Design of an Apparatus to Measure Friction Against Snow and Ice

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Abstract

The purpose of this is to design an apparatus to measure friction against snow and ice. The literature review includes different types of snow, the abrasive wear of ultra high molecular weight polyethylene (UHMPE) on ice, various existing frictional tests, slip prevention, dirt absorption on skis and contact angles of cross country skis. Also included as background information pertaining more directly to this thesis are the theories on frictional laws, snow properties which affect ski performance and standardized ASTM friction tests. The various concepts generated and how there were evaluated to derive at the final design is also lightly discussed. This is followed by the final design of the apparatus along with its individual components where the matter of manufacturability of the testing device is covered. Further details about the apparatus are also given such as the methods of reducing noise and device assembly. However, due to the limited scope of this project and since this is an initial design, this report includes only the general basic design and excludes small details such as wiring. This report also excludes a detailed Finite Element Analysis (FEA) along with detailed instructions of the assembly and methodology.
List of Symbols

$\beta$: Attack Angle

$a$: Half Cone Angle

$\mu$: Coefficient of Friction

$G$: Sliprate

$\omega$: Angular Velocity

$\nu$: Translational Velocity

$F_f$: Frictional Force

$N$: Normal Force

$F_w$: Wet Friction

$\eta_w$: Viscosity of water

$A_w$: Wet contact area

$h_{wf}$: Water film thickness

$F_c$: Compaction resistance

$\Delta z$: Vertical snow compaction distance

$F_i$: Impact friction

$\rho_{s,i}$: Initial snow density

$m$: Mass

$T$: Temperature
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1. Introduction

Although research is being conducted to decrease sliding friction against snow and ice and thus increase optimal performance of skis little research has been conducted to design a device which can measure the performance of these skis, that is, their coefficient of friction (COF). Some of the elements that alter the COF include curvature radii’s, weight distribution and base material. The curvatures and weight distribution are a property which allows a skier to manoeuvre around obstacles and sharp corners with ease [1]. The contact material is the main interest for this project as it plays a major role in decreasing the drag created between the ski and the snow and ice.

Previous studies reveal factors which impact the coefficient of friction on the ski’s base material. One factor which can vary the coefficient is the water film thickness that is created due to the initial frictional forces between the ski material and snow [2]. Although friction between two specific surfaces is a theoretical fixed value which should only depend on the normal force being applied, studies show that roughness is a contributor which can affect the results of the final coefficient of friction. Also important is the effect of resultant forces and moments which can introduce ploughing forces. Since this study is concerned about sliding friction, the above factors can be reduced or nearly eliminated using various methods which are discussed.

Applications of this study can be related to the design of not only of Alpine and Nordic skis but also snow vehicles such as skidoos or any other application which implements the use of skiing material that slides against snow and ice.
2. Motivation and Objectives

In order to aid the field of research and development in any industry which depends on the use of skis or other sliding mechanisms, it is of interest to not only design them for optimal performance but to also develop methods and experiments to test them. This is the primary foundation on which this project is built.

The objective is to design a repeatable, convenient and accurate apparatus to measure the sliding friction against snow and ice under realistic conditions. By repeatable and convenient, it is implied that the apparatus should be portable, and easy to assemble and use. The designed apparatus is also meant to nearly eliminate unwanted data such as additional frictional forces which were previously discussed in the abstract. Other sources of noises will be discussed along with their respective process to eliminate them. Although we are interested in designing a device which measures only and purely sliding forces, conducting the test in an outdoor environment still implies that the device be used under realistic conditions.

The device for this thesis is intended for outdoor use and so by realistic conditions, we imply that the snow properties are of that of real snow and the test be conducted in true cold environment temperatures. Since our initial focus was to design the apparatus for indoor use, we hope that future development will modify this device so that it can be used indoors where environmental conditions can be kept constant. Doing so will allow tests to be conducted all year long and won’t depend on fluctuating outdoor climates condition.
3. Literature Review
3.1 Snow Properties

Snow crystals can vary with time due to a large amount of inconsistent variables. Because of this large amount of variables, only those that have a drastic effect on the crystals shape and are relevant to this research will be studied. Such variables include temperature, thermal history, environment pressure, water content, weight on the snow and shear stresses caused by sliding [3, 4]. All these factors can vary the resulting coefficient of friction during testing. Understanding how these factors affect the overall crystal structure of a snow flake would allow essential properties of natural snow to be mimicked for indoor testing, for further possible development of the design.

The overall shape of the crystal affects the frictional forces. There exist a few types of snow crystals such as plate, dendritic, columnar and needle crystals, however we will study the dendritic crystal as it is the most commonly formed. Scientists and physicists suggest that the dendritic crystal snow flake is naturally created under a supersaturated atmosphere and then the centre hub of the crystal is formed under less supersaturated conditions, transforming the dendritic crystal into a complete hexagonal plate [4].

The geometry of the overall crystal can be affected by certain variables. The shape of dendritic crystals can be altered by adjusting the environment pressure and temperature. Decreasing temperature and pressure will obviously initiate the formation of the snow crystal, however decreasing either one of those atmospheric properties too much will result in the formation of water vapour and a decrease in the crystals density. Snow crystals crystallize faster at a
temperature just below the freezing point (-1°C to -5°C) than at lower temperatures (-10°C to -15°C) [4].

Another factor which alters the geometry of the crystal is its history. By that, we mean for how long the slow flake has been on earth. The geometry of the snowflake changes drastically from the moment it falls through the sky to the time it experiences shear stresses and additional weight due to vertical forces. The average specific gravity of a newly fallen snow flake increases from 0.06 to 0.9 when it is touches ground and is compacted [5]. This compacting is due to the build up of snow on top and other vertical forces along with shear forces due to, for example, skiers [5]. The history of a snow flake is highly variable and affects the degree to which the snow is compacted. As discussed later, this compaction introduces a new source of unwanted force.

3.2 Polyethylene Properties
In general, polyethylene is an odourless, lightweight and translucent solid. Certain additives can also be introduced to alter the stabilisation, slip and antiblock characteristics. The key feature of PE which pertains to this thesis is the slip additive.

Since skis require a tough layer between the sliding interface, skis are made of ultra high molecular weight polyethylene (UHMWPE) [3].

The slip additive is used in certain grades of polyethylene for flexible packaging film products to decrease the coefficient of friction. Against metal, this additive is needed in order to prevent the polymer from sticking against the surface.
Polyethylene in general is one of the most stable and inert polymers; that is, it exhibits very high resistance to chemical attacks. Furthermore, there are resins which makes the polymer insoluble, in all organic solvents although some absorption, softening or embrittlement may occur. The degree to which these solvents attack the polymer rely on the molecular weight of the polyethylene. The higher the weight, the more resistant it is to the attack [6].

Although polyethylene is resistant to water and water vapour, its molecular weight also influences the amount of environmental stress cracking due to detergents and silicone oil. With increasing temperatures the polymer can be dissolved and at lower temperatures, the oil causes the polymer to exhibit swelling, discolouration, and disintegration [6].

In addition to the properties mentioned, some of the reasons why UHMWPE, specifically speaking, is used for the application of skiing are because of:

1. High Impact Resistance
2. Outstanding Resistance to Wear and Abrasion
3. Very low coefficient of friction
4. Self-lubricating and non-stick surface
5. Excellent low-temperature properties
6. Energy Absorption characteristics

3.3 Surface Roughness of Ice
The amount of friction created is influenced by several factors, including the heterogeneous distribution of contact points which alters the ice roughness. These contact points depend on
the rheological and physicochemical state of snow, wax, polymer and morphology of the ski base. The amount of wear heavily relies on these factors.

To approximate real ice, two models of ice roughness with different topographies, R1 and R2, were created in a study [7].

![Ice Roughness Topographies](image)

*Figure 3.1 – Ice Roughness Topographies [7]*

Referring to the Figure 3.2, attack angle $\beta$ represents the angle between the sole and ice. The relationship between angles $\beta$ and $\alpha$ is given by $\beta = (\pi / 2) - \alpha$.

![Relationship Between $\alpha$ and $\beta$](image)

*Figure 3.2 – Relationship Between $\alpha$ and $\beta$ [7]*

According to simple friction laws, presented later, it is evident that the coefficient of friction increases with ice roughness increasing. However, experimental studies show that different
loads can alter the coefficient between two sliding surfaces although theoretically, it should remain constant [7].

![Graph showing friction coefficient vs. number of cycles](image1)

*Figure 3.3 – Friction Coefficient vs. Number of Cycles [7]*

This phenomenon is mainly due to changes in localized shear forces. These shear forces can vary depending on the true contact area which is dependent on how much the slider is pressed down, that is the vertical load applied being applied. Therefore, the difference between the true contact area and the experimental contact area is the reason for this discrepancy. This is shown in Figure 3.4.
From Figure 3.4, one can see that the true contact area is equal to the area which each asperity touches the sliding material, not the surface area itself. With each pass of the sliding material over the same surface, the contact area does increase due to localized melting of the snow and ice however introduces a water film which is discussed later. This increase in contact area is shown in Figure 3.5.

Below 273 K temperature, the contact area between the ice and sole is described as a solid/solid interface but with a water film in between. Above this temperature the ice is very
soft and the frictional force decrease. When the temperature decreases to 263 K, the polymer chains become very rigid, making the snow and ice much more difficult to scratch [7].

With mean attack angles of $\beta_{\text{mean}} = 0.95$ rad for R2 and $\beta_{\text{mean}} = 0.66$ rad for R1, it is seen that the morphology sliding on R1 undergoes smooth scratch lines that are very straight and parallel. But in experiments with R2, each ice asperity acts as an indenter which increases the coefficient of friction and the abrasive wear of the polymer.

All in all, this study showed that the coefficient of friction depends on the attack angle and temperature of the localized contact surface.

3.4 Slip Prevention on Icy Surfaces
A study was conducted to study the effect of footwear sole abrasive wear on slip resistance on ice, and to compare the slip resistance of abraded soles on melting and hard ice what that on lubricated steel [9]. The results showed that artificially abraded footwear is more slip resistant than the smooth one for use on hard ice. However, crepe rubber soling was highly recommended for use on hard ice if one wishes to increase slip resistance. In the case where
there is melting ice, crepe rubber is recommended to improve slip resistance since melted ice is much more slippery.

These results were confirmed after conducting experiments where the coefficient of friction was measured with the icy surfaces formed at room temperature and the footwear was not conditioned in the cold environment before the experiment. However, those that conducted this study say that the experiment would have been much more accurate had the equipment been placed in a cold chamber.

This can also be indirectly implied that ski soles which are made of a non rubbery material but with small abrasive surface would be a more suitable to decrease slip resistance.

3.5 Dirt Absorption on Ski Surfaces
In this study, the rate of surface contamination build up was measured of a ski running surface. The sole was made of a transparent base with a white background [10]. A light ray would be emitted onto the surface. The degree to which the incident light ray is reflected back from the white background would indicate how much the sole absorbed the film of dirt. Thus, the light loss would be the result of absorbance of the film of dirt. Along with grime surface scattering, the experiments also revealed grime thickness.

The performance of skis is also dependant on surface energy. Drop-shape analysis was utilized to determine this surface energy to measure contact angles. Contact angles are measured by fitting a mathematical expression to the shape of the drop. The slope of the tangent to the drop at the liquid-solid-vapour interface line is then calculated.
Results showed that waxed ski surfaces became dirtier sooner than unwaxed skis. As a result, waxed skis lose their sliding velocity sooner than unwaxed. However, waxed ski surfaces experience a higher initially velocity due to certain additives added in the wax to increase surface energy.

The experiment concluded that to maintain a perfect glide in skis, the base and the snow do not need any additional lubricant than that which is present, water.

### 3.6 Contact Angles on Cross Country Skis

Surface treatment consists of a mechanical base treatment and waxing. Perfluorocarbon, a hydrophobic wax, has a water contact angle limited to 120° [11]. However, the optimal mechanical running surface treatment may attain a water angle of up to 180°. This water contact angle is dependant by the forces exerted at the three phase contact line of the drop in the plane of the solid. The amount of water produced and its contact angle is determined by the roughness and surface energy [11].

However, no real relation was found between the roughness of the sample and hydrophobicity of the wax. Dry stone ground surfaces showed to have a lower contact angle than scraped surface. After the skis were applied with hot wax, the contact angle increased. One factor that may have hindered this result is the manual scraping that can result in some kind of rough surface with a high degree of hydrophobicity.
3.7 Common Friction Tests
In addition to researching on that relate to snow friction, other friction tests were researched. Although these may seem unrelated at the moment, the specific ones presented here were used to derive some of the concepts shown later.

3.7.1 Friction Test of Polyethylene on Ice
There currently exists a device which measures the friction of ski material such as polyethylene against snow and ice. This device consists of a rotating table which carries the ice annulus and two arms. One arm holds the ice surface preparation tools and the other arm carries the slider, or skidpad, along with the frictional force measurement unit and thermocouples which are integrated into the slider. This tribometer is designed and built with variable temperature, skid velocity and applied vertical load (Figure 3.7).

![Figure 3.7 – Tribometer [3]](image)

As the ice annulus revolves on rotating table, a polyethylene slider is lowered. Using the frictional force measuring units and the thermocouples, the friction force and the temperature close to the interface are recorded.
With the data collected, numerical models of the dry friction, heat conduction, phase changes, and the shearing of water films are compared to experimental results. Comparing the two will yield discrepancies between the two sets of results. The first discrepancy is found in the model of heat conduction from 3D to 1D, however this is corrected using a correction factor. Secondly, the test results assume a constant contact area \([3]\). This can hinder the true COF calculated. Third, although the COF is theoretically only depended on the friction force and the applied vertical load, the COF shows to vary with different loads as mentioned previously. Fourth, the test assumes that the slider has a uniform smooth surface that maintains constant contact with the ice which is not true. Again, a correlation factor is used to account for this discrepancy.

3.7.2 Friction Test of Rubber Tires on Ice

![Figure 3.8 – Tire Friction on Ice [12]](image)

This test consists of a wheel which is translated over by a rail on an icy surface (Figure 3.8). Attached to the centre of the rim is a link. A normal force is applied to the wheel by means of a load cylinder. As the wheel is pulled by the link, the sliprate of the wheel is monitored using sensors. This sliprate is represented as
Where $G$ is the sliprate, $\omega$ is the rotational velocity, $r$ is the radius of the wheel and $v$ is the translational speed of the wheel. As the wheel is pulled along the linear path, a braking force is applied on the wheel to maintain a steady rotational velocity. Since the radius and translational velocities are kept constant, the sliprate is also kept constant. Furthermore, it is seen from the figure below that the sliprate is roughly linear in the beginning of the tire contact.

![Figure 3.9 – Torque vs. Sliprate][12]

As the wheel begins to slide, it is evident that the curve becomes non-linear. This sliding is due to excessive translational speed and heat generation between the interfaces. To measure the change in braking force, a torque transducer is placed between the wheel and the braking...
system to measure the change in torque needed to keep the sliprate constant. Note that measuring the change in torque is linearly related to the sliprate which is influenced by the frictional force. The graph below shows this.

![Graph showing Braking Force vs. Slip Ratio](image)

*Figure 3.10 – Braking Force vs. Slip Ratio [12]*
4. The Coefficient of Friction and Frictional Laws on Polyethylene

The coefficient of friction is a dimensionless quantity used as a standard to classify the amount of friction generated between two materials and is symbolized as $\mu$. This value is normally obtained experimentally. Normal forces and frictional forces are measured and the coefficient of friction can be calculated using the following equation,

$$F_f = \mu N$$

Eq. 1

Where $F_f$ is the frictional force on an object due to sliding, $N$ is the normal force applied on the object and $\mu$ is the coefficient of friction.

The two most common types of coefficient of friction are kinetic and static. Kinetic friction classifies the frictional force of an object when moving for example a box sliding across the floor. However, a greater force is needed to initially move the box. This is because static frictional forces first need to be overcome before the sliding box can experience kinetic friction. This is graphically seen below where a box is being pushed from rest.
In some cases, there is no breakpoint as in Case 1 and instead the recording may resemble Case 2. Both quantities are measured similar ways using the equation above, the only difference being the type of frictional force being measured. That is whether $F = F_{\text{static}}$ or $F = F_{\text{kinetic}}$. We are mainly interested with the kinetic friction as we intend to analyze the friction of skis in motion. Such a number is crucial when conducting wear analysis on two materials being rubbed against each other. The application of such a simple equation is widely used throughout the automotive, aerospace, manufacturing, railroad and other such competitive industries. In addition to the above equation, certain industries will deal with additional factors that will affect the product they design. Such factors are shown in Figure 4.2.
That is why for our case, it is important to research on other frictional forces that come into play when an object is slid against snow and ice. Some factors include temperature, ice roughness, sliding speed, snow drag, compression of snow. The resistance caused due to compaction and snow drag go hand in hand [8].

As mentioned before, one factor is the amount of water produced when a ski or another object is slid against snow [3]. When sliding against snow at a really high velocity, a very fine film of water is produced between the ski and snow. This is the result of heat being generated due to the frictional forces. This thin film can aid the skier in a race as it gives a smoother slide down a hill at fast velocities [13]. This is done as the lubricant prevents the ski to make dry contact with the snow. However, too much water can create extra drag. This drag is introduced by a vacuum being created between the ski and snow and Van der Waal forces which exist between water molecules [3]. The amount of water created, however, can be controlled by selecting a proper load. Because the coefficient between two materials is theoretically fixed, it is evident from
rearranging the above equation that a greater normal force would result in a greater frictional force with a linear relationship. Therefore, more heat is generated and a greater amount of water is produced with increasing weight. If the weight of a heavy object or person is placed on the pair of skis, the focus is to minimize excessive heat generated. One method utilized in skiing is done by changing the colour of the ski base. For a lighter person, the shade of the skis can become darker to increase the temperature of the skis. The degree to which it affects the heat generated is unknown. The effect of this film can be visually seen in below [13]. The factors which govern this effect are split into two regions: ski and snow. In term of the skis, such factors include roughness, hardness, wettability, thermal conductivity, pressure distribution, vibration and flexion. From snow, temperature, density, water content, grain size, grain shape, thermal conductivity, roughness and hardness contribute to this phenomenon [3].

![Figure 4.3 – Friction Coefficient vs. Water Film Thickness][13]

This kind of friction is known as wet friction and is equated by:

\[ F_w = \frac{\eta_w f v A_w}{h_w f} \]
Where $F_w$ is the wet friction, $\eta_{wf}$ is the viscosity of the water at 0°C, $v$ is the absolute velocity of the water film, $A_w$ is the wet contact area, and $h_{wf}$ is the water film thickness [3]. This equation is only valid for those water film thicknesses below a certain threshold because the wet frictional force does not decrease with infinite water film thickness. There are also equations that can aid to determine the heat generated. The first equation represents heat on dry ice that is generated during the initial movement of the ski [14]:

$$P_{dry} = \mu N v$$  \hspace{1cm} \text{Eq. 3}$$

The other equation represents heat generated once a thin film is present by the shearing of water [14]:

$$P_{wet} = \eta_{wf} \left( \frac{\partial v}{\partial Z} \right)^2 h_{wf} \frac{\eta v^2}{h_{wf}}$$  \hspace{1cm} \text{Eq. 4}$$

Where $Z$ is the coordinate orthogonal to the sliding surface. This coordinate is used to determine the heat generated at a specific depth into the thin film.

Another friction factor that can be discussed at this time is the resistance due to snow compaction and drag. For this type of resistance, there are two separate equations. One equation relates to the compaction and compression of the snow underneath the ski, and thus depends on the vertical depth that a ski plunges. Its equation is as follows [14]:

$$F_c = \frac{\Delta z}{l} N$$  \hspace{1cm} \text{Eq. 5}$$
Where $F_c$ is the compaction resistance underneath the ski, $\Delta z$ is the vertical snow compaction distance, $l$ is the length of the ski in contact with the snow and $N$ is the normal force. The other kind of friction which resists the free motion of the ski can be called the ploughing friction. This kind of friction accounts for the impact resistance due to snow build up at the front of the ski. Such a phenomenon can be equated by [14]:

$$F_i = \rho_{s,i}w\Delta z v^2$$  \hspace{1cm} \text{Eq. 6}

Where $F_i$ is the impact friction due to snow, $\rho_{s,i}$ is the initial snow density and $w$ is the width of the ski.

Although some factors are difficult to account for using an equation, they shall be kept in mind when designing a method and apparatus. All frictional forces mentioned above are undesirable in this project because we are interested in pure sliding friction. Instead of eliminating these numbers within our calculations, the device will be design to reduce the torque and force dynamic effects which introduce these frictional forces. If one were interested in calculating all discussed forces that are applied on the ski, the forces would be summed to find the total frictional force, assuming there exists a water film:

$$F_T = F_w + F_c + F_i = \frac{\eta_w f v A_w}{h_w f} + \frac{\Delta z}{l} N + \rho_{s,i}w\Delta z v^2$$  \hspace{1cm} \text{Eq. 7}
5. Standardized Friction Testing

In order to conduct a test which involves wear, friction or erosion one must follow standards and protocols set out by the American Society for Testing and Materials (ASTM) [8]. These standards created by the ASTM are very generalized guidelines used to conduct any kind of testing on materials. Our specific standard is coded as ASTM G 115 which is a standard on conducting any friction analysis and recording the test data. It is vital that the standards of this code be discussed in order to design a method which follows the basic conventional friction tests. The basic theories, such as the kinetic and static friction are included in this standard and were discussed in the previous section and thus ASTM G 115 will be talked about lightly. A key note to be made is that ASTM G 115 is the fundamental standard that lays out other types of friction tests that are discussed below.

ASTM G 115 is one of the most basic standardized friction testing method used for sliding objects and may be used to test friction against snow and ice. The direct application of these standards does not complete our objective though. Recall, that our objective is to create a test apparatus and method using the aid of standards set out by the ASTM.

The ASTM G 40 standard contains a description which includes a theory of “stiction”. According to this stiction phenomenon,

A force between two solid bodies in nominal contact acting without the need for an external normal force pressing them together, which can manifest itself by resistance to tangential motion as well as to being pulled apart.

The study of stiction is of importance to us because the thin water film mentioned earlier can create a vacuum between the ski and the snow, pressing them together without an external
force. There is a wide range of standards for more specific cases and tribosystems and it is important that they are lightly mentioned. Some standards that follow may seem irrelevant to completing our objective, however each standard might have had a potential of aiding us in designing our friction test apparatus and thus are worth lightly mentioning. Furthermore, researching the following standardized friction tests enables one to carefully analyze and understand general testing devices and their methods.

5.1 ASTM D 3029 - Frictionometer
This device measured the friction of plastics against other plastics, metals or other materials at higher sliding speeds than a friction test covered by ASTM G 115 (Figure 5.1). A rotating disk is tangentially contacted by a spring-loaded rider. This rider is connected to a pendulum which rotates in the direction of sliding. How much this pendulum rotates in degrees is the measure of the sliding friction. The rider disk is about 25mm in diameter with a width of 1mm. This test simulates loaded Hertzian contacts, like sphere on sphere, at sliding velocity from 1 to 3 m/s. Hertzian contact stresses quantifies stresses between mating parts. This standard may be of interest as it is meant for tests involving high sliding speeds [8].

![Frictionometer Diagram]

*Figure 5.1 – Frictionometer [8]*
5.2 ASTM E 670 – Pavement/Tire Test
This standard is created to test tire friction on roads using actual tires. A three-wheeled trailer is towed behind a car as the test apparatus. Two of the wheels are skewed while being towed and water can be directed between the tire and pavement. Torque measurements are recorded which correlate with the ease of the tires sliding on the pavement specimen. This standard can aid us as the tire can be made of high density polyethylene and the road can be covered with natural or artificial snow along with a thin layer of water being sprayed to mimic the film created by the heat caused by skis rubbed against natural snow. Similar standards of this type of test include ASTM E 707 which uses a smaller tire on a pendulum which glances on a pavement material. ASTM E 510 uses two tires on a vertical spindle which skids on a pavement material [8].

5.3 ASTM G 133 – Reciprocating Block-on-Plane
This test is useful for analyzing friction characteristics of various combinations of materials that experience a reciprocating motion. Although skis do not experience a reciprocating motion, a coefficient of friction can still be obtained. There are already tests done to find a coefficient of friction between ice and polyethylene using a reciprocating motion. The friction force recording can be different as the reciprocating test includes the push and pull stroke. If the two are different, then it should be recorded which kind of a motion correlates to the obtained force value [8].
A large number of standards created are derived from the basic of ASTM G 115 standard. For example ASTM G 164 standard [8] is the same idea as ASTM G 115, however the plane is inclined. Figure 3.2 shows a few tribosystems that use the exact same formula as ASTM G 115 while others use it to derive their own separate coefficient of friction equation. This is true for all tribosystems shown except for Figure L-8 which is an ASTM G 143 (Capstan) [8]. This shows that although the previously mentioned standards may not seem useful to us at the moment, they can be reconfigured to suit our specific needs. Also, some of the standards shown here are used in our concept generation stage. As mentioned before, these standardized tests are to be used as guidelines for creating our test apparatus.
6. Design Parameters and Constraints
Before discussing the various concepts generated it is vital to discuss the general parameters and constraints all concepts must follow. Firstly, because our design is to be used outdoors, we assume that tests are being conducted on a non flat surface. The flatness of the area which our skidpad may slide over may not be flat and thus may contain small ditches and hills. Secondly, the testing apparatus, including on board sensors must be specially picked to operate in below zero degree Celsius temperatures. Third, in the case where the snow is too soft, there should be a limit on what the maximum normal force is. This force should be sufficient enough to obtain proper friction readings but not too big so as to create a compaction and ploughing effect. Fourth, since a low coefficient of friction is being calculated, the sensitivity is a major concern as small errors in the readings might affect the results significantly. As a result, all sensors will be within a minimum of 0.5% accuracy to ensure the proper data is obtained. Fifth, the translational speed should be large enough to mimic the true motion of a ski but not too large to generate excessive heat which can create a thick layer of water. Below are the assumed numerical parameters that govern the above concerns and will be followed when designing the final design. More specific details about the final design will be given in later chapters.

Vertical Displacement of Skidpad: ± 3cm
Minimum Temperature: -10°C
Normal Mass: 1lb. – 11 lbs.
Accuracy of Sensors: ± 0.5%
Skidpad Speed: 20km/hr = 5.56m/s
7. Design Concepts
Concepts below discuss various ideas that were generated for the overall apparatus. Each explanation is followed by a schematic representation of the concept. The first subchapter includes some of the overall concept designs which were brainstormed. The later subchapters present concepts on the placement of force/moment transducers and skidpad designs pertaining to the overall concept design selection done in chapter 7.1.

7.1 Overall Concepts
7.1.1 Concept A
This device consists of a motor mounted inside a jig which is bolted into the earth. Attached to the motor is an arm that extends downwards on an angle. At the end of the arm is the skid pad which makes contact with the snow surface. Attached to the bottom of the skid pad is the testing ski material. The main idea is that the motor supplies a torque to the skidpad which is under vertical load due to mass added. The force is then passed through this angled arm. Because the angle a is known, data can be collected by means of a force transducer attached to the arm.

Figure 7.1 –Concept A
7.1.2 Concept B
In this concept, the skid pad is rested on the ground attached to a cable. Connected on the other end is a device which pulls the cable. Tension in the cable is measured using a tensile force transducer or connecting a torque transducer to the motor. Such a concept complies with the ASTM G 115 standardized friction testing.

![Figure 7.2 – Concept B](image)

7.1.3 Concept C
This concept is similar to that discussed previously in the report on the topic of measuring friction of rubber tires on ice. The major difference between that device and this is that the wheel is now made of UHMWPE, the common ski material [7]. Also, the surface which it rolls on consists of not only ice but also snow. One great disadvantage that this introduces is the ploughing factor that may arise due to the circularity of the wheel.

![Figure 7.3 – Concept C](image)
7.1.4 Concept D
The final concept is similar to that of concept B. Instead of a device being mounted away from the skipad and pulling a cable, the skipad will be dragged back and forth using a track mounted motor overlooking the ground. Initial runs of the skipad will clear any light snow particles which may account for ploughing resistance. After these initial runs, frictional forces can be obtained by using a torque or force transducer located on the arm which attaches the skipad to the overlooking motor.

![Figure 7.4 –Concept D](image)

7.1.5 Selection of Overall Concept
Concept A was selected to be the simplest solution which fulfilled the projects objectives. There were a number of factors as to why this concept was selected over the others. Firstly, unlike concept B, the skipad in concept A can travel an infinite distance whereas the distance traveled using concept B is limited by the length of the cable. Although taught, the cable still introduces too many degrees of freedom which makes the motion of the skipad too flexible.
such that new sources of friction such as ploughing may be introduced. Although concept C has been utilized on surfaces with low coefficient of frictions and has shown to obtain accurate results, it simply does not depict the true motion of a sliding body. Although concept D depicts true sliding motion, an extremely large surface area would be needed. As a result, such a device can be expensive and too large which affects the portability. Nonetheless, a few of the concepts did contain small features of the general design which were better than concept A. For example, since concept A travels in a circular path, it experiences an almost pure sliding motion. All in all however, it was concept A which proved to fulfill its objectives, within constraints, under the simplest manners.

### 7.2 Placement and Type of Transducers
Before discussing some the more specific concepts generated, such as ideas to measure frictional forces and how to attach the skidpad, note that a small change was made to the figure shown above. Instead of creating an arm which extends downward at an angle, the arm now extends horizontally and then bends vertically downward, perpendicular to the ground.

#### 7.2.1 Torque Transducer
This device is a simple torque transducer. Initially, the idea was to attach the transducer on the motor shaft located in the middle. However, because there are internal forces and moments which can cause vibration. Using this idea would also require the tedious and somewhat inaccurate method of running the device without the skidpad contacting the ice and snow, obtaining the measured torque and then subtracting that value from the measured torque when the skidpad makes contact with the snow and ice. Instead, this transducer is should be attached to a link that is closest to the skidpad. It’s a similar idea when calculating the overall
performance of a car by conducting a dynamometer test at the wheels instead of at the crankshaft or elsewhere to obtain the true horsepower of the car or as in our case, the true torque applied to the skidpad. Therefore, for this concept it is ideal to attach the torque transducer at the 90° bend of the arm or lower. Furthermore, because these transducers contain strain gauges and therefore turn, a telescoping device will need to be implemented so that the skidpad remains in full contact with the surface.

![Torque Transducer](image)

**Figure 7.5 – Torque Transducer**

### 7.2.2 Force Transducer A
The force transducer is placed near the end of the vertical arm. A link is then connected between this transducer and a rigid attachment on the skidpad as shown. A key feature about this design is that it provides two degrees of freedom (DOF) using one parameter: the angle which the link makes with respect to the surface. This allows for the skidpad to tilt as well as translate vertically up and down. The fact that only one parameter is needed to tilt and
translate the device up and down also indicates this device requires fewer parts to manufacture and assemble for the ease of the user.

![Figure 7.6 – Force Transducer A](image)

### 7.2.3 Force Transducer B
This setup consists of attaching a miniature uniaxial force transducer that can be placed in between the ends of a shaft. In order to allow the skidpad to tilt, there is a pin connection at one of the 90° bends near the end of the vertical arm. Placing the pin connector at such a location will allow for the force transducer to be parallel to the skidpad at all times. The placement of this pin connection is crucial as it can introduce unwanted moments that may act on the skidpad and thus introduce a ploughing and compaction force component. For example, if the pivot point is placed further up from the skidpad, a greater moment is placed on it. Furthermore, the placement of the pin also influences how much the skidpad can tilt before it interferes with the vertical arm. To allow for the vertical translation, a slider mechanism is introduced. This mechanism consists of a shaft sliding inside a bored channel.
7.2.4 Six-Axis Force Transducer

This setup involves placing a six-axis force transducer along the vertical arm. A six-axis force transducer is simply used to measure forces in the transverse and uniaxial directions. It also measures moments. Pins can be placed in a similar manner as described above. The main reason why this idea was thought of was to eliminate the horizontal link or member which connects the skidpad to the vertical arm. Therefore, this type of transducer would be used to measure primarily transverse forces. The other data which it supplies can be neglected if the user wishes to do so in which case the cost of this product is expensive. Other transducers which measure forces only in the transverse direction were not found.
7.2.5 Selection of the Placement and Type of Transducer

After evaluating each concept briefly discussed, the setup which includes force transducer B was chosen. For the torque transducer, frictional forces would be attained by dividing the torque value obtained by the length of the vertical arm. Because this vertical arm rotates, due to the strain gauge inside the torque transducer, the length would not remain constant and calculating the frictional force would be a harder task than it should be. Therefore, the idea of using a torque transducer was rejected. The six-axis force gauge, although very simple to use, showed to be extremely expensive especially since it is not crucial for this device to measure moments and forces in the uniaxial direction. The price for this device showed to be around $9000 whereas a simple force transducer for our case should be no more than $2000 [15]. After doing a price check on this transducer, the concept was also rejected. Force transducer A showed to be a very simple design. Although this concept uses only one parameter to vary two degrees of freedom, it becomes a much harder task to evaluate the tilt and the vertical translation through only one parameter, θ. Therefore, since ditches and bumps are defined by curved contours on the surface of a vertical height of 3 cm, it would also be harder to measure this 3 cm difference in vertical displacement. Furthermore, due to two pivots points in the link, the vertical distance between the pins and the skidpad will need to be increased in order to prevent the skidpad from interfering the vertical arm. This increased length now increases the moment created by the frictional force and the force pulling the skidpad. Furthermore, because the snow depth varies from place to place and from day to day, there is no guarantee that the link will stay horizontal with respect to the skidpad when the device is placed on the ground as shown below. Because the force transducer being used measures forces in one direction, this causes a big problem in that only a component of the horizontal force, or in this case the
frictional force, will be obtained. The other component would contribute to a vertical force which would be too difficult and tedious to quantify. Using a sliding mechanism, the skidpad still obtains the same number of DOFs but also manages to keep the force transducer horizontal with respect to the skidpad at all times.

7.3 Skidpad Design

7.3.1 Skidpad Design A
The first design of the skidpad consists of counterbored holes of about an inch in thickness and each hole getting sequentially bigger as the depth of the skidpad increases. The purpose of these counterbored holes is to quickly place the weights onto the skidpad. These grooves also allow for the weights to fit inside a snug contour to reduce any movement of the weights. A curved nose is created at the front so that it doesn’t plough through any small bumps.
7.3.2 Skidpad Design B
The idea of this concept is quite similar. The major difference is how the weights are placed. Instead of creating counterbored holes to add successive masses, small indents are created on the top surface to hold the mass. Threaded fasteners are used to ensure the masses do not shift while the device is spinning.

![Diagram of Skidpad Design B](image)

**Figure 7.11 – Skidpad Design B**

7.3.3 Selection of Skidpad Design
Skidpad concept B was selected as the concept design. Although there aren’t many differences between the two concepts presented, concept A did reveal a major problem. Even though skidpad concept A contains a pivot point, each successive mass added to the counterbored holes would dramatically reduce the amount of tilt. This is because the masses would interfere with the vertical arm. Although there aren’t major differences between the two concepts, this subchapter goes to show small details such as these can impact the overall design.
Furthermore, when brainstorming concepts for the skidpad, it was kept in mind that plastic sheets and plastic slabs of polyethylene will be attached to the skidpad by means of using countersunk screws. Therefore, as a general note, each concept contains a removable recess portion at the bottom of the skidpad to accommodate for thin sheets that can be flexibly wrapped around the skidpad and also for thick sheets that will fill inside the recessed portion of the skidpad as it will be shown in the final design. The main reason for this recess is to prevent any ploughing.
8. Final Design
8.1 Overall Design

As mentioned before, the final design of the apparatus now contains a horizontal arm which elbows vertically down at a 90° angle. The detailed drawings of the apparatus and its exploded views are shown in Appendix A. Note, the addition of a second arm being attached to the overall design. The main purpose for this is to compare the coefficient of friction relative between two different types of sliding specimens. Doing this will allow one to observe which material has a lower or higher coefficient of friction. This also will allow one to check whether or not the sensors on both arms are functioning within the specified accuracy by swapping the two sensors from their respective skidpads and observing the difference between the results. More of this is discussed later. Also note that the shaft attached to the vertical arm passes through linear bearings. As a result, these two arms are needed to counterweight the arms to
prevent them from side sliding all the way down. Note however that this does not cancel out the weight of the masses being added on top of the skidpad.

These two arms also act as counterweights so that the mass of the arms, linear bearings, horizontal platform and the displacement gauge are not added to the overall weight being applied on the skidpad. Also note that each arm is about 2m in length which gives each skidpad a distance of about 6.3m to travel. This large circular path and distance not only allows the skidpad to clear snow dust and level most asperities but also prevents the individual snow crystals from being over sheared due to the skidpad. Thus, creating a bigger circle will mean that fewer rotations around the entire circle will be made to collect a sufficient amount of data without over shearing snow crystals. This large circular path also allows the skidpad to travel in an almost linear path.

In terms of measuring the frictional force, the final placement and type of transducer consisted of a combination between force transducers A and B. The major difference is that the pin joint in figure 7.6 has now been changed to a rigid joint and a linear channel is added to allow vertical translation of the skidpad. Also, the link now runs left to right from the vertical member, pushing the skidpad instead of pulling it. The reasoning for this is explained in the Dynamics section.

Because the idea of creating a bored feature might jam the shaft inside the channel, linear bearings were implemented to allow a smooth vertical motion of the skidpad without any vertical resistance. Note also that two linear bearings are used in the setup shown in figure 8.2. This type of placement is not only supporting the horizontal platform, but more importantly,
prevents the vertical shafts from rotating with respect to the arm. Furthermore, the vertical member connecting the skidpad to the horizontal platform is a yoke. This is done to prevent interference between the horizontal link containing the force transducer and this vertical member when the skidpad tilts.

![Figure 8.2 – Linear Bearing Assembly](image)

Also note the displacement transducer connected from the vertical arm to the horizontal platform. This transducer is there to calculate the vertical translation of the skidpad when it goes over a bump in a concave contour with a difference in height equal to or larger than 3cm.

Lastly, note the location of the pivot point which the skidpad tilts about. Placing the pivot point can make a few drastic changes to the dynamics of the skidpad and how the information is gathered. For example, putting the pivot point up at the top of the yoke member would create a major moment of the skidpad due to the large distance between the pivot point and the skidpad. Placing it to the other end of the horizontal link which contains the force transducer would greatly reduce the amount which the skidpad can tilt. Placing the pivot where it is shown reduces the moment drastically due to the small vertical distance, allows more room for the
skidpad to tilt and constraints the frictional force transducer to stay horizontal to the skidpad at all times.

8.2 Components of the Design
Presented below are the separate major parts of the overall apparatus. Engineering drawings are shown in Appendix B. Furthermore, note that prices for those components which added a great amount of cost to the overall apparatus, such as transducers, were obtained. Small costs such as fasteners were not. Also, some parts can be machined while others are bought off the shelf.

8.2.1 The Motor
A DC, brushless motor is utilized in this design. Furthermore, because experiments are being conducted in below 0°C temperatures, a motor which specifies its operating temperature to include the freezing temperatures is essential. The other specifications of interest are the torque and RPM. A motor with a low torque can be used since the force of friction is also low. The overall RPM should move the 2m arm at a tangential velocity of 20km/hr. Below are the specifications.

- Temperature Range: -29°C to 68°C
- Torque: 31.19 Nm
- Max RPM: 3300
- Cost: $500
- Model Number: HT-R86 [16]

8.2.2 The Motor Mount
The motor mount is made of steel to add weight. This is one of the few parts that need to be heavy in order to reduce vibration from the motor spinning. This part is custom made using a lathe and milling machine.
• Material: Carbon

8.2.3 Arm
The 2m long arm is made of carbon tubing. On the end of the tubing is a stub shaft which is used to connect one end of this arm to the motor and the other to the block. Because there aren’t any high loads being transferred, this stub shaft is connected to the arm by means of using an epoxy. Note that two arm are needed.

• Material: Carbon
• Cost: 2 x $34.43/foot x 6.67 ft = $ 451.84
• McMaster-Carr Part Number: 5287T16 [17]

8.2.4 The Block
The block connects together the arm and the torque transducer along with the vertical shafts which act as the channel for the linear bearings. It is made of aluminum for a sufficient amount of strength that the apparatus requires, and is a considerably lightweight material. This part is machined using a milling machine.

• Material: Aluminum

8.2.5 Linear Bearings
The two linear bearings are made of an aluminum shell. Both are frelon-lined closed bearings. The two grooves on the linear bearings are used to hold the bearing in place with the horizontal platform.

• Material: Aluminum shell and frelon liner
• Cost: 2 x $32.95 = $65.90
• McMaster-Carr Part Number: 5986K7 [17]
8.2.6 Horizontal Platform
The part is attached to the linear bearings. It is made of the same material as the block for the same reasons. This part is machined using a milling machine.

- Material: Aluminum

8.2.7 Displacement Transducer
The displacement transducer is used to identify any bumps or ditches with a height or depth of 3 cm. It is connected to an L bracket which is then connected to the block. The transducer also passes through the block for greater support. A crucial note to make is the reaction force it has on the skidpad. If this force is large, it can prevent the skidpad from translating up and down with ease. As a result a transducer with a very small reaction force was chosen. Because the displacement reference on the transducer varies for different depths of snow, it was crucial to not only have a mechanical range of 6 cm but also accommodate a certain amount of extra length so the apparatus can be placed in between a tolerance zone of snow depth. More about this limit is discussed later.

- Material: Aluminum housing, Steel rod
- Cost: $350
- Reaction Force: 0.5 N
- Mechanical Stroke: 10.4 cm
- Model: TEX 0100 [18]

8.2.8 Yoke Member
The yoke member is comprised of two steel parallel plates separated by pins. Note that one end contains two pins holes. This is done to create a rigid connection at the horizontal platform. As mentioned before, this member is a yoke is to prevent the link containing the force transducer to interfere when the skidpad tilts. This part is machined using a milling machine.
• Material: Steel

8.2.9 Force Transducer
This miniature force transducer is placed inside a horizontal link which connects the yoke member to the skidpad. The transducer is connected by means of male threads. One of the biggest concerns when researching for this transducer was to find one that is small and compact yet also measured forces in the appropriate range.

• Material: Aluminum
• Cost: $1,200
• Force Range: 0 – 2000 N
• Model: XFTC310 [19]

8.2.10 Weights
The weights used to apply the vertical force are standardized weights. Note that the weights of different mass have difference product numbers. The last two digits change from mass to mass.

• Material: Steel
• Mass Range: 0.5 oz. – 5 lbs.
• Min/Min Cost per weight: $27.13 - $105.94
• McMaster-Carr Part Number: 1908T(XX) [17]

8.2.11 Skidpad and Plastic Specimen
The skidpad is made of two parts. The first part contains the curved portion of the skidpad and the contours which help place the weights. These weights are screwed down to the skidpad. It also contains a recess to allow for thick sheets, or slabs, to be attached. The second portion is for thin, flexible sheets. This add-on is screwed onto the first portion. The plastic sheet can now be wrapped from the back of the skidpad and around the curved portion. Like the thick sheets, these too are screwed into the skidpad. Both portions of the skidpad are made using a milling machine. The plastic sheets and slabs however are bought off the shelf.
8.2.12 Fasteners, Bolts, Retaining Rings, Brackets

All fasteners, bolts, and the internal retaining ring are comprised of standardized sizes. Diameter lengths range from 0.25” to 1”. Note that these fasteners or bolts are not shown in the assembly or blown up assembly. Brackets are also made from standard sizes except for the one supported the displacement gauge [17]. The bracket which fixes the horizontal link containing the force transducer to the skidpad is machined by using a milling machine.

- McMaster-Carr Part Number for L-bracket Supporting Horizontal Platform: 1556A26 [17]
- McMaster-Carr Part Number for Supporting Yoke Member: 15275A53 [17]
- Bearing Model: R2CC Series

8.3 Sensitivity Analysis

This basic analysis is to present the error range for a specific frictional force obtained from the transducer. The accuracy of the force transducer is 0.5%. The true COF, µ, was assumed to be 0.5 while the load was 10N. This produced a frictional force of 5N.

\[ 4.975 < F_f < 5.025 \]

\[ 0.4975 < \frac{F_f}{N} < 0.5025 \]

Therefore, there is a margin of ±0.0025 in µ using the above force values.
8.4 Dynamics
This section covers the major forces and moments which act upon the skidpad. The biggest concern is the moment couple created by the frictional force, the horizontal force transferred by the motor and the perpendicular distance from the pivot point to the bottom of the skidpad. This creates a moment in the clockwise direction. There is a method of reducing that moment other than decreasing the vertical distance. Recall that the sensor is oriented in such a position that it pushes the skidpad instead of pulling it. This creates a moment in the counter clockwise direction due to the weight of the horizontal platform, the linear bearings and the yoke member, reducing the overall moment on the skidpad. Note that since a couple moment is a free vector, it can be applied anywhere on the skidpad. Below is the equation which represents the overall moment acting on the skidpad. The calculations can be found in Appendix C.

\[ \sum M_o = 0.613 \mu m - 0.0553W \]  

Eq. 8

Where \( m \) is the mass and \( W \) is the collective weight of the horizontal platform, linear bearings and the yoke member. Since \( W \), the weight, is fixed we can see that the moment on the overall body of the skidpad would increase with increasing mass, \( m \), added onto it.

8.5 Finite Element Analysis
A finite element analysis was conducted on the 2m long arm. Although it is made of carbon fibre, the amount of force transmitted is still of interest. This is due to the long 2m arm which can be a significant source of vibration.
ANSYS Workbench was used to conduct the analysis. Assuming the COF is 0.5, a mass of 1.02 kg is added and the displacement transducer is at maximum extension, a bearing force of 5 N was applied to one end of the arm due to stub shaft inserted. The other end, which also has a stub shaft, was constraint using a cylindrical restraint. In addition to this moment, a torsion moment of 2.155 Nm was also applied to the same end of the shaft that the bearing force was applied to. The other end was constraint using a compression restrain which is applied by the coupling clamp. Furthermore, the tensile strength used for the analysis is 120000 psi and the Young’s modulus is 2.1755e7 psi [6]. The deflection and von misses stress distribution is given below. The calculations and free body diagrams are shown in Appendix D. Note that the distance for the horizontal link, \( R_2 \), does not affect the torsion moment acting on the arm due to the linear bearings.

![Figure 8.3 – Von Misses Stress Analysis](image-url)
8.6 Noises

There are a few factors which contribute to unwanted data, or noises, that may arise in the results. The following discusses these noises and the methods of reducing them to obtain results with greater precision and accuracy. These noises include wind, uneven surfaces, and the transducers. Note that only a basic analysis has been done in this section due to the scope.

8.6.1 Wind

Because the arm is quite long at 2m, wind turbulence can create a moment about the centre of the motor. Thus, the frictional force would be directly hindered due to this wind factor. Furthermore, assuming worst case, wind can also blow over snow onto the path which the skidpads travel during the experiments. This introduces light snow dust and particles which need to be removed by allowing the skidpad to travel the circumference of the circles a few times once again and then restarting the experiment. A very cost effective and simple solution
to this problem is to enclose the apparatus by placing a plastic dome over. This solution will also eliminate other unknown environmental impacts that may be applied to the apparatus. However note that the experiment is still conducted under realistic conditions.

8.6.2 Uneven Surfaces
As mentioned lightly earlier, the displacement transducer is used to identify bumps and ditches. For our scenario, a bump or ditch is classified as a curved contour with a vertical height difference of 3cm from the reference surface. Another criterion that needs to be added is that a bump or ditch has the specified height change over a small rotation of the arm. For example, if the displacement transducer, and therefore the skidpad, translates by 3cm or more within about p/18 rad of the arm rotating, the computer will record that it just went over a bump at a specific angle of the arm. If it however, translates by 3cm or more over a span of p/2 rad then only the start and the end of the vertical translation is of concern. The region of travel in between that angle is still level. Since the angular velocity of the motor is known, the computer can calculate the angular distance travelled by observing the time it took for the displacement transducer to extend or retract by 3cm or more. For example, if it took 0.5 seconds to retract 3cm at an angular velocity of 2 rad/sec, then the angular distance travelled would be 1 rad. As a result, the computer would indicate that the skidpad just travelled over a bump due to the sudden change in height over a small angular distance. Using time and vertical translation alone is not sufficient to detect contours because the criterion is dependent on the change in height per radian, not time. Allowing the computer to find these contours instead of the user makes this device a more user friendly and accurate apparatus. Note however that the programming and detailed signal processing is excluded in this report due to the projects scope.
8.6.3 Transducers
Transducers can also introduce sources of error at times. Because this part is purchased off the shelf, there isn’t much, within the scope of this project that can be done. However, one can check to ensure that the force transducers are working within accuracy. First, frictional results are obtained by conducting the experiment on two different specimens, one on each skidpad. Because the transducers can be easily removed by means of removing and replacing the horizontal link, they are swapped and the experiment is run once more. Ideally, the differences between the two friction results ($F_A$ and $F_B$) for the two separate experiments (1 and 2) should be the same (ie. $F_A^1 – F_B^1 = F_A^2 – F_B^2$). If not, we can conclude that there might be a fault with the force transducers.

8.7 Assembly Process
The assembly process is presented in steps for easy instructions. One time assemblies which don’t need to be disassembled after experiments, such as inserting the stub shaft inside the carbon tubing using epoxy and inserting the force transducer into the horizontal link, have been excluded.

Step 1: Secure motor inside the motor mount.

Step 2: Remove the snow where the motor mount will need to be placed and secure the motor mount to the earth.

Step 3: Insert the motor shaft and secure it using a keyway.

Step 4: Secure the arm to the motor shaft by using a coupling clamp to secure the motor shaft to the already fitted stub shaft of the carbon arm.

Step 5: Attach the L-bracket which supports the displacement transducer onto the block.

Step 6: Attach the block onto the other end of the arm which has a stub shaft with external threads. These threads allow the block to be screwed on.
Step 7: Attach the displacement transducer by first inserting it through the top hole on the block and then securing it to the L-bracket using threaded fasteners.

Step 8: Attach the horizontal platform to the linear bearing mount which holds the bearing using a retaining ring. This connection is made using an L-bracket.

Step 9: Attach the horizontal link containing the force transducer to the skidpad using L-brackets and to the yoke member using a pin connection.

Step 10: Attach the yoke to the bottom of the horizontal platform using L-brackets and two pins on each side to prevent any movement, that is, to create a rigid joint.

Step 11: Screw in the vertical shafts into the block and slide the linear bearings through.

Step 12: Screw the end of the displacement transducer into the horizontal platform.

Step 13: Add weights and screw them down on the skidpad using fasteners.

Step 14: Repeat the above steps for the second arm.

8.8 Device Limitations
8.8.1 Snow Depth
The mechanical range for the displacement transducer is 10.4 cm according to the specifications provided by the supplier. Since this transducer needs enough rod length to retract and extend a minimum of 3 cm, there are limits on how deep the snow can be in order to allow the displacement, and thus the skidpad, to translate up or down. The two limits are shown in Figure 8.5.

For the minimum limit, one must note that with each successive pass of the skidpad, a small fine layer of snow is removed. Assuming the skidpad travels the circular path about maximum 20 times, a minimum snow depth of 2 cm is appropriate.
The displacement transducer is fully extended if the snow depth were to be 0cm (Figure 8.5). To find the maximum limit of the snow depth, we know that the transducer will have enough rod length to extend 3cm but it also needs enough length to retract a minimum of 3cm. Therefore, the maximum snow depth is $10.4\text{cm} - 3\text{cm} = 7.4\text{cm}$.

Snow Depth: $2\text{cm} < h < 7.4\text{cm}$

8.8.2 Environmental Conditions
As mentioned earlier, the minimum temperature parameter needed for the overall device was to be $-10^\circ\text{C}$. However, the coldest environment which this apparatus can operate depends on the coldest operating temperature the sensors or motor can endure. As a result, the motor had
a minimum operating temperature of -29°C whereas the sensors had a minimum temperature of -40°C. The maximum temperature is 0°C because snow melts at higher temperatures.

Temperature: 
\[-29^\circ C < T < 0^\circ C\]
9. Recommendations

Due to the scope of the project and because this is only an initial design, some aspects of the design were excluded. For example, because majority of the parts experience an angular velocity, there is a lot of vibration that can be further reduced. As a result, the signal should be processed to eliminate as much noise as possible.

Furthermore, only a preliminary finite element analysis was conducted on the arm. A more detailed analysis should be conducted on other parts such as the motor shaft and the holes in the motor mount which secure the entire device to the earth.

The resultant couple moments can also be further reduced to increase the accuracy of the device. The discrepancy caused by the moments can also be filtered out by either calculating the ploughing force and subtracting it from the results or by processing the signal or by further reducing the moment arms.

Increasing the tolerance zone on the snow depth can also be increased by means of developing the kinematics of the apparatus.

In terms of increasing portability, the numbers of components may be reduced. Doing so will also decrease the number of parts to assemble, making it more convenient for the user. This also reduces the overall mass of the apparatus.

Greater details about the classes of fasteners will also need to be studied. This report only included an assembly drawing which outlined the major connection links between parts such as L-brackets. Note that the purpose of the drawing was to give an overview of the device concept. Minor details such as fasteners and wiring were excluded.
Only a basic methodology was given throughout chapter 8. In order to conduct a proper test, further research should be done to increase the accuracy of the data obtained. This will also help in reducing the number of human errors during the experiment.

Also, a more detailed cost analysis should be done to estimate the overall cost of the apparatus. Due to the preliminary nature of this report, costs of machined parts were not obtained in this report and thus estimating the overall cost would be inaccurate.

Lastly, the structure of the arm is a very basic design which has not been optimized. A detailed FEA and further development on this part should be done to reduce vibration and deflection to increase the accuracy of this device.

Overall, this report outlined the major details of the apparatus. Therefore, further development of these and smaller details should be conducted before building the entire apparatus.
References


Appendix A – Final Design Drawings
A.1 Skidpad Design Closeups

Figure A.1 – Skidpad Design(a)
Figure A.2 – Skidpad Design (b)
A.2 Exploded Views

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<th>PART NUMBER</th>
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</tr>
<tr>
<td>2</td>
<td>M1.00</td>
<td>1</td>
</tr>
<tr>
<td>3</td>
<td>Motor Shaft</td>
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</tr>
<tr>
<td>4</td>
<td>Arm</td>
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</tr>
<tr>
<td>5</td>
<td>Coupling-Clamp</td>
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Figure A.3 – Exploded view of the motor
Figure A.4 – Exploded view of the vertical arm

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<td>Arm</td>
<td>1</td>
</tr>
<tr>
<td>2</td>
<td>Arm T O block</td>
<td>1</td>
</tr>
<tr>
<td>3</td>
<td>Block</td>
<td>1</td>
</tr>
<tr>
<td>4</td>
<td>Vertical Shaft</td>
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</tr>
<tr>
<td>5</td>
<td>Linear Bearings</td>
<td>2</td>
</tr>
<tr>
<td>6</td>
<td>Retaining Ring</td>
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</tr>
<tr>
<td>7</td>
<td>Bearing Mount</td>
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<td>Displacement Trans. Bracket</td>
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<tr>
<td>9</td>
<td>C-Bracket</td>
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<tr>
<td>10</td>
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<tr>
<td>11</td>
<td>L-Bracket</td>
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<tr>
<td>12</td>
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Figure A.5 – Exploded view of skidpad assembly

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<td>SkidpadBracket2</td>
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</table>
Figure A.6–Exploded view of the vertical arm and skidpad assembly
Appendix B – Engineering Drawings

B.1 Arm

Figure B.1—Arm
B.2 Arm to Motor Stub Shaft Connector

Figure B.2—Arm to motor stub shaft connector
B.3 Vertical Bearing Shaft
B.5 Skidpad Bracket
B.6 Horizontal Platform L Bracket

Figure B.6—Horizontal platform L bracket
B.7 Yoke Bracket

Figure B.7 – Yoke Bracket
B.8 Portion 1 of Skidpad

Figure B.8—Portion 1 of Skidpad
B.9 Portion 2 of Skidpad

Figure B.9—Portion 2 of Skidpad
B.10 Horizontal Link

Figure B.10—Horizontal Link
B.11 Yoke Member
B.12 Arm to Block Stub Shaft Connector
B.13 Bearing Mount

Figure B.13—Bearing Mount
B.14 Block

![Diagram of a block with dimensions and markings]

**Figure B.14—Block**
B.15 Coupling Clamp
B.16 Horizontal Platform

Figure B.16—Horizontal Platform
B.17 Displacement Transducer L Bracket

Figure B.17– Displacement Transducer L Bracket
B.18 Motor Mount

Figure B.18–Motor Mount
B.19 Weight

Figure B.19 - Weight
B.20 Yoke Pin

Figure E.20 – Yoke Pin
B.21 Force Transducer

Figure B.21 – Force Transducer [19]
B.22 Displacement Transducer

Figure B.22 – Displacement Transducer [18]

\[ A = 15.3 \text{ cm} \]

\[ B = 10.4 \text{ cm} \]
B.23 Motor

Figure B.23 – Motor [16]

$L_{\text{MAX.}} = 12.93$

$(x) = 11.56$
Appendix C – Dynamics Calculation

\[ \Sigma M = F_t r_1 - W r_2 = 0.0625 F_t - 0.0553 W \]
\[ \Sigma M_0 = 0.0625 \mu m g - 0.0553 W \]
\[ = 0.613 \mu m - 0.0553 W \]

*Figure C.1 – Dynamic Calculations*
Appendix D – Free Body Diagrams of FEA

Figure D.1 – Free Body Diagrams of FEA