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Fundamentals of Machine Design and Manufacturing: Design of a Compliant Winding Machine

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FUNDAMENTALS OF MACHINE DESIGN AND MANUFACTURING:

Design of a Compliant Winding Machine

by

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(Master of Science in Mechanical Engineering, M.S.M.E)

A Thesis Submitted to the College of Engineering Department of Mechanical Engineering in Partial Fulfillment of the Requirements for the Degree of

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This thesis is dedicated to my parents who have given me the opportunity of an education from the best institutions and endless support throughout my life.

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NOMENCLATURE

CWM = Compliant Winding Machine
CAD = Computer-Aided Design
SE = Sparton Electronic
CAM = Computer-Aided Manufacturing
CAE = Computer-Aided Engineering
NC = Numerical Control
DNC = Direct Numerical Control
MIT = Massachusetts Institute of Technology
CATIA = Computer-Aided Three-Dimensional Interactive Application
HEEDS = and Hierarchical Evolutionary Engineering Design System
m = Mass
m_c = Mass of Carriage
m_c = Mass of Bearings
W_{SH} = Weight of Spool Housing (Dereeler)
W_S = Weight of Spool
A = Area
F = Force
W = Weight
CM = Center of Mass
K_f = Stress Concentration Factor, Fatigue
K_s = Shear Stress
P = Load
a = Acceleration
g = Gravitational Acceleration
n / FOS = Factor of Safety
t = Time
ν = Poisson's Ratio
V_f = Final Velocity
V_i = Initial Velocity
ρ = Density
E = Young's Modulus
σ = Normal Stress
τ = Shear Stress
F_a = Force due to acceleration
F_g = Force due to gravity
r = the displacement vector

ABSTRACT

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Title: Design of a Compliant Winding Machine

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Degree: Master of Science in Mechanical Engineering, M.S.M.E

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A Sparton Electronics Compliant Winding Machine has been redesigned for improved performance. Thorough study of the existing machine identified issues such as the inconsistent diameter and pitch of the compliant wind, outdated equipment, as well as safety and ergonomic issues. A comprehensive methodology for the design of the new machine is presented through using CAD systems and advanced analysis tools, such as Finite Element Analysis, and HEEDS. Resulting, the machine is successfully re-designed to fully control the diameter and the pitch of the product via servo motors and a control panel. The safety and ergonomic issues are effectively resolved by relocating the linear system, spool and spool housings to the bottom of the machine. The height of the machine is also adjusted following OSHA regulations. In conclusion, CWM is successfully designed and manufactured to meet the requirements and specifications of Sparton Electronics.

INTRODUCTION

Chapter 1 introduces the current machine specifications and processes as well as a short background on Sparton Electronics. It highlights the requirements for the project and details the problem statements and objectives. Chapter 2 covers an introduction to manufacturing engineering and crucial machine design fundamentals. Chapter 3 describes Compliant Winding Machine. It details the proposed design solutions considering the principles and fundamental knowledge from machine design and manufacturing engineering literature previously covered in Chapter 2. Finite Element Analysis is provided to support the design structure in Chapter 4. Chapter 4 also highlights the structural optimization of the design. Chapter 5 presents the design comparison of the early design phases and the final design of the machine. It also describes the results achieved and the advantages of the new Compliant Wind Machine. In addition, it highlights the need for further research and studies.

CHAPTER 1

PROJECT INTRODUCTION

1.1 Overview

The purpose of this thesis is to resolve current problems and concerns regarding the “Compliant Winding Machine”, CWM, at Sparton Electronics. The compliant winder produces a product called the “compliant wind”, which is assembled in the sonobuoy manufactured by Sparton Electronics. The compliant wind consists of two components as seen in Figure 1.1 below:

1. A cable/wire
2. A compliant bungee cord, a flexible rubber like material

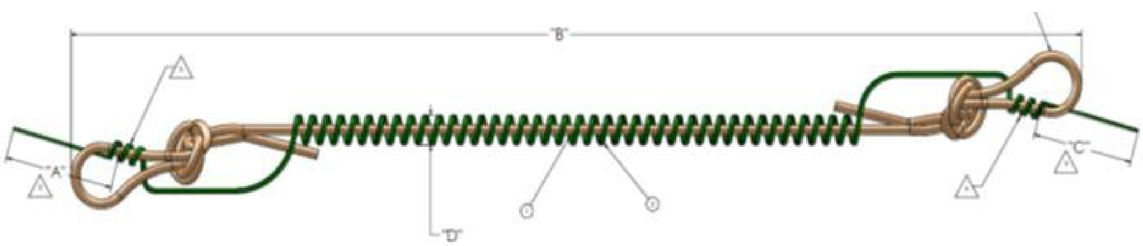


Figure 1.1: Compliant Wind Concept Shown with the Bungee (Rubber) Passing Through the (Green) Wound Compliant Cable

The spiral like shape of the cable is produced by the CWM. Once the cable takes its shape, the bungee cord is passed through to complete the product. The compliant wind is an essential component of the sonobuoy. The CWM machine’s task is to perform the “spiral” like cable wind that is eventually going to help deploy the buoy once in its place. However, the current CWM has several concerns that need to be addressed.

1.2 Current Machine Specifications and Manufacturing Process

The current CWM is 37 feet and 3 inches long, 3 feet wide and utilizes three independently controlled variable speed motor and brake combinations. All machine components are located on top of the table (workspace), and to the right of the work space. The current CWM has several mechanical components such as the compliant cable spool, winding arm, dog bone, motors, table, wind mechanism, track-and-trolley system, rotating arm, guiding rollers, and steel cable mandrel. All components play an important role in performing the cable wind. The machine has two operators producing and then assembling the complete product. The operator begins the process with pulling the cable from the spool and guiding it through the rotating arm that is attached to the spool. The cable is then tied to a “Dogbone” which is later attached to the trolley and track system.

Once attached, the rotating arm begins to perform the wind by forcing the compliant cable to wind around the steel mandrel. The headstock motor helps push the cable towards the tailstock motor, while the tailstock motors pulls the cable so the wind can be performed. Once the cable winding cable reaches the pre-determined length, the motor automatically stops and the cable snaps. The operator repeats the process by lifting the spool cover and pulling the compliant cable to tie to the Dogbone. The mandrel tension is controlled by a brake on the headstock motor. The rotation of the spool is controlled with an electric brake and the speed of the wind motor is being controlled via a discrete speed controller.

Although it performs a relatively simple process, the current machine has several weaknesses and concerns, which will be covered thoroughly in the following chapters with proposed solutions to the problems.

1.3 Company Overview

Sparton was founded over a century ago, and has been an industry leader since. It is known for its excellence in designing, developing and manufacturing some of the world's most complex devices [1]. Sparton Electronics has four main sonobuoy lines, all of which require the compliant wind. Currently, the company has four compliant winding machines to meet the production requirements.

1.4 Problem Statements and Objectives

One of the main issues provided by Sparton Electronics is the difficulty of controlling the diameter and pitch of the compliant wind. The proposed design will reduce both the fluctuation in the diameter and pitch of the product. The combination of three variables requires the operator to perform adjustment of each component individually to achieve the desired results. The wind travels with the help of the track and trolley system which is located on top of table and interferes with the operator's workspace. It is also a potential hazard to the wire as it may cut or nick the wire while transporting.

The existing design also raises many ergonomic issues such as the table height, the lifting of the spool housing cover as well as re-loading the spool. The operator constantly has to lift the heavy spool cover and replace spools weighing approximately 25 pounds above their waist. Sparton is concerned with this repetitive processes of lifting spool cover and replacing of the spools since it can cause serious shoulder and back injuries. In addition, the machine also consumes a large area of the floor due to its large size. In summary, the weaknesses of the current process are:

1. Issued with controlling the diameter and pitch of the wind.
    a. The pitch and diameter often fluctuate within a single wind.
    b. Individual adjustment required of the motors and the brakes which can be time consuming
2. The machine consumes a large amount of the floor space
3. The V-groove track and the trolley system clutter the work space and create potential for damaging the cable.
4. Excessive vibration of the spindle on the spool
5. Multiple Safety/Ergonomic issues

The overall objective is be to design a new CWM that will improve and/or reduce the current technical issues as well as comply with Occupations Safety and Health Administrations, OSHA guidelines.

CHAPTER 2

FUNDAMENTALS OF MACHINE DESIGN WITHIN MECHANICAL AND MANUFACTURING ENGINEERING

2.1 Background

When designing a machine or any of its components, there are many things to consider. However, there are some key concepts that must be kept in mind shown in Figure 2.1. The Design of Machine Elements presents the following list of fundamentals to be considered by V.B Bhandari [2]:

1. Goal or Purpose: An engineer must have the complete knowledge of the machine's components and processes.
2. Mechanisms: Selecting the group of mechanisms that will assist in providing the machine with the desired motion.
3. Load Analysis: After the concept design is completed, it must be analyzed with all the appropriate loads.
4. Material Selection: Selecting the right material can be tricky and requires special attention.

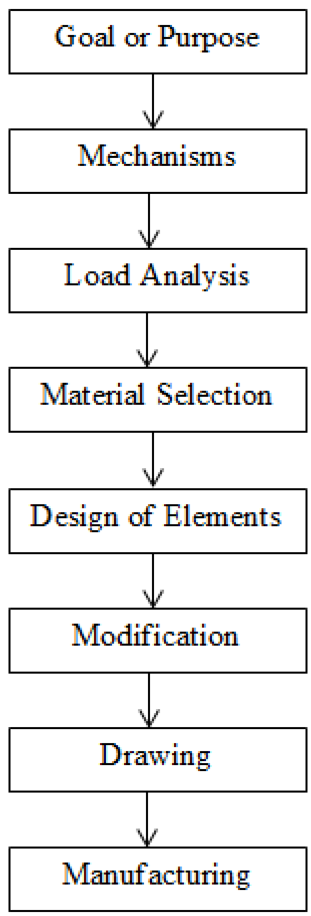


Figure 2.1: Design Considerations and Guidelines [2]

5. Design of Elements: A static and dynamic load case analysis must be conducted on each member by analyzing the forces acting on each element.
6. Modification: Once the design of elements is concluded, the elements may be modified for an optimized design.
7. Drawing: Detailed drawings of each component must be presented in order to turn the design from concept to manufacture.
8. Manufacturing: Once the design phase is successfully complete, the concept idea on paper is accepted for manufacturing.

A designer must reliably assess criteria used to characterize designs and their predicted performance. The Machine Design Data Handbook by Lingaiah and Iyengar lists the following basic decision making ingredients and their commonly practiced surrogates [3]:

Table 2.1: Basic Decision Making Ingredients [3]

Ingredient	Surrogate
Fact	Information
Knowledge	Advice
Experience	Ad hoc experimentation
Analysis	Intuition
Judgment	None

Many designers heavily rely on the surrogates listed in Table 2.1 rather than the basic ingredients these surrogates are derived from. By following the surrogates, the design is not reliable because it eludes the simplified and often crude decision-making replacements. In some situations where performance and economics are the requirements, "second hand" tools are no longer valid for selecting the best design concept.

In this thesis, therefore, it is required that producing valuable proofing by conducting scientific analysis is necessary. The Finite Element Method handbook details the benefits for considering such essentials [4]:

- Reproducible Results: Detailed calculations or finite element methods produce results that are easily reproducible at a later stage of the design. Intuition based decisions, on the other hand, are hard to re-derive by fellow peers.
- No ambiguity: Quantifying designs with meaningful numbers such as weight, stiffness, modal frequency, etc.
- Comparison: Fair comparison between very different designs.
- Analyzing Results: Results point out design challenges, therefore, it is important to analyze results for benefits and drawbacks.

Concept design selection must have a structured hierarchy whereby selection happens at different levels of details. It should be started with a coarse set of criteria which is subsequently refined until the most feasible and optimized design is identified.

2.2 Design Essentials

When designing a machine, it is important to first identify and understand the two categories of available machines - Fully automated and Semi Automated. Some companies require fully manual machines where others might require semi-automated or fully automated. It is common for manufacturing plants to have a mixture of all three – manual, semi-automated and/or fully automated machines [5].

This thesis will focus on semi-automated machine since the machine being designed will require some human assistance. Semi-automated and fully automated machines consist of three basic elements as per U.C Jindal's Machine Design textbook [6]:

a. Power – in order to operate and process the system
b. Instructions – a program of instructions is required to direct the process
c. Control System – to actuate the instructions

These basic elements are dependent on each other and are essential for the equipment to be productive in the final process. The relationship between the three elements is demonstrated in Figure 2.2.

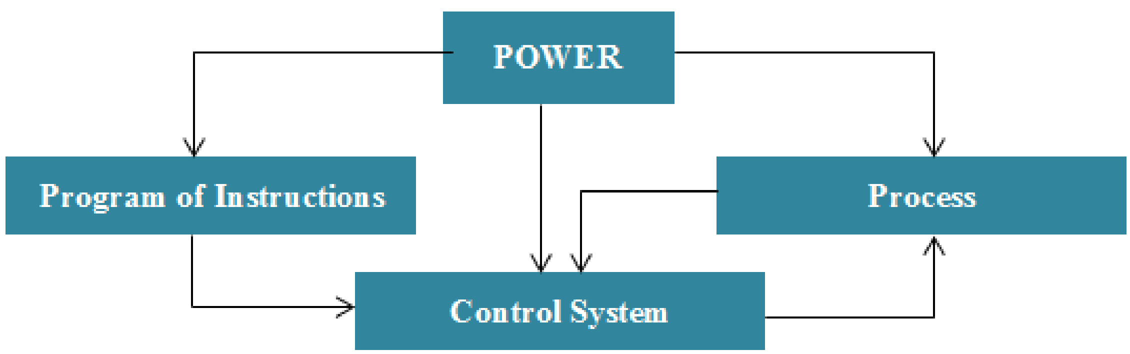


Figure 2.2: Basic Elements of an Automated Machine (Power, Program of Instructions, Control System) Resulting to the Final Process [6]

Above and beyond the basic elements of any manufacturing machines, there are a few common steps to be followed as part of the requirements. Following are some guidelines (not necessarily in the order to be applied), suggested by The Machine Design Handbook by Lingaiah and Iyengar [3]:

1. <u>Design Elements:</u> All design elements must be considered such as the machine geometry, applied forces, material, etc.
2. <u>Number of Machines Being Built:</u> The designing of the machine will sometimes depend on how many machines are being built.
3. <u>Lessons Learned:</u> Having past experiences and learning from the previous mistakes and lessons learned play a big role.
4. <u>Budget:</u> An impressive machine design is the one that helps get the finished product with all the major functionalities and highest possible quality at the lowest possible cost.
5. <u>Possible Mechanisms:</u> When designing the machine, the designer must consider all the possible mechanisms which will help the group of motions in the proposed design.
6. <u>Transmitted Forces:</u> The machine should be rigid enough so that under the effect of applied forces for which it is designed there is no deformation of the machine or machine elements beyond the specified limits.
7. <u>Material Selection:</u> It is important to thoroughly study material properties and characteristics before the material can be applied to the physical model.

8. Allowable Stress: If forces are being applied, it is obvious that the machine elements are subjected to stress. The stress and strain can be calculated by using the following equations [7]:

$$\text{Stress, } \sigma = \qquad (2.1)$$

Where,

$$\text{Force, } F = \text{Mass, m x acceleration, a} \qquad (2.2)$$

$$\text{Strain, } \varepsilon = \qquad (2.3)$$

Stress is proportional to load and strain is proportional to deformation expressed by Hooke’s law:

$$\text{Young's Modulus, } E = \qquad (2.4)$$

Young’s Modulus - often referred to as Modulus of Elasticity is commonly used for metals. It is important to consider all mechanisms, forces, and properties of material in order to achieve accurate results.

The machine elements and/or the machine should be strong enough to sustain all the forces it is designed for, so that it is not damaged or permanently deformed during its expected life time.

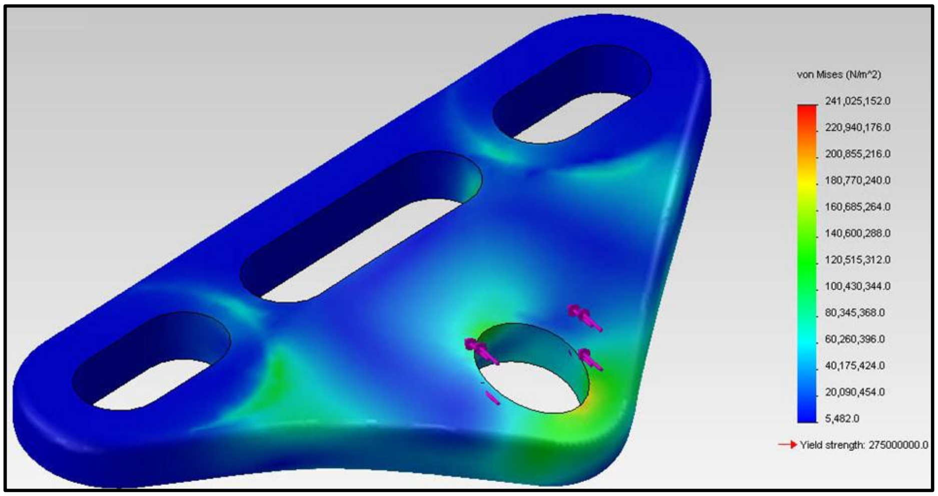


Figure 2.3: Results of a Finite Element Analysis Study on a Plate Showing the Maximum Stress and Deformation [7]

9. <u>Communication with the Customer:</u> To ensure that the final design meets the customer requirements, it is critical to work with the customer early in the development and track the requirements throughout the development cycle.

10. <u>High Output and Exceptional Efficiency:</u> Compact, affordable, and efficient is the general objective of any product.

11. <u>Durability</u>: The machine should be designed considering minimum maintenance and highest durability.

12. <u>Optimized Design:</u> The machine elements and machine should be strong, rigid and wear resistant with minimum weight. This can be achieved by selecting the appropriate kind of material and modifying the design parameters.

13. <u>Maintenance and Reliability:</u> Low maintenance and reliability are two important factors to be considered. The machine design should be simple enough, however,

reliable so that very little maintenance and servicing is needed. The cost of repairing the machine should be simple, affordable and quick. Use of standard parts is highly recommended.

14. Design for Manufacturing and Assembly: It is essential that all components or their sub-assemblies fit into each other to make the final assembly. Ease of assembling and disassembling the final product must also be considered for the sake of servicing and maintenance.

15. Safety: Safety rules are set by Occupational Safety and Health Administration, OSHA for almost all manufacturing sites in the U.S. For the safety of the operator, all hazards must be considered and be completely removed or at least be minimized.

16. Ergonomics: The machine and its elements should be easy to handle and access. While designing the machine the aesthetics and ergonomics of the machine should be given consideration without affecting its functionality.

17. Remarkable Performance: For the proper economic performance of the machine, appropriate mechanical, hydraulic, electrical, thermodynamics, and other principles should be applied.

By considering the mentioned guidelines, the engineer can greatly increase his/her design's efficiency. The following flowchart (Figure 2.4) represents the procedure and guidelines at a glance.

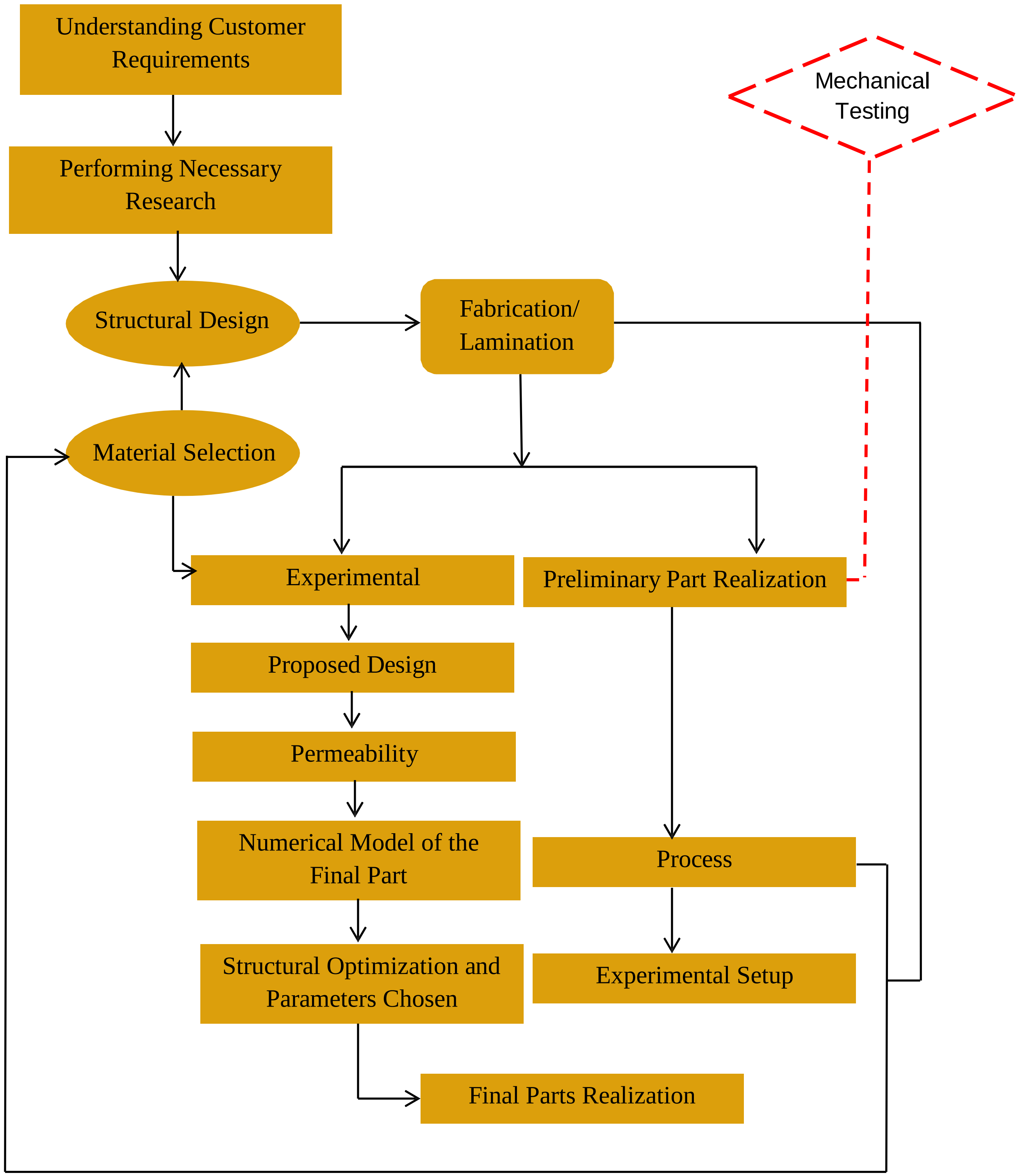


Figure 2.4: A Flowchart Representing the Design Process for Machine Design [3]

Using standard procedures such as the flowchart shown in Figure 2.5, to define the functionality of your device helps analyze the design thoroughly. The flowchart forms the basis one might require to frame the algorithm design [3].

2.3 Machine Design Challenges

Powerful competition is putting pressure on design engineers to deliver systems with higher output, reduced operating cost, and increased safety. Design engineers have switched from rigid, single-purpose machines relying purely on mechanical gears and cams to flexible multipurpose machines by adopting modern control systems and servomotors [8]. Although these improvements have made machines flexible, they have also introduced a significant amount of complexity to the machines and subsequently to the machine design process.

The technology has improved so much today and provided us with the resources such as computer aided design, CAD. A design defined in CAD software with dimensions, notes, and materials specified is much more powerful than a traditional drawing board sketch. It implies a comprehensive, well-organized design and determination of the engineer. It also helps visualize the final product to both the provider and the receiver.

2.4 Material Selection Process

When selecting a material, there are several things to be considered such as the material's physical and chemical properties, cost, manufacturing abilities and availability. Material properties determine what the material is capable of handling. Manufacturing abilities focus on what method can or will be used to make the part. Material availability deals with how accessible the material is. It is always best to have multiple resources to access and buy the material when needed [9].

2.5 Finite Element Analysis

It is important to consider all forces on all elements of the equipment during machine design. One way to do this is through Finite Element Analysis, FEA. Complicated stress analysis can be done comparatively quickly with the help of software rather than the traditional paper and pencil methods [10]. Finite element codes are a lot less complicated than many other programs. The Automation, Production Systems, and Computer Integrated Manufacturing textbook by Groover M. divides the analysis into three sections: (a) *Pre- processing*, (b) *Analysis*, (c) *Post Processing* [11].

(a) *Pre-processing:* The user begins with constructing the model he/she desires to be analyzed. The geometry created must be separated and divided into elements connected at nodes, discrete points, model and eventually analysis.

(b) Analysis: The model prepared is translated into data to be used for the analysis. It creates a code for itself which constructs and solves a system of linear/non-linear algebraic equations as:

$$K_{\mathrm{ij}} u_{\mathrm{j}} = f_{\mathrm{i}} \qquad (2.5)$$

where, u and f are the displacements and externally applied forces at discrete points otherwise known as nodal points. The formation of K matrix depends on the problem itself and what exactly is being solved.

(c) Post-processing: Final results of the problems are typically shown as colored contours representing stress levels on the model [11].

To produce meaningful FEA results, the design engineer must know the principles underlying the finite-element method. The engineer must also have the practical experience, a feel for design and sound engineering judgment [12]. The software relies on the information proved by the user; therefore, if the information provided to the system is incorrect, reliable results will not be achieved.

2.6 Factor of Safety, F.O.S

To have a feasible design, it is important to calculate its Factor of Safety, FOS, which determines the product's productivity. To decide upon the allowable tension or shear stresses acting on a material, the factor of safety must be considered, which can be calculated by using the below equation for brittle materials [13]:

Factor of Safety, n = (2.7)

For ductile materials:

$$n = \underline{\hspace{6cm}} \tag{2.8}$$

To calculate the design load, the following equation may be used:

$$\text{Design Load, } F_d = \text{Factor of Safety, } n \text{ x Working Load, } F \tag{2.10}$$

The engineer must take each component of an assembly, work out the loads, pressure, acceleration, bending moment etc. acting on each component. It is crucial that the appropriate material is selected to manufacture the product so that the FOS falls within the specified requirements. Within metals, it is important to select the right category of material as well. Furthermore, strength is a statistically determined quantity, because a material is never purely standardized and the mechanical properties vary slightly from sample to sample. To establish the minimum strength of a material, an engineer is advised to test the material experimentally.

2.7 Computer-Aided Design, CAD, Computer-Aided Manufacturing, CAM and Computer-Aided Engineering, CAE

With the rapidly growing technology being used in modern products, the engineers are making products compact with a much higher efficiency factor than ever before. With the help of the computer-aided design, CAD software engineers are able to design components within minutes. The model may be also be modified to the user's desires as many times without any costs.

Modeling with CAD software offers many advantages, however, also a number of disadvantages. Some positives include the ease of modifying a design without going back to the beginning. They can also be rotated on any axis enabling the design engineer to have the insight of the part/product. To design an effective part or a product, it is important to understand what CAD is unable to do. CAD systems function by their ability to classify geometrical concepts [14].

2.8 Structural Optimization through Computer Software

The basic idea behind optimization is to find the best way to design a product or a structure which can benefit the engineer or the designer from the available resources. The minimum requirements for a proficient structural design are that the response of the structure should be acceptable as per various specifications, that is, the design must be feasible. Most often, there will be many feasible designs, but it is imperative to choose the best design [22]. The best design may be in terms of minimum cost, mass, factor of safety, maximum performance, etc.

Structural Analysis is characteristically done by conducting Finite Element Analysis, FEA on the product geometries generated by Computer Aided Design, CAD software. The availability of affordable high performance computers and commercial Finite Element Software further promotes the integration of structural analyses with numerical optimization, which is, Structural Optimization reducing the need of the design by iterations manually. For this thesis, the objective is to minimize mass by minimizing the thickness of the carriage plates.

CHAPTER 3

THE COMPLIANT WINDING MACHINE DESIGN DEVELOPMENT

3.1 Proposed (New) Design Solution

The overall purpose of the new design was to design a machine while minimizing and/or eliminating the current issues. The new approved design utilizes two servo motors programmed to perform the mechanical tasks attached to a linear system. The servo motors are controlled through a touch screen system connected to one end of the machine. The linear system consists of linear guide profiles that are mounted on the bottom of the machine for safety and to increase the work space. The smaller of the two motors is located towards the bottom and under the top table connected to a structure called the “winding carriage”. The second motor is located on top of the table inside the “winding box”. The winding box and the winding carriage are connected through two stepped rods and travel simultaneously.

The proposed design offers the flexibility of winding two cables at faster speed rates as well as eliminated the need for two operators. The cable winds around the bungee rather than a steel cable mandrel, so the footprint of the process is also reduced. Currently, Sparton uses 4 CWMs to meet the production demand; with the new design the customer will be able to meet the production demand with 2 machines.

The current machine has a track-and-trolley system along with the 25 pound spool, spool housing and the motor that are located on the top of the table. Due to their current location, these machine components interfere with the operator’s work space. To reduce this issue, the track and the trolley system is eliminated by introducing a Roller

Pinion System RPS. The spools are re-located in dereelers (spool housings) located on top of the winding carriage and under the table for the ease of the operator. By having the spools on the bottom, the operator will have an open work space. By introducing the open spool housings, the operator does not have to repetitively lift and close the cover.

The roller and pinion system replaces the track and the trolley system and is located under the table. By moving the relatively large components under the table, the work space is opened up. For the safety of the operator, the machine will be closed off from all sides with the help of 80/20 framing material. The side panels/doors will automatically lock once the machine is in operation. It is also suggested to have an automatic switch that will enable the machine to shut off automatically if the any of the side panels are opened.

The smaller winding mechanisms are confined in a small winding box located on top of the table. The proposed design solution also introduces a much smaller rotating arm that will reduce the vibration. By implementing the proposed design, the objective is to have a dynamic “Compliant Winding Machine” that will minimize maintenance and down time. In addition, there are numerous other benefits to the proposed design, which are detailed throughout the thesis.

3.2 Part Description

The compliant winding machine has several assemblies which may be divided into four sub-assemblies:

1. The Winding Carriage

2. The Base / Frame
3. The Winding Box
4. The Control Box (With Touchscreen Panel)

These sub-assemblies contain several additional sub-assemblies within themselves. The use and interaction of these sub-assemblies within each other is the reasoning behind the new and efficient designed compliant winder. The components of each sub-assembly can be seen in *Table 3.1 and in Figure 3.2).*

Table 3.1: Showing Assemblies, Sub-Assemblies of the Compliant Winder Machine, CWM

Compliant Winding Carriage			
The Winding Carriage	**The Base/Frame**	**The Winding Box**	**The Control Box (With a Touchscreen Panel)**
A servo motor	Precision roller pinion and track	A servo motor	Control Box
A gear reducer	Side window panels	Toothed pulley and belt	Touchscreen panel
Base structure	Rectangular tubing	Shaft to drive the pulleys	Feedback controller
Dereeler Tensioner Stand	80/20 frame	Supporting pillow blocks	Motor cables
Spool	Leveling feet	Rotating winding arm	Adapters
Bearings	Swivel castors	Dogbone	
Drawer Slide Guides	Linear profile rail and bearings	Bearings	
	Table/Workspace		

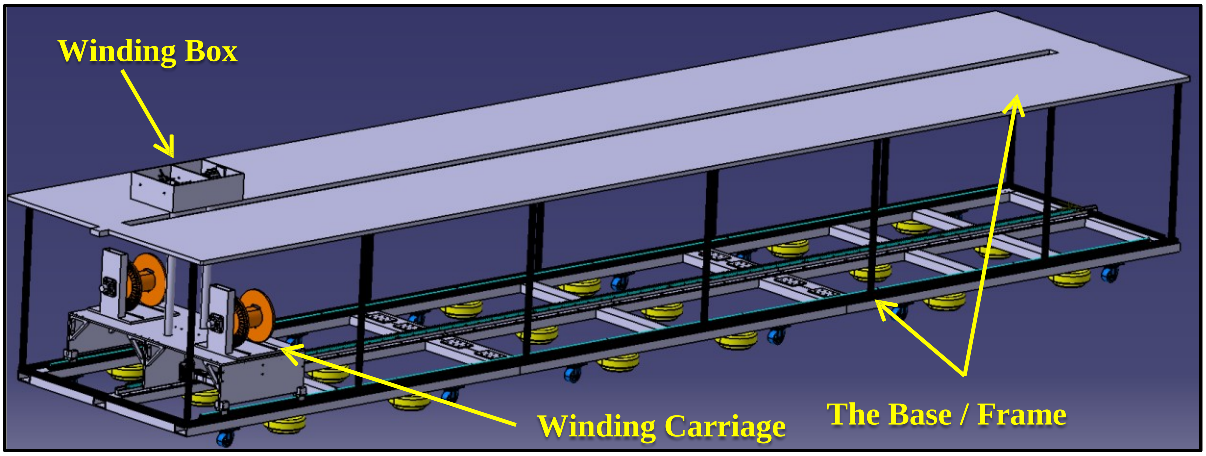


Figure 3.1: Showing Assemblies, Sub-Assemblies of the Compliant Winder Machine, CWM

**Note that the control box is not shown since this thesis will not be covering the details on this component. The side window panels are also not shown.*

3.2.1 The Winding Carriage

The winding carriage consists of several parts (Shown in Table 3.1 and Figure 3.1) and provides the linear motion the winding box needs in order to perform the wind. It is made out of four aluminum plates that complete its basic structure (Figure 3.2). The plates are assembled together with multiple bolts and supporting brackets. The use of aluminum brackets to connect all four plates rather than welding was due to the rigidness they were able to provide. Bolted brackets also offer the flexibility of disassembling the carriage easily in case of maintenance or any future design modifications. The entire structure (plates and the supporting brackets) is made out of 6061-T6 aluminum due to its physical and mechanical properties.

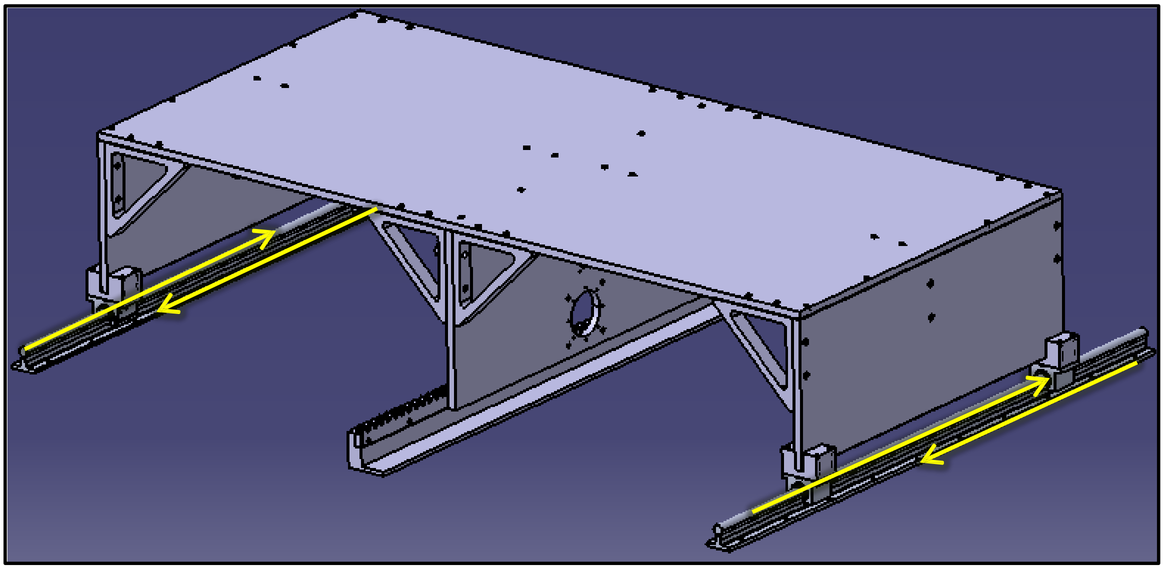


Figure 3.2: Isometric View of the Base Structure of the Winding Carriage Shown Without the Spool and Spool Dereelers on Top; The arrows indicate the linear motion it performs.

Selecting a material that is easily accessible provides Sparton Electronics the flexibility of purchasing new material in case the plates require any modifications or maintenance. The selected aluminum alloy has a density of 0.098 lb/in^3 and the ultimate strength of 300 MPa or 42,000 psi. The finite element analysis on the carriage with 6061-T6 aluminum is performed in the next section to assure that the material will not deform or fail under the given loading conditions.

The weight of the carriage is distributed on the linear ball bush bearings and linear profile rails while the motor, gear head and the roller pinion is rested on the roller pinion rack. To secure the carriage on top of the bearings, small aluminum blocks were placed to connect the bearings and the plates together, since the bearing did not provide enough support to hold the plates vertical.

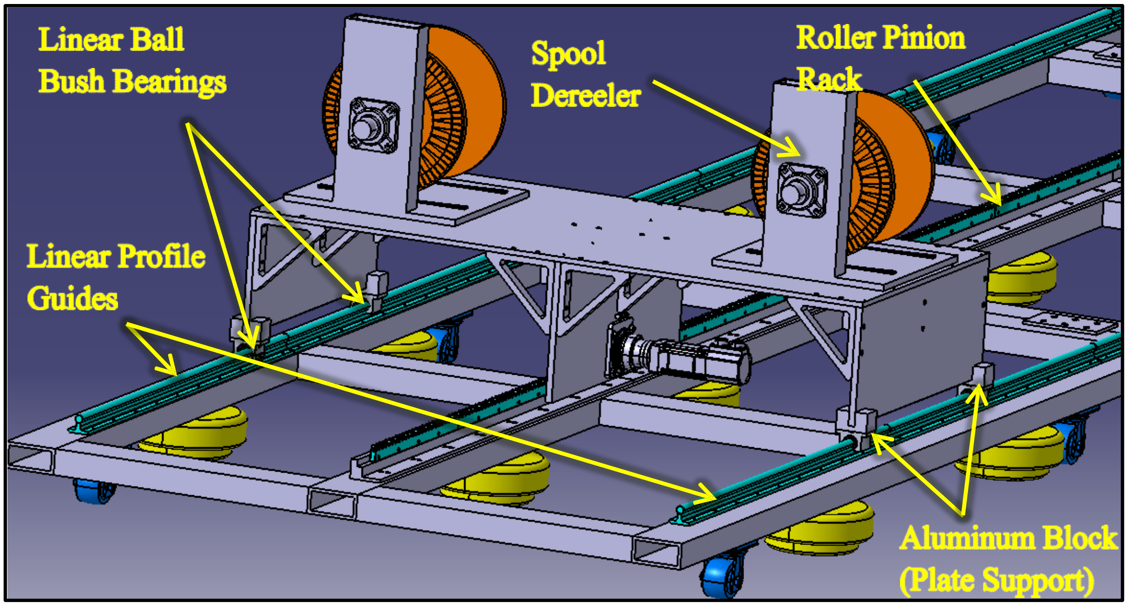


Figure 3.3: Isometric View of the Base Structure with the Linear Guide System of the Winding Carriage

The main purpose behind the designing the new machine with the wind carriage was to relocate all components of the machine away from the work space. Currently, the workspace is cluttered by the spool, motors, track-trolley system as well as other small components. The new design uses dereelers, also referred to as spool housing (Figure 3.4) that prevent the operator from lifting the spool cover repeatedly. As mentioned in Section 2.2 of this thesis, it is often best to select standard and readily available parts than incorporate customized designs. Hence, once the spool housing design was finalized, online research assisted in finding a similar

Figure 3.4: Broomfield Dereelers Tensioners – DT Series, DT-124 Model Heavy Wire Tensioners [15]

structure. The researched spool housing that complies with the CWM design requirement is a Broomfield Dereelers Tensioners - DT Series, DT-124 Model: Heavy Wire Tensioners. The DT-124 model has an adjustable air pressure regulator with manual shut off switch which allows freewheeling of the spool and instant return to the preset tension.

This dereeler supports spool diameters of up to 24 inches and load capacity of 450 pounds [23], whereas the spool used in the CWM is 11.50 inches in diameter and weighs 25 pounds at its maximum. This ensures that the selected component will safely carry the load of the spool. The Broomfield DT series wire dereeling and tensioning is designed and constructed to provide years of low maintenance and trouble-free service [15].

The spool dereelers are located at 37.5 inches from the bottom, and have a versatile "sliding drawer" mechanism, which offers the operator the flexibility of easily sliding out the housing in and out as many times without risking any bodily injuries.

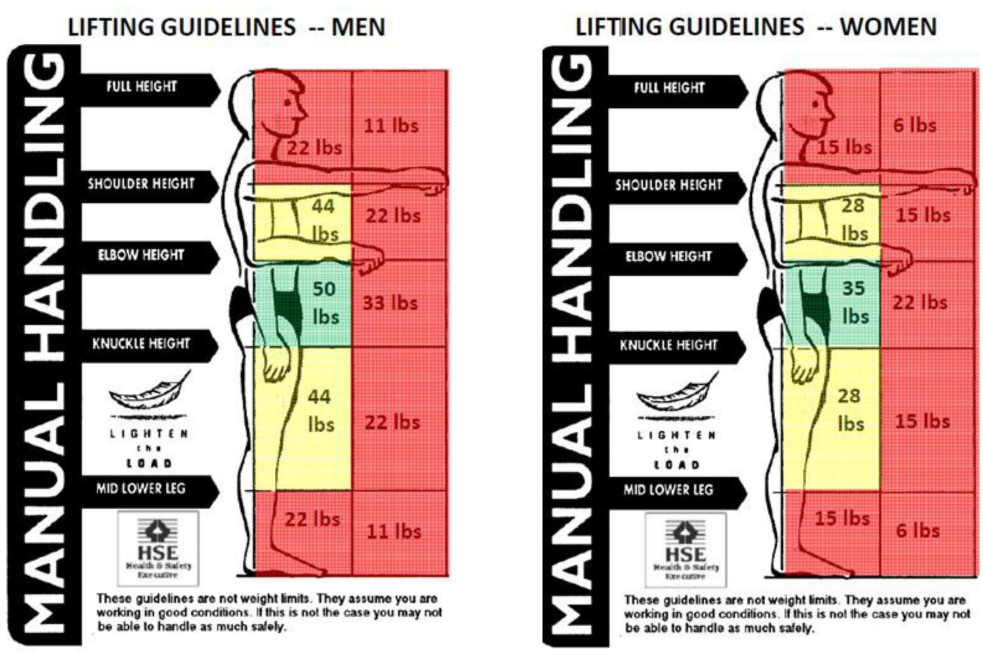


Figure 3.5: Lifting Guidelines Provided by Occupational Health and Safety Administration, OSHA [16]

Figure 3.5 shows a range of accepted weights that can be lifted by an operator without hurting his/her body. The calculated average of weights ranges between 22 pounds to 33 pounds to be lifted parallel to thighs for both men and women. The spool is located parallel to thighs and weighs 25 pounds, which falls between the ranges given by OSHA [16].

The location of the spool housing makes it easy for the operator to replace the spool without injuring his/her back. The operator can pull out the housing to load the spool, and easily push it back in, where it will automatically lock into its place. The mechanism used is an "Accuride Extra Heavy Duty, Full Extension Drawer Slide". The selected Model 9301 as shown in Figure 3.6, with 14 inch slide length and travel is a self-locking adaptable deign. The current mass of a dereeler with a full spool is approximately 115 pounds. When the slides are mounted flat (as in this case), the specified load rating as per the product's specifications is 150 pounds [17] each, which is well above the requirements of this design.

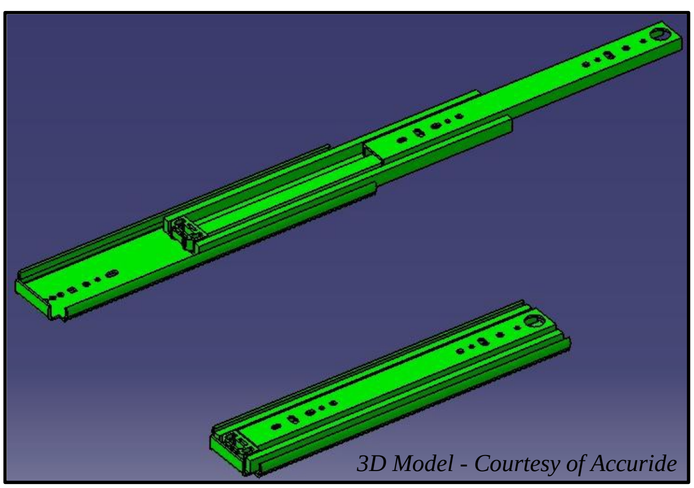


Figure 3.6: 3D Model of Accuride Model 9301 - Extra Heavy Duty, Full Extension Drawer Slide: Fully Extended Slide to 28 inches (Top); Fully Closed 14 inches (Bottom) [17].

By incorporating the wind carriage, the work space has opened up enormously. The new design has two compact and efficient servo motors that weigh less than 12 pounds when combined. Servo motors are known for their small size and exceptional performance. The selected motors were recommended by Sparton electronics. However, a thorough analysis and calculations were performed to ensure that the motors will operate and provide the necessary torque. The torque required to move the winding carriage was calculated considering the following information provided by the operator.

- Travel Time, t = 179.5 seconds
- Current Machine Length, L_{CM} = 33 feet and 3 inches

We know that,

$$\text{Velocity, } v = \frac{\qquad}{\qquad} \tag{3.1}$$

Hence,

$$\text{Average Velocity, } v_{ave} = \frac{\qquad}{\qquad} \text{ or } = \frac{\qquad}{\qquad}$$

$$= 2.22 \text{ in/sec or } 0.185 \text{ ft/sec}$$

Average Acceleration, a_{ave} of the current machine:

$$\text{Average Acceleration, } a_{ave} = \frac{\quad}{\quad} \tag{3.2}$$

$$a_{ave} = \frac{\qquad}{\qquad} = 0.0010 \text{ ft/sec}^2$$

The performed calculations give the average velocity that the machine would require to perform the wind. Using the current average velocity, v_{ave} = 2.22 in/sec or 0.185 ft/sec, the time can be calculated for the new machine. It is known that the new machine is 24 feet in length, including one foot of space on each side. The functional length of the new machine is 22 feet. Hence, substituting values, we get the following:

$$x = 264 \text{ inches or } 22 \text{ feet}$$

$$v_{ave} = 2.22 \text{ in/sec or } 0.185 \text{ ft/sec (For the new CWM)}$$

Using E. 3.1;

$$t_{ave} = \frac{}{} = \frac{}{} \text{ or } \frac{}{}$$

$$t_{ave} = 118.91 \text{ seconds}$$

Note that the cycle time is reduced from 179.5 to 118.91 seconds.

Average velocity of the machine is 0.185 ft/sec, and it takes 118.91 seconds to reach from point A to B. To better understand this, the velocity and time may be drawn as such (Figure 3.7):

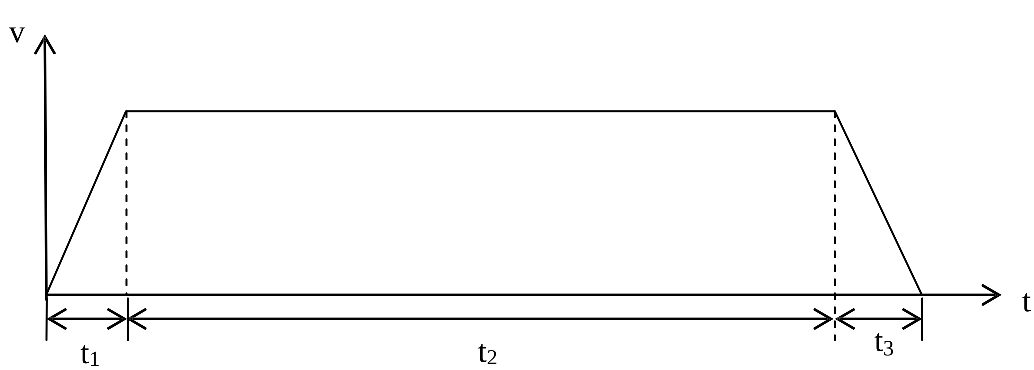


Figure 3.7: Velocity, v based on Time, t

Where,

t_1 = time it takes to accelerate

t_2 = constant acceleration

t_3= time it takes to de-accelerate

To calculate the acceleration, the distance for the machine to accelerate and decelerate must be identified. The distance to accelerate and decelerate is assumed to be 1 foot and 5 inches. Figure 3.8 below shows the velocity based on distance:

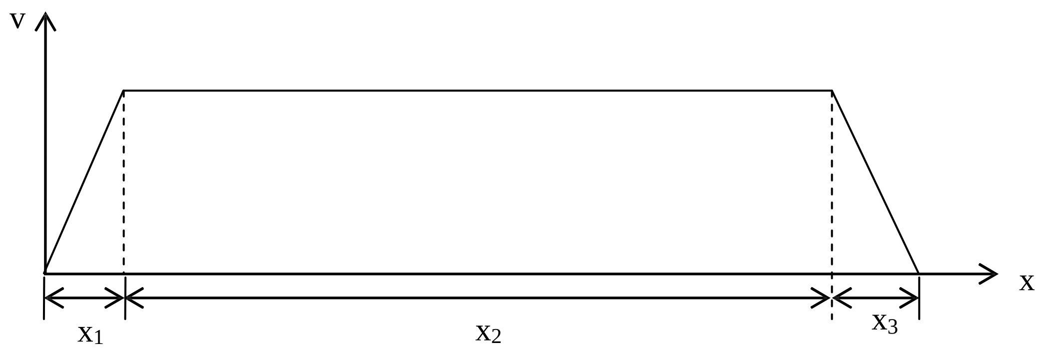


Figure 3.8: Velocity, v based on Distance, d

Where,

x_1 = distance to accelerate = 1.50 ft

x_2 = distance during constant acceleration = 22- (1.50) = 20.5 ft

x_3 = distance to de-accelerate = 1.50 ft

Thus, calculating time based on v_{ave} and distance, t_1, t_2, and t_3 can be calculated as:

$$t_1 = t_3 = \frac{}{} = \frac{}{} = 8.10 \text{ seconds}$$

$$t_2 = \frac{}{} = \frac{}{} = 110.80 \text{ seconds}$$

The acceleration, a can be calculated using Equation 3.3 below:

$$d = v_i t + \frac{}{} a t^2 \qquad (3.3)$$

$$1.5 \text{ ft} = 0 + \frac{}{} a\,(8.10)^2$$

$$a = 0.045 \text{ ft/s}^2$$

Since the acceleration and decelerate times are the same, the deceleration rate is - 0.045 ft/s^2

To calculate the torque required in order to move the carriage, the force was first calculated by performing the following calculations:

Load acting on the Carriage:

Weight of spool, W_S = 20-25 lbs.; 2 spools = 40-50 lbs.

Weight of spool housing, W_{SH} = 90 x 2 = 180 lbs.

Nuts & Bolts Weight, W_D = 10 lbs.

Total Weight, $W_T = W_S + W_{SH} + W_D$

$$= 50 + 180 + 10 \text{ (lbs.)}$$

$W_T = 240$ lbs.

Another factor to consider is the weight of the carriage, W_C, which is equal to 100 pounds. Considering a factor of safety, the total W_T is considered to be 350 lbs. It is known that $W = mg$, hence, to calculate Mass of the Carriage, m_c, Equation 3.4 was considered:

$$\text{Mass, } m = \qquad (3.4)$$

$$m_c =$$

$$m_c = 10.87\text{slugs}$$

It is known that the motor weighs 4 pounds, and approximately 15 pounds with the gearbox and self-aligning bracket. So, the Mass of the Motor, m_m can be calculated as:

$$m_m =$$

$$m_m = 0.466 \text{ slugs}$$

$$\text{Total Mass, } m = m_c + m_m$$

$$= 10.84 \text{ slugs} + 0.466 \text{ slugs}$$

$$m = 11.30 \text{ slugs}$$

The machine operates with four linear profile bearings as well as a Roller Pinion System, RPS system which carry the weight of the entire carriage assembly, including the spool and spool housing (dereeler). (Figure 3.9)

By Newton's second law, the force, F acting on a body is

$$F = ma \tag{3.5}$$

Since, the carriage weight and the bearing act as opposing forces, Equation 3.6 below may be used to conclude F_X:

$$\sum F_X = F_{CARRIAGE} + \sum \mu F_{BEARING} = 0 \tag{3.6}$$

$$\sum F_X = ma + \mu N = 0$$

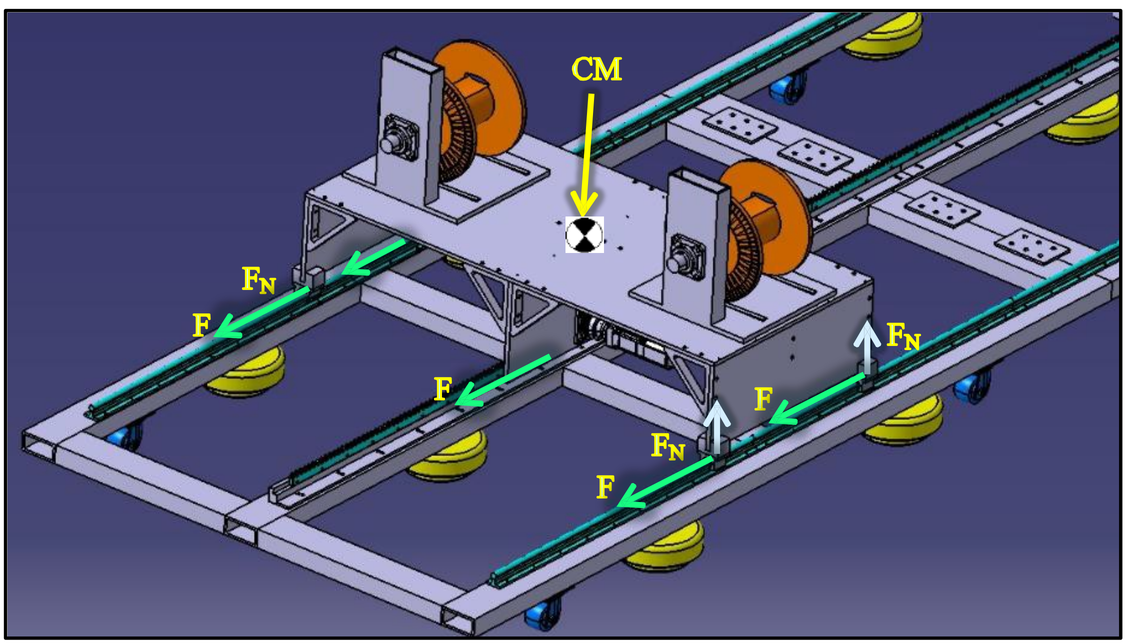


Figure 3.9: Applied Loads and Acting Forces Shown on the Wind Carriage, where F = Horizontal Force, FN = Normal Force and CM = Center of Mass

While, the force being applied by the bearings in the y-direction is:

$$\sum F_Y = 0; F_A + F_B - W = 0 \tag{3.7}$$

To perform further calculation, the friction of coefficient of the bearings must be known. The friction of coefficient of the pillow block bearings may be gathered from Figure 3.10. Since, the machine is using self-aligning ball bearings; the coefficient of friction ranges from 0.0008 to 0.0012 [18]. The average of the given minimum and maximum is used to calculate the coefficient of friction: 0.0010.

Type of bearing	Friction coefficient μ
Deep groove ball bearing	0.0010 to 0.0015
Angular contact ball bearing	0.0012 to 0.0020
Self-aligning ball bearing	0.0008 to 0.0012
Cylindrical roller bearing	0.0008 to 0.0012
Full complement type needle roller bearing	0.0025 to 0.0035
Needle roller and cage assembly	0.0020 to 0.0030
Tapered roller bearing	0.0017 to 0.0025
Spherical roller bearing	0.0020 to 0.0025
Thrust ball bearing	0.0010 to 0.0015
Spherical thrust roller bearing	0.0020 to 0.0025
<Ref> Plain bearing	0.01 to 0.02

Figure 3.10: Coefficient of Friction for Bearings [18]

Solving the force in x-direction using 3.6: the maximum acceleration can be calculated:

$$\sum F_X = ma + \mu N$$

Where,

$$N = mg = 11.30 \text{ slugs x } 32.174 \text{ ft/s}^2$$

$$N = 365 \text{ lbf}$$

$$\sum F_X = (11.30 \text{ slugs})\ (0.045 \text{ ft/ s}^2) + (0.0010)\ (365 \text{ lbf})$$

$$\sum F_X = 0.508 + 0.365$$

$$F_X = 0.873 \text{ lbf}$$

Combining the calculated results, the torque required from the servo motor can be calculated given the equation below:

$$\tau = Fr \tag{3.8}$$

Where,

τ = torque

F = force

r = the displacement vector

Since the motor is attached to the RPS system, the vector of displacement in this case is the radius of the pinion roller. The radius of the roller pinion is 1.36 inches.

Hence,

$$\tau = 0.873 \text{ lbf x } 1.36 \text{ inches}$$

$$\tau = 1.19 \text{ lb-in (or 1.2 Approximately)}$$

The motor specifications provided by Emerson show that the motor is able to provide 12.5 lb-in or 1.4 lb-ft, which is below the required torque [21]. The reader must note that if more torque is ever required by the customer, the attached gear reducer has a ratio of 25:1, which is able to increase the output by 25 times. From the chart provided by APEX (Figure 3.11), the given data shows AD064 nominal output of torque to be 60 Nm or 531.04 lb-in [20].

Model No.		Stage	Ratio[1]	AD047	AD064	AD090	AD110	AD140	AD200	AD255
Nominal Output Torque T_{2N}	Nm	1	4	19	48	130	270	560	1,100	1,700
			5	22	60	160	330	650	1,200	2,000
			7	19	50	140	300	550	1,100	1,800
			10	14	40	100	230	450	900	1,500
		2	20	19	48	130	270	560	1,100	1,700
			25	22	60	160	330	650	1,200	2,000
			35	19	50	140	300	550	1,100	1,800
			40	19	48	130	270	560	1,100	1,700
			50	22	60	160	330	650	1,200	2,000
			70	19	50	140	300	550	1,100	1,800
			100	14	40	100	230	450	900	1,500
			16	19	48	130	270	560	1,100	1,700
			21	22	60	160	330	650	1,200	2,000
			31	19	50	140	300	550	1,100	1,800
			61	19	50	140	300	550	1,100	1,800
			91	14	40	100	230	450	900	1,500
Emergency Stop Torque T_{2NOT}[2]	Nm	1,2	4~100	3 times of Nominal Output Torque						
Nominal Input Speed n_{1N}	rpm	1,2	4~100	5,000	5,000	4,000	4,000	3,000	3,000	2,000
Max. Input Speed n_{1B}	rpm	1,2	4~100	10,000	10,000	8,000	8,000	6,000	6,000	4,000
Micro Backlash P0	arcmin	1	4~10	-	-	≤1	≤1	≤1	≤1	≤1
		2	20~100	-	-	-	≤3	≤3	≤3	≤3
Reduced Backlash P1	arcmin	1	4~10	≤3	≤3	≤3	≤3	≤3	≤3	≤3
		2	20~100	≤5	≤5	≤5	≤5	≤5	≤5	≤5
Standard Backlash P2	arcmin	1	4~10	≤5	≤5	≤5	≤5	≤5	≤5	≤5
		2	20~100	≤7	≤7	≤7	≤7	≤7	≤7	≤7
Torsional Rigidity	Nm/arcmin	1,2	4~100	7	13	31	82	151	440	1,006
Max. Bending moment M_{2KB}[3]	Nm	1,2	4~100	42.5	125	235	430	1,300	3,064	5,900
Max. Axial Load F_{2aB}[3]	N	1,2	4~100	990	1,050	2,850	2,990	10,590	16,660	29,430
Service Life	hr	1,2	4~100	30,000*						
Efficiency η	%	1	4~10	≥97%						
		2	20~100	≥94%						
Weight	kg	1	4~10	0.7	1.2	3.0	5.6	11.9	31.6	56.1
		2	20~100	1.0	1.6	3.7	7.3	15.9	36.9	70.4
			16~91	1.0	1.4	3.5	6.5	15.5	34.2	67.2
Operating Temp	°C	1,2	4~100	-10°C~90°C						
Lubrication				Synthetic lubrication oils						
Degree of Gearbox Protection		1,2	4~100	IP65						
Mounting Position		1,2	4~100	all directions						
Noise Level(n_1=3000rpm, No Load)	dB(A)	1,2	4~100	≤56	≤58	≤60	≤63	≤65	≤67	≤70

Figure 3.11: APEX 064 P2 Gearbox Performance Specifications [19]

The servo motor recommended by Sparton is an Emerson Control Techniques NTE-212-CONS-0000 (As shown in Figure 3.12). Figure 3.13 shows the two servo motors being used in the concept design.

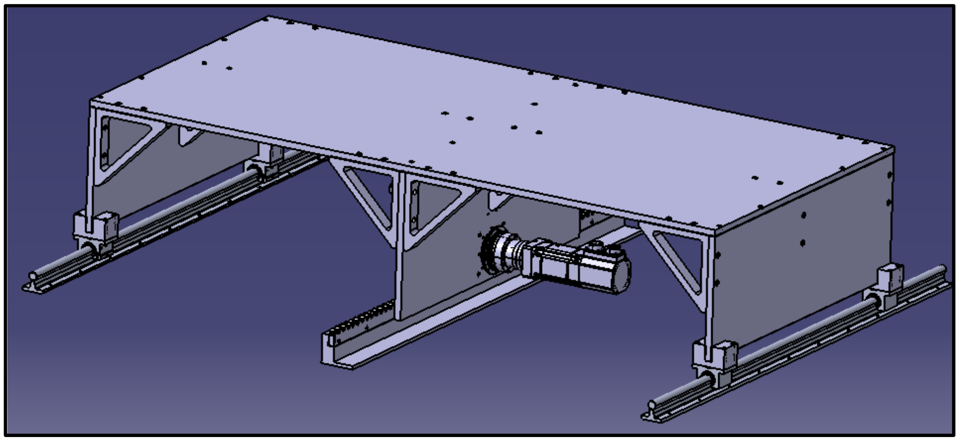


Figure 3.12: Isometric View of the Base Structure with the Emerson NTE-212-CONS-0000 Servo Motor and an APEX Gear Reducer on the Winding Carriage

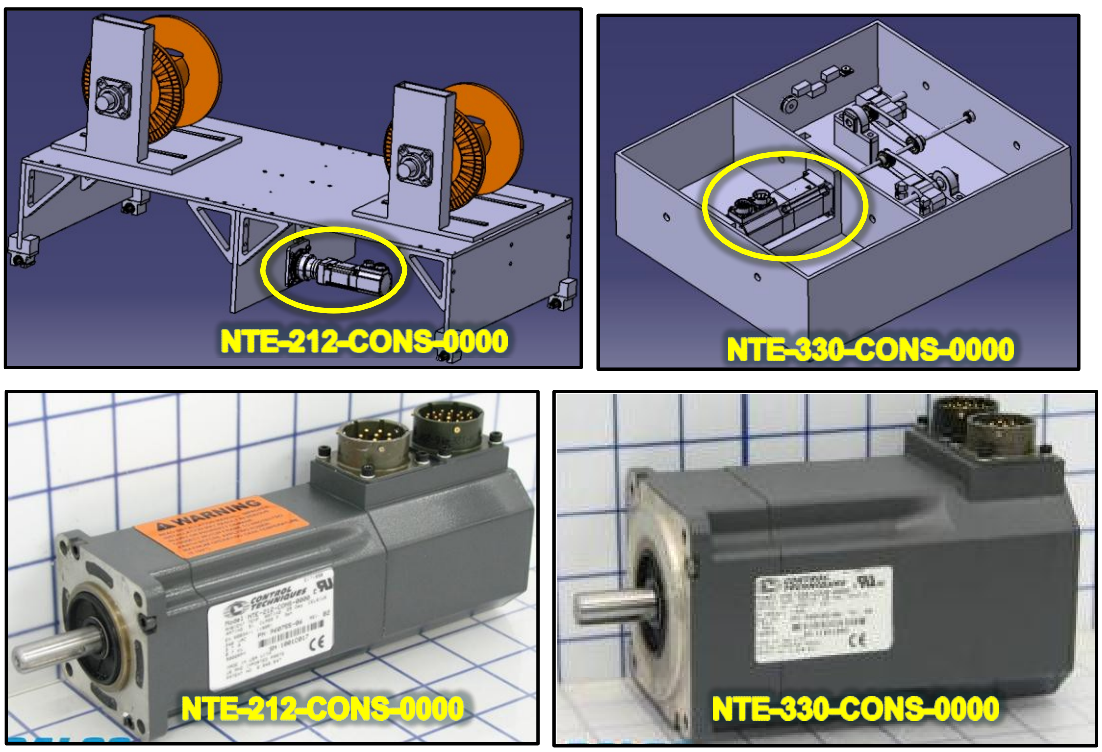


Figure 3.13: NTE-212-CONS-0000 [21] Shown on the Wind Carriage (Left Top) with the Actual Motor Shown on Bottom Left and NTE-330-CONS-0000 [20] Shown on the Winding Box (Top Right) and Actual Motor Shown on the Bottom Right

Given the design parameters, the velocity, acceleration, forces acting and the torque was calculated in this section. The calculations were performed to ensure that the final design will fulfill the customer specifications. The winding carriage and the base of the machine are the driving factors of the CWM and depend on the each other. The winding carriage works with the base/frame section of the machine. The following section will explain the base/frame and its components.

3.2.2 The Base/Frame

The base of the machine consists of several components as shown in Table 3.1, however, the functioning and the most essential components are the linear profile rails with (4) pillow blocks, and the roller-pinion system. The use of modern system of roller – pinion system along with a linear profile guide system eliminates the need for the track-and-trolley mechanism. The roller-pinion system used in this thesis (Figure 3.14) is a Nexen Group product. Nexen offers an advanced technology that revolutionizes linear and rotary motion control. The Roller Pinion System, RPS is an innovative pinion consisting of bearing-supported rollers and a unique tooth profile. The use of rollers supported by the bearing gives a smooth motion along the face of each tooth on the rack as shown in Figure 3.14 [22].

The RPS tooth design is conceptually different from traditional gearing. It is similar to a cam and follower system. Teeth on the rack are designed so that the roller is always in contact with two or more teeth. This makes the system roller backlash-free in both directions. [22]. The RPS offers unlimited length, hence perfect for the 24 foot complaint wind machine. The variations shown in Figure 3.15 represent minor errors occurring throughout the pinion's travel.

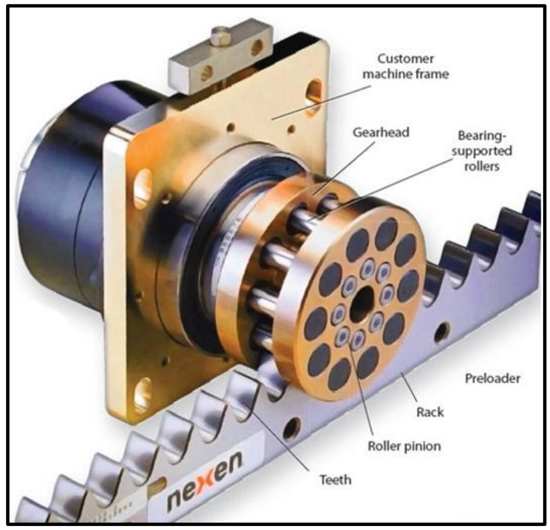


Figure 3.14: A Roller-Pinion System, RPS with Its Components Shown from Nexen Group Used in the New Compliant Wind Machine [22]

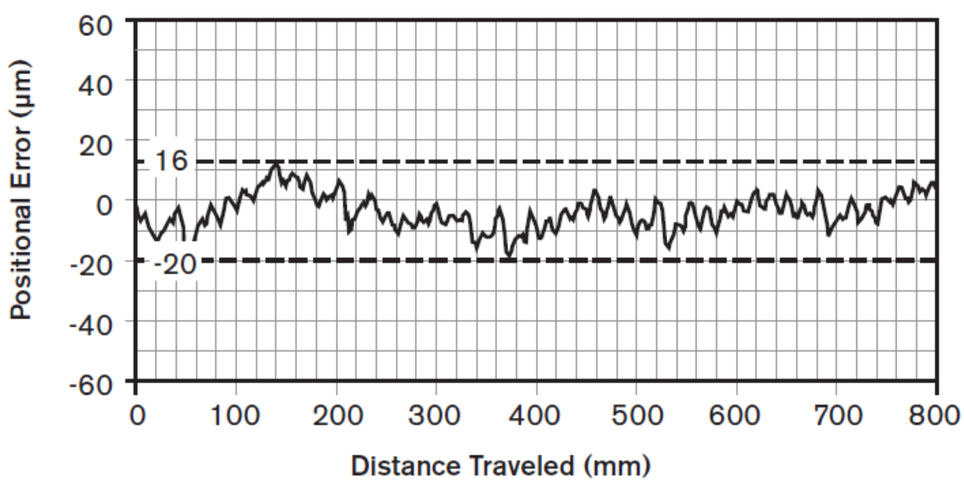


Figure 3.15: Regardless of the Distance Traveled, Positional Accuracy Remains Constant With the RPS System [22]

The advantages of the RPS system may be seen it a glance in Figure 3.16 below [30]:

INDUSTRY PROBLEMS	Ball Screws	Traditional Rack/Gear & Pinion Systems	Belt Drives	Chain Drives	Linear Motors Direct Rotary Stages Direct Drive Motors	nexen ROLLER PINION SYSTEMS
Low Accuracy			⬢	⬢		High Positional Accuracy
Backlash / Vibrations	⬢	⬢	⬢	⬢		Near-Zero Backlash
High Cost	⬢	⬢			⬢	Economical, Efficient Components
Dirty Operation	⬢	⬢	⬢	⬢		No Dust Emissions
High Maintenance	⬢	⬢		⬢	⬢	Little to No Maintenance
Low Load Capacity			⬢		⬢	High Load Capacity
Noisy	⬢	⬢	⬢	⬢		Quiet: pinion rollers glide smoothly along teeth
Low Speed	⬢	⬢				High Speeds (up to 11 m/sec)
Magnetic Field					⬢	No magnetic field
High Wear/ Low Life	⬢	⬢	⬢	⬢		Long Life (up to 36 million meters)
Limited System Length/Size	⬢		⬢	⬢		Custom Rack Sizes & Modular Components

Figure 3.16: Advantages that Overcome Common Problems Found in Traditional Drive Systems [22]

The following data in Table 3.2 is provided by Nexen Group which helped determine the proper RPS model size [22]:

Table 3.2: Typical Friction Coefficients and Shock Factor to Determine the Required Data [22]

Typical Friction Coefficients (μ)

Profile Guide Rail	0.005
Ball Bearing Guide Rail	0.02
Polymer Bushing Guide	0.1
Bronze Bushing Guide	0.2

Shock Factor[3] (K)

Shockless Smooth Operation	1.0
Normal Operation	1.2
Operation with Impact	1.5
Operation with High Impact	2.5

From the calculations performed previously, Compliant Wind Machine requirements are as follows:

Table 3.3: Specifications Required to Determine the Proper RPS Model Size [22]

Required Data for RPS Selection		Data
Weight to be Driven (W)	lb	350 lbs
Maximum Velocity (v)	ft/s	0.185 ft/s
Acceleration Time (t) or Known Acceleration (preferred)	seconds ft/s^2	0.047 ft/s^2
Shock Factor (K) See table above		1.2
Other Forces (F_1), (F_2) etc.	N	0 N
Frictional Coefficient (μ)		0.005
Angle from Horizontal (θ°)	°	0 °
Travel Distance	ft	22 ft
Cycles Per Day		140

Where,

Weight, W = weight to be driven

Velocity, v = maximum velocity

Time, t = acceleration time (Figure 3.17)

Shock Factor, K = smoothness of operation

Other Forces, F_1, F_2 etc. = forces such as cutting forces, springs, counter balances, fluid dampening, wind resistance etc.

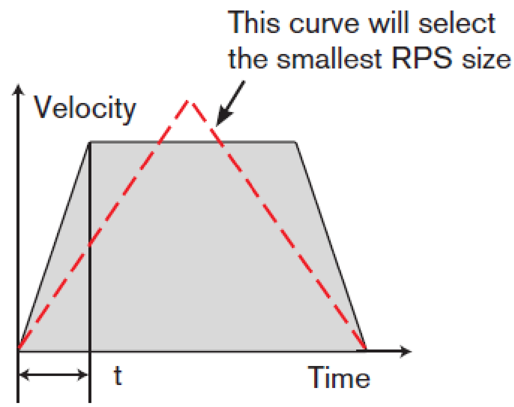


Figure 3.17: Acceleration Based on Time (t) [22]

Cycles per Day = cycles per day assuming going the full travel distance and returning to home each time

The calculated results were then entered in the Nexen Group’s online system, which automatically generates the proper roller pinion size. The results received from Nexen’s online system recommended the RPS16.

The specifications of RPS 16 are shown in Figure 3.18:

Roller Pinion Size	Number of Rollers	Max RPM	Max Torque [1] (Nm) Dynamic / Static	Distance per Revolution (mm)	Pitch Circle Diameter (mm)	Product Number	Series	Base Material/ Coating	Mount Style	Bore Size (mm)	Mass (kg)	Moment of Inertia kgm^2x10^{-4}
16	10	1500	61.1 / 61.1	160	50.9	966687	B	Nickel	Flange	N/A	0.8	4.0
						[illegible]	B	Nickel	Shaft	20	0.7	[illegible]
						966759	B	Stainless	Flange	N/A	0.8	4.0
						966761	B	Stainless	Shaft	20	0.7	3.9
						966715	C	Nickel	Flange	N/A	0.9	4.2
						966659	C	Nickel	Shaft	20	0.8	4.12

Figure 3.18: The Selected Roller Pinion Recommended by Nexen Group’s Online System [22]

Since, RPS 16 is divided into several categories; product number 966759 was selected to fit the needs of this machine. The selected model was first narrowed down to material (stainless steel) and then the mounting style. Flange mount was selected conforms to ISO 9409 specifications. Once the rack and the roller pinion were selected, a Roller Pinion Gearhead, RPG had to be selected to complete the RPS package. The RPG was selected by referring to the chart provided by Nexen (Figure 3.19).

Pinion Size	Adapter w/ Pinion (not required in some applications)	Pinion Preloader	Customer Provided Gearhead						
			Alpha/ Wittenstein	APEX	Mijno	Neugart	SEW-Euro	Sumitomo	Stöber
RPS16	N/A	N/A	N/A	AD047	N/A	N/A	N/A	N/A	N/A
RPS20	RPS16 & 966688	960851	TP004	AD064	BDB 085	PLFE/N 64	PSBF221/2	N/A	PH/A/KX 321/2
RPS25	RPS20 & 966676	960850	TP010	AD090	BDB 120	PLFE/N 90	PSBF321/2	PNFX080	PH/A/KX 421/2
RPS32	RPS25 & 966674	960852	TP025	AD110	BDB 145	PLFE/N 110	PSBF521/2	PNFX250	PH/A/KX 521/2
RPS40	RPS32 & 966668	960853	TP050	AD140	BDB 180	PLFN 40	PSBF621/2	PNFX450	PH/A/KX 721/2
RPS4014	RPS40 & 966698	960854	TP110	AD200	BDB 250	PLFN 200	PSBF721/2	N/A	PH/A/KX 821/2
N/A	RPS4014 & 966701	N/A	TP300	AD255	BDB 300	N/A	N/A	N/A	PH/A/KX 912/23

Figure 3.19: Gearhead Compatibility Chart [22]

The preloader has high precision ground surfaces and an adjuster allows the pinion to be moved up or down into the rack while keeping the pinion properly oriented to the rack. Figure 3.20 details the RPS and RPG components.

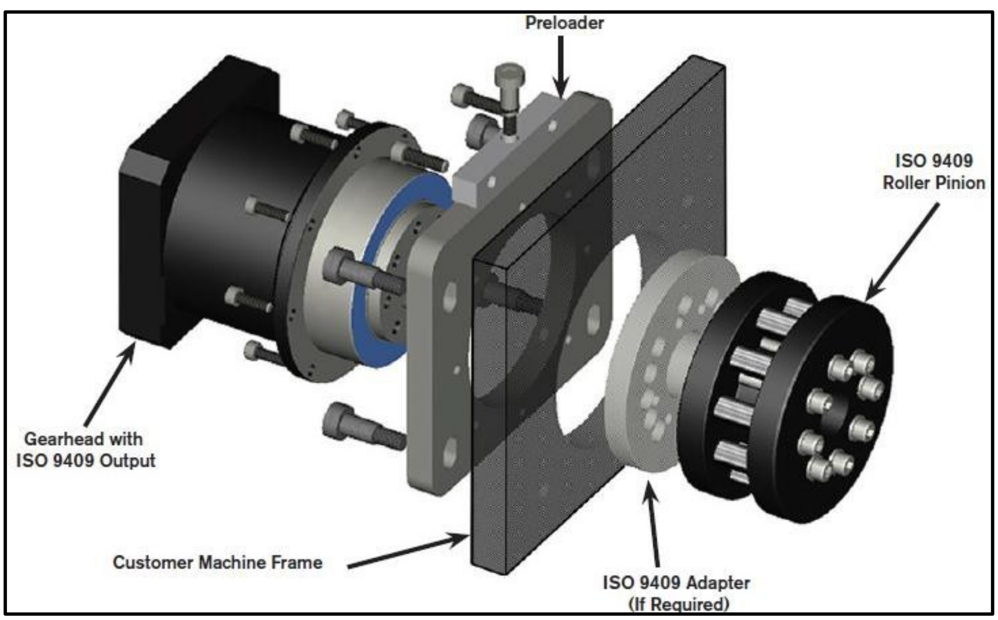


Figure 3.20: Apex Dynamics AD064-0025-P2 Series Servo Motor Reducer with Roller Pinion [22]

The Roller-Pinion System on the machine gives the winding carriage the linear motion it requires in order to produce the compliant wind. The speed is adjusted by the computerized interface. The servo motor is connected to the carriage with a gear reducer as mentioned previously. A gear reducer is a mechanism that transmits the motor's rotational speed at a higher torque. The reducer used in this application is an Apex AD Series – AD064-0025-P2 that has 25:1 ratio [21] (Figure 3.21).

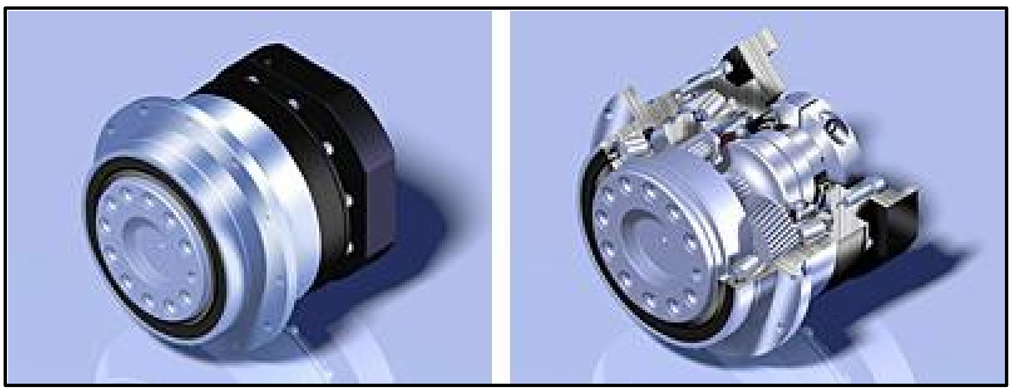


Figure 3.21: Apex Dynamics AD064-0025-P2 Series Servo Motor Reducer [21]

The sides of the carriage rest on the "RoundRail Linear Profile" guides with four ball bush bearings. The Thomson Linear RoundRail Profile was selected due to their many advantages. These linear profile guides are often used in applications where smooth, accurate and linear motion is essential. When selecting the linear guides and bearings, a few considerations were taken into account such as:

- Load/Life
- Smoothness of Travel
- Installation Process
- Travel Accuracy
- Speed & Acceleration
- Maintenance

• Rigidity • Environment • Availability

Thompson offers a variety of linear guides and bearing pairs. The two major types of linear guides are round rail bushing bearings and profile rail bearings. In order to apply the correct type of linear guide and bearing pair in this particular application, it was important to consider the advantages and capabilities of each. A key advantage of round rail ball bushing bearing systems is their ability to accommodate torsional misalignment caused by inaccuracies in carriage or base machining or machine deflection with little increase in stress to the bearing components. The self-aligning-in-all-directions design of round rail bearings is forgiving of poor parallelism and variations in rail height. As a result, these bearings allow for smooth travel [23]. Table 3.4 below outlines typical performance characteristics of round and profile rails and where one is superior.

The selected linear guides and bearings are the Super Smart Ball Bushing Pillow Blocks with the 60 Casing Shafting (Figure 3.22). Another key feature of the Super Ball Bushing Bearing is its ability to “roll”. The bearing plate is designed with the radius of its outer surface smaller than the inside radius of the precisions outer ring (See Figure 3.21). This key feature allows the bearing plate to compensate for torsional misalignment and evenly distribute the load on each of its two ball tracks. The component assures maximum load capacity and travel life.

The linear pillow block bearings used in CWM are 0.625 in bore and 1.750 wide (See Figure 3.24) since it fulfills the load requirement of 350 pounds. To ensure safety and zero failure, 2 bearings were used on each of the edge of the side plates (4 bearings total). Using four bearings provided a total load capacity of 2480 pounds (620 lbs. X 4).

Table 3.4: Typical Performance Characteristics of Round and Square Profile Rails, Also Where One is Superior [23]

Attribute	Round	Square
Performance		
Load Capacity	medium	**high**
Accuracy	medium	**medium - high**
Stiffness	medium	**high**
Available preload	yes (light)	**yes (light - heavy)**
Single guide supports moments	no	**yes**
Same load capacity in all directions	no	**available (typical)**
Seal effectiveness	**high**	medium
Smoothness	**high**	medium - high
Drag	**low**	medium - high
Total Cost of Ownership		
Ease of installation	**high**	medium
Required precision of mating component	**low**	high
Self aligning	**yes**	no
Life Expectancy	medium	**high**
Material Cost	**low**	medium
Design Flexibility		
Ability to span gaps	**yes**	no
Can be used as structural frame member	**yes**	no
Ability to use as single rail & guide	no	**yes[1]**
Compactness (load capacity to size)	medium	**high**
Ease of modification/Interchangeability	**high**	low
Rail Mounting		
Available end support mounting	**yes (preferred)**	no
Available continuous support mounting	yes	**yes (preferred)**
Available mounting from top of rail	yes	**yes (preferred)**
Available mounting from bottom of rail	**yes**	**yes**

1. Depending on the application. Most square rail applications use dual rails.

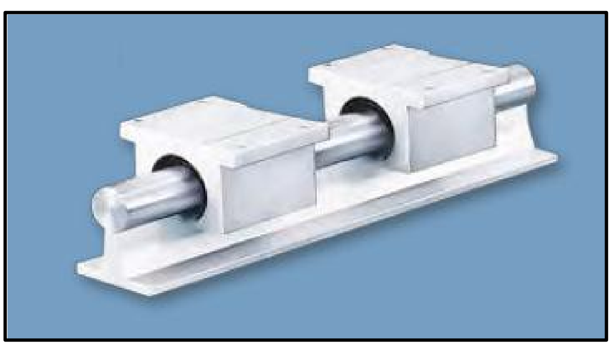


Figure 3.22: Thomson RoundRail Linear Guides Shown With Super Smart Ball Bushing Bearings [23]

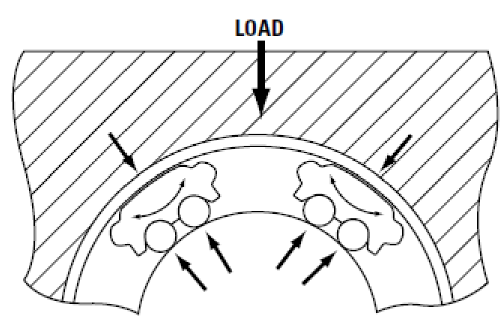
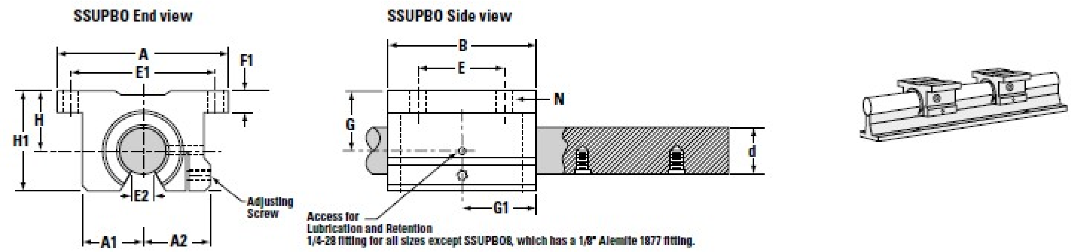


Figure 3.23: Close-up of Double Track Bearings Showing How They Self-Align (roll) to Evenly Distribute the Load on Each of Their Two Ball Tracks [23]

The super smart ball bush bearing technology provides twice the load capacity of the standard ball bush bearing and six times the load capacity of the conventional linear bearings. They also provide eight times the travel life of the standard ball bearing and two hundred and sixteen times the travel life of the conventional linear bearings. Figure 3.25 shows the load comparison and the life comparison between standard, conventional and super smart ball bushing bearings.

The selected round bearings offer a self-aligning capability of up to 0.5^{o} compensating for inaccuracies in base flatness or carriage machining. The RoundRail combined with the self-aligning feature eliminated the need for derating factors commonly required for linear guides. It also offers travel speeds of up to 10 ft/sec without a reduction in load capacity. Light-weight, wear-resistant and engineered polymer retainers and outer sleeves assist in reducing inertia and noise [23].



Super Smart Ball Bushing Pillow Blocks (Open Type) and 60 Case LinearRace (Dimensions in inches)

Part Number (3)		Nominal Diameter	H ±.003	H1	60 Case LinearRace Diameter d
Super Smart Ball Bushing Pillow Block	60 Case LinearRace*				
SSUPBO8	1/2 L PD	.500	.687	1.13	.4995/.4990
SSUPBO10	5/8 L PD	.625	.875	1.44	.6245/.6240
SSUPBO12	3/4 L PD	.750	.937	1.56	.7495/.7490
SSUPBO16	1 L PD	1.000	1.187	2.00	.9995/.9990
SSUPBO20	1 1/4 L PD	1.250	1.500	2.56	1.2495/1.2490
SSUPBO24	1 1/2 L PD	1.500	1.750	2.94	1.4994/1.4989

* 60 Case begins on page 170.

Part Number (3) Super Smart Ball Bushing Pillow Block	A	A1	A2	B	E ±.010	E1 ±.010	E2 min.	F1	G	G1	N Hole	N1 Bolt	Pillow Block Mass lb	Dynamic (††) Load Capacity lb_f
SSUPBO8	2.00	.69	.75	1.50	1.000	1.688	.31	.25	.50	.89	.16	#6	.23	360
SSUPBO10	2.50	.88	.94	1.75	1.125	2.125	.34	.28	.55	.95	.19	#8	.41	620
SSUPBO12	2.75	.94	1.00	1.88	1.250	2.375	.41	.31	.67	1.08	.19	#8	.51	1130
SSUPBO16	3.25	1.19	1.25	2.63	1.750	2.875	.53	.38	.87	1.45	.22	#10	1.03	1900
SSUPBO20	4.00	1.50	1.63	3.38	2.000	3.500	.62	.44	1.15	1.83	.22	#10	2.15	2350
SSUPBO24	4.75	1.75	1.88	3.75	2.500	4.125	.74	.50	1.28	2.02	.28	.25	3.29	3880

(††) The Dynamic Load Capacity is based on a rated travel life of 2 million inches. The actual Dynamic Load Capacity can be affected by the orientation of the bearing or the direction of the applied load. For dynamic load correction factors see polar graphs below.
(3) For part number description and specifications see page 36.

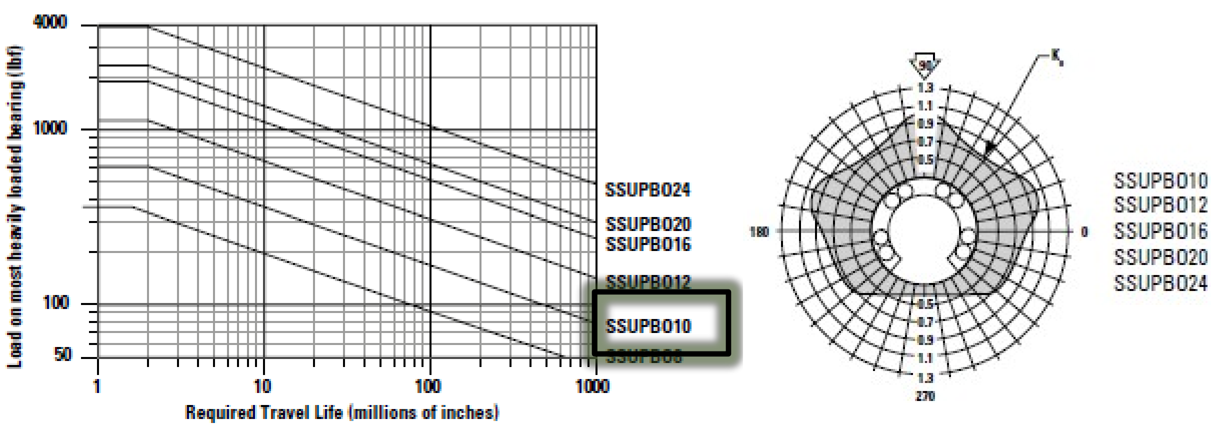


Figure 3.24: Available Super Smart Ball Bushing Pillow Blocks Sizes for Continuously Supported Applications (Top); Load/Life and Polar Graph with Lines Indicating Limiting Load for Given Ball Bushing Bearing (Bottom) [23]

The selected round bearings offer a self-aligning capability of up to 0.5^{o} compensating for inaccuracies in base flatness or carriage machining. The RoundRail combined with the self-aligning feature eliminated the need for derating factors commonly required for linear guides. Travel speeds of up to 10 ft/sec without a reduction in load capacity. Light-weight, wear-resistant and engineered polymer retainers and outer sleeves assist in reducing inertia and noise [23].

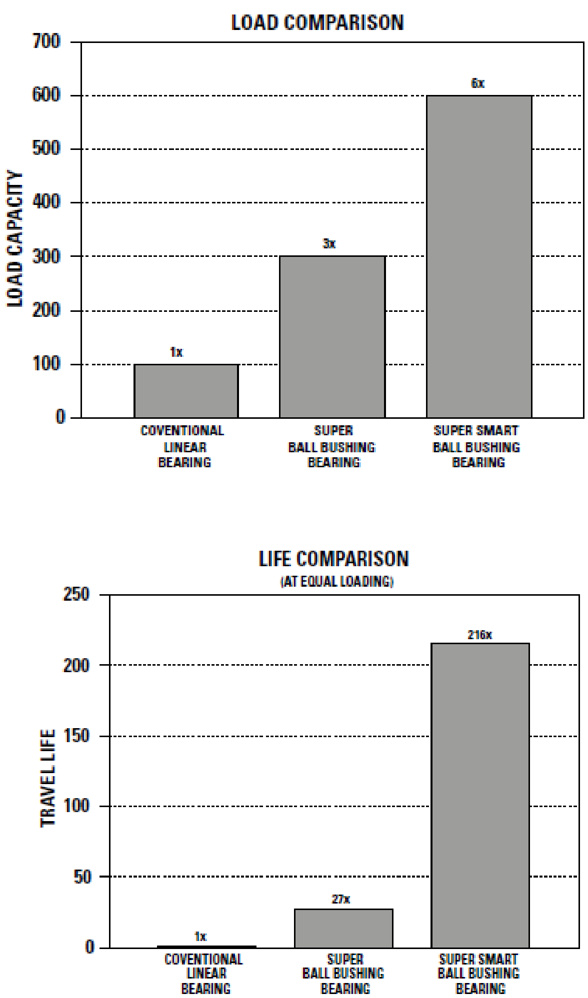


Figure 3.25: Comparing the Load and Travel Life between the Conventional, Standard and Super Smart Ball Bushing Bearings [23]

The linear profile is bolted to the base/frame of the machine, which is structured with simple 6 X 4 inches steel rectangular tubing with a wall thickness of 0.25 inches. The selected tubing gives the machine a rigid structure. The base is to be made in 8 foot sections to provide flexibility to the customer for future expansion of the machine. Leveling a machine that is 24 foot can be challenging, hence, leveling feet are to be installed to level the machine to compensate for any inconsistencies in the floor.

The base of the machine can be broken into 3 foot sections. This is done so that the customer can easily assemble/disassemble the machine. Cross braces at the end of each 8 foot will help the customer assemble/dis-assembly the machine easily. This also gives the flexibility to further expand the machine in the near future if the customer desires. Swivel castors are installed in case the machine ever needs to be relocated. Figure 3.26 shows the base along with its components.

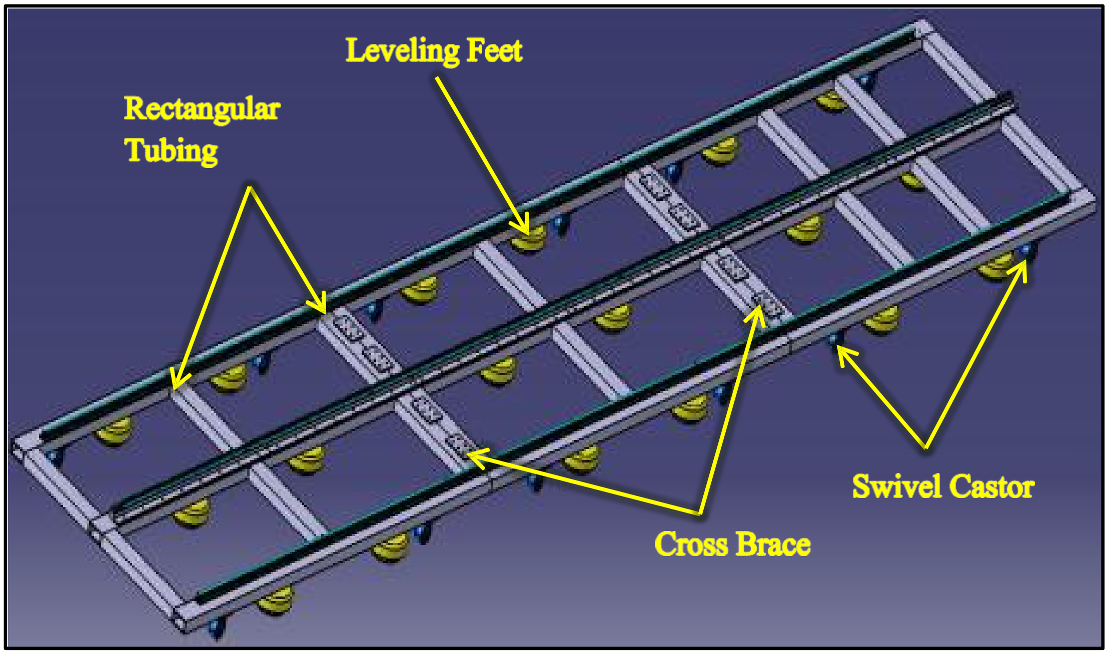


Figure 3.26: The Base of the Compliant Wind Machine shown With Leveling feet and Castors

The work space of the machine is supported with the 80/20 material, mainly due to customer desires. 80/20 is a framing system consisting of extruded beams of 6105-T5 aluminum alloy as shown in Figure 3.27.

Figure 3.27: 80/20 Material: Extruded Beam of 6105-T5 Aluminum Alloy [24]

The t-slotted profiles create the foundation for the profile's assembly technology. The "T" shaped slots allow for infinite positioning along the axis [24]. 80/20 will assist in framing the outside structure of the machine. It will support the top table, also referred to as the workspace. 80/20 will also help close in the machine in the future, if the customer desires.

3.2.3 The Winding Box

The winding box performs the compliant wind and travels with the winding carriage. It consists of several components – the winding head, toothed pulleys and belts, supporting pillow block, rotating shaft, multiple bearings, winding arm and a servo motor. Figure 3.28 shows the components of the winding box. The winding box structure supports and locates all the components that assist in winding the cable.

The servo motor used in the winding box is a NTE-330-CONS-0000. The NTE-330-CONS-0000 motor offers low inertia and weighs only 7.5 pounds. It offers a constant torque of 3.6 Nm or 2.65 lb-ft. [21]. The servo motor and the belt pulleys assist in the shaft's rotational movement.

The winding box is connected to the wind carriage; hence, they both travel at the same speed. The winding box is located above the working table and the attached to the winding carriage with two stepped rods. There are two stepped rods have two guide rollers attached that control the tension and straighten the cable as it unwinds from the spool.

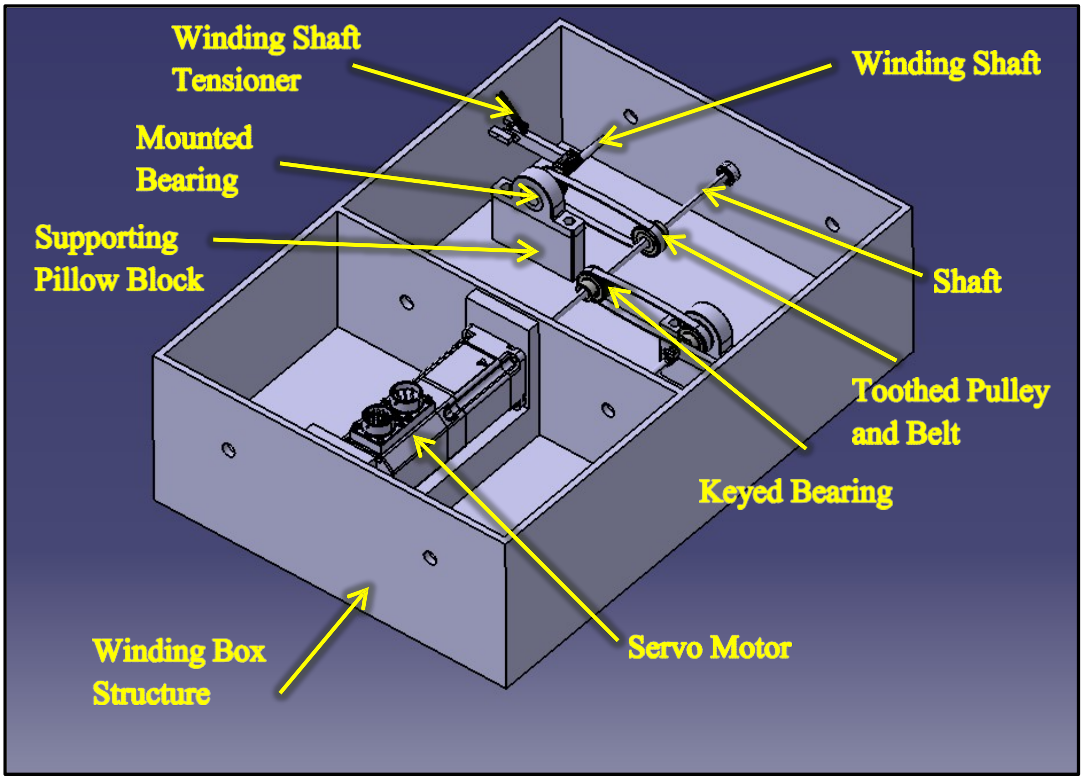


Figure 3.28: Components of the Winding Box Shown

The process starts with the winding shaft tensioner in an unlocked position (upwards). The operator begins the process by pulling the cable upwards from the spools, and bungee cord through the holes in the side panels of the wind box (Figure 3.29). The operator guides the cable through the roller guides, and onto the winding shaft. Once the cable is in its place, the operator pushes the winding arm tensioner downwards to lock it into its place.

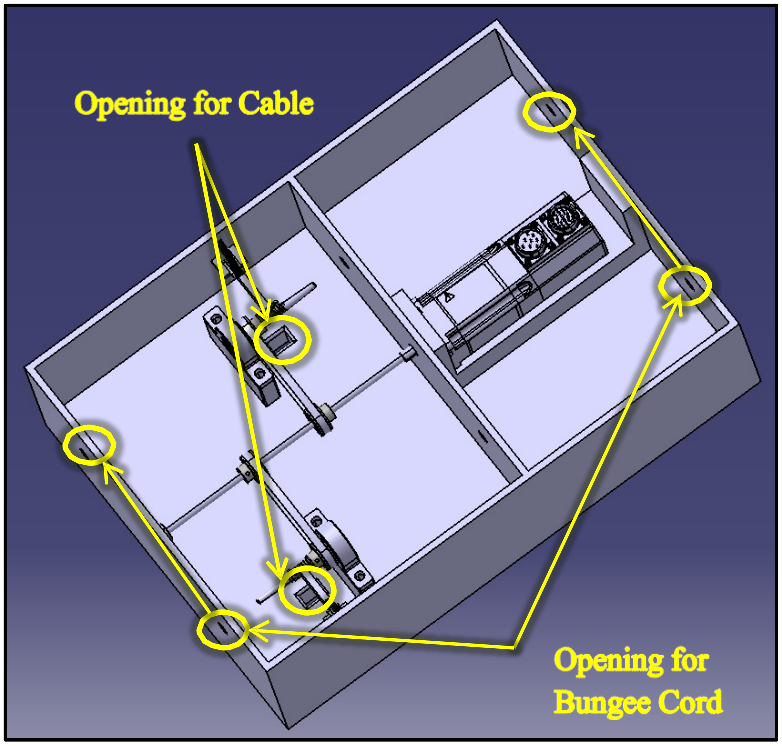


Figure 3.29: Inside of the Winding Box Shown

The wind carriage and the winding box travel from point A to point B while winding the cable and stretching the bungee (Figure 3.30).

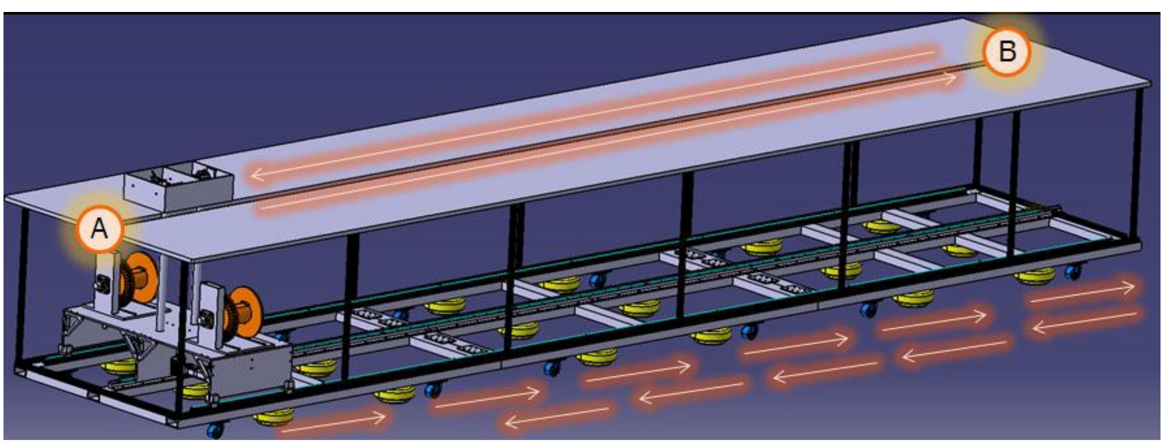


Figure 3.30: The Final Design of the Compliant Wind Machine Shown with A Representing the Initial Position and B Presenting the End Position

The winding box travels to the other end of the table finishing the wind around the previously stretched bungee cord. The marked position “A” represents the home position and “B” represents the away position. The wind is being produced while the wind box travels from position A to B. Once the wind is manufactured, the box once again travels back to position A repeating the process.

The wind box is conveniently located in an ergonomically friendly location, right above the workspace. According to the Department of Labor and Industry, an ergonomic workspace is 40.9 inches in height for an average male/female, if he/she is working standing up [16]. Keeping this in consideration, the height of the workspace is kept at 41 inches.

3.2.4 The Control Box (With a Touch-Screen Panel)

The wind box and the carriage are dependent on the Control Box. The touch screen computer interface gives the operator the flexibility of changing or modifying the pitch and diameter of the wind as the wind is being processed. Having a touch screen control panel gives the operator a tremendous amount of flexibility and significant less room for error. The process for the wind is mostly automatic and is controlled by this computer panel. The machine is programmed to move at a speed provided by the user/operator.

CHAPTER 4

PART ANALYSIS

4.1 Finite Element Analysis, FEA

Structural components may be determined to fail by various modes depending upon buckling, deflection, natural frequency, strain, or stress. Strain or stress failure criteria are different depending on whether they are considered as brittle or ductile materials. In order to assure a feasible design, finite element analysis was performed on the winding carriage following the steps mentioned in Section 2.6. The analysis was performed in three steps:

1. Pre-processing
2. Analysis
3. Post-Processing

During pre-processing, the CATIA model was first transferred to CATIA's Generative Structural Analysis, GSA where a customized mesh of 0.10 inches was applied to the winding carriage (Figure 4.2) separating the constructed geometry into elements connected at nodes, or otherwise known as discreet points. The applied material was 6061-T6 Aluminum Alloy with the following properties (Figure 4.1):

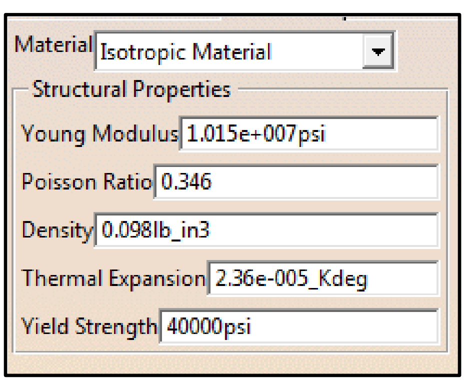


Figure 4.1: 6061 – T6 Aluminum Alloy Properties Used to Perform the FEA

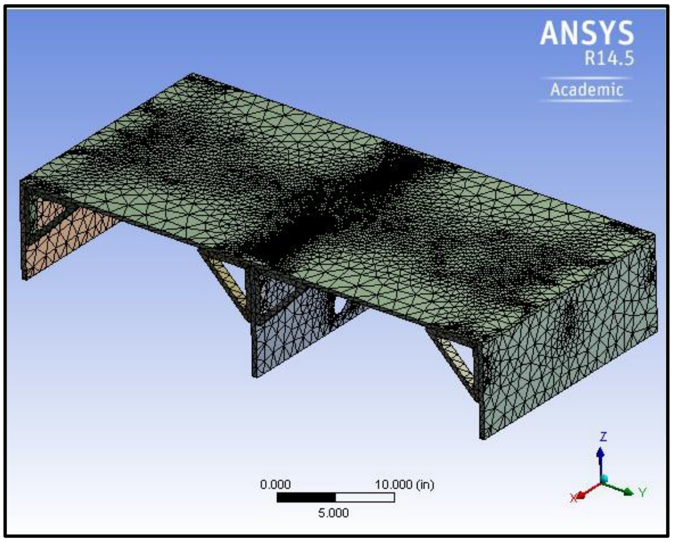


Figure 4.2: Model of the Wind Carriage Shown With 0.10 Mesh Applied in ANSYS

The analysis was then performed by applying the loads and fixing the plate where necessary. The top plate of the carriage is fixed to the side and middle plates with aluminum brackets. The brackets on the carriage were applied as fixed points, and a load of 250 pounds was applied to the top plate. Figure 4.2 the applied load of 250 pounds (weight of the spools and dereelers).

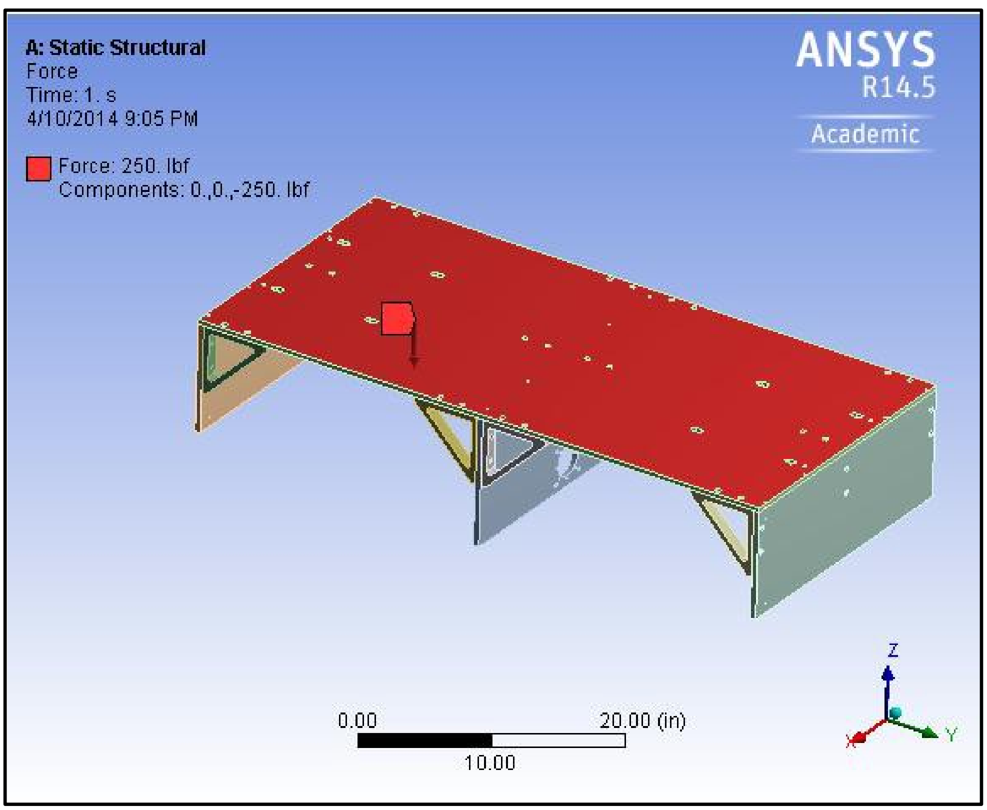


Figure 4.3: Model of the Wind Carriage Shown With Applied Load

Once, the pre-processing and the analysis was performed, the final result also referred to as post-processing was obtained. Figure 4.4 shows the maximum stress of 217.42 psi and a minimum stress of 0.063 psi acting on the carriage.

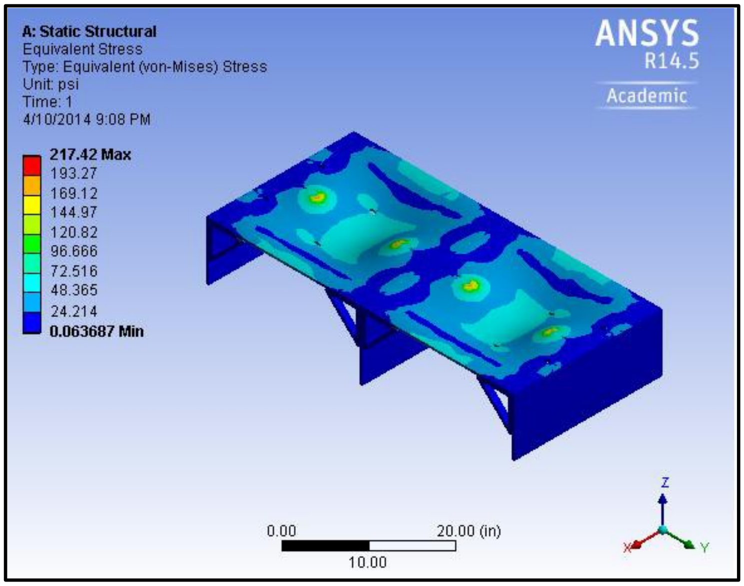


Figure 4.4: FEA of the Wind Carriage with the Von Mises Stress Shown

The total deformation of the carriage can be seen in Figure 4.5 below:

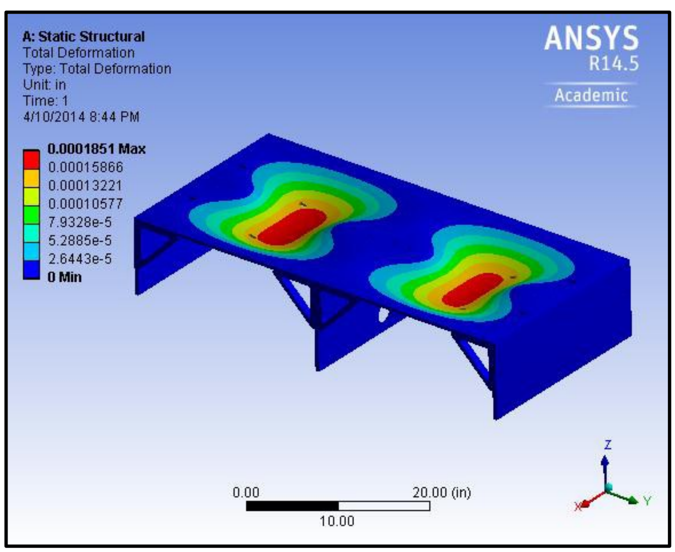


Figure 4.5: FEA of the Wind Carriage with the Total Deformation

Referring to Figure 4.5, it can be seen that the maximum deformation occurs where the spool and the dereelers are located. Once the wind carriage was analyzed, the linear bearings and rails had to be fully analyzed because the entire carriage rests on the base of the machine. Referring to the previously recorded data Section 3.2.2, the linear bearings used on the new CWM are capable of handling 620 pounds of load individually. The four bearings together are capable of handling 2480 pounds of load. Considering that the load of the carriage is 350 pounds, this assures that the bearings will not fail.

After the FEA was performed, a structural optimization of the carriage was performed to further optimize the design. However, this structural optimization was solely performed for learning purposes, and the results achieved will not be incorporated in the actual machine design or fabrication.

4.2 Bearing Damage Analysis

Bearing damage analysis is conducted on the round rail ball bush bearings considering the product's specification. According to Thomson Linear, Inc. the selected bearings can travel 2 million inches before complete failure given the 620 pounds load [23]. Note that in this application, the load acting on the bearing is 350 pounds. Based on the total travel life of 2 million inches and 36960 inches travel life per day, the following calculations were done to compute the travel life in days.

Total Travel Life, $T_T = 2000000$

Total Travel per Day, T_{D} = (Machine Length, X) x (Cycles per Day, C)

= 264 inches x 140 cycles per day = 36960 inches/day

Travel ~~Life, TL =~~ = 54.11 Days

From the provided data by Sparton, it is known that Sparton runs 7 hours a day and 4 days a week, total number of weeks calculated are 13.5 weeks. It is recommended that Sparton replaces or services these bearings every 13.5 weeks or 3 months approximately.

4.3 Structural Optimization through HEEDS

To prepare for the optimization problem, each plate of the carriage was carefully analyzed in ANSYS then exported to HEEDS to be optimized structurally. Upon analyzing, it was determined that the primary objective would be to minimize mass by reducing the thickness of the side, middle and top plate. The constraints that were kept in mind when optimizing were the load of the spool and dereelers. The objective becomes to minimize mass such that the factor of safety remains well above 2. Hence as per the outlines of structural optimization, our problem can be formulated as under:

Objective Function:

To Minimize Mass

Subject to Constraints:

Factor of Safety, $F.O.S \geq 2$

(i.e., the parts/components that undergo active loading can withstand twice the prescribed loads before the failure begins)

To optimize a design in HEEDS, the first crucial step that is needed is the set of parameters to be modified. For the CWM machine's wind carriage, the critical values provided were the thickness of each plate as previously mentioned. The constraints specified were the factor of safety and mass. The plate's results are based on 100 iterations and the best design result is shown in Table 4.1.

Table 4.1: Comparison Between the Current and Optimized Design

Part Name	**Current Design, Thickness**	**Optimized Design, Thickness**
Top Plate	0.50 in	0.25 in
Middle Plate	0.50 in	0.21 in
Side Plates	0.50 in	0.26 in

The baseline thickness for each plate is 0.50 inches, when performing the analysis; the minimum thickness was kept at 0.10 and maximum at 0.90. The following data was extracted from HEEDS, detailing the top 10 feasible designs. Referring to the below Figure 4.6 and above Table 4.1, it can be seen that the thickness of the plates were reduced at an average of 0.24 inches in all three cases.

Parameter Optimization Post Processor

Performance | Agent Variables | Agent Responses | Designs | Parallel Plot

Top Plate

Designs

Ranking	1	2	3	4	5	6	7	8	9	10
Evaluation #	8	38	96	4	56	23	63	11	31	14
Design Source	CurrentStudy	CurrentStudy	CurrentStudy	CurrentStudy	CurrentStudy	CurrentStudy	CurrentStudy	CurrentStudy	CurrentStudy	CurrentStudy
Design Flag	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE
performance	-0.500047	-0.510048	-0.590056	-0.600057	-0.650062	-0.660062	-0.690065	-0.800076	-0.900085	-0.940089
FOS	15.0	15.0	15.0	15.0	15.0	15.0	15.0	15.0	15.0	15.0
Mass	29.9278	30.5264	35.3148	35.9134	38.9062	39.5047	41.3004	47.8845	53.8701	56.2643
Thickness	0.25	0.255	0.295	0.3	0.325	0.33	0.345	0.4	0.45	0.47

Performance | Agent Variables | Agent Responses | Designs | Parallel Plot

Middle Plate

Designs

Ranking	1	2	3	4	5	6	7	8	9	10
Evaluation #	47	88	87	42	68	12	54	62	22	99
Design Source	CurrentStudy	CurrentStudy	CurrentStudy	CurrentStudy	CurrentStudy	CurrentStudy	CurrentStudy	CurrentStudy	CurrentStudy	CurrentStudy
Design Flag	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE
performance	-0.583981	-0.59998	-0.615979	-0.631979	-0.647978	-0.663978	-0.695977	-0.711976	-0.727976	-0.743975
FOS	15.0	15.0	15.0	15.0	15.0	15.0	15.0	15.0	15.0	15.0
MASS	5.33362	5.53488	5.73615	5.93742	6.13869	6.33996	6.54123	6.74249	6.94376	7.14503
Thickness	0.212	0.22	0.228	0.236	0.244	0.252	0.26	0.268	0.276	0.284

Performance | Agent Variables | Agent Responses | Designs | Parallel Plot

Side Plate

Designs

Ranking	1	2	3	4	5	6	7	8	9	10
Evaluation #	47	88	87	42	68	12	54	62	22	99
Design Source	CurrentStudy	CurrentStudy	CurrentStudy	CurrentStudy	CurrentStudy	CurrentStudy	CurrentStudy	CurrentStudy	CurrentStudy	CurrentStudy
Design Flag	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE	FEASIBLE
performance	-0.583981	-0.59998	-0.615979	-0.631979	-0.647978	-0.663978	-0.695977	-0.711976	-0.727976	-0.743975
FOS	15.0	15.0	15.0	15.0	15.0	15.0	15.0	15.0	15.0	15.0
MASS	6.54123	6.74249	6.94376	7.14503	7.3463	7.54757	7.74884	7.95011	8.15137	8.35264
Thickness	0.26	0.268	0.276	0.284	0.292	0.3	0.308	0.316	0.324	0.332

Figure 4.6: The Top Ten Feasible Design and Results Obtained from HEEDS

Currently the carriage contains weighs approximately 90 pounds, the thickness and the mass of the carriage was reduced to approximately 48 pounds. Figure 4.7 below shows the comparison of the thicknesses between the final design and the optimized design. Figure 4.8 presents the change in mass.

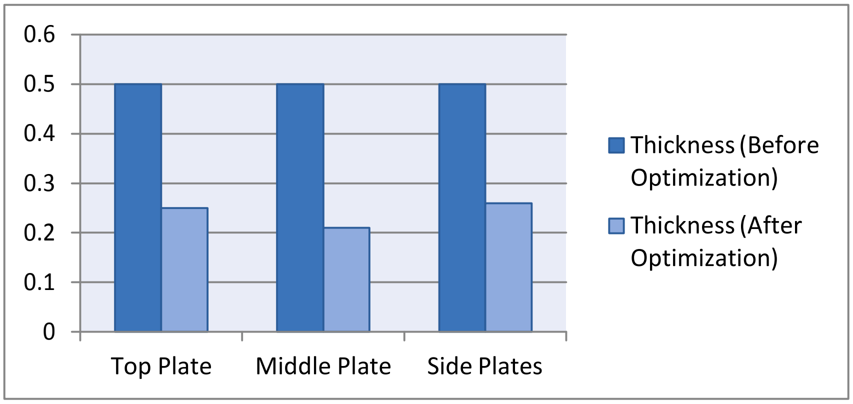


Figure 4.7: The Comparison of Thickness Extracted from HEEDS for the Top, Middle and Side Plates of the Carriage

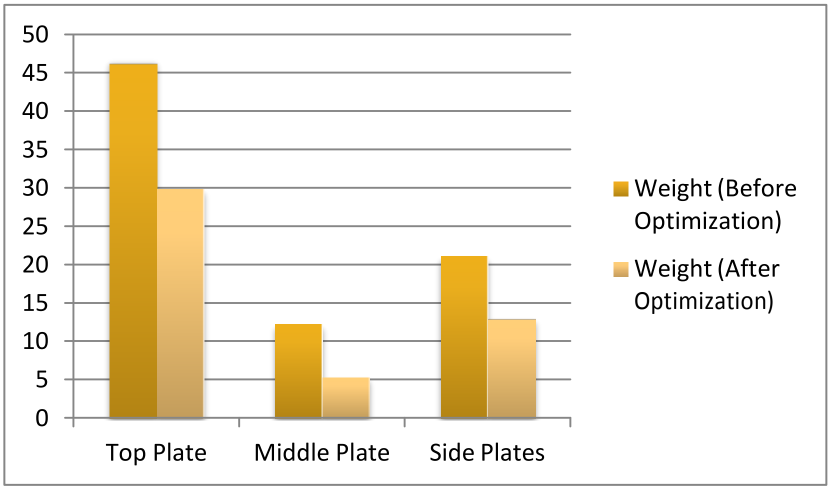


Figure 4.8: The Comparison of Mass Extracted from HEEDS for the Top, Middle and Side Plates of the Carriage

In conclusion, all the design aspects of the new compliant wind machine and its components have been thoroughly explained throughout this chapter. The following chapter will explain the results achieved and the need future research/study.

CHAPTER 5

CONCLUSION AND FURTHER RESEARCH

5.1 Results Accomplished

The objective of this thesis was to re-design the “Compliant Wind Machine”, CWM while minimizing the current problems. The new design is successfully able to solve the current problems. The problems and their solutions are summarized below:

1. Controlling the diameter and pitch of the wind.
   a. Fluctuation of the pitch and diameter
   b. Time consuming individual adjustment required of the motors and the brakes

The diameter and the pitch of the wind are controlled via two servo motors as well as a control box. A winding shaft with grooves is attached to the winding box to give accurate pitch of 0.1 inches and an outer diameter of 0.3 inches. A tensioner is also provided to control the tension of the cable. The control box unit is programmed to set the diameter and the pitch of the wind. In addition, the touch screen and fully controllable computer interface give the operator the flexibility to control and modify the compliant wind. It also controls the speed of both servo motors, hence the speed of the winding carriage and the wind box.

2. The machines large size takes a large amount of space (6276 square feet)

The machine size is reduced from 33 feet to 22 feet (2688 square feet)

3. The V-groove track and the trolley system cluttering the work space and creating potential for damaging the cable.

A modern system of linear profile guides and an RPS system is installed below the workspace to assist in performing the linear function of winding carriage.

4. Multiple Safety/Ergonomic issues

The height of the workspace is adjusted to 41 inches as recommended by OSHA. The lifting of the heavy spool cover is eliminated by placing the spools on the bottom of the machine without a cover. The operator no longer has to lift the 25 pound spool above his/her chest. It is now conveniently located at OSHA recommended level. Safety was assured by relocating the spools and adding maintenance-free spool housing to the bottom of the table.

It is highly recommended to further assure the safety of the operator, that the machine is fully closed from all sides. Installation of 80/20 is provided so that side window panels may be installed, which must be controlled strictly. The opening and closing of the side window panels must be controlled so that the machine automatically shuts off once they are opened or tempered with. It is highly recommended that the operator does not try to insert any of his/her body parts anywhere in/near the machine, (especially inside the winding box since it has small components) once the machine is in operation. The operator is to be fully attentive when operating the machine. Like any machine, it is important to stay clear out of the machine’s moving parts. The operator must not temper or try to service any of the machine’s parts without a professional maintenance crew member present.

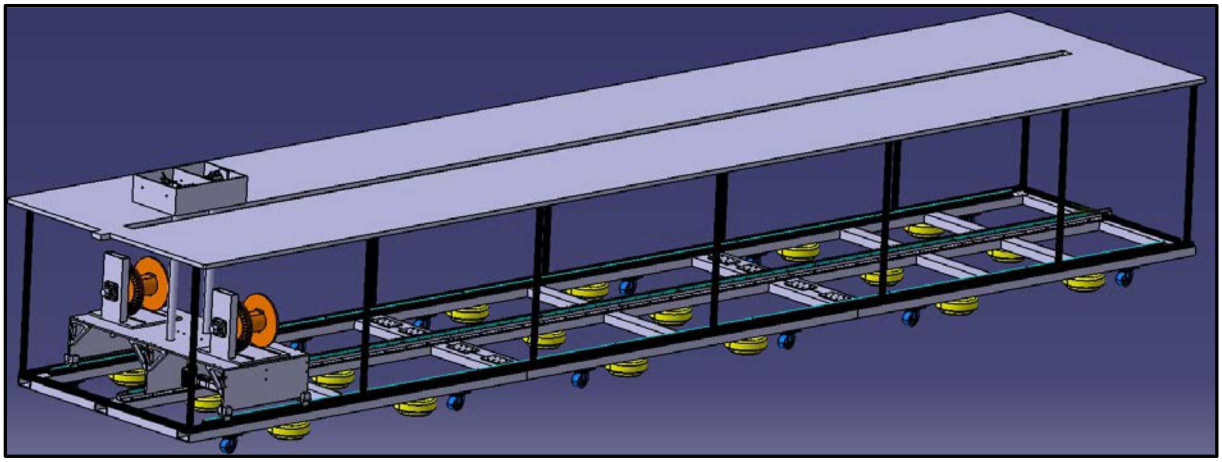


Figure 5.1: Final Design of the Compliant Wind Machine

The project is successfully completed (the design and manufacturing phase) as shown in Figure 5.1. An 8 foot section of the machine is currently being manufactured and will be ready for testing in the near future.

5.2 Further Studies / Research

As mentioned previously, only an 8 foot section of the machine was fabricated, however, not tested. Hence, the “Compliant Wind Machine”, CWM will require further research. Further research is required on manufacturing the wind box and its components. Future research is to be performed during the testing phase of the machine and perhaps furthering the fabrication process of the entire 24’ complaint winder. Research may also include conducting detailed frequency, modal, fatigue and stress analysis on the entire machine. Also, the performed FEA is based on static loading; it is advised that dynamic loads are also analyzed. It is also recommended to study and record the new machine’s process cycles and time for 6 months after the machine has been in production.

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