

September 2009

Saint-Gobain High Force Grinder Fixture System

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Saint-Gobain High Force Grinder Fixture System

A Major Qualifying Project Report:
submitted to the Faculty of the
WORCESTER POLYTECHNIC INSTITUTE
in partial fulfillment of the requirements for the
Degree of Bachelor of Science
by

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Abstract

Currently, the Saint-Gobain China Grinding and Technology Center in Shanghai uses a low force grinder with an accompanying fixture system to test abrasive belts. A similar system capable of higher force grinding is desired and our team was given the task of designing the high force grinder fixture. To do this, we fully constructed original designs using SolidWorks for some parts, and for others, researched industrially available components. We analyzed each element, compared with several options, and assembled a final design based on our analyses. We compiled our work and supplied Saint-Gobain with a final concept for the high force grinder fixture.

Acknowledgements

We would like to thank several people for their assistance throughout the course of our project. Firstly, we would like to thank our advisor Professor Kevin Rong from Worcester Polytechnic Institute for providing us an incredible experience in China and guiding us through our work along the way. We would also like to thank our WPI co-advisor, Wenjing Lou and Huazhong University of Science and Technology co-advisers Jiajun Yang and Prof Gao Liang for their insight on our work. We would like to especially thank our project liaison at Saint-Gobain, Jinsheng Wang, and his supervisor, Alex Wang, for letting us help them with this remarkable project.

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1.0 Introduction

In order to evaluate abrasive belts, Saint-Gobain's China Grinding and Technology Center uses a robotic-like machine to assist in grinding. This machine, known as the grinding fixture, is used to apply a metal workpiece against a grinder using constant force. The current design works well for the companies past testing requirements of their abrasive belt products, but with a new product emerging, a stronger and more complex grinding fixture is needed.

The current design is a low force grinder fixture. It is pneumatically driven and capable of providing a constant force into the grinder of up to 200lbf with low precision. It is able to move the fixture in two directions and is controlled through a Programmable Logic Controller (PLC) card. The PLC card can be connected to a force sensor when the operator wants to measure the grinding force of the workpiece. The data from the PLC card is relayed to the operator through a laptop. In order to meet the requirements for testing Saint-Gobain's new product, the new high-force grinding fixture must be capable of providing 400lbf, move the fixture in all three directions, and do so with a precision of $0.1\mu\text{m}$. The machine should also have a central control system with an easy to use human-to-machine interface.

Ultimately, the objective of this project is to design a new fixture which can evaluate Saint-Gobain's abrasive belts at a higher force and with greater precision than the previous design. Additionally, the design must meet more specific requirements provided by Saint-Gobain while remaining as user-friendly and reliable as possible. There are many other requirements to be considered in the design which will be discussed in the following sections.

When designing a new fixture for Saint-Gobain, the design team's goal was to provide a means for evaluating abrasive belts of the high force grinder. The team's objectives were to

provide Saint-Gobain with a completed design that fulfills their requirements and additionally provide a list of other potential options, which they may choose to use if they wish. The success of the project has significant implications for Saint-Gobain, whose Research and Development group will use the fixture to test new products in the near future.

2.0 Problem Statement

Saint-Gobain's China Grinding and Technology Center (CGTC) has the task of developing and evaluating abrasive belts. To do this in the past, CGTC has used a ten horsepower low force grinder and a feeding mechanism to test the performance of the abrasive belts with a variety of metal workspaces and grinding motions. This fixture automates and replaces traditional manual grinding, where the customer's technician would physically hold the workpiece and remove material by pushing it into the grinder. The feeding mechanism which holds the workpiece is capable of applying a force of 200lbf in the normal direction. Below in Figure 1 is a picture of the current low force grinder and fixture.



Figure 1: Current low force grinder and fixture

To meet the expanding needs of customers, Saint-Gobain requires a thirty horsepower high force grinder and a fixture that can evaluate grinding belts at double the force of the low force grinder fixture. The previous system was pneumatically powered and even by appearance

seemed unstable and lacking precision. The problem presented to our team is to design a feeding system for the new high force grinder, which can provide a constant grinding force of 400lbf in the normal (x) direction, 400lbf in the tangential direction (y), 200lbf in the axial direction (z).

The goal of designing the high force grinder is to provide a better means to accurately test the performance of the abrasive belts than was possible with the current low force grinder fixture. While the most obvious requirement is the increase in grinding forces, Saint-Gobain's upgrade to the high force grinder also included many other requirements. Most notably, these included the ability to grind in all three directions, clamp four geometrically different workpieces with the same chuck jaws, maintain a high level of accuracy, and provide detailed data from testing, while maintaining a safe environment for the operator. All other aspects of design were at our discretion to promote efficiency of the facility. Further details about specific design requirements are shown later in the report.

3.0 Background

In order to better understand the scope of the project, we first researched the successful business history of Saint-Gobain leading to their current status. We highlighted their products, given special attention to abrasives.

3.1 Saint-Gobain History

Saint-Gobain is known around the world as producer of a wide range of industrial materials. With its diverse product line, Saint-Gobain is a global leader in many of the fields which it supplies. The French company, founded in 1665 became a leader in mirror manufacturing industry after less than a decade. The company produced solely mirrors until the 1860's, when they began to produce other products such as "glass panes for skylights, roofs and room dividers, thick mirrors, semi-thick glass for windows, laminated mirrors and glass, and finally embossed mirrors and window panes" (Saint-Gobain Group, Our History, 2009). In the nineteenth century, Saint-Gobain expanded its operations across Europe to nearby countries Spain, Italy and Germany. Saint-Gobain continued to expand into the twentieth century, increasing its product line to include household glassware such as jars, bottles, and tableware, which would be further complimented by an emergence into the construction industry with the manufacturing of fiberglass insulation. (Saint-Gobain Group, Our History, 2009).

Dedicating itself to research Saint-Gobain still maintained focus on its primary product, and made several significant ground breaking advances in glass manufacturing. In the 1910's, Saint-Gobain developed several new glassmaking techniques including a dipping process which reduced shattering of automotive glass. Two other important techniques that Saint-Gobain

pioneered was a glass bending technique and a process to coat glass with aluminum (Saint-Gobain Group, Our History, 2009).

From 1950 to 1969, Saint-Gobain profits increased ten percent each year and expanded its number of employees from thirty-five thousand workers to one-hundred thousand workers. The company continued to integrate and since 1970, has merged and horizontally integrated with many companies worldwide. In 1993, Saint-Gobain made a substantial purchase in Worcester, MA of one of the country's leading abrasives manufacturers, Norton Company. Norton is over one hundred years old and has close ties with Worcester Polytechnic Institute (Saint-Gobain Group, Our History, 2009).

3.2 Saint-Gobain Today

Today, Saint-Gobain is a diversified company who manufactures goods from fields including high performance abrasives, semi-conductors, construction materials, and glassware (Saint Gobain, 2009). The company is able to be a leader in its field in part to both its long standing presence in its fields, and an ambitious research program. In fact, just in 2007, Saint-Gobain increased its research program staff from two thousand to three thousand employees in 2007 alone. They operate fourteen research centers worldwide and applied for over three hundred patents in 2008 (Rueters, 2009).

In the last decade, Saint-Gobain has positioned itself as an arising competitor in the habitat and construction market. This sector has great promise and room for expansion, and Saint-Gobain is aimed at being a world leader here also. Currently, about 75% of their gross sales is on the habitat and construction markets. Products they are most focused in include high performance insulation, lighting, and fuel cells (Saint-Gobain Markets).

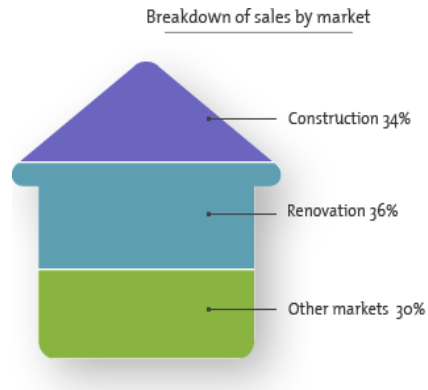


Figure 2: Breakdown of markets

Despite its efforts in research, sales volume in Worcester has decreased approximately forty percent in the last year, and the company as a whole has suffered a 14.8% loss in sales in the first quarter of 2009 alone. While a large company, at two hundred thousand employees, the company's 2008 gross sales is approximated at 43.8 billion Euros. Saint-Gobain's stock, which trades on the Paris Stock Exchange under the symbol 'SGO,' like most companies, has experienced a significant loss over the past two years. Since 2007, SGO stock has lost approximately thirty percent of its value (Businessweek.com, 2009).

3.3 Saint-Gobain Research

Research and innovation is the building block that Saint-Gobain has developed its company with since its founding and is at the heart of its business strategy. Since 2004, Saint-Gobain's research staff has increased almost 60%, from 2,000 to 3,500 in 2008 and expanded funding 10% each year therein. Their research team cooperates with over 200 Universities and laboratories and applies for over 300 patents annually. Their total budget is over 530 million USD and is spread out throughout fourteen worldwide research and development sites (Saint-Gobain Group, 2009).

Saint-Gobain spends a great deal of their resources on a limited number of “Major strategic projects” which aim to boost them into emerging markets with revolutionary technology. The company engages in a great deal of risk in their research in development since the capital needed to fund these unpredictable projects can be quite high (Saint-Gobain Group, 2009). While a majority of their money and resources are intended to improve existing products, Saint-Gobain also has a great interest in pursuing new markets to expand their enterprise (Saint-Gobain, Research).

3.4 Saint-Gobain Products

Due to the large size of the company, and the many products they produce, Saint-Gobain is divided into five different departments.

One of their main areas is Flat Glass production. Saint-Gobain is currently ranked N° 1 in France and N° 2 worldwide for the production of flat glass. They have a wide range of applications for their glass products ranging from self cleaning flat glass to flat glass for visual and light control. Some of their outstanding products in this area are the SGG ALBARINO extra-clear patterned glass, SGG DIAMANT float glass for production of photovoltaics, and the SGG PRIVA-LITE technology controllable transparency. (Saint Gobain, 2009)

High-Performance Materials also make a huge part of Saint-Gobain’s products. Their high-performance materials are divided into the following sub categories: ceramics, grains and powders, crystals, plastics, abrasives, and textile solutions. Within these categories there is the innovative NOVELIO ® paintable wall covering, SOFC (Solid Oxide Fuel Cell), particle filters, semiconductor sapphire, and super abrasives. (Saint Gobain, 2009)

Saint-Gobain’s Construction Products consists of insulation, gypsum, industrial mortars, exterior products, and pipes. The pipeline designs are mostly for drinking water supply networks

and waste water drainage. They also supply green materials for home and do plasterboard recycling. (Saint Gobain, 2009)

In the Building Distribution department they have a number of stores that are targeted toward different clients such as clients with the need of renewable energy, energy saving, wood protections, saving time, and others. (Saint Gobain, 2009)

The Packaging department could be considered the creative side of the company. In order to gain market in the packaging industry, Saint-Gobain uses innovative glass packaging designs to meet the customer's demands. To emphasize their environmental policy, Saint-Gobain packaging sector recycles used glass containers and their own production waste to create brand new bottles. (Saint Gobain, 2009)

3.5 Saint-Gobain Abrasives

Abrasive products can be broken down into two main categories: bonded and coated. Bonded abrasives are more commonly known as grinding wheels, which consist of multiple layers of abrasive grains and bond and are relatively more complicated to make. Coated abrasives, such as sandpaper, contain only one abrasive layer and have a simpler composition. Another category, called superabrasives, is sometimes used when describing these products. However, almost all superabrasives are used exclusively in grinding wheels and are only used on a coated abrasive in an exceptional situation (Webster, 2004).

3.5.1 Coated Abrasives

Coated abrasives consist of three components: the backing, the bond, and the abrasive grain. The backing is the initial layer in a coated abrasive and can be one of four options. Each type has different characteristics which gives the product characteristics like strength and/or flexibility

and limits it to various conditions such as dry versus wet surroundings and high versus low pressure. These four kinds of backing are paper, cloth, fiber, and film. Paper is the lightest weight and weakest backing that is flexible and usually requires a dry environment (unless it is specially treated). Cloth backings span a range of weights and can be used in wet or dry operations depending on the material. The cloth can be made out of natural materials such as cotton or synthetic materials like polyester and rayon and are kept together by being either woven or stitched. Fiber is a very strong and rigid backing which operates well in dry conditions and under high pressure. Film is a plastic material which is flexible yet brittle. It is good for low pressure and heat sensitive operations (Davis, 1995).

The next component in a coated abrasive is the adhesive bond, which is applied in two layers. The main purpose of the first layer, called the making coat, is to hold the abrasive grains to the backing maintaining a specified density of particles per area. The second layer, known as the sizing coat, is meant to cover a certain amount of the grain particles, and therefore, controls the roughness of the product. Resin bonds, a type of organic bond, are the most commonly used for both layers in coated abrasives. The material for resin bonds is readily available and these types of bonds can be mixed to have various properties that prove to be useful for a range of situations. Finally, and most importantly, is the layer of abrasive particles. All synthetic materials, including aluminum oxide, silicon carbide, and zirconia alumina, are used in coated abrasives. However, only a few natural materials, such as garnet, emery, flint and crocus, are used because they cannot grind metals, carbides or ceramics (Davis, 1995).

3.5.2 Characteristics of Abrasives

Several properties of the abrasive grain are used to describe the material, including grit size, chemistry, crystal structure, durability, sharpness, modulus of resilience, hardness, toughness, and friability. The grit size defines the size of the individual abrasive particles. As the grit size increases, the particle size decreases. Understanding the chemistry of the abrasive is important to avoid attrition resistance, the reaction between the abrasive and the workpiece. The crystal structure will help determine the materials durability and determines how the grain will break. A material's durability indicates its ability to hold up under high pressure. Sharpness measures how well an unworn grain is able to cut a workpiece. Sharpness is another property that depends on crystal structure. The modulus of resilience of a material predicts how much energy is required to fracture it, and it is calculated using the ultimate tensile strength and the elastic modulus. Hardness measures a materials resistance to plastic deformation, whereas toughness describes a material's ability to plastically deform before breaking. Hardness is a particularly important characteristic for abrasives because an abrasive must be harder than its workpiece in order for it to grind effectively. Finally, friability defines how likely a material is to crumble. This may be a useful property for some grinding wheels because, as the grains crumble, they reveal new edges and maintain the abrasive's sharpness. However, friability does not benefit a coated abrasive because it only has one layer of abrasive grains, meaning there is not enough abrasive if the material crumbles too easily (Davis, 1995).

4.0 Goal Statement

Before work began on designing the fixture, our team evaluated the problem, reviewed the background research, and created a goal statement. This one sentence statement clearly and concisely identifies what needs to be accomplished, but in general terms so that no possibilities would be limited from our designing. Below is the goal statement for this project:

To provide a means to accurately evaluate abrasive belts on a high force grinder.

5.0 Performance Specifications

The next step of the design process was to arrange a list of performance specifications, which will outline the requirements of the fixture and describe what the design must do. This section is intended to constrain the problem so it can be solved. After completing the design, it may be evaluated by how well it satisfies these specifications. The design team organized this section into three parts: General, Mechanical System, and Controls System.

5.1 General

1. The fixture must be able to apply a constant grinding force of 1780 N in the x-direction
2. The fixture must be able to apply a constant grinding force of 1780 N in the y-direction
3. The fixture must be able to apply a constant grinding force of 890 N in the z-direction
4. The fixture's x-axis must have a minimum stroke distance of 250mm
5. The fixture's y-axis must have a minimum stroke distance of 230mm
6. The fixture's z-axis must have a minimum stroke distance of 230mm
7. The fixture should be able to move the work piece linearly in three directions simultaneously

8. The fixture should be able to apply grinding forces in two directions simultaneously
9. The resolution of position detection of each axis is $1\mu\text{m}$
10. The fixture should have a precision less than $10\mu\text{m}$ on all axes

5.2 Mechanical System

1. The fixture must be self standing on a provided platform
2. The fixture must be automatically driven by electric motors
3. The fixture should not touch any part of the high force grinder
4. The fixture should have a working lifespan of over ten years
5. The fixture should be water and corrosion resistant
6. The fixture should not be affected by dust or small chip buildup
7. The fixture should have minimal chip deflection onto bearings, motors, and sensors
8. The fixture should be easily cleaned, maintained, and allow for the easy replacement of interchangeable parts
9. The fixture should be able to accommodate the installation of safety glass
10. The fixture's internal forces should be optimized
11. The mechanical structure should withstand forces from all three grinding modes simultaneously while deforming less than $10\mu\text{m}$ in any direction
12. The mechanical structure should not experience frequencies exceeding 100 Hz and vibrations should be minimized
13. The mechanical design should consider user-friendly placement of control components
14. Bearings must withstand the maximum force applied in each direction listed above
15. Bearings should utilize lubricant and be replenished easily
16. The chuck must be automatically driven

17. The chuck should be user-friendly and allow work pieces to be clamped easily
18. The chuck's jaws should be able to hold work pieces with dimensions 180x12mm, 6x25mm, 26x26mm and cylinders of a radius less than 30 mm and minimum depth of 25mm
19. The number of moving parts should be minimized
20. Cost of materials and fabrication should be minimized

5.3 Controls System

1. The control system should operate with 220 AC Volts, three phase voltage at 50Hz
2. The control system should command the actuation of the chuck
3. The control system should be able to control the time for grinding
4. The control system should be able to operate in three modes: plunge grinding mode, vertical grinding mode, and horizontal grinding mode
5. Plunge Grinding Mode should be able to complete the following process: clamp workpiece automatically, move slowly towards belt, touch the belt and begin grinding in the positive x-direction at a constant force of 1780N, return to starting position when grinding time has elapsed
6. Vertical Grinding Mode should be able to complete the following process: clamp workpiece automatically, move slowly towards belt, touch the belt and begin grinding in the positive and negative y-directions at a constant force of 1780N, return to starting position when grinding time has elapsed
7. Horizontal Grinding Mode should be able to complete the following process: clamp workpiece automatically, move slowly towards belt, touch the belt and begin grinding in the positive and negative z-direction at a constant force of 890N, return to starting position when the grinding time has elapsed

8. The control system should be able to measure the normal force, tangential force, and axial force to be used in each of the different modes of control.
9. The control system should be able to display values for force, velocity, and position.
10. The control system should be able to control, adjust, and display the following variables:
force, velocity, and position.
11. Grinding force should be kept constant after its value has been set. The velocity of the workpiece should fluctuate according to the set value.
12. The control system should have a position accuracy better than 10 μ m
13. Parameters in each mode should be easily adjustable in the interface
14. The interface should be user-friendly

6.0 Methodology

The goal of this project was to design a fixture system that can evaluate abrasive belts for Saint-Gobain's high force grinder. The following objectives were used to implement the project. We identified the problem, classified the fixture system components into two categories, mechanical and control. We selected best design components from within both categories, designed certain sections of the fixture, and then integrated both the mechanical design with the control design into a functional model. By doing so, we were able to select the best components for the system, and provide a preliminary model to Saint-Gobain.

6.1 Identifying Problem

This section addresses the problems with the current fixture for low force grinder currently present in Saint-Gobain Shanghai Research Center. We identified the problem by visiting Saint-Gobain Shanghai Research Center and looking at low force grinder fixture. Discussions with our liaisons aided us in determining the pros and cons of the current machine. The information provided by Saint-Gobain gave us guidelines that guided us through the design process

6.2 Classifying Components

This section addresses the necessary components that the fixture system needs in order to function. To tackle the problem, we decided to break into two teams, one to take care of the mechanical components and the other the control components. By dividing both the components of the fixture system and ourselves into two, we were able to look at each category in more detail and keep efficiency at a maximum.

6.3 Choosing Design Components

We first collected data using books available to us and internet sources to find already available options in the market that would fulfill our needs. Then, we contacted manufacturers to get more information by requesting their catalogs and datasheets. The next step was Interviewing HUST professors to get their opinions and to learn from their personal experience. After gathering all this data, lists of top options were made, along with decision matrices to select the best option. Dr. Jinsheng Wang was then contacted to confirm and comment on our choices. Based on the feedback from Dr. Jinsheng Wang, we adjusted our selections.

6.4 Designing

Besides choosing components, we also needed to design the mechanical structure and the human-to-machine interface. For the structure, we used the trial and error strategy by first designing a model based on our experience and then using the program ANSYS Workbench to evaluate it. This process was repeated multiple times until the structure was optimized. For the human-to-machine interface, we experimented with two different software and hacked codes to be used by the interface. The programs using different software were then compared and the better program was selected.

6.5 Compiling Fixture System

Once all components were selected and designing was done, a complete model including sensors and motors was created in SolidWorks. Further testing was done to verify that the requirements were met. Finally, we created two-dimensional engineering drawings that Saint-Gobain can use in the future to fabricate a prototype.

7.0 Results

After implementing our methodology, we gathered and recorded the results of our work. Here we list the various options we considered for each component of the fixture design and our final selection.

7.1 Mechanical Design

The mechanical design team had two main tasks: selecting mechanical components and designing the structure. The various components that needed to be selected include the clamping system, driving components, supporting and guiding structures, and accessories. The criteria for selecting many of these components depended on the requirements stipulated by Saint-Gobain. For designing the structure, the mechanical design team was responsible for dimensioning major components, such as the plates and brace, and optimizing the overall structure to increase stiffness and eliminate points of concentrated strain.

7.1.1 Chuck

The clamping system is one of the most important aspects of the high force grinder fixture. Its purpose is to automatically clamp a workpiece securely using a stationary chuck, and to prevent movement of the workpiece while tests are being conducted on the abrasive belt. In this section we will demonstrate the process used to choose the chuck and all of its elements.

7.1.1.1 Driving System for the Chuck

One of the requirements set forth by Saint-Gobain is for the chuck to be an automatically driven system. This means that the clamping force of the chuck should come from a separate

source, such as an electrical motor, pneumatics, hydraulics, etc. For this driving system, the team considered all three options and weighed their positive and negative qualities.

Our design options were quickly set after communication with Saint-Gobain when visiting CGTC to inspect the low force grinder fixture. When speaking with our sponsor, Dr. Jinsheng Wang, he indicated to the design group that he would like to see designs for both electrically and pneumatically driven systems, but due to the additional costs and inconvenience a bulky standalone hydraulic system would bring, this option should not be used. Dr. Wang also indicated to us that a power supply for the electric motors would be simple and that the facilities contained a reliable pneumatic system which maintained pressures between 0.6 and 0.8 MPa and has ample capacity for powering a chuck.

In further analyses between the electrical and pneumatic systems, we considered two major factors. Firstly, the weight of the chuck would be of consideration since it could reduce the load of y-axis. We evaluated the mass of the two driving systems which could provide a clamping force of comparable force. In the case of the electric motor, a screw drive, motor, and potentially a gearbox would need to be mounted directly to the chuck. This put this system at a clear disadvantage, because any additional mass mounted on the chuck itself would put additional stress on the y-axis of the structure. The pneumatic driving system however did not have this problem. While the pneumatic actuator would be inside the chuck, all other components of its system could be mounted externally from the fixture. Hoses would attach to the chuck to provide air pressure to the actuator, but the regulator and valves would be mounted away from the fixture and pressure would be provided by the pre-existing shop air system.

Another factor that was considered is the commercial availability of chucks. A vast majority of the chucks available from manufacturers are hydraulically or pneumatically driven and there

is a comparatively narrow selection for electrically driven chuck systems, a fact indicating that pneumatic chucks are far more developed in industry than their electrically powered counterparts. A minor worry was that adding more components using shop air would put further strain on the system since we were unsure of its stability. However, after further communication with Dr. Wang, the shop pneumatic system is very capable, and since the time of actuation is very short, shop strain may be considered minimal. In order to determine which driving system to use, we utilized a design matrix and scored both systems on a scale which we determined weighed based on practicality. According to the results, we found that the pneumatic system would be greatly preferable. The team agreed to use this system and the decision has been confirmed by Saint-Gobain. The decision matrix used is shown below in Table 1.

	Clamping Force	Mass	Strain on Shop	Availability	Total
Full Score	6	6	2	5	17
Pneumatic	5	6	2	5	16
Electrical	6	2	1	2	11

Table 1: The decision matrix for chuck driving system

7.1.1.2 Manufacturer Selection

After determining that the fixture would utilize a pneumatically driven chuck, we began to search for manufactures that produced suitable chucks. Three manufactures were found that were eager to offer design suggestions for the fixture. These companies were Northfield Precision from New York, USA; Changzhou Beut Machinery Manufacture Co., Ltd from Changzhou, China; and Huhhot Machine Tool Accessory Plant from Huhhot, China. All three manufactures offered good products that seemed to be good prospects to meet our design requirements.

Northfield Precision offered two design options to meet the design requirements. The first of which is the CV-650, a 6” diameter chuck which is capable of jaw forces ranging from 220N to 25kN. The chuck has a clamping actuation of 1mm and requires only 0.55MPa of pneumatic

pressure for full jaw force, which is adjustable by an external regulator. With steel jaws, the chuck weighs less than 5.5kg, has a depth of 107mm, and can be custom manufactured for a side air-port in order to mount the chuck easily. With a high precision of 0.64 μ m repeatability, this model was quoted quite high at 48,000 Yuan.

Northfield also offered a second chuck, the CV-800, which is the upgrade to the smaller 6” chuck. This chuck offers a much greater jaw force, with a range of 220N to 38kN. The chuck however weighs significantly more at approximately 21kg and is slightly larger with a length of 122mm. All other features for this chuck are the same as the CV-650 model and it was quoted for 62,000 Yuan.

The design team was also interested in Beut, a Chinese company, from Changzhou. This chuck, the BK160SQ, has a maximum jaw force of 47.3kN. The jaw actuation has a much higher range of 2 to 170mm and requires between 0.4 and 0.8MPa of pneumatic pressure. It weighs much more than its Northfield counterparts at 14kg and is 129mm long. The chuck has a precision of 30 μ m repeatability and is priced very reasonable at 5,900 Yuan.

The third company the design team considered is Huhhot Machine Tools. Their chuck, the K51T-160, is slightly different from the others, since it requires an individual pneumatic cylinder to be mounted inside the chuck. The max jaw force for the chuck is 22kN, and its clamping range is 0-160mm. Its net weight is the heaviest considered at 19kg and its length is also the longest at 182mm. The precision of the chuck is 50 μ m repeatability and it has been quoted for 4700 Yuan.

Once again, in order to help determine which chuck to use, the design team developed a comprehensive design matrix which considers many important design features and assigns each a value based on its importance. All the companies offered chucks with excellent precision and quality, so the most important factor in choosing between manufacturers became price. Beut

offers the best design, with a very acceptable cost, so the design team chose this company. The design matrix showing our selection is below in Table 2.

	Precision	Mass	Quality	Price	X length	Total
Full score	4	3	6	7	3	23
Northfield	4	3	6	2	3	18
Buet	4	2	4	6	3	19
Huhhot	3	1	4	7	1	16

Table 2: Decision matrix on chuck manufacturer

7.1.1.3 Design of Chuck Jaws

In addition to choosing a chuck manufacturer, the design team was also tasked with developing a top chuck design, also known as jaws, which could hold all four workpieces that Saint-Gobain will be using. The jaw design was important to Saint-Gobain because the current chuck design on the low force grinder fixture required the time consuming task of switching sets of jaws when different workpieces were being used. In our design of the jaw, we were most interested in three fundamental aspects, which were user-friendliness, compatibility, and adaptability.

In order to make the chuck user friendly, we sought a design which would allow for the easy replacement of workpieces and remove the step of time-consuming jaw changeovers. We wanted to design the chuck to be compatible with the four workpieces and be able to securely clamp them without deformation or slipping. Furthermore, we wanted the chuck to be adaptable to other workpiece geometries in case Saint-Gobain wanted to use other shaped workpieces in the future.

7.1.1.4 Two-jaw Chuck

The first jaw design that the design team investigated was the simple two-jaw chuck. This jaw originated from the concept of the parallel vice, which was discarded early in the design process, since the concept would have been used in conjunction with an electric motor.

Depending on the jaw geometry, the two-jaw chuck can hold a vast variety of workpieces.



Figure 3: Two-jaw chuck

The standard two-jaw geometry, has two small jaws with flat faced tips. They can hold a wide variety of workpiece geometries, but have a very small clamping surface area and do not provide a great deal of support for larger workpieces. To fit all the workpieces, including rectangular, square, and cylindrical, while maintaining the best adaptability, modified jaws are needed.

7.1.1.5 Four-jaw chuck

After examining the advantages and disadvantages of the two-jaw chuck, the team decided to focus on another type of widely used top-chuck design, the four-jaw chuck. The self-centering four-jaw chuck builds upon the concept of the two-jaw chuck by adding an additional two jaws perpendicular to the first two horizontal jaws. These chucks have many good qualities, including

an ability to add two more points of contact onto the workpiece, but also drawbacks such as clamping range restrictions.

In the case of using the four-jaw chuck to clamp Saint-Gobain's four workpieces, the design works better for some workpieces than others. Workpieces 1 and 4 are perfect for this design due to their symmetric nature, but workpieces 2 and 3 are a much more difficult case. The biggest obstacle is finding a set of four jaws that can fit all four workpieces. In the case of the sets of jaws that will fit an 8-inch chuck, the clamping stroke was too small for workpiece 3; it is simply too long to be clamped by the same set of jaws as all the other workpieces. Workpiece 2 has a similar problem, except that its shortest side is too small to be clamped by the same set of jaws as the other workpieces. Figure 4 shows a picture of a typical four-jaw chuck.

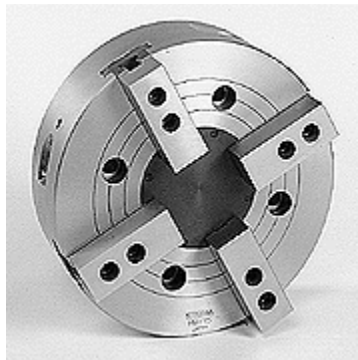


Figure 4: Four-jaw chuck

The team abandoned this idea because of its inability to be compatible with all four workpiece geometries. While this configuration fits cylindrical workpieces well, it is not efficient enough to clamp rectangular and square workpieces with unusual dimensions. After investigating the various standard jaw configurations, the design team developed several designs which would work well to specifically clamp the four workpieces. The five designs are shown in the following sections.

7.1.1.6 Design 1: Chuck with Interchangeable Jaws

The concept of this design is to use a pair of two standard jaws to clamp cylindrical workpieces and use modified jaws, which are very similar to a parallel vise, to clamp square and rectangular workpieces. Changing the jaws would be necessary when switching between workpieces.

In this design, the diameter of the bottom chuck is around 160mm, which is slightly smaller than the height of workpiece 3 (180mm), but still provides more than enough surface area to clamp adequately. The width of the two standard jaws would be approximately 25-30mm, and because of the v-slot on the jaw, it would be adaptable to workpieces having a wide range of diameters. Figure 5 below shows what this jaw design would look like.

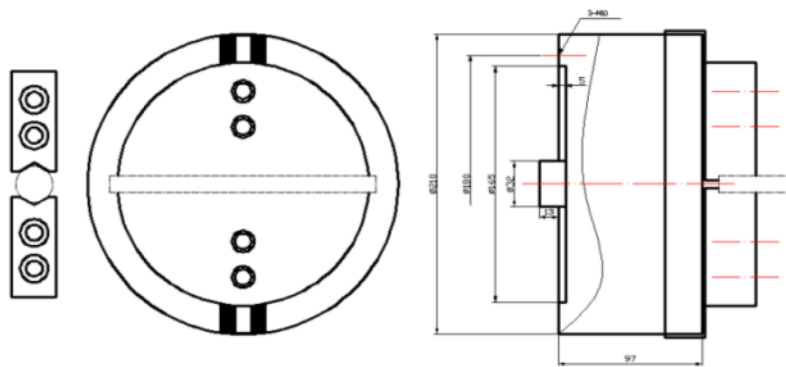


Figure 5: Interchangeable jaws (Beut Company)

7.1.1.7 Design 2: Jaws with V-slots and backings

This design aims to precisely fit every given workpiece in the center of the jaws and provide support on three directions of the grinding force. A rear backing is provided on the jaws to reduce the necessary clamping force required during PGM grinding. The workpiece will be clamped by the two jaws and inserted fully into the jaws so that its end is flush with the backing. Two V-slots are placed in the middle of the jaw to adapt cylindrical workpieces.

The V-slot was designed to optimize compatibility with the workpieces. According to the chart from the Handbook of Chuck Design, in order to clamp cylindrical workpieces of diameter less than 35mm with V-blocks, the span of the slot should be 24mm. Inside of the V-slot is a rectangular section cut out to clamp workpiece 2 horizontally. Additionally, workpieces 1 and 3 would not be effected by the V-slot. Since workpiece 1 is a square with sides of 26mm, it would clear the V-slot and sit on the flat edge of the jaws. One new feature seen implimented into this design is the L-shape at the bottom of the right jaw. This overhang is 15mm long and designed in order to assist the operator in placing workpiece 3 in the same location everytime. The model of the jaw is shown in Figure 6.

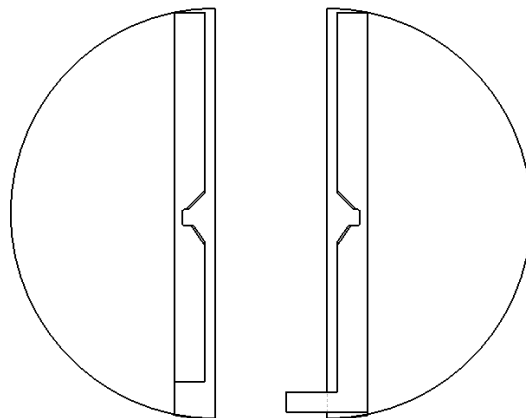


Figure 6: Chuck with V-Slots and Backings

After designing the jaw, the design group conducted a FEM analysis with ANSYS Workbench. We did this in order to understand more about the jaw force and how it would affect the workpiece. In order to clamp the workpieces, especially during PGM+HGM and VGM modes, clamping forces over 13,000 N may be necessary. It was important to do this analysis to ensure that the workpiece would not be damaged by the high forces. The results of the analysis showed that no large stress concentrations existed and can be seen in Figure 7.

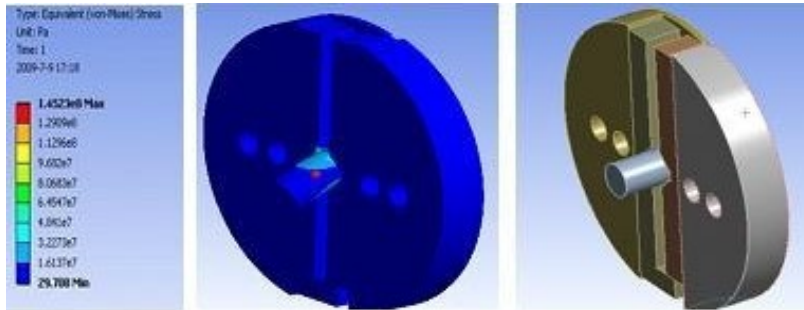


Figure 7: FEM on Design 3

As this idea was evaluated by the design team, it became apparent that there were significant disadvantages to this design. The main disadvantage is that the V-slot is not very user-friendly and that there were other possibilities that could clamp each workpiece more securely and easily. To improve this, the design team developed the idea to dramatically increase the size of the V-slots and use steel inserts machined to exactly fit the shape of the workpieces. With the insert concept, it is possible to clamp workpieces in the center of the jaws, which will decrease the stress concentration to an acceptable 228MPa. This stress concentration is a great improvement over previous analysis where workpiece 2 clamped horizontally with contact points on the angle of the V-slot. With this configuration, stress concentrations on the jaw reached critical levels with a max stress of 1234.4MPa. FEM analysis on central clamping is shown in Figure 8, while the insert concept is show below in Figure 9.

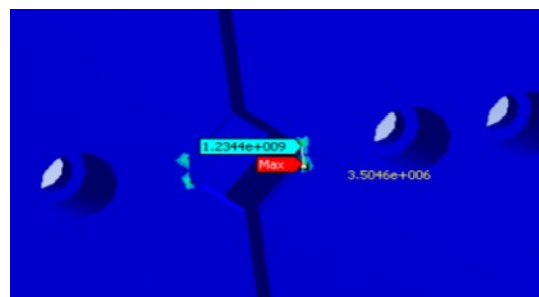


Figure 8: FEM on central clamping

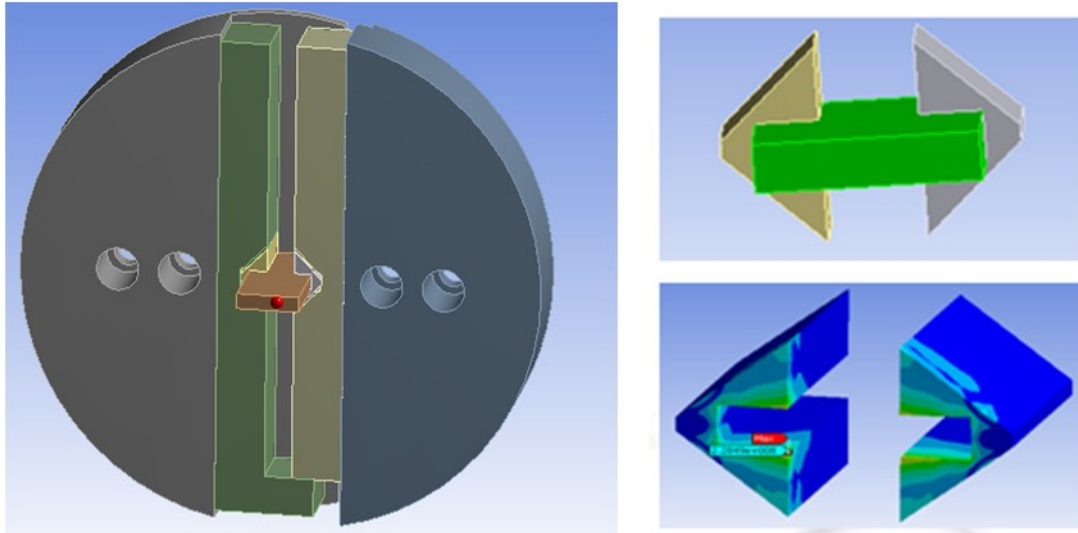


Figure 9: Preliminary Insert Concept

7.1.1.8 Design 3: Screw-in Insert Design

Evolving from the preliminary concept of using inserts in *Design 2*, the screw-in insert design makes an excellent substitute for replacing entire jaws. The chuck used with this jaw design is the single jaw chuck from Northfield Precision. The chuck and jaw design has one side of the jaw attached rigidly to the base, while the other side actuates. The design uses two screw-in inserts which are designed to fit each workpiece exactly. The benefit over *Design 1*, where three different jaws need to be changed, is that changing inserts takes a fraction of the time (one minute opposed to ten). Essentially, the design is an excellent mix of being user-friendly, compatible to all workpieces, and easily adaptable for new workpiece geometries.

Because the workpieces are all placed in the center of the chuck, there is no concern over torque on the jaws, so the diameter of the chuck can be smaller. For this jaw design, a Northfield CV-650 would be used, which has a diameter of 152.4mm (6 inches). Circular slots are machined on

the jaws to place inserts; the slots in the inserts could be custom made according to different shapes of workpieces. The demonstrative drawing of *Design 3* is shown in Figure 10.

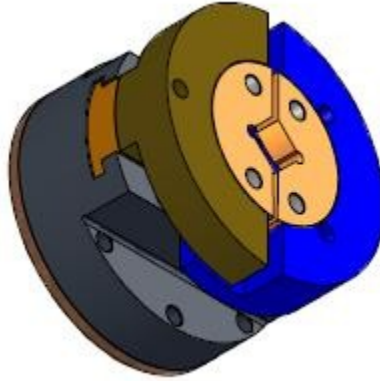


Figure 10: The Screw-in Insert Design

7.1.1.9 Design 4: the Put-in Insert Design

The concept of this design is very similar to *Design 3*, which uses different inserts for every workpiece. The difference lies in the connection between the inserts and the jaws: the insert is simply put in instead of screwed in. The advantage of this design is that it would reduce the time of changing inserts, but the disadvantage is that the inserts will not stay secure as they would in *Design 3*.

Like in *Design 3*, all the workpieces are clamped in the center of the chuck, so there is no torque on the jaws. Since torque is not a concern, the design will use a smaller chuck with a diameter of 160mm.. The width of the inserts is 15mm, leaving 5mm space between the slot on the jaw for the inserts and the screw hole on the jaw. The height of the insert is 40mm, which is slightly larger than the 26mm side of workpiece 1 and allows there to be more area for machining. The inserts are positioned on the jaw with 5mm thick by 5mm long keys and matching indents on the inserts. The demonstrative drawing of *Design 4* could be seen in Figure 11.

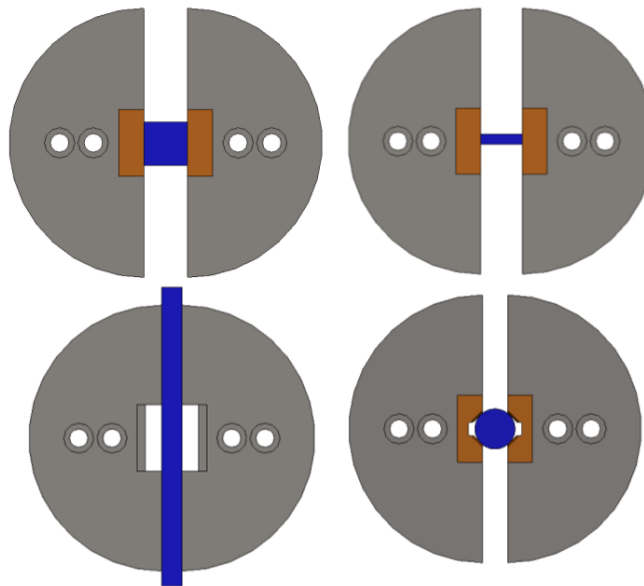


Figure 11: The Put-In Insert Design

7.1.1.10 Design 5: Specifically Machine Jaws

The concept of this design is to machine slots in the jaws designed specifically for the four workpieces requested by Saint-Gobain. While this design will clamp each workpiece perfectly, an obvious disadvantage of this design is its inability to adapt to future workpieces. Also, due to the inability of this jaw design to clamp the workpieces directly in its center, the torque created while grinding becomes a major concern, requiring a larger chuck. While other designs will require only a 6-inch diameter chuck, this jaw design will require the stronger 8-inch chuck from Northfield precision. Below in Figure 12 is a design developed by Northfield Precision which will exactly fit each workpiece.

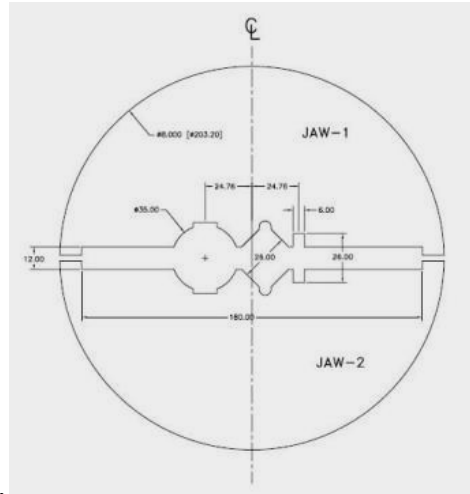


Figure 12: Fit-all Jaws (Northfield)

7.1.1.11 Decision Matrix

In making the decision of which design to use, four major factors are considered: adaptability, switch time, torque and repeatability. Adaptability is divided into three sub-categories: adaptability for the first three workpieces, other rectangular workpieces, and cylindrical workpieces. It is clear that all designs could perfectly fit all three given workpieces. However, *Design 2* and *Design 5* have limited adaptability if the dimensions of the rectangular workpieces are adjusted, whereas *Design 1* and the two insert-designs could fit them very well. For cylindrical adaptability, *Design 1* and the two insert-designs could adapt to a wide range of cylindrical workpieces, while *Design 2* could only fit a limited range of cylindrical workpieces with its V-slot and *Design 5* could only adapt to cylindrical workpieces within a small range of diameters. As for switch time, *Design 5* and *Design 2* do not have parts that need to be changed when switching between workpieces, resulting with the shortest possible switch time. *Design 1* and *Design 3* both need the operator to remove and then replace screws in order to switch between jaws or inserts when changing workpieces. Since there are no screws, *Design 4* requires

very little time for the operator to change between inserts. However, a concern lies that when there is no need to change the inserts because the same workpiece is being used repeatedly, *Design 4* would have less precision in repeatability since the inserts are not secure. The decision matrix is shown below in Table 3.

	Adaptability			Switch Time	Torque	Precision	Total
	Given	Rectangular	Cylindrical				
Full Score	6	4	4	8	6	5	33
(1) Interchangeable Jaws	6	4	4	6	6	5	31
(2) V-slot and backs	6	2	2	8	3	3	24
(3) Screw-in Inserts Z	6	4	4	6	6	5	31
(4) Put-in Inserts	6	4	4	7	6	3	30
(5) Machined Jaws	6	2	2	8	5	5	28

Table 3: Decision matrix on jaw geometry

7.1.2 Rail Configurations

The first task that the design team had to do was to decide how to arrange the three axes. The first step was to list out the six possible arrangements, which are (bottom - middle - top). They include X-Y-Z, X-Z-Y, Y-X-Z, Y-Z-X, Z-X-Y, Z-Y-X. The team quickly decided that the x-axis should be on the bottom, since the x-rails have to accommodate the longest stroke of 250mm and the PGM force of 1780N (400lbf). It is impractical to have the y-axis on the bottom because it would make the y-axis, the axis that has to resist gravity, have to lift much more weight than necessary and cause an excessive amount of preloaded stress. With the z-axis on the bottom, it would be difficult to properly support the x-axis. If the z-rails were spread far apart, there would be a lot of deformation in the middle of the plate supporting the x-rails and if the z-

rails were close together, the deformation would occur at the ends of the plate that would support the x-rails.

The team quickly established that the x-axis would be on the bottom. This eliminated most of the possible options, leaving only two arrangements: X-Y-Z and X-Z-Y. The team developed four design concepts based on these two arrangements. A schematic of each design is shown below in Figure 13.

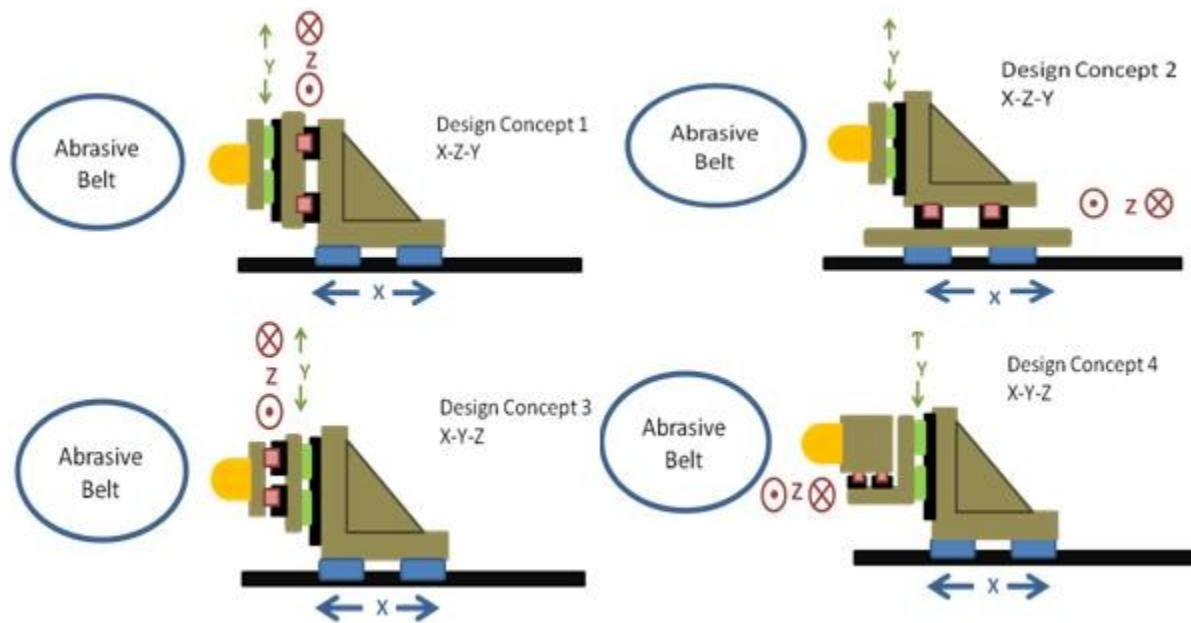


Figure 13: Schematics for the initial four overall design concepts

Design Concepts 1 and 2 have the axes stacked as X-Z-Y, but the z-rails are mounted on a vertical plane in Concept 1 and on a horizontal plane in Concept 2. Similarly, design Concepts 3 and 4 are both X-Y-Z, but the z-rails are mounted vertically in Concept 3 and horizontally in Concept 4.

Evaluation of the four design concepts is based on the direction of forces and the size of moments exerted on the axes. The effects on each axis are evaluated separately under different grinding modes, called Plunging Grinding Mode (PGM), Horizontal Grinding Mode (HGM) and Vertical Grinding Mode (VGM). There are three possible directions that these grinding forces

can be applied to the rail. These directions are normal, perpendicular, and parallel and are defined as follows in Figure 14.

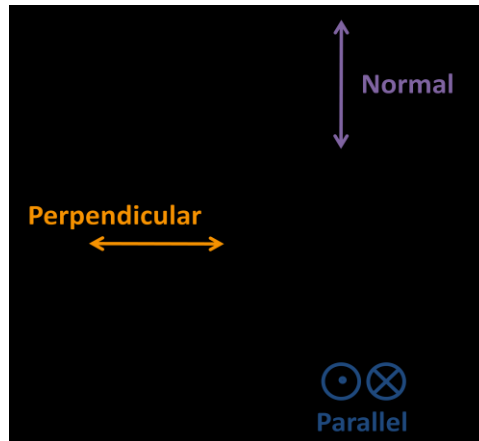


Figure 14: Illustration of parallel, perpendicular, and normal forces

Direction of Force	Plunge			Vertical			Horizontal		
	X	Y	Z	X	Y	Z	X	Y	Z
Concept 1		N	N	N		⊥	⊥	⊥	
Concept 2		N	⊥	N		N	⊥	⊥	
Concept 3		N	N	N		⊥	⊥	⊥	
Concept 4		N	⊥	N		N	⊥	⊥	

|| = Parallel ⊥ = Perpendicular N = Normal

Table 4: The direction of grinding forces on the x, y, and z-rails

As a first step in analyzing and comparing these concepts, we listed the direction of the force on each set of rails (Table 4). Since the results were inconclusive, we estimated the size of the moments of force created by the grinding forces that acted on each set of rails for all of the design concepts as a better way of evaluating them. We did this by considering the distance from the grinding forces to the rails. All of the estimated moments are shown in Table 5. Since having no moment produced is best, these boxes are colored green. The small moment boxes are colored

blue because they are still acceptable. The medium and large moments are colored yellow and red, respectively, since they are progressively worse.

Moments of Force	Plunge			Vertical			Horizontal		
	X	Y	Z	X	Y	Z	X	Y	Z
Concept 1	L	0	0	L	S	M	M	S	M
Concept 2	L	0	L	L	S	L	M	S	S
Concept 3	S	0	0	L	M	S	M	S	S
Concept 4	S	0	S	L	S	S	M	S	S

0 = No Moment S = Small M = Medium L :=Large

Table 5: size of the moments produced by grinding forces acting on rails

A summary of Table 5 can be seen in Table 6 below. It shows that the majority of the moments created in Concepts 3 and 4 are relatively small, where as the majority of the moments created in Concepts 1 and 2 are relatively big. However, before a decision could be made, more characteristics needed to be evaluated.

Moments of Force	Total M			
	0	S	M	L
Concept 1	2	2	3	2
Concept 2	1	3	1	4
Concept 3	2	4	2	1
Concept 4	1	6	1	1

Table 6: Summary table of moments acting on each model

The next step was to use the software SolidWorks to create preliminary 3-D, solid models of each concept so that we could judge the size of the machines more easily. The team used these models to determine if the machine would be user-friendly, meaning the operator can easily switch the workpiece being used. Another characteristic we looked for was if the fixture was

well-balanced by determining how much weight was on the y-rails. A chart (Table7) was created to summarize all of the major pros and cons for each concept.

	Pros	Cons
Concept 1	<ul style="list-style-type: none"> • Less mass sitting on Y-rails • User-friendliness similar to #2 & #3 	<ul style="list-style-type: none"> • Force of gravity acts perpendicular to Z-rails
Concept 2	<ul style="list-style-type: none"> • Less mass on Y-axis • User-friendliness comparable 	<ul style="list-style-type: none"> • Has the most large moments • Plunge force acts perpendicular to Z-rails
Concept 3	<ul style="list-style-type: none"> • Smaller moments than #1 or #2 • Only large moment acts on X-rails 	<ul style="list-style-type: none"> • More mass on the vertical rail
Concept 4	<ul style="list-style-type: none"> • Moments are small 	<ul style="list-style-type: none"> • Large amount of preloaded stress • Would need to extend chuck out further • Least user-friendly

Table 7: Summary of pros and cons of rail configurations

Finally, each concept’s stiffness was considered. In order to test the stiffness of a concept, a SolidWorks model was needed for deformation analysis using ANSYS Workbench. Only two models, which are Concept 2 and 3, were selected for this type of analysis. When strictly considering the pros and cons chart in Table 7, which does not include deformation, Concept 3 appears to be the most favorable option. However after fully dimensioning and optimizing Concept 3, the ANSYS analysis resulted with too much deformation, which made us realize that deformation was an important criterion for selecting a rail configuration. We then chose to model, optimize, and analyze Concept 2 because this configuration is more balanced than the other figures and it seemed as though it could resist deformation better. Additionally, the arrangement of Concept 2 and 3 differ greatly, whereas Concept 1 is very similar to Concept 3 with both the y-axis and z-axis mounted in front of the brace. Therefore, the design team assumed that they would receive similar deformation results from Concept 1 as we obtained from Concept 3. Concept 4 was not considered since all of the previous evaluation of rail configurations indicated this to be the worst concept and there is an extra structure mounted in

front of the brace. The process and results of the stiffness analysis for both Concept 2 and Concept 3 are discussed in detail later in this report in the ANSYS Analysis section. Since Concept 2 outperformed Concept 3 in every test that was run, Concept 2 was considered to be much stiffer than Concept 3, and therefore stiffer than Concept 1 and 4 as well.

To select the best concept, the team made a decision matrix (shown as Table 8). We considered the moments created, user-friendliness, the weight on the vertical axis, and stiffness to be the criteria. The team thought that reducing the moments created on the fixture was more important than user-friendliness and the weight on the vertical axis and therefore gave that category a larger scale of six. User-friendliness and the weight on the vertical axis were considered equal, so they each had a scale of four. We decided that stiffness was the most important criterion and gave this category a scale of eight.

	Moment	User-friendliness	Weight on vertical axis	Deformation	Total
Full Score	6	4	4	8	22
Concept 1	3	3	3.5	3	12.5
Concept 2	2	3	3.5	6	14.5
Concept 3	5	3	2	3	13
Concept 4	5	1	1	1	8

Table 8: Decision matrix for rail configurations

As shown in Table 8, Concepts 3 and 4 have the highest rating for the moments created since they both have only one large moment. Concept 2 has more large moments than Concept 1, thus it received the lowest score. The next criterion is user-friendliness. Since Concept 4 requires an extra structure to mount the z-rails horizontally on top of the y-rails, it is not very user-friendly. The operator would have to reach around the extra structure, making it more difficult to exchange workpieces. The other concepts all rank the same value for user-friendliness because none of them have that extra structure and the machine can just be oriented so that the chuck is closer to the operator when the workpiece needs to be changed. When considering the weight on the vertical axis, Concept 4 is the worst because of the extra structure mentioned above.

Concepts 1 and 2 are ranked the highest since they only hold the weight of the chuck on the y-axis. Concept 3 is ranked moderately since it has the added weight of the z-rails but does not include the extra structure Concept 4 has. As described above, Concept 2 proved to have better stiffness than Concept 3. Concept 3 and Concept 1 are similar in configuration and weight distribution, thus we gave them the same score. Concept 4 has much more weight mounted on the front of the machine and we figured this will have the most deformation. When comparing the total scores in the right column of Table 8, we concluded that Concept 2 is the best option.

7.1.3 Driving Components

A very important component of the fixture design is the driving and transmission mechanism. The working condition and requirements, stipulated by Saint-Gobain for the driving component are shown in the following table, Table 9.

Driving, Transmission	Resistance	Load	Range	Distortion Under Load	Lubrication & Dust Proof
X-axis	As low as possible	1760N	0-250mm	<10um	Grease; dust proof accessories needed
Y-axis	As low as possible	1760N (plus gravity)	0-230mm	<10um	Grease; dust proof accessories needed
Z-axis	As low as possible	880N	0-230mm	<10um	Grease; dust proof accessories needed

Table 9: List of requirements for the diving and transmission mechanism

The design team decided to order this mechanism from external manufacturers because these manufacturers can provide a highly efficient product that would be much cheaper and perform much better than one designed by ourselves. Two types of driving and transmission mechanisms were considered, which are the ball screw and the ACME screw (conventional screw), because they are commonly used in industry in many applications similar to the HFG fixture design. According to research, the ball screw performs at a high efficiency, which is more than two times

the efficiency of the ACME screw. A graph depicting the efficiency of each is shown in Figure 15 below.

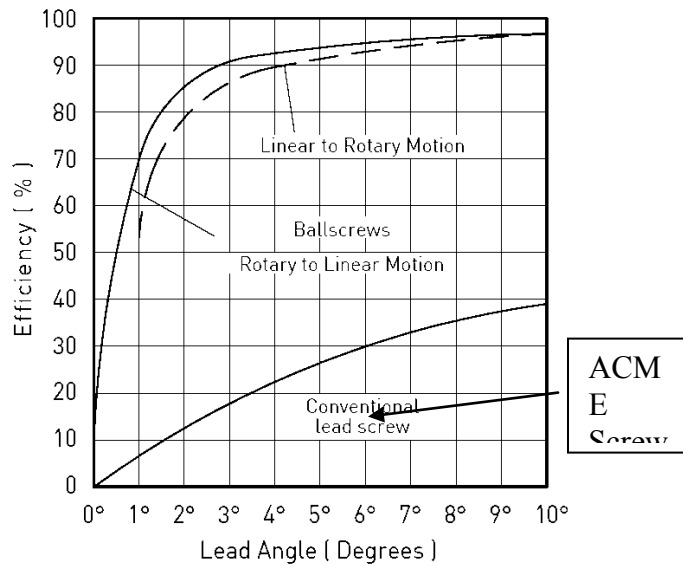


Figure 15: Graph of the efficiencies
(HIWIN Linear Motion Products & Technology, 2008)

The ACME screw is capable of functions that the ball screw cannot do, such as a self-lock function (meaning it is unable to be reversely driven). However, there is no requirement for a self-lock function. We chose to use the ball screw for each axis because it has a high efficiency and the resistance is low (HIWIN Linear Motion Products & Technology, 2008).

After investigating multiple companies, we chose a ball screw manufacturer, called HIWIN, which provides many options for ball screws. HIWIN offers products comparable to the other companies in the field and offers a detailed catalog including clear descriptions of each product. The next task we had to do was choose a specific series of ball screw. The code and features of each HIWIN ball screw series are shown in the following table (Table 10).

Series Code	Flange	Circulation Tube Within Diameter Measurement	External Circulation	Double nut	Offset pitch (lead) preload	High lead
FSV	•		•			
FSW	•	•	•			
FDV	•		•	•		
FDW	•	•	•	•		
FSI	•	•				
RSI		•				
FDI	•	•		•		
RDI		•		•		
PFDW	•	•	•	•		
PFDI	•	•		•		
OFSW	•	•	•		•	
OFSI	•	•			•	
FSH	•	•	•			•
DFSV	•		•			•
PFDW	•	•	•	•	•	•

Table 10: List of HIWIN ball screw series and their features

The features that each series may include are a flange, circulation ball tube within nut diameter, external circulation, double nut, offset pitch (lead) preload, and high lead. A flange would be located on the nut and can help ensure the reliability of the connection between the nut and the moving part (HIWIN Linear Motion Products & Technology, 2008). In our design, this feature is necessary since positioning accuracy is important and we need to attach the nut to a movable plate above it so that the workpiece can move in all three directions.

A circulation ball tube allows the balls in the nut to continuously rotate throughout the nut, which is essential for achieving smooth movements. These circulation tubes can be located either outside or inside the outer diameter measurement of the nut. A nut with circulation tubes located within the nut diameter keeps the shape of the bracket attaching the nut to the movable plate simpler and smaller, which is an attractive feature since this bracket has limited space in our fixture design. External circulation is when the circulation tubes are located on the outside of the nut and require a slightly larger mounting hole, and thus a larger bracket. The external circulation design has more turns inside the nut, enabling it to withstand larger loads, so it is

actually a better choice when the bracket size is unrestricted. However, our design only provides limited space for the bracket, as mentioned above (HIWIN Linear Motion Products & Technology, 2008).

The next feature is a double nut. A second nut can increase the capacity considerably, but it also shortens the range that the nut can travel on the ball screw at the same time. This is an unnecessary feature since the capacity of a single nut is sufficient but the offset pitch preload eliminates backlash between the ball screw and the nut. This feature is important for open-loop and semi-closed-loop linear driving systems (HIWIN Linear Motion Products & Technology, 2008). Since a closed-loop controlled system is required for the HFG fixture, offset pitch preload is not necessary.

The last feature offered by HIWIN is a high lead. The lead of a screw is the distance between threads and it affects the positioning accuracy, velocity, and output force for a ball screw. A high lead enables faster motion by sacrificing output force (HIWIN Linear Motion Products & Technology, 2008). Considering the requirements of output force up to 1780N and velocity of 50.8 mm/s, the lead should be around 5mm to satisfy both requirements, meaning that the high lead feature is unnecessary. According to the requirements and limitations of our fixture design, we decided to use the FSW series because it offers the proper features.

The next step was to pick the appropriate model for our design. To select the correct model, we needed to calculate the rated capacity required by our fixture design by following the equations provided by HIWIN. This process explained further in Appendix A. Later we compared our calculated rated capacity with the rated capacity of the various ball screw models and picked the model that is capable of performing under the rated capacity required by our fixture design. We selected model FSW20-5B2, which has a rated capacity of 14,896N for the x-

axis and y-axis, which have calculated rated capacities of 10,410N and 12,228N respectively. We chose model FSW20-5B1 for the z-axis because it has a rated capacity of 8,208N, which is sufficient for handling the calculated rated capacity of 5,205N.

7.1.4 Guiding and Supporting Structure

The guiding and supporting structures are essential for creating structural stiffness while allowing precise movements. These components are capable of constraining the motion of five degrees of freedom (DOF) and reducing resistance against the desired linear motion. The requirements of the guiding and supporting structures that will be used in the high force grinder fixture system are shown in Table 11.

	Load applied to the work piece			Stroke	Precision			Lubrication & Dust proof
	X-direction	Y-direction	Z-direction		X-direction	Y-direction	Z-direction	
X-axis	1780N	1780N plus gravity	890N	250mm	<10 μ m			Grease; Dust proof accessories needed
Y-axis				230mm				
Z-axis								

Table 11: Requirements of the guiding and supporting structures

There are two kinds of linear guiding and supporting structures that are widely used in numerical control tooling machines. One of these is linear bearings, which are mainly used in situations with low cutting forces, such as electric discharge machines, engraving machines, and timber working machines. Another option is linear guideways, which are designed to withstand far larger loads than linear bearings. Because of their high capacity, linear guideways are widely used in CNC lathes, milling machines, and grinders.

To choose between linear bearings and linear guideways, we listed the advantages and disadvantages of both options (see Table 12).

	Pros	Cons
Circular linear bearing	<ul style="list-style-type: none"> • Low assembling precision • Low cost 	<ul style="list-style-type: none"> • Long rails result in high deformation • <u>Capacity requirement not met</u> • Poor dust proofing
Linear guideway	<ul style="list-style-type: none"> • <u>Good capacity and stiffness</u> • High-quality dust-proof features 	<ul style="list-style-type: none"> • Requires high assembling precision • Expensive

Table 12: The advantages and disadvantages of linear bearings and linear guideways

Considering the loads that will be applied to the fixture are 1780N, 1780N plus gravity, and 880N in the x, y, and z-directions respectively, we decided to use linear guideways as the guiding and supporting component for all three axes of the fixture because they can withstand these loads.

After investigating multiple manufacturers, we decided to use guideways made by HIWIN, one of the most popular and qualified guideway manufacturers. HIWIN produces many series of guideways that can meet different needs. There are many models that differ in size within each series. Next, we needed to choose a specific series and a specific model. For choosing a series, we considered the typical usages of each one. The typical usages are shown in Table 13. The bolded and underlined words represent usages that are similar to the high force grinder fixture.

Series	Typical usage
HG	Machine Centers, NC Lathes, <u>Grinding Machines</u> , <u>Precision Machining Machines</u> , Heavy Cutting Machines, Automation Devices, Transportation Equipments, Measuring Equipments, <u>Devices Requires High Positional Accuracy</u>
EG	Automation Devices, High-speed Transportation Equipment, Precision Measurement Equipment, Semiconductor Machining Equipment, Woodworking Machinery
RG	Automation Systems, Transportation Equipment, CNC Machining Center, <u>Heavy-duty Cutting Machines</u> , Injection Molding Machines, <u>CNC grinding machines</u> , Plano Miller, <u>Devices Requiring High Rigidity</u> , <u>Devices Requiring High Load Capacity</u> , EDM

Table 13: Typical usages of HIWIN guideway series
(HIWIN Linear Motion Products & Technology, 2008)

Table 13 indicates that both the HG series and RG series can meet the requirement and withstand the grinding loads. After further investigation, we found that the major difference

between the HG series and the RG series is that RG series uses cylinder rollers instead of steel balls to reduce friction. Therefore, the RG series guideways have a larger capacity when compared with HG series models of similar size.

To make a choice between RG series and HG series, we estimated the required rated capacity for the guideways needed for our design. The method we used to do this is explained in Appendix B. The approximate value of rated capacity is around 20kN, which is within the range of the HG series (11.38-208.36kN). Clearly, the rated capacity of the HG series is sufficient to withstand the load, and thus it is unnecessary to use the RG series. This discovery allowed us to select the HG series.

The selection of the model from HG series is based on the equations that involve life span and load (see Appendix B). We choose HGH 25CA for the x and z-axis and HGH 20CA for the y-axis.

7.1.5 Dimensioning

A major task in designing the structure of the machine is dimensioning its components and the spaces between them. This process requires multiple iterations as the size of one part affects the forces and strain on another. To accomplish this task, we first chose preliminary dimensions and then optimized them using calculations and running analyses in ANSYS Workbench.

7.1.5.1 Rail lengths

To start dimensioning the fixture, we first set appropriate lengths for each set of rails because these dimensions only depend on a few factors and are needed to size other features. The factors to determine rail length are the stroke, the span of the supporting blocks, and safety. The required stroke distance is fixed and depends on the rail set. The y and z-rails require a stroke of

230mm and the x-rails require a stroke of 250mm. The next factor is the span of the supporting blocks. Each rail has two supporting blocks that glide along the rail and hold a steel plate. The span of the supporting blocks is the distance from the far end of one of these blocks to the opposite side of the other and is the factor that varies the most. This measurement is highly dependent on the spacing of the blocks, which is explained later in this section. The x, z, and y-rail sets allow 384mm, 334mm, and 277.5mm, respectively, for the factor of the span of the supporting blocks.

The last factor is meant to incorporate safety measures, such as physical stoppers and covers for the rails. Physical stoppers are necessary to ensure that the plates will never go too far and fall off of the machine. Each stopper uses 50mm of rail space on each end, resulting in a total of 100mm per rail. The covers we suggest, which is discussed in the Recommendations section, are able to fit in the 50mm space that the stoppers create because the covers are situated over the stoppers. Therefore, a safety distance of 100mm was used for dimensioning the rails.

After adding these factors up, we chose standard rail lengths offered by HIWIN that could fulfill our needs. This resulted with a rail length of 820mm for the x-rails, 700mm for the z-rails, and 640mm for the y-rails.

7.1.5.2 Spacing of rails and guideway blocks

Choosing the correct distance between the rails in each set and the two supporting blocks on each rail depends on the maximum force on the supporting blocks and the strain in the plate supported by the rails and blocks. The first step was to calculate the maximum force on the supporting blocks for each rail set using Microsoft Excel because these computations are dependent on many variable factors and needed to be recalculated every time our fixture design changed. With the Excel file (Appendix D), we chose optimized the maximum force on the

blocks by adjusting the spacing. Next, we imported the fixture design into ANSYS and optimized the stress and deformation occurring around the rails by once again changing the spacing between the rails and blocks appropriately. Finally, we inputted information for the new, improved model into the Excel file to either repeat the process or verify that the rails we chose are still sufficient. After repeating this process multiple times, the resulting distances between the rails (measured to and from the center of each rail) are 380mm, 320mm, and 220mm for the x-rails, z-rails, and y-rails, respectively. The distance between the supporting blocks on a single rail (measure to and from the center of each block) are 300mm for the x-rails, 250mm for the z-rails, and 200mm for the y-rails.

7.1.5.3 Plates and brace

Besides the length and spacing of the guideways, the other main components are the plates and the brace. In our fixture design, there are five steel plates, which are the x-plate, horizontal z-plate, vertical z-plate, y-plate, and sensor plate. Various factors affect the dimensions of these plates and vary depending on the plate. The x-plate is simply sized just to mount the z-rails on top of it, making it 355mm by 700mm. The horizontal z-plate extends in the x-direction just enough to sit on top of both z-rails. In the z-direction, the horizontal z-plate is wide enough to support the brace and the vertical z-plate, resulting with the horizontal z-plate being 368mm by 340mm. The vertical z-plate is high enough to support the y-rails on top of it, yet the plate extends past the rails in the z-direction. This feature makes the plate as wide as the brace and allows the brace to have a rib behind each y-rail, which stiffens the machine. The dimensions of the vertical z-plate are 640mm in the y-direction and 310mm in the z-direction. For the y-plate, the sizing is dependent on the sensors on top and the guideway blocks underneath. The sensors do not overlap with the guideway blocks for ease of assembly purposes. This results in the y-

plate being 264mm in the z-direction and 260mm in the y-direction. The last plate is the sensor plate, which is used to mount the chuck on top of the sensors. The resultant size of the sensor plate is 180mm by 180mm square. The sensor plate also has a counter borehole with a diameter of 130mm in its center. A disc sits on top of the counter bore and inserts into the back of the chuck. This allows for precise positioning of the chuck.

Finally, we used ANSYS analyses to confirm the dimensions discussed above and to determine the size of the brace and the plate thicknesses. The dimensions of the brace are 550mm in the y-direction, 280mm in the x-direction, and 340mm in the z-direction. Generally the brace is 40mm thick except for the ribs. As for the plates, they are all 30mm thick, except for the sensor plate, which is only 20mm thick.

7.1.5.4 Spacing between the plates

In order to fit the transmission system between the plates, the guideways needed to be mounted higher. To do this, we placed steel bars underneath the z and y-rails and welded them to the plate supporting the rails. These steel bars are as long as the rails they were supporting, have a width of 40mm, and raise the rails 47mm away from the supporting plates. This final dimension was chosen to make room specifically for the bracket holding the ballscrew nut. Also, the steel bars are design with a 3mm high and 10mm wide lip to easily locate the rail position. For the x-rails, the pedestal is cast iron and has a unique shape to raise the x-rails enough to provide adequate space without requiring steel bars.

7.1.6 Analysis and Optimization

In order to optimize and analyze the stiffness of our fixture design, the design team used ANSYS Workbench specifically, a software which has a friendly user-interface and is used for designing, simulating, and optimizing.

7.1.6.1 Analysis Process

The design team followed a simple process for optimizing our fixture design. First, we simplified the model in SolidWorks, and then imported it into ANSYS to run tests. If some weak parts were identified, we made the necessary improvements in SolidWorks and then put it into ANSYS again to repeat the tests. If the result met our requirement and/or the structure could not be efficiently optimized further, the process was finished. If not, we repeated the steps until we reached one of the conditions mentioned before.

In order to conduct a test in ANSYS, preparation was necessary. The first step of the preparation was to simplify the model, as mentioned before. This was an important step because it greatly influences the analyzing time and the accuracy of the results. To simplify the model, the design team suppressed unimportant features, such as holes and chamfers. Also, we used simple shapes like cylinders to replace complex components like the chuck and the motors. These simple shapes were created to have the same size, material, and weight as the components they replaced.

The second step included fixing the position of the chuck at the most remote, yet still realistic, grinding position. Fixing the position of the chuck at a designated point allowed us to compare several tests with each other since the testing conditions were consistent. The remote position was selected to ensure we were evaluating our fixture design in the worst case scenario.

Finally, we imported the SolidWorks model into ANSYS and set conditions in order to analyze the structure. The conditions include the connections between parts, the materials, the fixed surface, and the location and magnitude of the applied forces.

7.1.6.2 Optimization of Concept 3

The first model that we tested was an extremely simple version of Concept 3 that did not include any linear driving mechanisms. For the ANSYS analysis of this model, we had to set the testing conditions as described above. For measuring the displacement, we fixed the bottom of the pedestal and applied forces with a magnitude of 1780N (400 lbs) in the x-direction, 1780N (400 lbs) y-direction, and 890N (200 lbs) in the z-direction. Even though the grinding forces are not at these maximum values all the time, we used the biggest force in order to get the worst possible deformation. We applied these forces over a circular area with a diameter of 10mm located in the middle of the chuck to represent the grinding point. Lastly, we imposed the effects of gravity over the entire structure. The result of the deformation of this initial model for Concept 3 is shown in Figure 16.

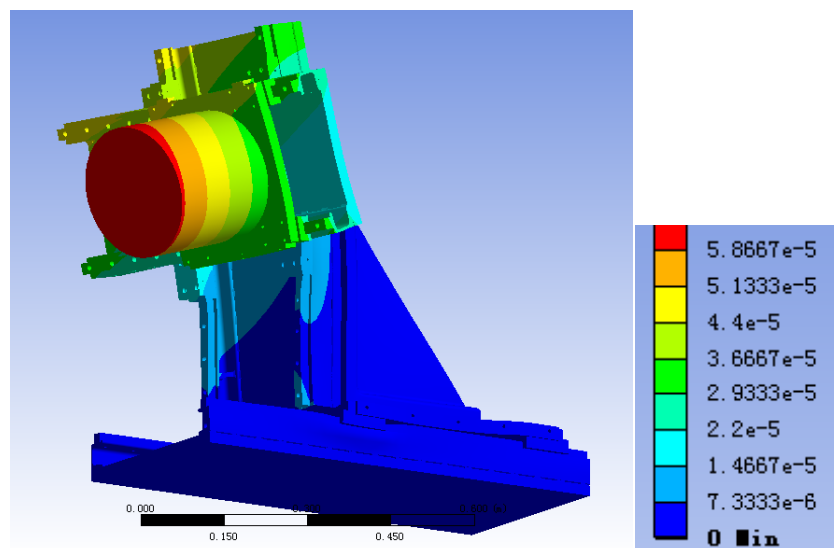


Figure 16: ANSYS analysis of the initial model for Concept 3

The purpose of testing this extremely simplified model, which lacks important features such as ball screws and joints, was to evaluate the various structural components and identify locations of high stress concentration. As seen in Figure 16, the max deformation is about $65\mu\text{m}$, which is much more than the requirement, indicating we have to optimize the whole structure. When analyzing the distribution of the strain, stress, and deformation of each part, several weak points stand out. First is the C-channel, whose long, thin, and open structure is too weak to resist the torsion and bending inflicted on it effectively. As shown in Figure 17, the C-channel is greatly distorted. The second is the horizontal x-plate (seen in the top right of Figure 17), which is much longer than the cast iron brace, causing the plate to provide more support than the guideway block. This is an inefficient design because it causes the plate to support the brace instead of the blocks, which are designed to handle such high loads. The third is the vertical x-plate (bottom left in Figure 17), which is underneath the y-rails, is not supported well enough, causing it to bend. The fourth is the mounting method between the y-guideway blocks and the y-plate (bottom right in Figure 17). Here, only a strip of metal that was only partially covered by the y-plate was used to connect the block to the plate. High levels of strain occurred at the point where the y-plate ended because the force is applied to a small area.

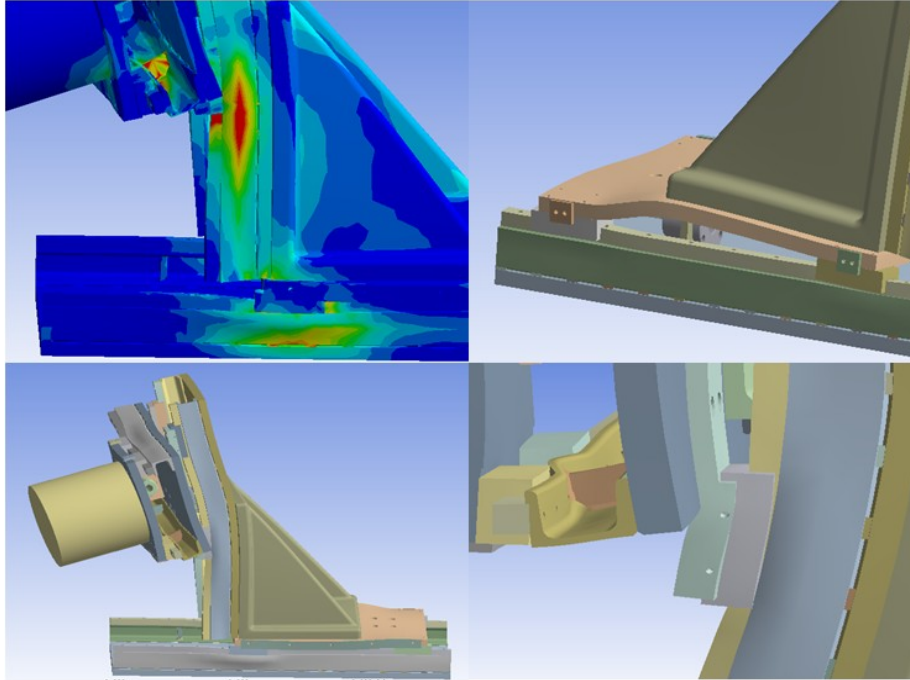


Figure 17: Problems of the initial Concept 3 model

To improve these weak parts and others, we changed either the material used or the shape of the part. First, we replaced the C-channels with solid steel bars, which increased stiffness, but also increased mass. Second, we increased the height and length of the brace and aligned the ribs with the y-rails. Increasing the height provides more support to the vertical x-plate. With the brace being longer, grinding forces are applied directly onto the x-guideway blocks instead of on the middle of the plate. Finally, putting the ribs behind the y-rails provides more support without increasing the size or weight of the brace since the grinding forces are transmitted to the brace through the y-rails. Third, we changed the steel pedestal and the C-channels for the x-rails to be one cast iron part. Fourth, we increased the height of the y-plate to ensure that the blocks on the rails were completely covered. This expanded the area which the grinding forces are applied and reduced the stress concentration. After making these improvements, we tested the model in ANSYS again, and the result is bellow in Figure 18.

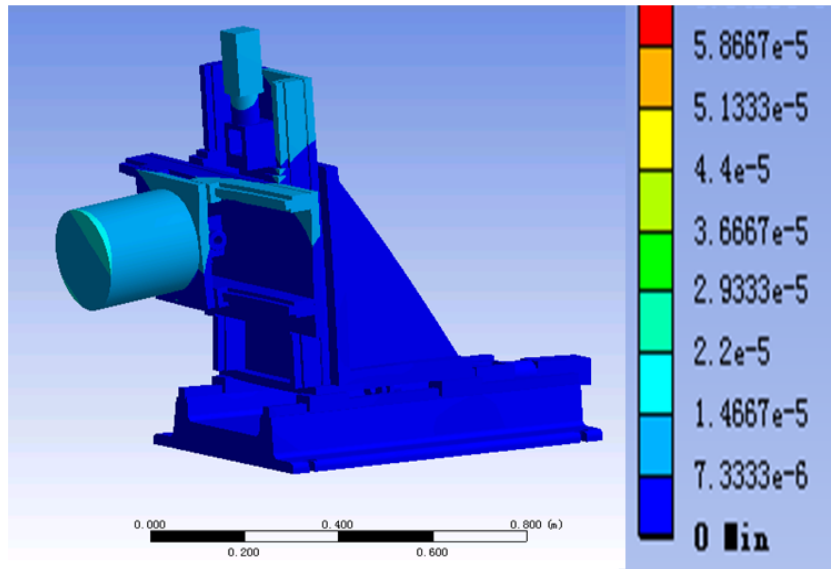


Figure 18: Result of the improvements made

Since we used the same color scale to represent the deformation, we can see clearly that the new model is drastically better with a max deformation is about 14.5 μm . Even though reducing the deformation from 65 μm to 14.5 μm was relatively easy, the design team had a lot of trouble optimizing the design any more to further increase the stiffness of the overall structure. During one attempt, for example, we once tried to make the whole brace solid, which means we filled the space between the two ribs. The calculated max deformation only improved by about 1 μm , yet the weight of the entire machine doubled. We also tried inserting ribs to help support certain structures. For example, we mounted two ribs on the section of the y-plate that extended outside of the z-rails in order to support the solid steel bars underneath the z-rails. Each rib was put directly above each y-rail as an attempt to stiffen the part under the most strain. However, these ribs only reduced some stress on the extended part of the y-plate and made no difference to the overall stiffness. With the results of making these changes, we realized that we had optimized the structure as best we could since additional alterations greatly complicated the design while barely stiffening the fixture.

7.1.6.3 Analysis accuracy

After communicating with Professor Rong and Dr Wang Jinsheng, we discovered that the method for testing the deformation outlined above may be overestimating the stiffness. This is because when designing, we merely made all the connections to be bonded, when in reality, the connections are very complicated. For example, the connections between the blocks and the rails can resist perpendicular forces, but not the force parallel to the rails. This force can only be counteracted by the ball screws. In order to obtain more realistic results, we changed the connection settings between the rails and blocks to “no separation,” which means the two parts cannot move away from each other but they can slide against each other.

After these changes, we found that the maximum deformation, which was newly calculated to be 65 μ m, occurred at the ball screw. The ball screws have large axial deformation, which contribute greatly to the deformation of the whole structure. On the fabricated fixture, there will be position sensors that will detect any axial deformation (the directional deformation of the ball screw that is parallel to its axis). Therefore, it is possible that the axial deformation does not need to be included in the calculations for determining the total deformation of the fixture. However, since our experience with ANSYS is limited and we cannot find an accurate way to exclude the axial deformation, we decided to use the ANSYS analyses comparatively. This means that we used ANSYS to determine which structure, out of two options, is stiffer, but we do not claim that the numeric results of each test is an accurate measurement of the true deformation. The numeric results were solely used to give us insight into optimizing the structure and to compare different configurations. This topic is discussed further in Appendix G.

7.1.6.4 Optimizing Concept 2

After finishing the analysis of Concept 3, we started to dimension and optimize Concept 2. Similar to Concept 3, we first created a model with rough estimates for the distance between the rails and the spacing of the guideway blocks. Then, we calculated the force on each block to determine if these dimensions are reasonable, made changes as necessary, and imported the model into ANSYS.

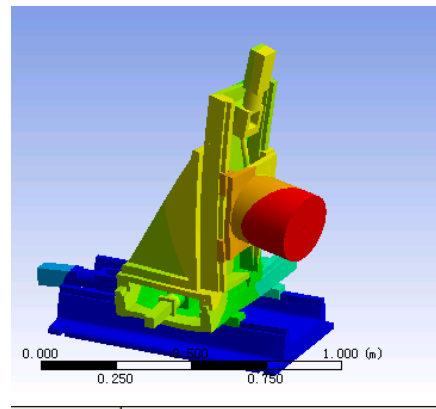


Figure 19: Results of initial analysis of Concept 2

For the purpose of optimizing the structure for stiffness, we tested the model in ANSYS and reviewed the results to find the weak points of our initial design for Concept 2. In Figure 19, one can see that the y-plate and the x-plate bend significantly. The reason for the bending of the y-plate is that the y-plate is much bigger than the chuck. Therefore, the grinding forces are transmitting through the chuck and acting on the middle of the plate instead of on the guideway blocks. However, the mechanical design group knew that force sensors needed to be mounted to the y-plate and might require it to remain larger than the chuck. For this reason, we did not change the size of the y-plate significantly.

When studying the effects on the x-plate, we noticed a small dip in the middle of the plate between the supporting points of the x-rails. Thinking that the distance between the x-rails was

too large and causing the deformation, we decreased the distance from 380mm to 350mm, expecting the deformation to improve. However, the results were different than we expected with only a 3 μ m increase in the total deformation. Still not convinced with these results, we feared that if the structure was located in the center of the z-rails, instead of being located off center and over one of the x-rails, that the dip in the x-plate would grow larger since the forces would be applied more centrally. We fixed the workpiece to be located in the middle of the x-rails and ran another ANSYS test. The results showed that the total deformation improved when the machine is located centrally, which makes sense and is consistent with our original reasoning for locating the grinding point off center for our stiffness analysis. Finally convinced, we left the distance between the x-rails to be 380mm. We ran similar tests to determine the best distances between the x-blocks, the z-rails, and the z-blocks. The largest improvement made from these sets of tests was increasing the distance between the z-rails from 313mm to 340mm. Other optimizations were required, such as enlarging the brace, in order to obtain our final model for Concept 2.

7.1.6.5 Comparisons Between Concepts 2 and 3

The goal of the next set of tests was to compare Concept 2 with Concept 3. To obtain the most accurate results, we set the characteristics, which are shared by both configurations, to be the same. These characteristics included materials, plate thicknesses, position of the grinding point in relation to the pedestal, the distances between the ball screw nut and the ball screw housing on all three axes, and connections between surfaces.

The position of the grinding point was fixed in the x-direction so that the distance between the front block and the front of the pedestal was 100mm, in the y-direction so the grinding point is in the middle of the y-rails, and in the z-direction so the grinding point is 115mm away from the middle of the z-rails at the extreme end of the stroke. After importing the model into

ANSYS, we change the materials so that most parts were structural steel, which has similar qualities to 1050 steel. Parts that were assigned different materials include aluminum housings to hold the motor and ball screw, the cast iron brace, and the cast iron pedestal. The chucks and motors were assigned custom materials in order to make them the correct weight. To connect the various surfaces in the model, we set the connections between the blocks and the rails to be “no separation.” All other connecting parts were bonded together.

To determine which concept is better, we decided to do two comparisons between Concept 2 and 3. For the first, we set the every plate in Concept 2 (y-plate, vertical z-plate, horizontal z-plate, x-plate) and Concept 3 (z-plate, y-plate, vertical x-plate, horizontal x-plate) to be 20mm thick. For the second comparison, every plate was 30mm thick. The result of both comparisons is shown in Figure 20.

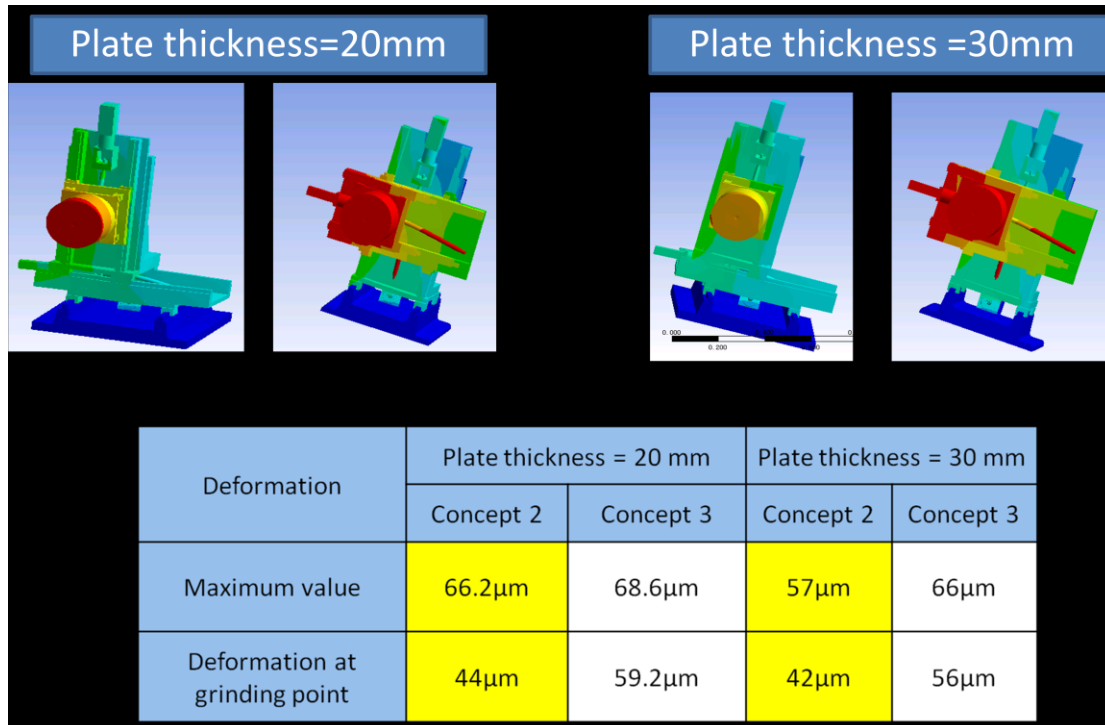


Figure 20: Results of the comparative analysis

It is easy to see that Concept 2 was better in every situation. At first, the difference between the total deformations of both concepts did not differ significantly. However when the plate thicknesses were increased to 30mm, Concept 2 showed great improvement, whereas Concept 3 did not. Also, it is clear that the deformation at the grinding point is much better in Concept 2 than in Concept 3. Since thicker plates seemed to be beneficial for Concept 2, we considered making the plates for this concept to be 40mm thick to see if the stiffness would improve more. The deformation of Concept 2 with 40mm thick plates was so similar to the deformation with only 30mm thick plates, that we believed it to be unnecessary to increase the plate thicknesses since it would needlessly add cost and mass to the fixture.

Since all the tests were done with the ratio of the normal force to the tangential force being 1:1, the design team was concerned that this produced an unreasonably favorable condition for Concept 2 since normal grinding conditions usually have a normal to tangential force ratio of 3:1 (Xiong, Yan, & Zhang, 2006). We quickly tested both concepts again, except this time with a force ratio of 3:1. The force distribution used for this test was 1780N normal force, 593N tangential force, and 780N axial force. The results showed that Concept 2 is better even in this situation.

When all of the comparative testing is considered, Concept 2 clearly outperforms Concept 3 in stiffness. Therefore, we chose our final design to be Concept 2 with 30mm thick plates.

7.1.7 Accessories

The design team has found two other items to our fixture design. The first one is protection for the rails and the second is protection for the operator.

7.1.7.1 Rail Protection

We considered two methods for protecting the rails from grinding particles: wipers and covers. The rail wipers would be custom design and fit on either end of the guideway blocks. As the blocks moved over the rail, the wipers would simply push the grinding particles out of the way. We inquired about the products called Way Wipers (shown in Figure 21) offered by KabelSchlepp America, Inc. These products included an aluminum frame that is 4.5 mm wide and a height in the range of 18mm to 40 mm (Kabel Schlepp *Accessories*). The frame holds a polyurethane strip against the rail that pushes particles and fluids away (Kabel Schlepp *Guideway*).



Figure 21: Way Wipers
(Kabel Schlepp Accessories)

The second option is the covers (Figure 22), which would have material resistant to hot particles covering both rails and the space between them. One end of the covers is attached to the end of the rails and the other end would be attached to the plate moving over the rails (Gortite, 2008). The design team preferred this method because it is simpler than the wipers and it prevents the grinding particles from ever touching the rails, a quality that is beneficial for keeping the rails optimally lubricated. Another benefit is that these covers will also protect the components in between the rails, such as the motor, ballscrew, and position sensor. Finally, this option makes the machine easier to clean because the grinding particles cannot fall into any

small crevasses within the mechanical structure and can be simply wiped off of the top of the covers.



Figure 22: Covers on a machine
(Gortite, 2008)

For our design, we suggest using six covers in all in order to completely cover the rails on both ends. We investigated and obtained quotes from two companies: Centryco and Gortite. Both companies provide the dimensions and an estimated price for the covers needed in our design. The total price of all six covers from Centryco is 832.64 USD (5688 Yuan), whereas Gortite offers all six for 750 USD (5124 Yuan). Therefore, we chose the Centryco covers as the best option.

7.1.7.2 Safety Shields

When the project team first visited the Saint-Gobain Research Shanghai (SGRS) workshop, we instantly noticed that there was nothing to restrict grinding particles from flying away from the workpiece and hitting the operator. Seeing this as an area needing improvement, we researched available safety shields online and discovered the company called OMRON Scientific Technologies, Inc, which offers a variety of transparent safety shields for various machine setups. All of the safety shields are comprised of polycarbonate capable of withstanding

impact. The shield configurations range from a simple square cut of polycarbonate supported by a single stand to polycarbonate walls supported by an aluminum frame with swinging and sliding doors.



Figure 23: Flexible-Arm Shield
(ORMON Scinetific Technologies, Inc.)

The project team suggests that Saint-Gobain incorporates the simple Flexible-Arm Shield with a magnetic base shown above in Figure 23. This design uses a flat $\frac{3}{16}$ inch polycarbonate shield, with sizes ranging up to 305mm by 305mm (12in x 12 in), to deflect airborne grinding particles. The flexible-arm is comprised of spring steel protected by a vinyl outer surface. The circular magnetic base has a 76.2mm (3in) diameter, adheres to any flat ferrous surface, and can be easily moved by the operator (ORMON Scinetific Technologies, Inc.). We believe this is the best option because it is the simplest choice (presumably indicating a low price), it has high adjustability and can be positioned to block the most amount of airborne particle, and it can be easily moved out of the way during workpiece change outs.

7.1.8 Final Overall Design

After selecting the best options for the various components and determining the dimensions discussed above, we completed SolidWorks model to represent all of our decisions. Figure 24

shows our final SolidWorks model. It is easy to see how the model reflects Concept 2 with the X-Z-Y rail configuration, since the x-rails are shown in blue, the z-rails in red, and the y-rails in green.

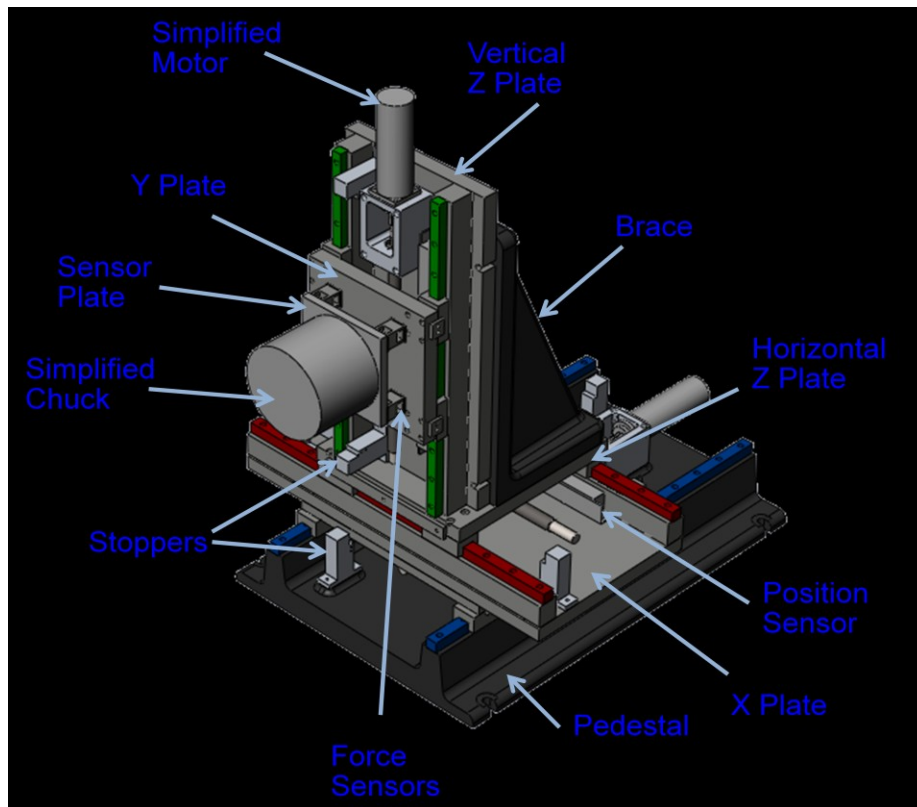


Figure 24: Final SolidWorks model of the fixture design

Besides the components already discussed in the Mechanical Design Results section, the model also includes housings, the motors, force sensors, and linear sensors. The housings are aluminum boxes designed by the mechanical design team to support the ball screw and motor and to ensure they are in line with each other. The motors and sensors are control components that affected the mechanical structure. The motors added weight to the structure, which needed to be considered during analysis. The sensors needed to be properly placed in order to perform optimally. The position sensors are parallel to the ball screw, as seen in Figure 24, to detect all movement in the specified direction. The force sensors are placed behind all four corners of the

sensor plate in order to accurately detect and measure the grinding forces. With this final model, we were able to produce two-dimensional drawings and supply them to Saint-Gobain for when they decide to fabricate a prototype.

7.2 Controls Design

Before starting the design of the control system our team made an overall system diagram that outlines the important components of the system.

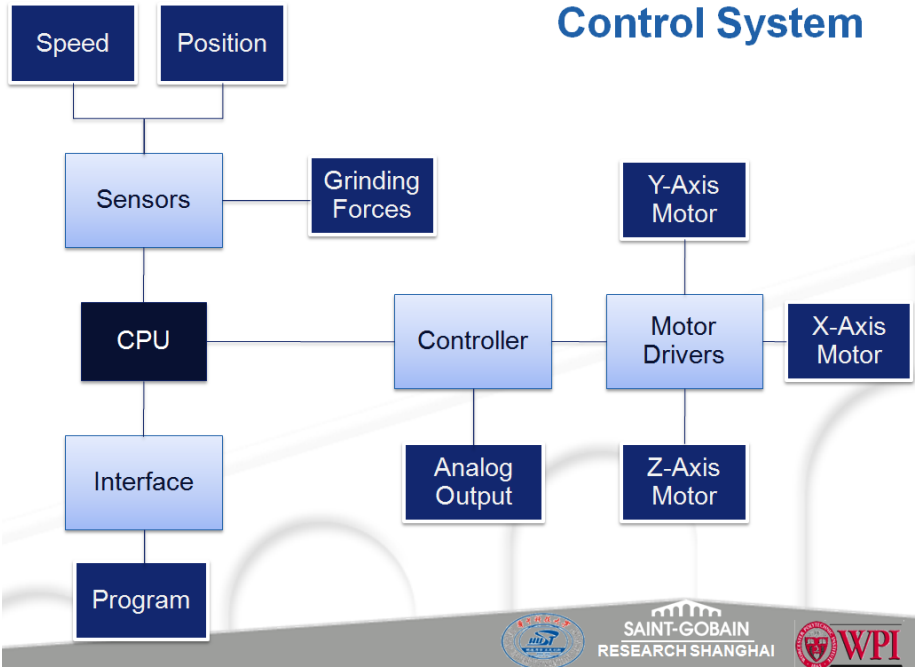


Figure 25: Overall System Diagram

Figure 25 contains the main components of our system and the things that they need to have. In the center of the system is the central processing unit (CPU) that manages all of the system operations. The sensors acquire essential information of the system such as speed, position, and grinding forces information and send it over to the CPU. The CPU analyzes these signals and then sends commands to the controller which controls the motor drivers that gives power to the three motors. The interface is used to get input data from the used and display information to the user.

7.2.1 Motor selection

Motors provide driving force to the fixture system; the proper choice and placement of motors is essential to the performance of the fixture. One of the requirements for the control system of the fixture is to provide constant feeding force to the three axes. Therefore the selected motors should be able to output constant torque. Ordinary AC and DC motors, and step motors require torque sensor feedback to keep feeding force constant. Servo motors on the other hand have built-in encoders that are able to keep constant torque when the servo speed is lower than the rated speed. Due to the integrated function of constant torque mode, it is easier to control both the torque and the speed of servo motors compared to other electrical motors. (Chen & Wang, 2005) To choose the motors we looked at different ratings of each motor. Ratings such as inertia, torque, speed, and power were all considered in our selection.

Servomotors are connected with ball screw directly or indirectly through gearboxes. Servomotors can output constant torque when they are used in the constant torque mode and torque can be converted to force through the ball screws. The accuracy of torque (torque ripple)

in constant torque mode is about 1%. Accordingly, the accuracy of the feeding force is at most 1%.

Servo motors are able to provide enough driving force that is required to move the fixture. We specify the servomotor performance requirements based on our calculations which are available in the Appendix C and can be taken as reference. The following performance specification must be met:

$$1 \geq \frac{J_L}{J_M} \geq 5$$

$$T_{\text{rated}} \geq T_L$$

$$T_{\text{max}} \geq T_m$$

$$N_{\text{rated}} \geq N_{\text{max}}$$

$$P_{\text{rated}} \geq P_L$$

Where

J_M is the rotor inertia moment,

J_L is the full load inertia moment converted into equivalent on the motor shaft,

T_{rated} is the rated torque of the servomotor,

T_L is the continuous load torque converted into equivalent on the servomotor shaft,

T_{max} is the maximum torque of the servomotor,

T_m is the maximum load torque into equivalent on the servomotor shaft from the load torque profile converted,

N_{rated} is the rated rotational speed of the servomotor,

N_{max} is the maximum load speed converted into equivalent on the servomotor shaft,

P_{rated} is the rated power of the servomotor,

P_L is the load power on normal working conditions.

There are other factors which should be taken into consideration in motor selection. Factors such as degree of protection, size of motors, type of construction, encoder system, length and cross-section of power cable. These factors later become important because motor suppliers often offer a big array of choices and these specifications helps to narrow down our choices.

Our first step in the motor selection process was to work on the performance requirements of the motors.

One of the most important requirements for the motor was the clear rail configuration given to us by the mechanical design team. The clear rail configuration affects the force and inertia moment distribution of the fixture. Their first option for the chosen clear rail configuration was for Concept 3. Concept 3 did not work as was expected therefore the clear rail configuration was changed to Concept 2.

We first selected the servomotors based on the third rail configuration given to us by the mechanical group.

To achieve the requirements we had two options, to use reduction gearboxes as shown in Figure 26 and to not use it as shown in Figure 27.

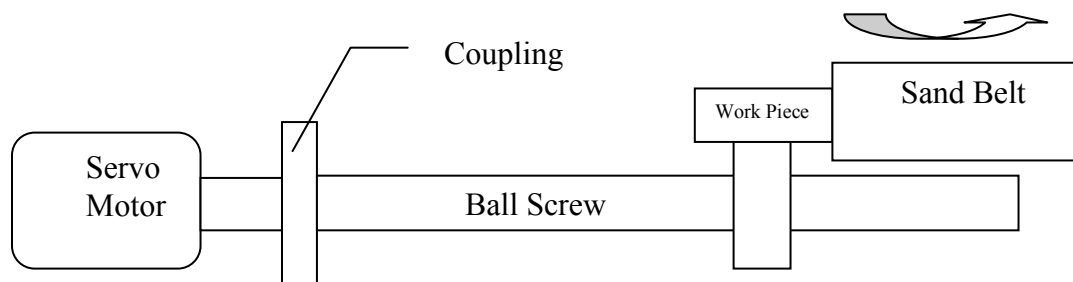


Figure 26: Driving model with no gearbox



Figure 27: Driving model with gearbox

The requirements converted into equivalents on the servomotor shaft are different according to our choice of reduction gearboxes usage. Reduction gearbox lowers the speed of the output rotor while magnify the output torque. We first calculated the requirement equivalents without using reduction gearbox. After that, we selected a gearbox with a reasonable reduction ratio to formulate another possible solution. Finally, the two options were compared to select the best solution. The calculation process for the selection of motors using the motor requirements can be found in Appendix C.

After going through all the calculation process we came up with two tables for each axis that included the best motor options with and without gearbox. These tables allowed us to compare all options side by side; we were then able to select the best options using the tables.

The following tables show our process of motor selection for Concept 3. The motor selection tables for the X-axis are as follow:

Performance Property	X-PGM	MSMD/750W	SM80-024-30LFB
Load Torque-- N· m	2.41	2.4	2.4
Max Torque-- N· m	3.86	7.1	7.2
Max Torque for selection (Safety=1.5) -- N· m	5.80		
Inertia Moment for selection--Kg· m ²	(0.46~2.27)*10 ⁻⁴	0.97	1.06
Max speed-RPM (Rated/Max)	609.6	3000/5000	3000
Max Power-W	153.6	750	750

Price--RMB	-----	6300	3200
Evaluation		All requirements met; High cost	All requirements met; Low cost

Table 14: X-axis no gearbox

Performance Property	X-PGM Requirements	MSMD/200W
Reduction Ratio	4	
Load Torque—N·m	0.61	0.64
Max Torque—N·m	0.70	1.091
Max Torque for selection (Safety=1.5) -- N·m	1.04	
Inertia Moment for selection—Kg·m ²	(0.03~0.14)*10 ⁻⁴	0.14*10 ⁻⁴
Max speed-RPM (Rated/Max)	2439	3000/5000
Power-W	153.6	200
Price—RMB	-----	3900
Gear Box		Taiwan VGM
Price—RMB		1377
Total Price—RMB		5277
Evaluation		All requirements met; reasonable cost

Table 15: X-Axis with gearbox

As can be seen from Table 14 and 15, the servo motor SM80-024-30LFB meets all the requirements for the X-axis and has the lowest cost of 3,200 RMB therefore we chose this motor to drive the X-axis. The motor selection tables for the Y-axis are as follow:

Performance Property	Y-VGM	MSMA/2000W	SM 130-060-25 LFB
Load Torque-- N·m	5.07	6.36	6
Max Torque-- N·m	6.52	19.1	12
Max Torque -- Safety=1.5-- N·m	9.78		
Inertia Moment for selection--Kg·m ²	(0.46~2.27)*10 ⁻⁴	3.46*10 ⁻⁴	12.6*10 ⁻⁴
Max speed-RPM	609.6	3000/5000	2500
Max Power-W	323.3	2000	1500
Price--RMB	-----	7800	4500
Evaluation		Motor inertia moment is a little big	Motor inertia moment is too big

Table 16: Y-axis no gearbox

Performance Property	Y-VGM	MSMD/400W	SM 130-040-25LFB
Reduction Ratio	5	5	5
Load Torque-- N· m	1.02	1.3	1.0
Max Torque-- N· m	1.07	3.8	3.0
Max Torque for selection (Safety=1.5) -- N· m	1.61		
Inertia Moment for selection--Kg· m ²	(0.02~0.09)*10 ⁻⁴	0.26*10 ⁻⁴	8.5*10 ⁻⁴
Max speed-RPM (Rated/Max)	2439	3000/5000	3000
Power-W	323.3	400	400
Price--RMB	-----	4300	4200
Gear Box		Taiwan VGM	PL120-5
Price--RMB		1377	2700
Total Price--RMB		4677	6900

Table 17: Y-axis with gearbox

Tables 16 and 17 shows that it is the motor MSMD/400W with a 5 reduction ration gearbox is the best option for the Y-axis because it has a comparatively low price and meets all the requirements. The motor selection tables for the Z-axis are as follows:

Performance Property	Z-HGM	MSMD\750W	SM80-024-30LFB
Load Torque-- N· m	1.61	2.4	2.4
Max Torque-- N· m	2.33	7.5	7.2
Max Torque for selection (Safety=1.5) -- N· m	3.49		
Inertia Moment for selection--Kg· m ²	(0.46~2.27)*10 ⁻⁴	0.87/0.97	1.06
Max speed-RPM (Rated/Max)	304.8	3000/5000	3000
Max Power-W	51.1	750	750
Price--RMB	-----	6300	3200
Evaluation		All motor requirements satisfied. Not working under the rated point	All motor requirements satisfied. Not working under the rated point

Table 18: Z-axis no gearbox

Performance Property	Z-HGM	MSMD/100W	SM 80-013-30 LFB
Reduction Ratio	10	10	10

Load Torque-- N· m	0.16	0.32	1.3
Max Torque-- N· m	0.17	1.091	3.5
Max Torque for selection (Safety=1.5) -- N· m	0.26		
Inertia Moment for selection--Kg· m ²	(0.03~0.14)*10 ⁻⁴	0.14*10 ⁻⁴	0.61*10 ⁻⁴
Max speed-RPM (Rated/Max)	3024	3000/5000	3000
Power-W	52	100	400
Price--RMB	-----	3740	3000
Gear Box		Taiwan VGM	PL80-10
Price--RMB		1377	1600
Total Price--RMB		5170	4600
Evaluation		All requirements satisfied	Motor inertia moment is over requirement

Table 19: Z-axis with gearbox

As can be seen from Table 18 and 19, the motor **SM80-024-30LFB** meets all requirements and has the lowest price. After choosing the rail configurations for Concept 3 we had to go through the same process for Concept 2.

Performance specifications	Motor Requirements	Panasonic MSMD/200	Motor Requirements	YASKAWA SGMAJ-02A
Rated Torque(N*m)	0.47	0.64	0.47	0.637
Maximum Torque(N*m)	0.95	1.91	0.95	2.33
Rotor Inertia Moment(Kg*M ²)	0.13-0.64	0.16	0.06-0.28	0.26
Rated Speed(RPM)	2438	3000	2438	3000
Rated Power(w)	132.9	200	132.9	2000
Gearbox Name	-----	Taiwan VGM PG 60-4	-----	NEUGART PLE 40-4
Total Price		5277		8700

Table 20: X-axis Concept 2

Table 20 shows the most important performance factors that were taken into consideration while choosing the motors Concept 2. Only motors that satisfied all the factors were chosen.

Selection	X(PGM)	Y(VGM)	Z(HGM)
Load(N*m)	0.47	0.47	0.12
First Option	Panasonic MSMD/200	Panasonic MSMD/200	Panasonic MSMD/50
Second Option	YASKAWA SGMAJ-02A	YASKAWA SGMAJ-02A	YASKAWA SGMAJ-A5A

Table 21: Final motor selection

After doing all comparisons we decided to use the servomotor MSMD/200 for both the X and Y axis, Z axis uses the MSMD/50 servomotor all of which are from Panasonic Company.

7.2.2 Sensors

To meet the precision and control requirement of the new machine, we researched and analyzed sensors that are able to measure grinding forces, and the speed and position of the worktable. This section goes over the selection of all the sensors needed for the control system.

7.2.2.1 Grinding Force Measurement

Currently in home and abroad the grinding forces are usually acquired by using dynamometers (Li, 2001). There are different kinds of dynamometers, such as mechanical, hydraulic, capacitive, inductance type, resistance strain gauges and piezoelectric crystal. Mechanical and hydraulic dynamometers can only measure average grinding forces because of their low rigidity and high inertia. Capacitive and inductance type dynamometers are able to measure the instantaneous value of grinding force, but because they are confined to their

complex structure when measuring multiple component forces, they are harder to work with. Hence, these four dynamometers are not applicable to measure grinding force in our system.

Strain gauges are used to measure the strain on an object. Widely used in aviation, ships, mechanics and architecture; strain gauges are small in size (Kang, 2006). They require less space on the mechanical system. Fast dynamic response is one of their characteristics, which allows them to measure the grinding force in real time with little delay. They have a high measuring accuracy and are easy to use.

Resistance strain gauges dynamometer is a kind of resistance dynamometers (Li, 2001). When combined, multiple strain gauges form an electric bridge. While grinding, strain foils will be in tension or compression, changing their electric resistance. This creates a change in the resistance of the bridge hence changing its output voltage. By using the calibration curve between grinding force and output voltage, the value of grinding force can be acquired. Resistance strain gauges dynamometers have high sensitivity, large range and high accuracy, but they are not precise while measuring grinding force with changing action point.

Piezoelectric crystal dynamometer has high sensitivity, accuracy, rigidity, natural vibration frequency, good linearity, noise immunity and no inertia (Li, 2001). It is especially applicable to measure dynamic force and instantaneous force. Currently, the widely used piezoelectric crystal dynamometers most come from a Swiss company-Kistler. The basic working principle is described as below. When the dynamometer is applied by external forces, the internal quartz crystal will have directive piezoelectric effect, then piezoelectric sensors and charge amplifier are used to output charge, and we can measure the force by measuring the output charge.

But there is an obvious disadvantage for piezoelectric crystal sensors, they will cause charge leakage in continuous cutting process, which will cause zero drift and influence measuring accuracy (Li, 2001).

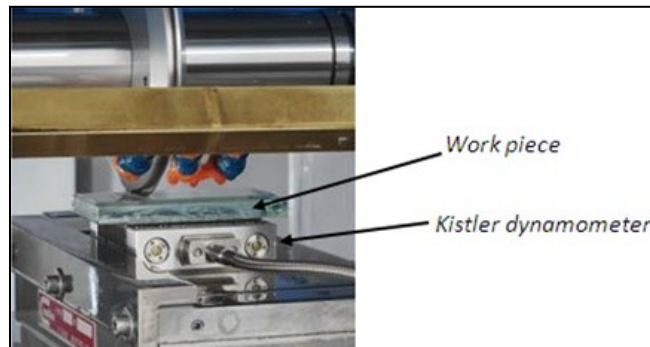


Figure 28: Kistler Dynamometer

Figure 28 shows where Kistler dynamometer is placed during normal use. Kistler dynamometer is usually put between work piece and worktable or chuck. Consisting of piezoelectric sensors, Kistler dynamometers have a high natural vibration frequency of 500 - 6000 Hz (Liu, 2009). Their accuracy is of 0.5%, compared to other piezoelectric crystal dynamometers they have a higher accuracy. They have a resolution of milli Newton magnitude and a price of 200,000 – 300,000 RMB. When they are placed, the surface contacting Kistler dynamometer should be grinded to meet the requirement for contacting. Each built-in sensor in the Kistler dynamometer has a limit of moment for it to withstand. Therefore there is a requirement for the distance between the grinding point and the measuring surface of the dynamometer (L). For most products of Kistler, L should be within 200mm. Usually, Kistler dynamometer is placed between work piece and chuck or worktable. According to our design, the chuck is designed with a special structure to adapt to different shape of work piece, making it unable to place the Kistler dynamometer between the work piece and chuck. It can only be

placed after the chuck. Considering the thickness of the workpiece, L is over 200mm. Kistler dynamometer does not meet all of our requirements.

Three component grinding dynamometers from Dalian University of Technology (DLUT) use piezoelectric sensors to measure grinding forces (Qian, 2009). It is usually placed between the chuck and the workpiece. They have a natural vibration frequency of 3 KHz. Compared to Kistler dynamometer it has a lower accuracy of one percent and resolution of one Newton at a cheaper price from 180,000 RMB.

The sensor block from Exact Sensor Company uses strain gauges placed between the chuck and the Y-plate to measure the grinding forces (Wang, 2009). Their accuracy is from 1-3% and resolution is of 0.2 Newton with a price of 9,000 RMB.

We custom designed four blocks to strategically place the strain gauges on them. When rightly placed, the four blocks and the strain gauges are able to measure the three components of the grinding forces. The four blocks are placed between the chuck and Y-plate. We use strain foils to detect the strain caused by grinding force, then we use dynamic strain indicator to acquire output signals of electric bridge formed by strain foils and process them and transfer them to computer for displaying.

We narrowed down our options of strain gauges to two. One option was the four blocks method in which we need to decide the dimensions of the block and how to place the strain gauges, the other one was using a design provided by Exact Company. In our own design, as we choose strain foils and dynamic strain indicator, if we don't consider the errors related to strain foils, then we can achieve an accuracy of 0.1% after calibration and a resolution of 24bits. Besides, it is easy to transfer the data from dynamic strain indicator to computer. According to

the discussion above, we can get the following decision matrix and we choose the four blocks method.

	Accuracy	Resolution	Price	Compatibility	Calibrating	Total
Full Score	4	6	8	6	4	28
Kistler	4	6	4	3	4	21
DLUT	3	5	6	0	3	17
Exact	1	5	8	6	2	22
Four blocks	2	6	7	6	2	23

Table 22: Summary of Sensors

Considering how we place the strain foils, we have two general ideas. One is we place the strain foils on the mechanical structure directly. The other is we design a measuring block, then put the strain foils on it and connect the block to the mechanical structure. Because it is hard to find a good position on the mechanical structure to place the strain foils we need to design a block. The block could only be placed between the chuck and Y plate. Usually the grid length of a strain foil is 2~3mm. When you use strain foils to measure force, the radius of curvature of the measuring area should be at least 5 times that of the grid length. On the other hand, we don't want to have more moment on the measuring block; otherwise, it will result in a deformation of the whole mechanical structure. Because there are three component forces to measure, we need to build a special block.

The whole measuring system for grinding force consists of two big parts-hardware and software. On hardware, we need four specialized steel blocks, 24 strain foils and one dynamic strain indicator. As to software, we need a special program to process the data and display them.

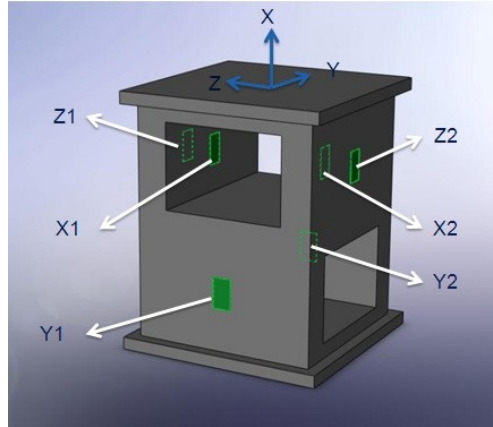


Figure 29: Strain gauge positions on a measuring block

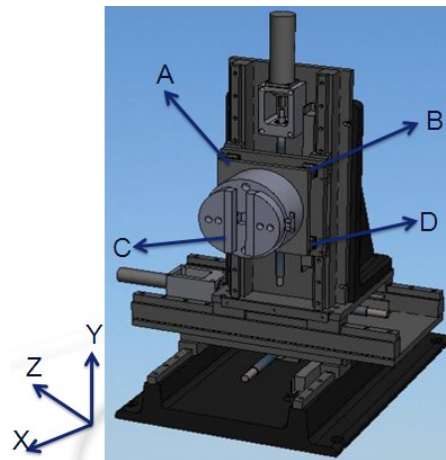


Figure 30: Where to place the four measuring block

On one measuring block, there are six strain foils. We use four blocks to measure, so in total we need 24 strain foils. Figure 29 shows how to place the six strain foils on a block. We use strain foils named X1 and X2 to measure normal force, Y1 and Y2 to measure tangential force and Z1 and Z2 to measure axial force. Figure 30 shows how to place the four blocks named A, B, C, D in the whole structure. We put them between the chuck and Y plate.

According to Appendix F, the strain foil we choose is BF350-2AA (11) N1-X. Our options were limited while choosing a strain indicator. The grinding process is dynamic and grinding forces are constantly changing this forced us to choose a dynamic strain indicator. The strain

indicator we choose is IOTECH6224, it is dynamic, has a resolution of 24bits and accuracy of 0.1% after calibration.

The software we need will be provided free by the company which sells the dynamic strain indicator.

7.2.2.2 Speed sensors

The only requirement for the speed sensors was of maximum speed reading. To measure the speed of the worktable we decided to use the incremental photoelectric encoders embedded in the servomotors. Using those encoders meets the requirements of maximum speed reading and also eliminates extra costs.

7.2.2.3 Position sensors

When choosing the position sensors we looked at four different types, linear scale, differential transformers, transducers, and magnetic scales.

Linear scale is a common linear position measuring component that is widely used in CNC machine tools to realize close-loop control. It is based on the Moire Fringe caused by the diffraction of the light, as the width of the Moire Fringe could directly reflect the placement of the moving part of the scale. The advantage of linear scale is that it has very high reading speed and it is thus very competent in dynamic measurements. The scale has two parts, the fixed part and the moving part. When it is being used, the moving part of the scale is connected to the component of which the displacement we want to measure. (Engineering Sensor Technology, HUST)

The linear variable differential transformer (LVDT) is a type of electrical transformer used for measuring linear displacement. The transformer has three solenoidal coils placed end-to-end

around a tube. The centre coil is the primary, and the two outer coils are the secondary. A cylindrical ferromagnetic core, attached to the object whose position is to be measured, slides along the axis of the tube. As the core moves, these mutual inductances change, causing the voltages induced in the secondary to change. The coils are connected in reverse series, so that the output voltage is the difference between the two secondary voltages. When the core is in its central position, equidistant between the two secondary, equal but opposite voltages are induced in these two coils, so the output voltage is zero. LVDTs are commonly used for position feedback in servomechanisms, and for automated measurement in machine tools and many other industrial and scientific applications. (Engineering Sensor Technology, HUST)

A transducer consists of two non-contacting elements, a scale and a slider for the linear transducer. Position Transducers are a “printed circuit” form of electrical resolver. The printed circuit transducer patterns can be produced on almost any substrate material. The patterns are bonded onto the substrate material and the resulting elements are attached to the customer’s fixed and moveable system parts. The most common transducer application uses inductive coupling between the moving patterns (Inductosyn Company).

A magnetic railings ruler is similar to a linear scale which also uses grid to detect the displacement of the moving parts. The difference between a magnetic railings ruler and a linear scale is that it is using magnetic grids instead of optical structures. Magnetic railings rulers are also widely used for position feedback in servomechanisms and for automated measurement in machine tools and many other industrial and scientific applications (Engineering Sensor Technology, HUST).

	Accuracy	Resolution	Price	Output Type	Compatibility	Total
Full Score	10	10	10	8	10	48
Differential Transformer	9	8	7	6	5	35
Inductosyn	7	5	10	6	10	38
Linear Scale	10	10	9	8	10	47
Magnetic Railings Ruler	5	9	9	8	10	41

Table 23: Position Sensor Decision Matrix

When we made the decision matrix, shown in Table 24, five factors were mainly considered: accuracy, resolution, price, output type and compatibility. Among these four types of linear rulers, linear scale had the highest accuracy while magnetic railings ruler had the lowest accuracy. Scores were rated according to their highest values. The price of Inductosyn was the cheapest while the differential transformer was the most expensive. For the output type, the linear scale and the magnetic railings ruler outputs digital signals directly which is more favorable for the PLC or motion control card to deal with while others output analog signals which need to be transferred to digital signal before being analyzed. The differential transformer's installation dimensions are large and do not fit our design well therefore we gave it a compatibility of 5 while others 10. Based on the total score of the table we selected the one with the most score, the linear scale.

According to our requirement for position sensors, the linear scale we chose was WTB1 from Wan Hao company. It has a resolution of 1um and an accuracy of $\pm 6.25\mu\text{m}$ which meets our requirements. We chose 250mm to be the nominal length for the sensors.

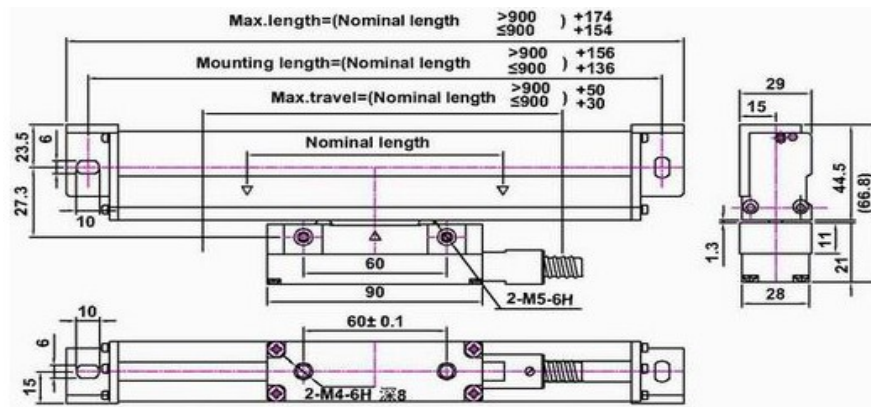


Figure 31: Mounting dimension of linear scale

6.2.2.4 Limit switch

To prevent the worktable from moving beyond the stroke of the machine, position limiting switches are necessary. The designed fixture for the high force grinder has three axes. Each axis needs two position limiting switches, in total the whole machine needs six. The worktable only moves in slow speeds of less than 2"/s and there are no impacts therefore the simplest and cheapest but rather reliable position limiting switch will work for our design. Microswitch were chosen as our position limiting switch due to their simplicity and low price.

7.2.3.Controller Selection

To link the motor drivers to the computer, we need a controller. The controller acts as a bus between the motor drivers and the industrial computer. Based on the selection of the motors for all three axes, new requirements were created for the controller. One requirement was that the controller needs to have three analog signal output. Having these analog outputs will help control the three servo motors in our design to function in the constant torque mode. The other requirement was that the analog output must have a resolution of 0.1lb (0.445N) since the resolution of the feeding force produced by the motor is required to be 0.1lb (0.445N). For our

design, there are two kinds of possible controllers, one is the motion control card (MCC) and the other is the programmable logic controller (PLC).

The MCC is widely used for complex motion control. (Deng, 2007) A motion control card connected to a PC can have a friendly human-machine interface (HMI) that is capable of using higher programming languages like Visual Basic and Visual C++. A MCC can transfer motion functions reliably. It is convenient to modify and add new functions with a MCC because it has more advanced input/output capabilities than a PLC. (Shanghai Ruilei Automation equipment Ltd., 2008) It is also convenient to transport software into a MCC. A MCC has higher real time ability, meaning it has a shorter reaction time between when it receives and sends information, than a PLC since a PLC's operating mode is scan-round, which allows time for scanning the program during each step.

On the other hand, a PLC uses logic judgment to control simple motions and has been used in industry for a long time. (Shanghai Ruilei Automation equipment Ltd., 2008) It is easy to expand a PLC's function and convenient to route wire for it. A PLC has a relatively short developing cycle, meaning that it is faster to program and install, and its configuration software can monitor industrial sites. A PLC has highly reliable I/O signals because it uses optoelectronic isolators to separate the external circuit in industrial sites from the circuit inside the controller. The input ports use R-C filters, and every module uses shielding protection to avoid noise. Hardware makes up the core of the PLC, allowing high stability. A PLC has good self-diagnostic capabilities because the CPU can instantly take effective measures once any abnormal phenomenon is detected.

A PLC has a simple and flexible system configuration, which includes an abundant amount of I/O cards that can be combined in various ways. (Siemens, 2007) It has interface modules for

HMI and is capable of the communication needed to form industrial local networks. A PLC's components, including the CPU, the electrical source, and the I/O ports, are all modularized and different modules can be connected by cables. A simple comparison between the characteristics of a PLC and a MCC are showed in Table 26 below.

	PLC	MCC
Speed	Low	High
Application	Simple, fixed motion trace	Complex motion linkage and interpolation
Compatibility	Low	High
Reliability and Stability	High	Relatively low
Precision	Relatively low	High

Table 24: Simple comparison between a PLC and a MCC

From Table 26, we were able to conclude that a MCC has better real time ability, is compatible with our control system, and has high precision. Whereas, a PLC can only do simple motion control like logic judgment and action control. For these reasons, we chose a MCC as our controller.

MCCs with analog output have both Analog/Digital outputs. (TRIO Motion Technology, 2006) They can control AC/DC servo motors and stepper motors as well as converters, pneumatic servo cylinders, hydraulic servo cylinders, and the combination of them. These MCCs combine motion control and logic control, can realize various kinds of motions easily, and have good expansibility with function modules capable of various combinations. There are several possible running modes available. Multitasking is possible, meaning complex programs can be divided into several independent tasks that are done simultaneously. These MCCs are capable of receiving various types of positioning feedback signals from sensor . Various kinds of interfaces with servo driver can be used, such as CANopen, DeviceNET, Profibus, SERCOS, SLM, and general servo interfaces. Two specific motion control cards with analog output were

investigated: the TRIO PCI208 Digital Motion Control Card and TRIO MC206 Digital Motion Control Card.

The PCI 208 motion control card is based on PC and PCI bus control. It uses an independent 120MHZ DSP microprogramming technique and can do calculations quickly. (TRIO Motion Technology, 2006) Also, it can combine motion control with logic control, is able to move in multiple directions simultaneously, and has good expansibility. Since the PCI208 is capable of multitasking, it can compensate for mechanical backlash and reduce any possible errors. The program responsible for compensating for backlash can be written into the PCI208's memory. This program works by running simultaneously with the main motion control program and monitoring the motion instructions continuously. Once the backlash compensation program detects an instruction that reverses the direction of motion, it will send its own instructions to correct the error caused by backlash. The PCI208 is the main-daughter board structure. Additions, such as the daughter board extensions called P182, P183, P184, and P185, can perfect the function of the PCI208. The software of Motion Perfect 2 can do the programming, off-line development, and on-line controlling and monitoring.

The MC206 is small in size, uses the Field Programmable Gate Array (FPGA) technique, and is easy to mount. (TRIO Motion Technology, 2006) The MC206 has 16 I/O which use optical isolation to ensure high reliability. When using the Motion Perfect 2 software, the MC206 can develop the program fast, modify motor's parameters on-line, and diagnose running state.

Table 27 compares the capabilities of the PCI208 with the MC206.

	PCI208	MC206
Number of Axes	2~8	1~5
Max interpolated axes (Virtual axes included)	8	8
DSP speed (MHz)	120	120
Simultaneous programs	8	8
Available memory	PC+250KB	256KB
Opto-isolated input	20	8

Assignable opto-isolated I/O	10 (output)	8
Analog output and input	16-bit±10V output ; 4 input	16-bit ± 10V output; 1 input
Max No. of Daughter Boards	0	1
Servo loop update rate	250us	250us
PCI bus	Standard	N/A
Serial ports	0	4
USB	N/A	Standard
Ethernet	N/A	Optional
DeviceNet	N/A	Standard
ModBus	N/A	Standard
Price	RMB ¥ 16917	RMB ¥ 17342

Table 25: Comparison of the two MCCs with analog output

From the table, it is clear that both MCCs are capable of multitasking with up to eight programs running simultaneously, including the program to compensate for backlash. Also, both MCCs have analog output of 16 bit. Since $10V \cdot 400lb / 2^{16} = 0.061lb < 0.1lb$, both MCCs meet the accuracy requirement. As for the two MCC models, the PCI208 can be inserted into a PCI slot of a PC, enabling it to use the PC's memory, while the MC206 cannot. The PCI208 also has more I/O ports than the MC206. The PCI208's communication is based on a PCI bus, which is faster than a USB or RS485. (TRIO Motion Technology, 2006) Finally, the PCI208 has a lower price than the MC206. Considering all of this information, we chose to use the PCI208 as our controller

7.2.4 Human-to-Machine Interface

A human to machine interface consists of superficial programs with good layout, hidden logic functions, data input, and data output. The interface is the link between the user and the control system for the machine. There are many software available that can be used to make an interface. Some of these programs include Microsoft Visual Basics, Microsoft Visual C++, Microsoft Visual C#, LabVIEW, Java, Matlab, and others. After doing further research on interface software and programming languages, we discovered that C++ language can realize big

and complex logic that other programs are inefficient in doing. Another benefit is that the C++ compiler is more aggressive for finding and correcting bugs for small and large programs alike. (Eckel, 2000)

In the beginning of our project we were supplied with the MPC2810 motion control card, DM4240A step motor, and DMD402A stepping motor driver. To decrease the development time of our experimental control system we chose to use Microsoft Visual C++ to create our first control interface. Visual C++ uses a computer language that is commonly used and at the same time uses visual aid to help and improve the hacking of programs. Visual C++ uses Microsoft Foundation Classes (MFC) helps to write applications faster and easier. (Junjie, 2001) These MFC represent objects such as windows, dialog boxes, menu bars, status bars, control bars, and window controls from which Windows applications are created. In addition, the MFC provide general purpose classes for handling strings, arrays, and linked lists which keeps the flow of the program more organized.

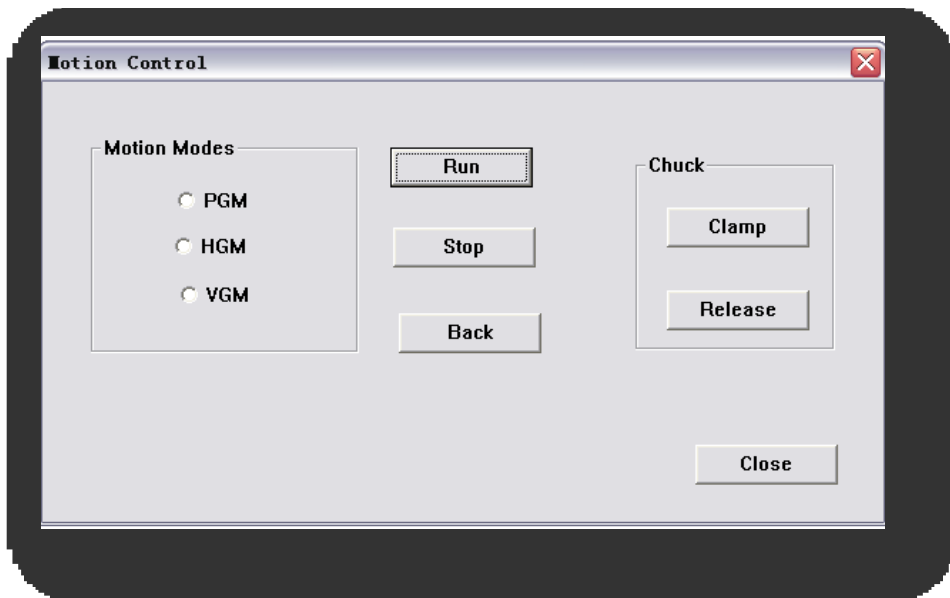


Figure 32: Experimental Interface

Using Visual C++ Microsoft Foundation Classes (MFC), Dynamic Link Library (DLL), and header files provided by both Leetro, and Microsoft we were able to link the supplied step motor to the computer via the motion control card and stepping motor driver. Our experimental Visual C++ interface is made up of several interactive push buttons and radio boxes as can be seen in Figure 32. These push buttons and radio boxes are used to control the motion of the step motor. Using one step motor we were able to realize the forward motion and backward motion of the motor. We also achieved the forward-stop-backward-stop motion series with using the step motor.

Due to the lack of more motors and motor drivers we were unable to realize the required motion modes for our control system. We were also unable to simulate the constant feeding force mode due to the lack of a servo motor.

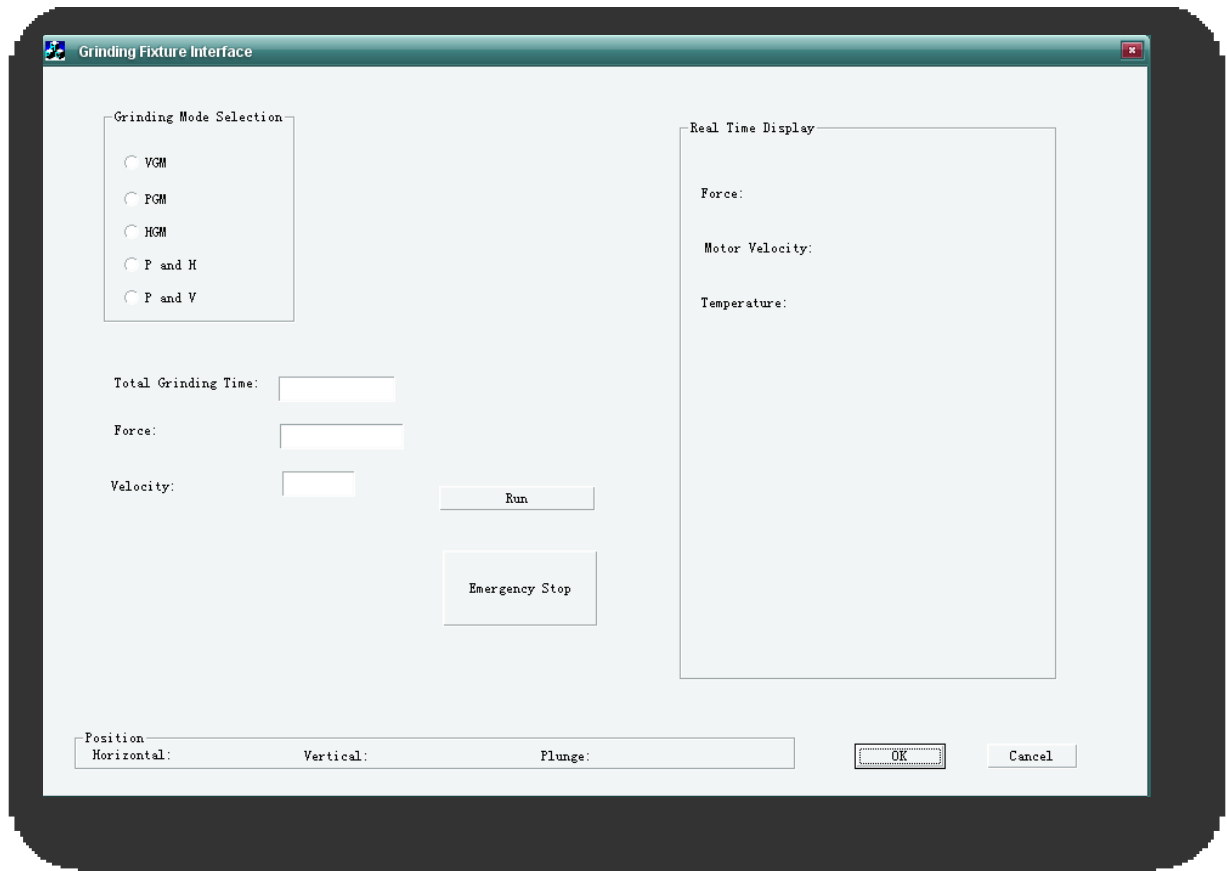


Figure 33: Requirement Fulfilling Interface

An interface richer in functions was created to fulfill the requirements of Saint-Gobain. This new Visual C++ interface displays all the required system information in one screen. The radio box on the left side of the screen is used to select the desired grinding mode. The user has the option of choosing between Plunging Grinding Mode (PGM), Vertical Grinding Mode (VGM), Horizontal Grinding Force (HGM), or combinations of either mode. The three input boxes in the middle are for the input of feeding force, total grinding time, and velocity. The right side and the bottom of the screen are used to display real time signals from the force, velocity, position and optional temperature sensor.

A shared library (DLL) is a very important part of program design based on Windows. (Xiong, Xing, Zuo, & Zen, 2008) The DLL contains functions that allow the interface to communicate with the motion control card. The building of the DLL has no specific

programming language or compiler, it is universal. As long as the DLL complies with the norms of the development environment and programming strategies of the hacker, any interface will be able to call it correctly. Our designed machine requires complex motion control, for which a DLL containing a number of specific functions is required. These functions can be created using the DLL that Trio Motion Technology Ltd. Company provides with the motion control card. Functions such as board initialization, operating conditions initialization, the three-axis operation, location of the three-axis, motion calculation of three-axis, stop and zero can all be created by calling the DLL provided by the company.

LabVIEW (Laboratory Virtual Instrument Engineering Workbench) is a development environment based on diagram block language launched by National Instruments. (Rui, 2008) LabVIEW is widely used in testing and industrial control environments. In LabVIEW users can build independent executable files that can be run without the development environment. Besides, it can call the DLL built in other development environment. This system has great capability and specialty of short development time.

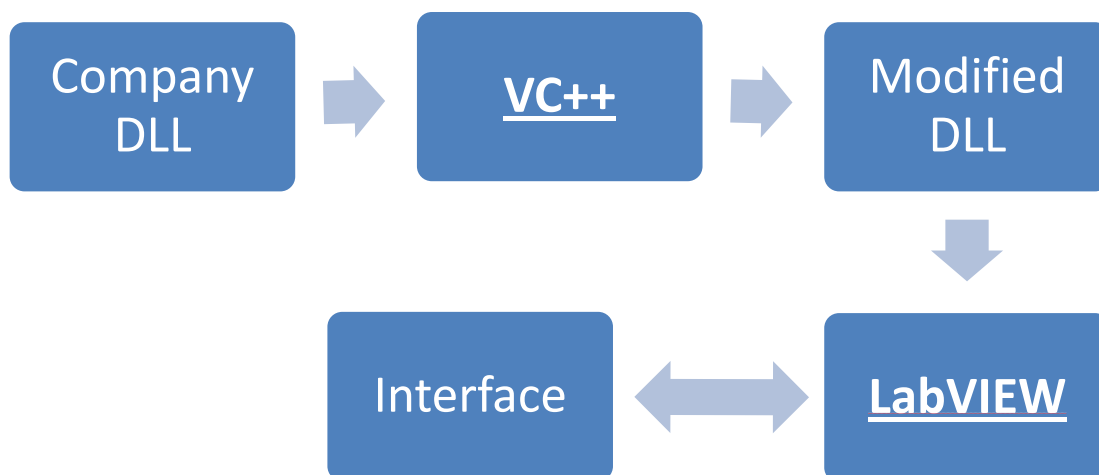


Figure 34: Flow chart of interface

The flow chart in Figure 34 shows the control procedure that should be used in order to develop the interface using LabVIEW. Using VC++ the DLL provided by the motor company is modified. The modified DLL is then used by LabVIEW to program the control interface. LabVIEW creates the front panel window and then calls the modified DLL to realize the motion control for the system.

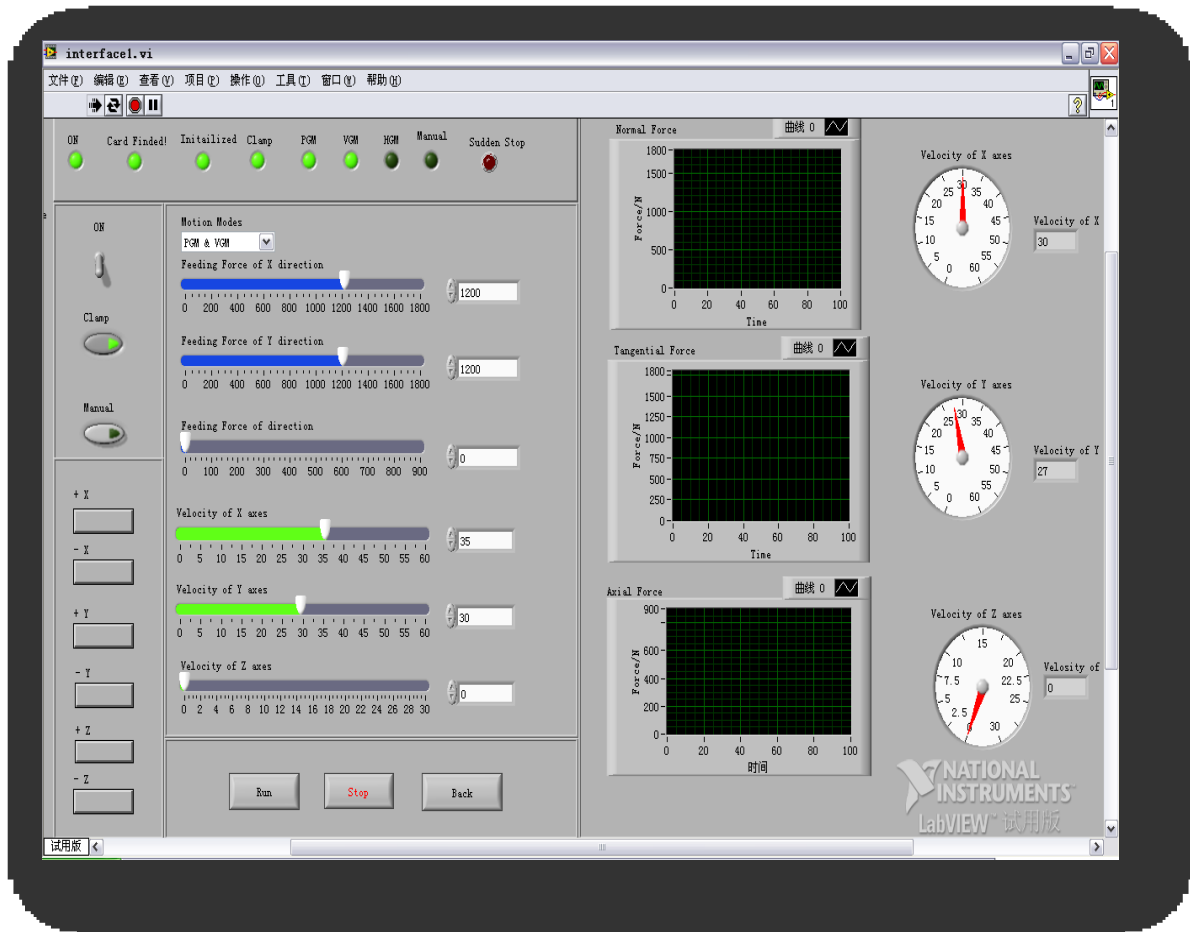


Figure 35: LabVIEW Interface

The front panel window of our designed interface is shown in Figure 35. The upper panel has a row of indicative lamps that show the real time status of components of the control system which includes the fixture, chuck clamp, and motion modes. When lighted the component is on otherwise it is off. The clamp button on the top left corner can make the chuck clamp or release

the workpiece. The Manual button is used to change the grinding mode from automatic to manual control. The automatic mode includes the five motion modes PGM, VGM, HGM and combination of PGM and VGM, PGM and HGM. The scroll bars are used to set the feeding force of X, Y, and Z direction as well as the velocity. The graphs and the speedometer on the right side of the panes are used for displaying the grinding force and velocity acquired from the sensors.

By using LabVIEW we were able to cut down on our program development time. We created a user friendly interface that is able to input values to the system and display feedback from the system. Our interface meets all requirements including grinding mode selection and clamp control.

8.0 Conclusions and Recommendations

After researching the various components of the fixture and designing the remaining aspects, we chose the options that would perform the best and were cost-effective. Our final design is a compilation of all of these components.

8.1 Mechanical Design

The mechanical design team was responsible for the driving mechanism, the guiding and supporting structures, the chuck, and the overall structure of the fixture. For the driving mechanism, ball screws from HIWIN were the best option since they have a high efficiency and are made to accommodate similar working conditions to the HFG fixture. Guideways from HIWIN were chosen as the guiding and supporting structures because they are able to withstand the high forces that will be inflicted on them, unlike the linear bearings. As for the chuck, the Changzhou Beut Machinery Manufacture Co., Ltd was chosen as the manufacturer for its good value and cost effectiveness. To supplement the Beut chuck, interchangeable jaws were chosen for their superiority in adaptability, magnitude of torque created, and precision. For the overall structure of the fixture, we decided to use Concept 2 for the rail configurations because of its stiffness.

8.2 Control Design

The control team needed to research the vital components including motors, sensors, controller, and human-to-machine interface. Servomotors were found to be the best option for our fixture design. We chose Servomotors produced locally in order to keep the cost of the fixture down. For the sensors we chose the ones that met all the requirements. Strain gauge

sensors were chosen for the force measurement, servomotor encoders for the speed measurement, linear scale sensors for the distance measurement, and microswitch for the position-limiting sensor. Motion control card with analog output was chosen over programmable logic controllers due to their higher accuracy and response speed. LabView's professional look and easy coding made it our first choice among all other human-to-machine interface software.

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