumped mass model, it was concluded that further work would build upon the results obtained from the Abaqus model. While the results obtained from the lumped mass model were useful in obtaining a starting point in the design space, the Abaqus results would provide more accurate predictions of the actual Tension Spoke design.

8.4 Summary

Using static analysis techniques, the force-deflection curve for the Tension Spoke design was observed. A mathematical equation was fit to match the curve so the force deflection behavior of the face could be simulated correctly through mathematical analysis. Next, a lumped mass model of the ball and club was created, allowing the dynamic behavior of the Tension Spoke design to be predicted. Using the results from the lumped mass model as a starting point, a simplified model of the design was created and tested using Abaqus software. The results of the Abaqus analysis were found to be different from those obtained from the lumped mass analysis, and it was concluded that further work would be based upon the results obtained from Abaqus.
The next step in the design process involved creating a more detailed model of the Tension Spoke design and analyzing it using Abaqus. The next chapter will present this process and the results it yielded.
Chapter 9: Concept Validation

Prior models used to simulate the behavior of the Tension Spoke design were greatly simplified to allow more efficient parametric analysis of alternative design configurations. In order to more fully predict the potential performance of the Tension Spoke configuration, a more advanced model was generated. This model was selected based on a design obtained from previous simulations with a predicted peak COR of 0.825 and likely robust behavior. This report will describe this model and will present the results obtained from a dynamic analysis of the new model.

9.1 Dynamic Model Specifications

The model used a total mass of 200 grams, the standard weight for a typical driver. The geometry (including dimensions) is shown on the following page in Figure 9.1. The model was constructed of deformable elements made of titanium and were assigned an unscaled modulus of elasticity of 110 GPa and an unscaled density of 4.5 grams per cubic centimeter.
One hundred and twenty four layers of thin titanium sheets, the total number of layers required to build this specific Tension Spoke design, were modeled as five layers for computational efficiency (as apposed to previous models that used only one sheet to model the one hundred and twenty four layers). In order to maintain proper stress values and dynamic response, the layers were assigned their true thickness (0.066 mm) but employed mass and stiffness scaling. The scaling factor was 24.8 (124/5). The beams were modeled using S4R shell elements. Each layer had ten beams extending radially from the center. Each beam was 34.7 mm long, 3 mm wide and 0.066 mm thick. The center section of each layer was deformable as shown in Figure 9.2.
The face was modeled as a rigid body. A rigid beam was used to connect the face to the first layer. The layers were connected with a deformable solid circular titanium shaft (B31 elements) with an eight mm diameter (see Figures 9.1 and 9.3). The total moving mass was calculated by summing the mass of the face, rigid shaft, deformable shaft, layer center sections, and one third of the mass of the beams. The model had a total moving mass of 24.9 grams.

The spokes were attached, at the perimeter, to a rigid body cage that maintained spacing between layers (see Figure 9.3). The cage also made it possible to have assignable mass and rotational moments of inertia for the model. These values were chosen such that the entire model had its center of gravity 31.6 mm behind the face, \( I_{xx} = 250 \text{ kg mm}^2 \), \( I_{yy} = 381 \text{ kg mm}^2 \), and \( I_{zz} = 283 \text{ kg mm}^2 \) (these properties were obtained from TaylorMade and are
typical of current drivers). Because of symmetry in this model, only $I_{yy}$ had an effect on performance.

![Figure 9.3 Entire model including rigid body cage](image)

**Figure 9.3** Entire model including rigid body cage

### 9.2 Discussion of Results

Using the model described in the previous section, dynamic analyses were performed using Abaqus to observe the performance of the configuration in terms of COR and maximum spoke stress values. A ball model was fired into the face of the Tension Spoke model at 0, 5, 10, 15, and 20 mm offsets and the results of these analyses will be presented in this section.
9.2.1 COR Distribution

In order to observe the COR performance characteristics of this configuration, a distribution of COR values across the face of the design was generated using the data output from Abaqus (see Figure 9.4 below).

![COR Distribution Across Club Face](image)

Figure 9.4 COR distribution from center of face to 20mm from center.

This configuration was designed to exhibit a COR of approximately .825 when impacted by a ball at the center of its face (as is exhibited in Figure 9.4). At a 5mm offset from the center of the face, the losses in COR were negligible. As the offset was increased, the COR of the impact decreased. In order to better understand the level of attenuation of the COR values, a table was constructed showing the percentage of the maximum COR (the COR exhibited at a zero offset) that the club exhibited at each offset (see Table 9.1).
Observation of Table 9.1 shows that the Tension Spoke club configuration tested in this simulation performs very well. Even at a 20mm offset, the club still produces a COR value of .802, or 97.13% of the maximum designed COR of the club. These early results, with an un-optimized club configuration are promising in that they show evidence that the Tension Spoke design could be very forgiving to players who do not consistently hit shots off the center of the club face.

9.2.2 Stress Distribution

Another important characteristic of the design that was observed from the analyses was the stress level found in the spokes upon impact with the ball. In order for the design to be feasible, the stresses in the spokes cannot exceed the yield strength ($\sigma_y$) of the material of which they are constructed (in this case titanium with $\sigma_y \approx 1 \text{ GPa}$). The stress level was observed at a node at the center of a spoke on the first layer of spokes (nearest to the face). This node was selected on the spoke directly opposite the location where the ball impacted the face (see Figure 9.5).
Figure 9.5 Model showing the element where stress levels were observed.

This node was selected upon observation of the stress distributions within the spokes as a location where high stress levels were exhibited. The results of the analysis can be observed below (see Figure 9.6).
The results of this simulation indicate that stress levels in the spokes are below the yield stress of titanium for all impacts except those at the 20mm offset, where the stress level exceeds 1 GPa by only 3.8%. Since this only exceeded 1 GPa by such a small amount and the design tested was not optimized for stress minimization, it was concluded that the design is feasible.

This output also presents an important trend for future analyses. It shows that the stress levels in the spokes increase linearly as the ball impact offset is increased. This signifies that future designs need only be designed to acceptable stress levels (below the yield...
stress of the spoke material) at 20mm offset impacts (or the max offset that the club will allow), allowing more efficient, less computation intensive analyses.

### 9.3 Summary

A more advanced model of the Tension Spoke design was created, including five layers of spokes and a supporting cage, and was analyzed at ball impacts at 0 to 20 mm offsets. The results of this analysis confirm that the design is feasible and warrants further investigation.
Chapter 10: Initial Prototype Plans

This section includes a presentation of a possible physical configuration of the Tension Spoke design. This is presented not as a final design, but in an effort to show how the design could be constructed physically.

10.1 Prototype Design

It is proposed that the spokes be made from very thin sheets of titanium due to its high yield strength to modulus of elasticity ratio (Wooley, 2003). The entire spoke structure, called a spoke ring is displayed below in Figure 10.1.

![Figure 10.1 Drawing of possible spoke ring configuration](image)

These spoke structures could then be stacked, with thin spacers in between them to keep them from rubbing, until the desired spoke cross sectional area was achieved. The
perimeter of the spoke structure could then be clamped by tightening bolts in the holes of the perimeter of the spoke structure. The floating face could then be attached similarly by inserting it through the center of the spoke ring and tightening it. Once clamped, this spoke “cartridge” could be used to validate the results obtained from the Abaqus output (see Figures 10.2 & 10.3 below for the entire Tension Spoke prototype assembly).

A similar cartridge design could be used for the actual golf club design. Refer to the figure below for possible Tension Spoke club configurations.

Figure 10.2 Expanded assembly view
Figure 10.3 View of assembled Tension Spoke cartridge.

Figure 10.4 Two different renderings of possible Tension Spoke club configurations
10.2 Summary

To demonstrate that the Tension Spoke design can be physically manufactured, initial prototype plans were presented. These plans should provide the basis for a functional prototype of the design that could be used to validate the results obtained from the computer simulations presented in previous chapters.
Chapter 11: Conclusions and Recommendations

Early simulations show that the Tension Spoke club design is not only feasible, but also that it has the potential for enhancing performance over current club designs. While there are design issues that still need to be overcome, early evidence suggests that the design warrants further investigation. Following is a description of the benefits and challenges that the Tension Spoke design faces at this point in its development.

11.1 Benefits

The following are a number of different benefits that the Tension Spoke design offers.

11.1.1 Completely New Mechanism/Design Space

Current club designs utilize essentially the same mechanism, a large thin clubface, to enhance the COR of the impact between the ball and club. Since the Tension Spoke design is not dependant on the face to store all the impact energy, it opens the door for a completely new design space and set of variables that engineers can optimize to enhance golf club performance.
11.1.2 Increased Active Face Area

As a result of the floating face mechanism utilized by the Tension Spoke design, theoretically the active face area can be as large as the entire clubface. This characteristic could be very attractive for lower skilled players that lack the ability to consistently hit the ball off the center of the face.

11.1.3 Increased COR at Higher Swing Speeds

Early results indicate that the Tension Spoke design reduces the attenuation of the COR of a club at higher swing speeds. This is particularly beneficial for highly skilled players, allowing them to achieve increased COR with PGA conforming clubs.

11.1.4 Strain Energy Stored in Tension

Previous attempts at utilizing tension to store the strain energy of the ball/club impact have failed. By orienting the tension elements orthogonal to the impact force, a feasible configuration was generated. Since tension utilizes the entire cross section of the material to store energy, it is a much more efficient use of mass that current designs that rely mainly on bending.

11.2 Challenges

The following are a number of different challenges that the Tension Spoke design presents at this point in its development.
11.2.1 Weight

With an optimal moving mass of around 25 grams, one of the major design issues will be constructing the floating face mechanism to be both light and extremely stiff.

11.2.2 Weight distribution: MOI consequences

Current club designs allocate about 50 grams to the clubface, which is distributed evenly across the face. Since the Tension Spoke design utilizes a floating face design, more weight will be located towards the center of the club. This redistribution of the weight will have the effect of lowering the rotational moment of inertia, allowing the club to rotate more upon off-center impact with the ball.

11.2.3 Prototyping/Manufacturing

Since this design differs greatly from current club designs, new manufacturing techniques will need to be employed to build the club.

11.2.4 Spoke Stress Levels

It is essential that each of the 100+ set of spokes be tensioned equally. If the spokes are not tensioned equally, unequal stress levels will form within the spokes upon impact, causing some spokes to be overstressed and break. This will be a key design/manufacturing issue that will need to be addressed.
11.3 Recommendations for Future Work

It is recommended that efforts be made to construct a physical prototype of the design to validate the results obtained from theoretical analyses. With this data, the Abaqus model could be improved to better predict the performance of the actual design, and work could begin on optimizing the configuration.
Appendix
Face Force Deflection Study

It was anticipated that there should exist some rate of face energy absorption upon impact with the ball that will provide optimal impact efficiency. In an effort to explore this idea, a number of different curves were analyzed using the closed form solution obtained from the lumped mass model. This section will present the results obtained from this analysis and will conclude with a discussion of the affect this had on the efficiency (COR) of the ball club impact.

A.1 Results

Six different nonlinear face configurations were analyzed. As was explained in the main body of this report, the characteristic force deflection equation for the face was defined as $F = k \times \alpha$, where “alpha” specified the shape of the force deflection curve. The analysis included six different curves with their corresponding alpha values which ranged from 0.5 – 3 varying in increments of 0.5. Six plots documenting the results obtained from the analysis are included on the following pages (see Figures A.1-A.6).
Figure A.1  Output results for lumped mass analysis with alpha=0.5. The legend displayed here for alpha=0.5 is valid for all six plots.

Figure A.2  Output results for lumped mass analysis with alpha=1. Refer to Figure A.1 for legend.
Figure A.3  Output results for lumped mass analysis with alpha=1.5. Refer to Figure A.1 for legend.

Figure A.4  Output results for lumped mass analysis with alpha=2. Refer to Figure A.1 for legend.
Figure A.5  Output results for lumped mass analysis with alpha=2.5. Refer to Figure A.1 for legend.

Figure A.6  Output results for lumped mass analysis with alpha=3. Refer to Figure A.1 for legend.
A.2 Discussion of Results

The plots of the results obtained from the lumped mass analysis of these different face curves show a number of different trends.

The first observation that can be made from the data is that there appears to be no “optimum” face deflection curve (within the range of curves that were analyzed) that results in extremely high COR values. It can be observed however that for smaller alpha values, the maximum COR region (within feasible design limits) is slightly higher in value than for the larger alpha configurations. One drawback to achieving the high COR designs with the smaller alpha values is that they require a small moving mass (<20g) to obtain these COR levels. This limitation on moving mass would make designing an actual physical model very difficult.

While there was no curve that resulted in extremely high COR values, there was an interesting trend that occurred as the alpha values of the force deflection equation were increased. The output plots clearly show that as the alpha values are increased, the high COR design region moves to include higher moving mass values and also becomes less sensitive to changes in face stiffness (k). If this trend is valid, it could allow the design of club configurations that have higher moving mass levels. In the case of floating face designs, this would be very beneficial in that it would allow enough moving mass to construct these heavier face designs.
References


