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Designing and Building an Automatic Chamfer Grinder

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DESIGNING AND BUILDING AN AUTOMATIC CHAMFER GRINDER

A Capstone Experience/Thesis Project

Presented in Partial Fulfillment of the Requirements for

the Degree Bachelor of Science with

Honors College Graduate Distinction at Western Kentucky University

By

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ABSTRACT

Modern day manufacturing is a demanding environment with a constant need for process improvement. As automation becomes more advanced, there are fewer jobs that must be completed by a human. In the case of Stupp Bridge Company, a local manufacturer of steel bridge girders, their workforce is highly skilled, so replacing mundane tasks with automation allow the skilled workers to focus on the difficult jobs. One such task is grinding a chamfer onto every leading edge of each girder flange, eight edges in total, ranging from 20 to 200 feet long. The purpose of this project was to design an automatic chamfer grinding system. To ensure maximum design potential, an entire semester was spent planning and designing the system. This has carried over into the current semester, and a number of techniques are being used, the largest of which is 3D computer aided design using Solidworks. Once a design is finalized, and company approval obtained, the building process will began. Concurrent with assembly will be component testing, to confirm that the chosen components will perform as required. By the completion of this semester, a finished device will be given to Stupp Bridge Company to be put into their everyday operations.

Keywords: Grinder, Steel Plate Girder, Bridge Girder

Dedicated to my family, friends, teachers, and professors who have supported me through the years and always pushed me to be the best I could be.

ACKNOWLEDGEMENTS

This project would not have been possible without the support of WKU engineering and Stupp Bridge Company. The faculty of WKU engineering is always working their hardest to ensure that our senior projects have actual use and are not without meaning. Stupp Bridge was generous enough to task a group of college students with a challenging problem, and had faith that we would complete the task. They also supported the project financially, which was a generous sum of money. I specifically thank Derek Clemons, Stupp Bridge Company's plant manager, for serving as the industrial contact and always being willing to meet with us and help in any way he could.

I would additionally like to thank Dr. Chris Byrne, the faculty advisor for this project as well as this thesis. His support and weekly meetings helped immensely to keep the project on track and focused.

Finally I would like to thank my friends and family for their unwavering support in all of my endeavors.

VITA

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FIELDS OF STUDY

Major Field: Mechanical Engineering

Major Field: Applied Mathematics

Minor Field: Systems Engineering

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PROJECT DESCRIPTION AND COMPANY BACKGROUND

Stupp Bridge Company is a locally based manufacturer of steel bridge girders that has been in business for over 150 years. Their facilities are state-of-the-art, providing a superior product on time and within budget. As a company they place an emphasis on quality, cost efficiency, and dependable delivery. To maintain maximum efficiency on the production end, they have a rail system for the delivery of raw materials. Once delivered, the materials are entered into a bar code tracking system – ensuring that they are completely tracked and traceable throughout the entire manufacturing process. In order to produce girders that are unquestionably up to bridge code, they put every piece of raw material through a pre-cleaning system. From there, Computer Numerical Controlled thermal and plasma cutters cut the pieces to shape. The cut material is then welded together either by skilled workers or by modern welding machines, with any holes being drilled by portable CNC drilling machines. The final process is a post-manufacturing shot blasting, to produce a uniform finish and get rid of any slag, and then painting (if required).

A smaller part of the manufacturing process is to grind a chamfer on every flange edge, as previously discussed. Even though this is a small process, the extreme significance of it cannot be understated. The grinding must be performed to relieve the stress risers and ensure the bridge is up to the task of supporting people, cars, or trains.

Currently the grinding is done with a handheld angle grinder, which necessitates the use of an employee. Since there are eight edges that must be ground per girder, of lengths up to 200 feet, this process takes a considerable amount of time for the operator. Time is money, and each employee at Stupp is highly skilled, so their time would be much better allocated in performing a skilled task – which grinding is certainly not. To this end, the project was to design a machine that would automatically grind the chamfers onto the flanges.

An automated grinding process has several key advantages over a manual process. First and foremost is operator safety, as grinding is not a very ergonomic task. The angle grinders currently in use weigh up to 12 pounds, must be held at a specific angle for proper operation, and are used for extended periods of time. This is very taxing on a person. If this process was automated, there would no longer be a risk for human injury. In addition, the operators get paid good money for a particular skill, and when they are doing a mundane task such as grinding it costs the company more money than necessary. Finally, the automatic system could grind the chamfers significantly faster than a human can. Once again, this saves money and is a much more efficient use of resources.

EXPLANATION OF STRESS CONCENTRATORS

To understand the need for this grinding system, it is important to understand what a chamfer is and why it is necessary on every longitudinal edge of a steel plate bridge girder. There are three basic options when a square edge is present on a part: leave it, chamfer it, or fillet it.

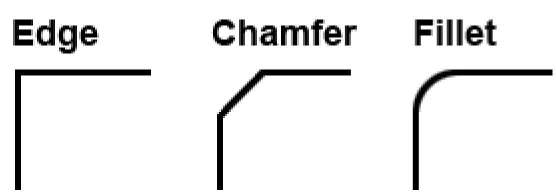

Figure (1) – Edge Finishes
http://www.neilblevins.com/cg_education/rounding_the_edges/edges.gif

The last two options, chamfering and filleting, each have more options associated with their size. A chamfer can be specified with either the two lengths defined, or a length and the angle. Both of these are equally acceptable, but oftentimes only one length with be specified. In this case, it is implied that the angle is 45°, which causes both lengths to be the same. This is the most common chamfer in normal parts. A fillet is simply specified by the radius of the quarter circle in the cross section.

When a part is envisioned by its designer, in most cases an edge finish will be specified. This could be as simple as “breaking” the edges, which means to put a small 45° chamfer on the edge, or as complicated as putting a full fillet of specified diameter on the edge. Depending on the method of production for the piece, one of these may be more feasible than others. For a part that is 3D printed, for example, any reasonable edge finish is possible since it is no more difficult for a rounded edge to be printed than a square edge. For a part being produced using manual processes, however, putting a fillet is more complicated. The machine operator would have to be constantly adjusting every axis of his machine, which takes considerable time and skill. A chamfer, on the other hand, simply requires tilting the head of the mill or clamping the part at an angle. Due to the relative simplicity of creating a chamfer, they are usually more favorable than a fillet. On a part as large as a steel plate bridge girder, it is only practical to use a chamfer, and the easiest way to create the chamfer is by grinding.

Knowing that the process of grinding the chamfer currently takes considerable amounts of time, it begs the question as to why it is even necessary. Would it not be simpler to just leave the square edge? The answer to this question is yes, it would be easier, but it is not an option in the case of a bridge girder. For a part that sees a constant loading, such as a bridge girder, small imperfections in the edge can have large consequences. These small imperfections are called stress risers, and it is important to understand just how big of a deal they are. The stress in a part can be thought of as a flow, which can be represented by trace lines. When this flow moves through a part, any abrupt change in the parts geometry can cause the stress traces to “bunch up,” which in turn causes the amount of stress to increase drastically. For a visual example, look at Figure (2).

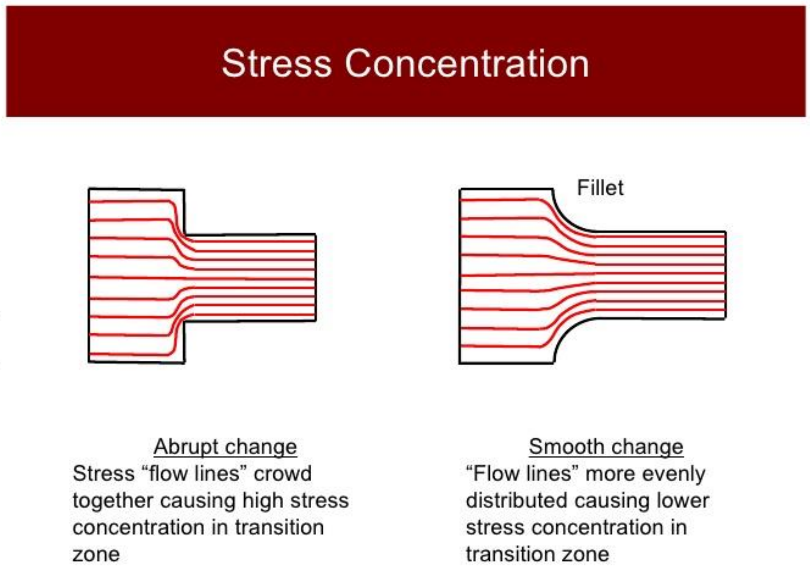

Figure (2) – Stress Flow Lines
http://www.slideshare.net/engCETL/tta104-section-7

In order to explicitly analyze the extra stress caused by stress risers, a simple plate with a bending moment on it will be considered, such as what is shown in Figure (2) above. This will allow for a simplistic answer that would extend to a full bridge girder with minimal effort. The basic equation for bending moment stress is shown in Equation (1).

$$\sigma_{ave} = \frac{Mc}{I}$$

Equation (1) – Average Bending Moment

This equation shows that the average stress, σ, is equivalent to the applied moment, M, multiplied by a ratio of geometrical values, c and I. This is relatively intuitive, that the stress would depend on the load and the shape of the part, however a layman would most likely assume that material properties would also play a role. The way a part is analyzed for failure is by comparing the average stress value given by Equation (1) to the yield strength, σ_y, or the ultimate tensile strength, σ_{UTS}, of the material being used. If the average stress is higher than the yield strength, than the part will permanently deform under maximum stress conditions. If it is higher than the ultimate tensile strength, it will fracture and fail. When a stress riser is present, the bending moment stress equation is modified by a stress concentration factor, K_c, as shown in Equation (2).

$$\sigma_{max} = K_c \sigma_{ave}$$

Equation (2) – Maximum Stress with Stress Concentration Factor

It can be seen that the concentration factor is a straight multiplier on the average stress, which means it can drastically change the amount of loading that could cause yielding or failure. Now the stress concentration factor must be discussed. A long time ago, in a research lab far, far away a selection of charts was produced that allow for the concentration factor to be easily determined. The chart for this specific analysis is shown in Figure (3).

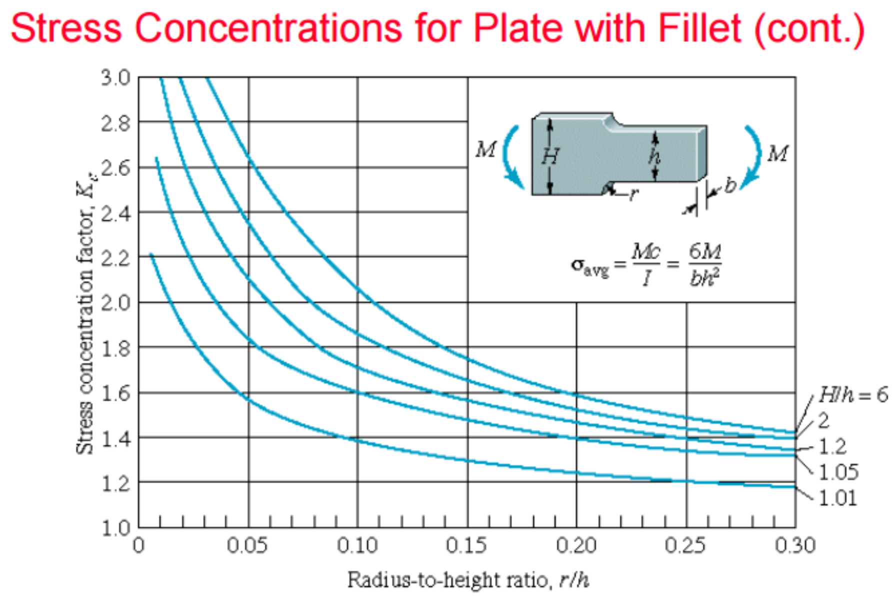

Figure (3) – Stress Concentration Chart for Filleted Plate
http://www.ux.uis.no/~hirpa/KdB/ME/stressconc.pdf

These charts can be slightly confusing at first, but notice that the several different curves just represent different height ratios. The horizontal axis is a ratio of the fillet radius to the smaller height. Using this and the height ratio, you can determine the stress concentration factor from the vertical axis. The most important part of this chart is to notice the characteristic of the stress concentration factor to increase as radius decreases. In fact, if we let the radius approach zero, i.e. a square edge, the stress concentration factor sharply increases up to three or more. This means that the presence of a square edge can decrease the allowed load by up to three times! It is obvious that if this was not accounted for in the design of a structure that it could lead to unexpected failure.

In the context of bridge design, the stress risers are imperfections in the edges of the steel plate girders. It has already been shown that if left unattended, the stress risers could cause the bridge to fail unexpectedly, potentially leading to human injury or death. In order to mitigate this risk, it is common practice to grind a chamfer on all the leading edges. This puts a uniform surface on each edge. Thinking about the stress trace lines again, it becomes clear that the lines would be able to traverse the full length of the beam without interruption.

PROJECT REQUIREMENTS

Developing good requirements is one of the most important steps in the design process. Once the general scope of the project is understood, it is necessary to record specific requirements that fully define the project. Without these requirements as guidelines, it is hard to not sway from the original design intentions. The successfulness of the completed design can be judged based on how closely it fulfills the requirements. To ensure exactness and a fully defined problem, the requirements can be broken down into three distinct areas: functional, performance, and constraint. Functional requirements define what the project must do. Performance requirements define and quantify how well the project must accomplish its intended function. Constraint requirements capture operational, environmental, and safety constraints. Together they give the project the highest chance of success. Below you can see the different requirements that were written for this project.

1. <u>Functional Requirements</u>

    1.1. The device shall grind reduce stress risers by grinding a chamfer into the flange edges.

    1.2. The device shall accommodate multiple girder sizes.

        1.2.1. The device shall accommodate plate thicknesses ranging from 0.5” to 2.5” thick.

        1.2.2. The device shall accommodate girder lengths ranging from 20’ to 200’ long.

    1.3. The device shall automatically stop at the end of the girder.

        1.3.1. The device shall have multiple stopping redundancies.

    1.4. The device shall have the ability to make multiple passes automatically.

    1.5. The device shall have a user adjustable chamfer depth.

2. <u>Performance Requirements</u>

    2.1. The device shall grind a chamfer of at least 1/16” by 45°.

    2.2. The device shall have a cycle time of at most 5 minutes per edge.

    2.3. The device shall be able to continuously operate for at least 30 hours per week.

3. <u>Constraint Requirements</u>

    3.1. The device shall be operated by a single person.

    3.2. The device shall weigh less than 50 pounds.

    3.3. The device shall cost less than $10,000.

    3.4. The device shall be protected against dust.

DISCUSSION OF DESIGN

Over the two semesters that this project was undertaken, my main task was the overall design of the system. This was broken up into two semesters, with the design iteratively changing throughout. Brainstorming the frame was a very involved process. In order to accommodate such a wide range of flange thicknesses, the mounts holding the grinders and motors must be adjustable. Since they have to contact both sides of the flange at once, however, this proved harder to design than anticipated. Initially a concept similar to a vice was explored. The idea was to have ACME threaded rod with a handle on one side to use as a manually operated clamp. When this idea actually started being planned, it became clear it was not going to function as intended. Since both sides have to adjust independently around the flange, there is no fixed point available for the threaded rod to be attached to. With some further thought, it was realized that a non-adjusting piece would have to sit centrally on the top of the flange. The idea of using a central shaft as a pivot point for the motors and grinders was introduced, and it quickly became evident that it would work very well. This will allow full adjustability of each, no matter the flange thickness. In addition, it will easily accommodate various designs for holding a clamping force at the motors.

Once the plan for the central shaft came together in the first semester, a plan for the rest of the system came together. The majority of it was simply designing mounts to interface with the chosen motors and grinders, and ensuring that nothing interfered over their full range of motion. In order to have equally distributed forces, idler wheels had to be placed directly opposite the motors. There were some other minor design considerations that had to be incorporated into the frame as well. Hose routing was a concern, but was managed by running a flat plate the full length of the device, directly above the central shaft. A ½'' solid walled pipe is run along this length, with 90° elbows at each end and a tee fitting in the center. On the perpendicular port of the tee there is a ball valve, which then leads to another five inches of hard pipe. At the end of this there will be a quick connect for the flexible airline. The buttons and electrical components enclosure will be mounted on top of the pipe as well.

One of the biggest design challenges was figuring out a clamping mechanism for both the motors and the grinders. Due to the large clamping force on the motors, it had to be a simple solution that did not require large amounts of strength. The idea of a simple extended handle with a swinging lock mechanism, similar to how a pair of hand shears lock closed, was decided upon. In order to know how long to make the handles, the average human grip strength was researched and a target value of 75 lbs of squeezing force was chosen. This is significantly less than an average male grip strength, and right around average for a female. For the grinders, a screw mechanism was implemented. The operator will simply twist the knob until the abrasive cone wheel contacts the material. The will allow for exact depth control, as well as keeping the grinder from jumping around. The first semester design can be seen below, with more detailed views in Appendix B.

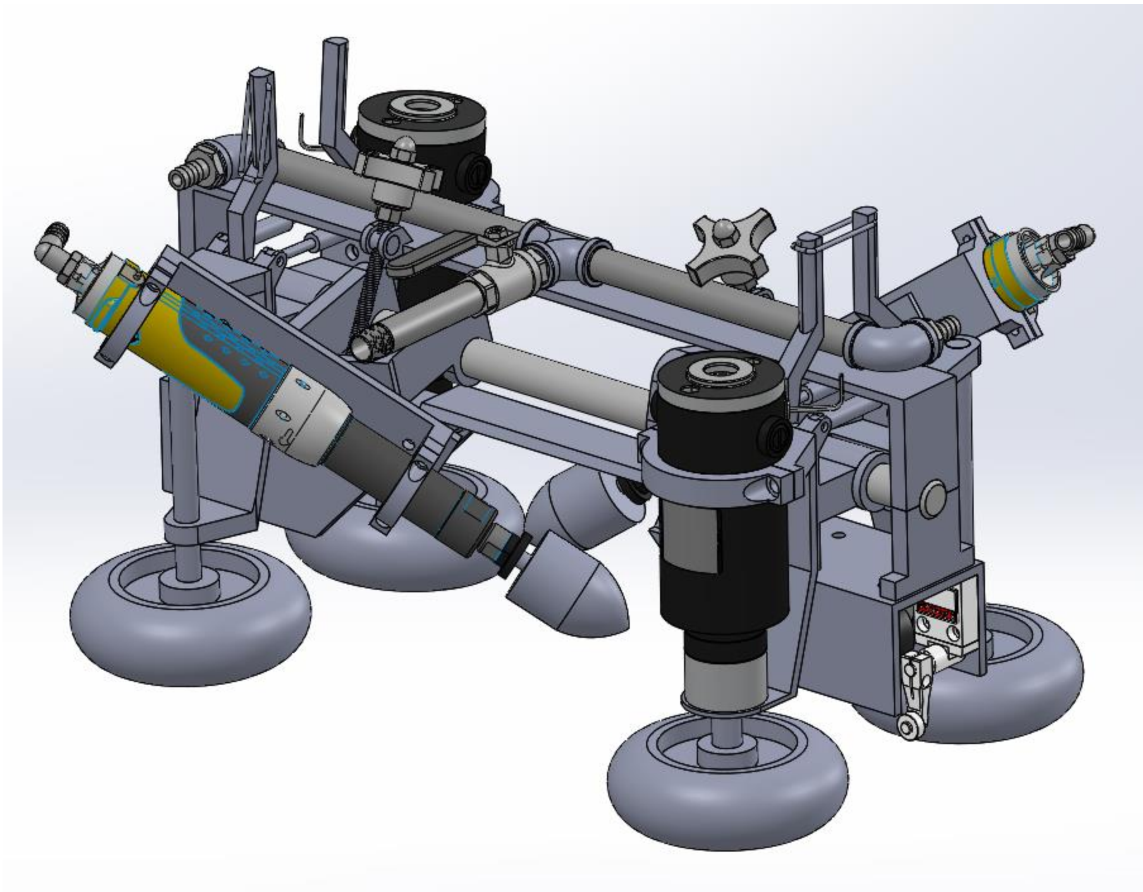

Figure (4) – First Semester Isometric Device View

At the end of the fall semester, a project proposal document was given to the industrial contact. Since there were three separate teams working on different design concurrently, he had to choose one design to move forward. The design from my team was chosen, but as is often the case the original design was not practical to have produced. It relied too heavily on really complex parts that would have been prohibitively expensive to have machined. Due to this, I had to start a major redesign of the system. The central shaft concept was retained since it was the key to the first semester design. Using that as my base, I started to design the second concept with more attention to keeping it reasonably easy to produce. This was done in two major ways: using off the shelf components and making sure any machined parts were easy to make. Every custom part was designed to be flat so that it could be easily cut out on a CNC waterjet cutting machine. The parts then all bolt together to create the full system. The grinders and motors from the first semester were retained, as well as the basic tensioning method for the motors. A major design change that actually made the device simpler was the elimination of a tensioning method for the grinders. It was determined through testing that the grinders had enough mass to cause gravity to be an adequate downward force. The final design can be seen below in Figure (5), with more detailed views in Appendix C.

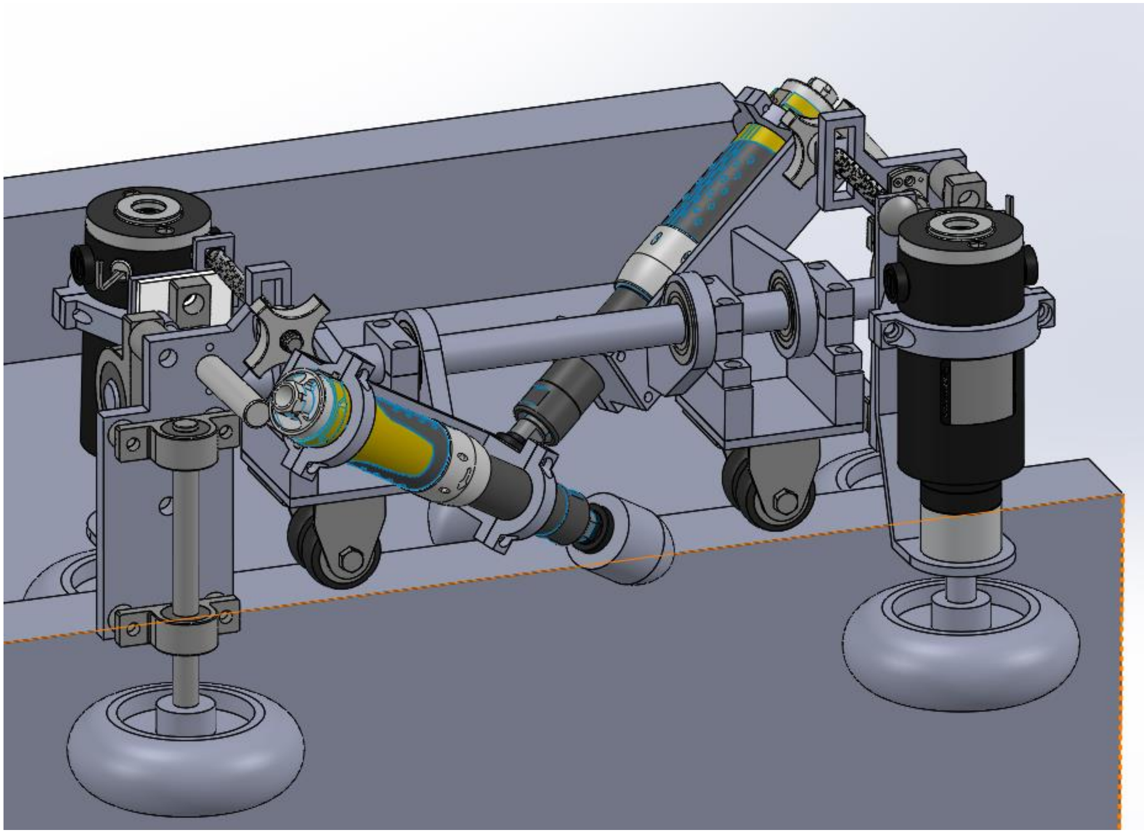

Figure (5) – First Semester Side Device View

JUSTIFICATION OF DESIGN DECISIONS

Over the course of any project, decisions will be made that have specific reasoning behind them. In most cases, the best way to make decisions is to lay out the options in a non-opinionated manner and decide based on objective data. Using hard data as the basis for decisions ensures that personal opinion and prejudice do not get in the way of choosing the best component. Unfortunately, hard data does not always exist for the specific problem being solved. When this is the case, oftentimes the only way to make decisions is based on personal experience, or on the experience of other respected peers and superiors. For a project such as this, since there really is not a similar device on the public market to use as a comparison, a lot of decisions had to be made based on experience and the advice from others. In making several decisions the system in question was modeled as accurately as possible, and mathematical equations were used to try to determine what was needed. The math, however, only tells what sort of numerical specifications are needed – not which specific model or brand will perform best. In the next few paragraphs all the major decisions will be outlined, with justification. These will be separated into three categories: grinding subsystem, drive subsystem, and controls subsystem.

Grinding Subsystem Decisions

1) Abrasive grinding or cutting

This was the very first decision that had to be made, as it is the foundational aspect of the project. Since there is not a true way to objectively determine which is better without testing, prior knowledge of both processes was relied upon. This led to choosing abrasive grinding for several reasons. By definition cutting would require a more precise force to be pushing the cutting bit into the material. It would also skip and chatter as it moved along, possibly stalling on a locally harder section of the steel. Further, the direction the device was moving would determine whether the device was climb or conventional cutting, two styles that produce different results with varying levels of difficulty. Using an abrasive grinding stone, on the other hand, should mitigate some of these challenges.

2) Pneumatic or electric

Once it had been decided to use a grinder, the best power source had to be determined: electric or pneumatic? This topic was discussed extensively, and a lot of time was spent on research. The findings seemed clear that for this application pneumatic would be the best choice. There are many important reasons why this is the case. The first of these is that a pneumatic motor stays inherently cool from the air running through it, no matter how long it is operated.

Since this will be used for long durations at a time without stopping, there was concern that an electric motor would overheat. Secondly, a pneumatic grinder can be repeatedly stalled without any damage. Occasional stalling is almost guaranteed, and that could seriously harm an electric motor. Dust is also a concern, and since a pneumatic grinder is sealed there is no way for dust to cause an issue.

3) Vane or turbine style

This discussion proved to be somewhat challenging, but the true deciding factor was the weight and size constraints of the project. Even though turbine style grinders are more powerful, they bring with this power a significant increase in weight and size. From previous conversation in the conceptual design review, it was felt that using a less powerful grinder with more grinding passes made more sense than using the turbine style. To this end, it was decided to use a vane style grinder.

<u>Drive Subsystem Decisions</u>

1) Pneumatic or electric motor

For the drive motors, the discussion of comparing the pros and cons of electric or pneumatic occurred again. In this case, it was determined that an electric motor makes more sense. The biggest concern for the drive motor was having the necessary startup torque to get the device from a static state to a dynamic state. Most air motors have a very small starting torque simply because of how they operate. Knowing this, it pointed strongly to an electric drive motor. Upon even further thought, it was realized that a direct current electric motor would be easiest to drive both forward and backward, and thus that is what has been specified in the design.

2) Torque necessary to drive the entire system

Determining the necessary torque rating for the motors was somewhat challenging, as this was a decision that had to be justified with math. To set up the governing equations, the system was broken down into simplified pieces – assuming that the total normal force of the device pressing into the flange would be the major force that had to be overcome by the motors. There were several components acting as the total normal force, including the device weight and the forces imparted by the grinding itself. The exact equations and calculations are show below.

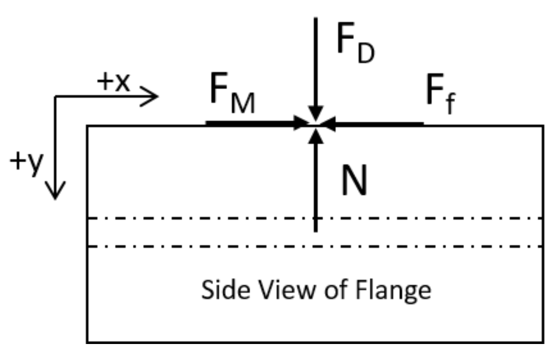

Figure (6) – Motor Torque Free Body Diagram

The forces shown are defined as follows: F_D = downward force, F_M = force required to move, F_f = frictional force, and N = normal force. Summing the forces in the x and y directions gives:

$$\rightarrow + \sum F_x = F_M - F_f = 0$$
$$\Longrightarrow F_M = F_f$$
$$\downarrow + \sum F_y = F_D - N = 0$$
$$\Longrightarrow F_D = N$$

The frictional forces are equivalent to the coefficient of friction multiplied by the normal force.

$$F_M = \mu \, N = \mu \, F_D$$

From here the downward forces must be determined. This will be the total weight of the device, W_D, and the force from the grinder on the flange, F_G.

$$F_D = W_D + 2 \, F_G$$
$$\Longrightarrow F_M = \mu(W_D + 2 \, F_G)$$

Now some of the forces must be converted to torques, where: τ_G = torque from grinder, r_G = radius of grinder cone, τ_M = torque from the motor, and r_M = radius of wheel.

$$F_G = \frac{\tau_G}{r_G}$$
$$F_M = \frac{\tau_M}{r_M}$$

Finally substituting and solving for τ_M gives:

$$\frac{\tau_M}{r_M} = \mu(W_D + 2 \, \frac{\tau_G}{r_G})$$

$$\tau_M = r_M\, \mu(W_D + 2\, \frac{\tau_G}{r_G}$$

The following assumptions will be made to get a final motor torque value: μ = 1 for maximum frictional effects, r_M = 3in which is the diameter of our chosen wheel, W_D = 50 lbf which is the maximum target weight of the device, r_G = 1in which is the radius of the chosen abrasive cone wheel, and τ_G = 9 lbf-in which is the stall torque of the grinder. These values imply:

$$\tau_M = (3)(1)(50 + 2 * 9)$$

$$\tau_M = 204\, lbf - in$$

Since there will be two drive motors, each motor will need a rated torque value of at least 102 lbf-in.

3) Specific motor selection

When it came time to choose a specific motor, many different styles and manufacturers were reviewed. A main factor being looked for was a motor driven by 24 VDC, as this is the simplest power level to produce. Higher voltage DC systems require more complicated motors controllers and such. The motors also needed to be able to output the calculated torque values. Other aspects desired were lightweight, compactness, sealed from dust, and made in the USA. After spending many hours searching manufacturer's catalogs, a company called Groschopp Incorporated was discovered. They have a wide range of gearing and sizes available, as well as the ability to fully seal the motors from dust. When contacted for a quote, they were immediately responsive and gave the pricing in less than 24 hours.

4) Motor tensioning mechanism

Tensioning for the motors is to be accomplished by dual mounted gas springs. These were chosen due to being a closed system. The motor clamping force was calculated by taking the maximum torque of the motor and seeing how force this translated to the steel, and then calculating the amount of normal force it would take (including an assumed friction value) to keep the wheel from slipping. Since there is mechanical disadvantage in play due to the pivot, the moment-couple calculation was done to see how much force the spring needed to supply to give the calculated normal force. These free body diagram and calculations are shown below.

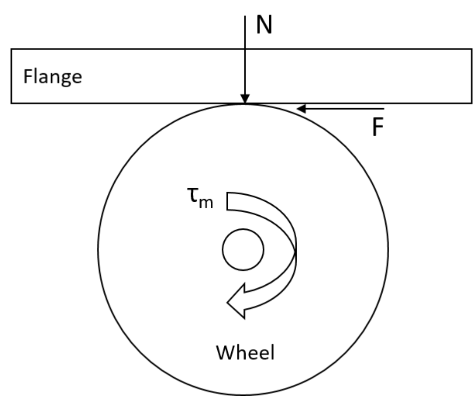

Figure (7) – Wheel Normal Force Free Body Diagram

In this diagram, τ_m is the maximum motor torque, r_w is the radius of the wheel, N is the normal force, F is the force imposed on the steel, and μ is the coefficient of friction. Then the equations for torque and normal force are:

$$\tau_m = F\, r_w$$
$$F = \mu\, N$$
$$\Longrightarrow \tau_m = \mu\, N\, r_w$$
$$\Longrightarrow N = \frac{\tau_m}{\mu\, r_w}$$

Assume μ=1 for maximum frictional effects, r_M=3in, W_D=50 lbf which is the maximum target weight of the device, r_G=1in, and τ_G=9 lbf-in. Which implies that:

$$N = \frac{106}{(0.75)(3)}$$
$$N = 47.1\ lbf$$

Thus a normal force of around 50 lbf must be applied at the wheel. To calculate how strong the spring force needs to be, the moment around the pivot will be summed using this 50 lbf normal force. Setting this equal to zero will then give what the spring force needs to be at the mounting location to produce the need normal force. In this case, M is the moment, N is the normal force, F_S is the spring force, d_N is the distance from the pivot to the normal force, and d_S is the distance to the mounting location of the spring. Consider the simplified interactions shown below:

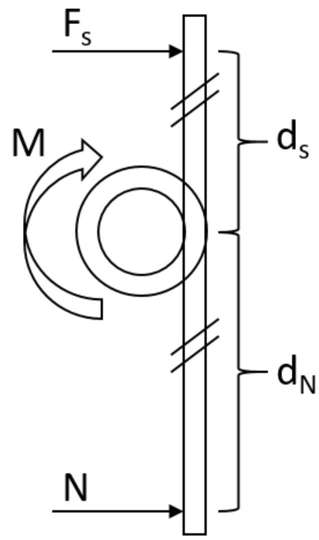

Figure (8) – Clamping Force Free Body Diagram

Taking the sum of the moments gives:

$$\sum M = N\,d_N - F_s\,d_s = 0$$
$$\Longrightarrow N\,d_N = F_s\,d_s$$
$$\Longrightarrow F_s = \frac{N\,d_N}{d_s}$$

Taking N = 50 lbf, d_N = 7 in, and d_s = 2 in gives:

$$F_s = \frac{(50)(7)}{2}$$
$$F_s = 175\ lbf$$

This shows that a spring force of 175 lbf is needed. In order to get this amount of force over a small amount of space, a gas spring will be used on each side. A concern about the ramping up effect of gas springs was brought up in the detailed design review. There was concern that in different parts of the stroke there would be differing force values, which is accurate due to the compression of the gas inside the tube. However, the rated force values are the minimum values, which happen at the most extended point. This means that the force will only get stronger as the spring is compressed, which will cause the clamping force to become greater – not a bad thing. For this reason, the gas springs were left in the design.

DESCRIPTION OF SPECIFIC COMPONENTS

Atlas Copco LSR38 S120-CW/2 Pneumatic Straight Grinder

Figure (9) – Atlas Copco Grinder

During the grinder selection process, many different brands were considered. Due to Atlas Copco already having a large presence in Stupp's plant, as well as their impressive performance characteristics, it was decided to use this brand. By the design of the system, it was obvious a straight grinder would have to be chosen in order to keep the form factor of the device small. This is due to needing to use a cone style abrasive wheel, instead of a disk style that is used on angle grinders. Further, straight grinders tend to be more compact and lighter weight than their angled counterparts.

This specific model was chosen for a number of reasons. It is lightweight, merely 3.5 pounds, which is helpful in keeping the total weight of the system below the 50 pound limit. The total length is barely over a foot, allowing the device to stay as compact as possible.

The grinder power, compared to the ones in current use at Stupp Bridge, is considerably less. On the other hand, though, the speed of the grinder is double what is currently being used. Whether or not this less powerful straight grinder would be adequate was an ongoing debate for several days. It was decided that it would be, for the following reasons. The first is that its use will be in a much more controlled manner. Since all the grinding is being done by hand currently, the force by the grinder being pushed into the steel will fluctuate based on how hard the operator pushes. If the operator pushes with a great amount of force, large horsepower grinders are necessary to keep from stalling. On this device, however, the pushing force will be constant and controlled so as to not stall the grinder. This is important because it allows for a less powerful grinder to be used with the same results.

Groschopp Inc. 24VDC Motor

Figure (10) – Groschopp Inc. Motor

Selection of the drive motors was another aspect of the project that took considerable time to figure out. A big initial hurdle during the research stage was sifting through all the Chinese manufacturers of low quality motors. It seemed that these were the only ones that showed up on Google without more specific search queries. Once the search results were specified to only include products made in the USA, it was time to go through the numerous manufacturers to find the motor with the right balance of weight, size, torque, and speed. This was somewhat challenging, due to the fact that a high torque rating was necessary, but slow speeds usually come as a result of this high torque. In addition, higher torque oftentimes brings larger size and weight with it. Fortunately, we found a semi-custom manufacturer that produces exactly the type of motors we needed for this project.

Groschopp, Incorporated is an electrical fractional horsepower and gearmotor producer located in Sioux City, Iowa. They have been in the business in one form or another for over 80 years. Due to their efficient one-plant facility, they are able to offer semi-custom arrangements of any motor they produce. The motor found to serve our purposes best is an inline shaft planetary gearmotor. It is driven with a 24VDC source current. This was one of our primary parameters for the motor, as it was undesirable to run any sort of special motor controllers. A 24VDC system can be run using simple relays or PLC outputs, and is also a common voltage for power supplies. The stock motor is a good mixture of torque and speed, able to produce over 106 lb-in of torque at 86 RPM. This torque rating is greater than the minimum required torque that was specified above, giving the device a little extra leeway. At 86 RPM, with a wheel diameter of six inches, the device will be moving at just over two feet per second. This gives a single pass cycle time of 10 to 100 seconds, for 20 to 200 foot long girders, respectively. To ensure that the motor is robust enough to handle the dusty manufacturing environment at Stupp, Groschopp will be modifying the motor to

IP65 protection standards. This means that the motor will be “totally protected against dust ingress” and “protected against low pressure water jets from any direction, with limited ingress permitted.” This will keep the motors running for a long time with minimal maintenance.

OVERVIEW OF DEVICE OPERATION

A main goal of this system was for it to be extremely simple to use. Ignoring all the negative reasons to have an operator perform a grinding operation, it does not get any simpler than having them perform the task. Since this device is being made to take the place of the operator, it must be easy to implement or else it will not get used. If they have to spend ten or fifteen minutes setting the device up and getting it ready to use, then they will eventually relegate it to the trash bin and just go back to doing it by hand. For this reason, every design decision was made in line with making an intuitively easy system operation.

To start the process, an operator will set the device on one end of the flange to be ground. This should be fairly trivial, as it will be light and compact enough to be easily moved. The bottom casters will rest on the flange, and then the operator will engage the clamping mechanism. This will cause the wheels to compress against the steel, locking the device on. Once the operator pushes the start button on the control panel, the device will traverse the length of the beam. It will automatically stop when it reaches the end, and the operator will then remove it from the finished flange and place it on the next one to be ground.

While setting the device up there are a couple things that require a bit of care. The first will be to ensure that the limit switches are oriented appropriately. Basically, they need to be turned so that both of them are pointing away from the device. In other words, if the operator is facing the device, the switch on the left should be pointing left, while the switch on the right should be pointing to the right. Even though the switches would still stop the machine with the lever oriented in either direction, it will traverse farther if they are not oriented as described above. To be absolutely as safe as possible, the levers will need to be pointed outwards.

When the device is actually moving and grinding, nothing actively needs to be done by the operator. However they should keep a passive eye on it, just in case a motor or grinder stalls out. Everything has been chosen so that the risk of this happening is minimal, but it will always be somewhat of a possibility. For this reason, the operator should just be aware and be ready to hit the emergency stop and to turn off the air if this happens. Since everything is fused, if the motors stall a fuse will be blown and will need to be replaced. If the grinders stall, the device's power will need to be cycled.

TESTING AND FURTHER WORK

Now that we are reaching the end of the second semester of the project, we are nearing completion. Even though the device is not put together yet, we have done quite a bit of individual component testing. This was done to ensure that there were not any major faults with our design or the components chosen. Atlas-Copco loaned us a test grinder, which allowed us to verify that it would be powerful enough for the task. A few of the grinding cones were also purchased. Using a wooden jig that mimicked the real design, we did testing on a scrap piece of steel from Stupp Bridge Company. We were mainly interested in proving that the grinding cone would not wear too fast. If the cones wore really quickly, it would be too expensive and cause too much hassle for Stupp’s operators. Fortunately, through this testing, we determined that a single cone should last for at least eight full length girders.

During the assembly process there were a few minor issues. The pillow block bearings have the capability to account for shaft misalignment, and because of this they are not staying perpendicular to the central shaft when the clamping force is applied. To remedy this, spacers were made that contact the rigid sides of the bearings, ensuring that they stay perpendicular to the shaft.

The gas spring picked for the device based on the calculations ended up being too firm. It proved to be impossible to compress in any reasonable manner. A simple fix was used in which a coil spring was put on the clamp releasing screw. This allows the clamping mechanism to function in the same manner as originally intended.

In order to test the device in a realistic fashion, a wooden test beam was made. A two inch thick by twelve inch thick beam was placed in holders that help it with the thing side facing up. Since it was around ten feet long, it allowed us to practice moving the device. During the first test, a small tracking problem revealed itself. It was determined that the idler casters on the bottom of the device, being two pieces, were causing the device to move from side to side while driving. This problem was solved by replacing the caster wheel pairs with a single roller made out of Delrin, a type of plastic. To test the grinding action, thin steel plates were attached to each side of the wooden beam.

By the time the device is presented to Stupp Bridge, it will be fully operational. It is important to keep in mind that this was a first iteration of a device that has no equivalent on the commercial market. For this reason the completed device might have some issues that can only be found through extensive use. It is the hope of the team that it will prove useful for Stupp Bridge, but it is also expected that further improvements will be made by future senior project teams or by Stupp's engineers.

APPENDIX A

FUNCTION FLOW BLOCK DIAGRAM

The functional flow block diagrams for both the fully automated and the semi-automated design scheme are shown below. Even though these systems are both relatively simple processes that happen in a mostly linear manner, the flow diagrams still help in visualizing the device operation.

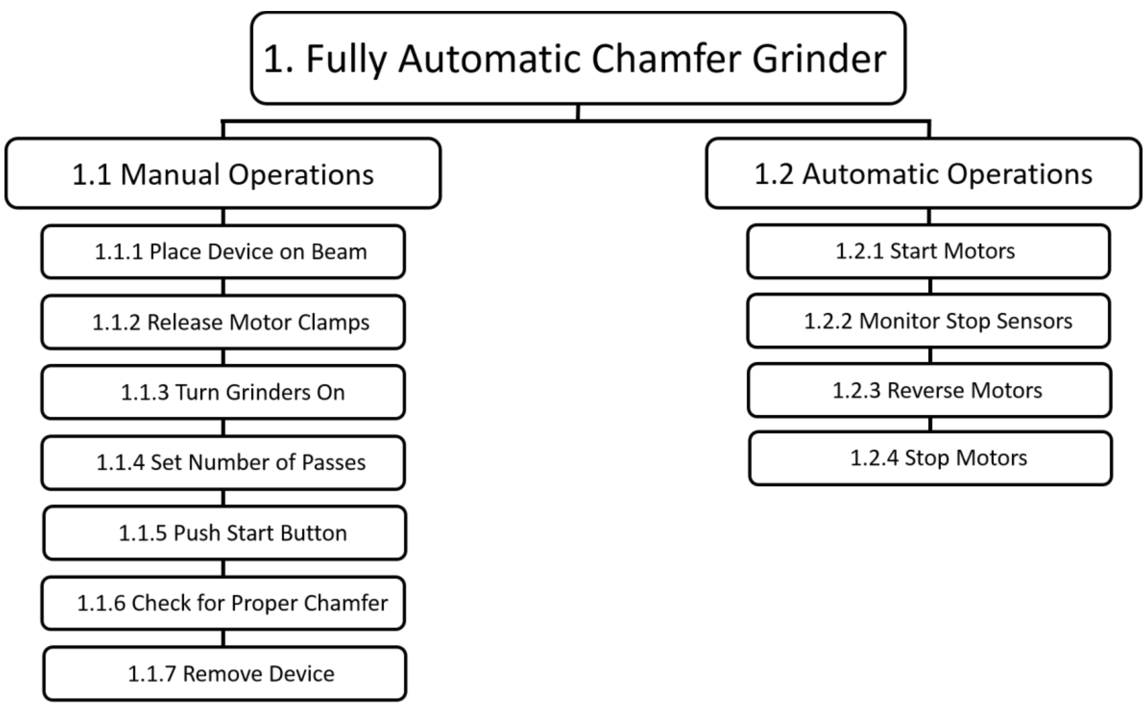

Figure (11) – Functional Flow Block Diagram Hierarchy

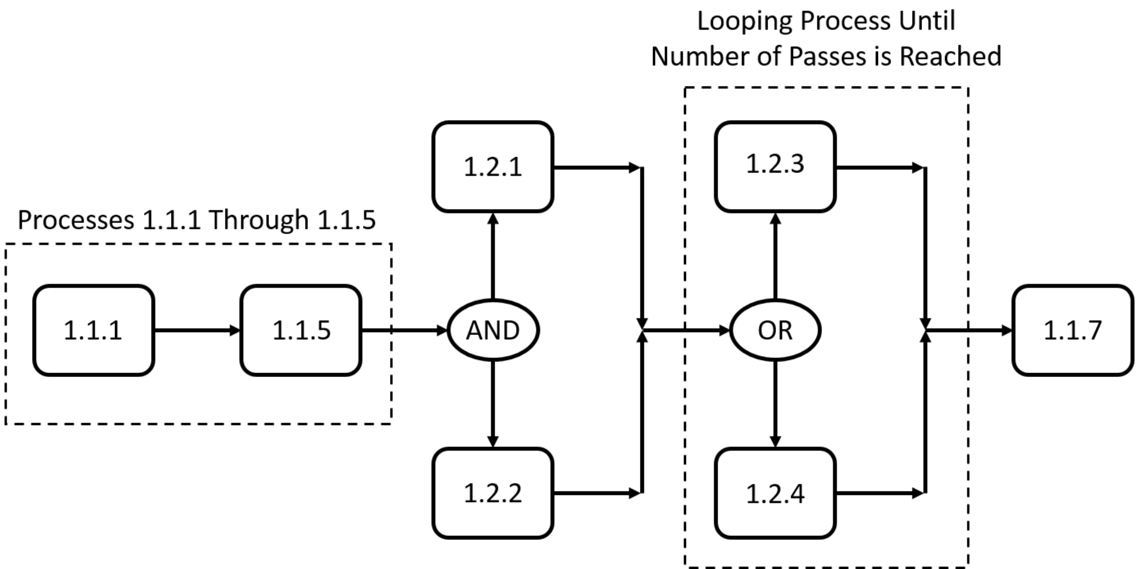

Figure (12) – Function Flow Block Diagram

APPENDIX B

DETAILED FIRST SEMESTER DRAWINGS

Using SolidWorks to model the design was a very large part of the work done. All drawings are done to scale, using parts that were either drawn by me, provided by a manufacturer, or pulled from an open source website. All models shown can be shown with additional detail if requested, and dimensioned drawings for fabrication shops are available as well. To save space, they are not included in this document.

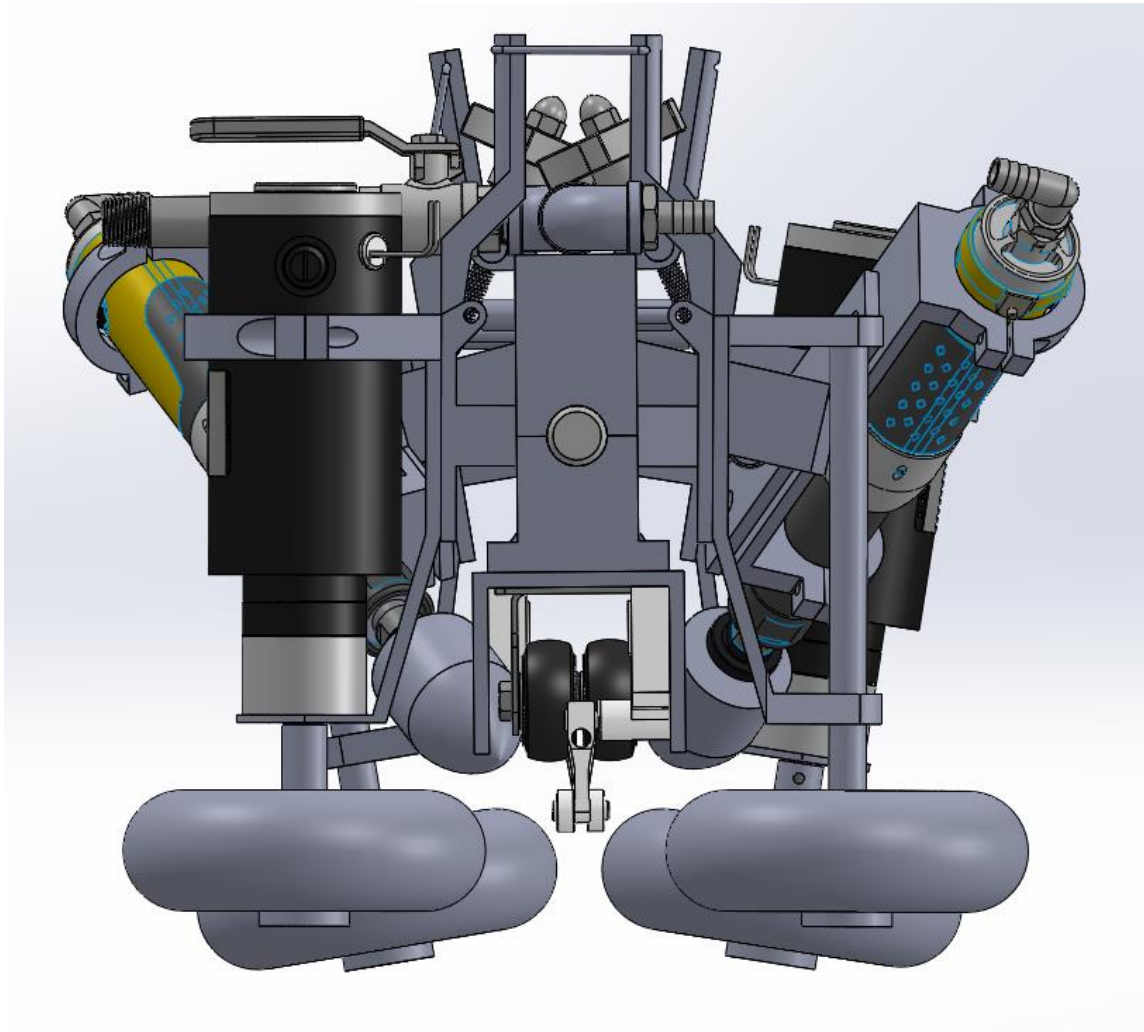

Figure (13) – First Semester Front View

Figure (13) shows a frontward view of the device. It shows how the motors and grinders pivot from the central shaft, as well as showing the bottom idler wheels that contact the top of the flange. In addition, the clamping mechanism for holding the motors open can be seen. The motor in the foreground is clamped open, while the motor in the background is not. The limit switch for this end is also in view.

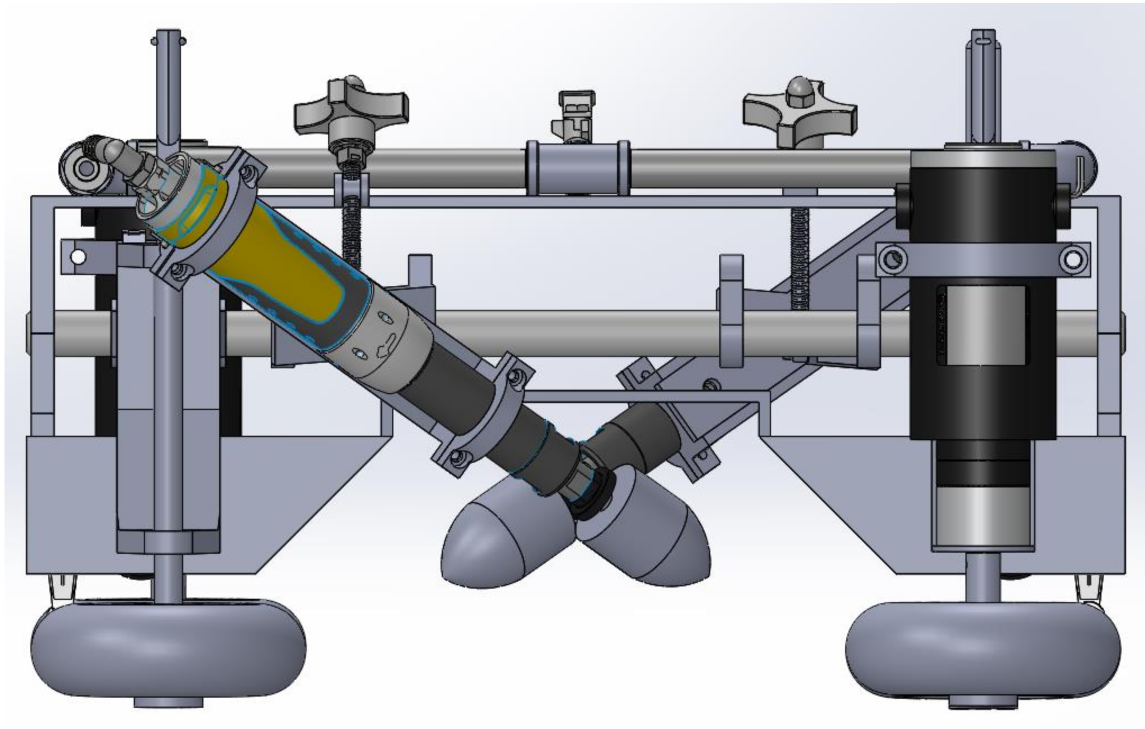

Figure (14) – First Semester Side View

The side view in Figure (14) clearly shows the length of pipe along the top, as well as the grinder tensioning dials. This view makes it very apparent how symmetric this design is.

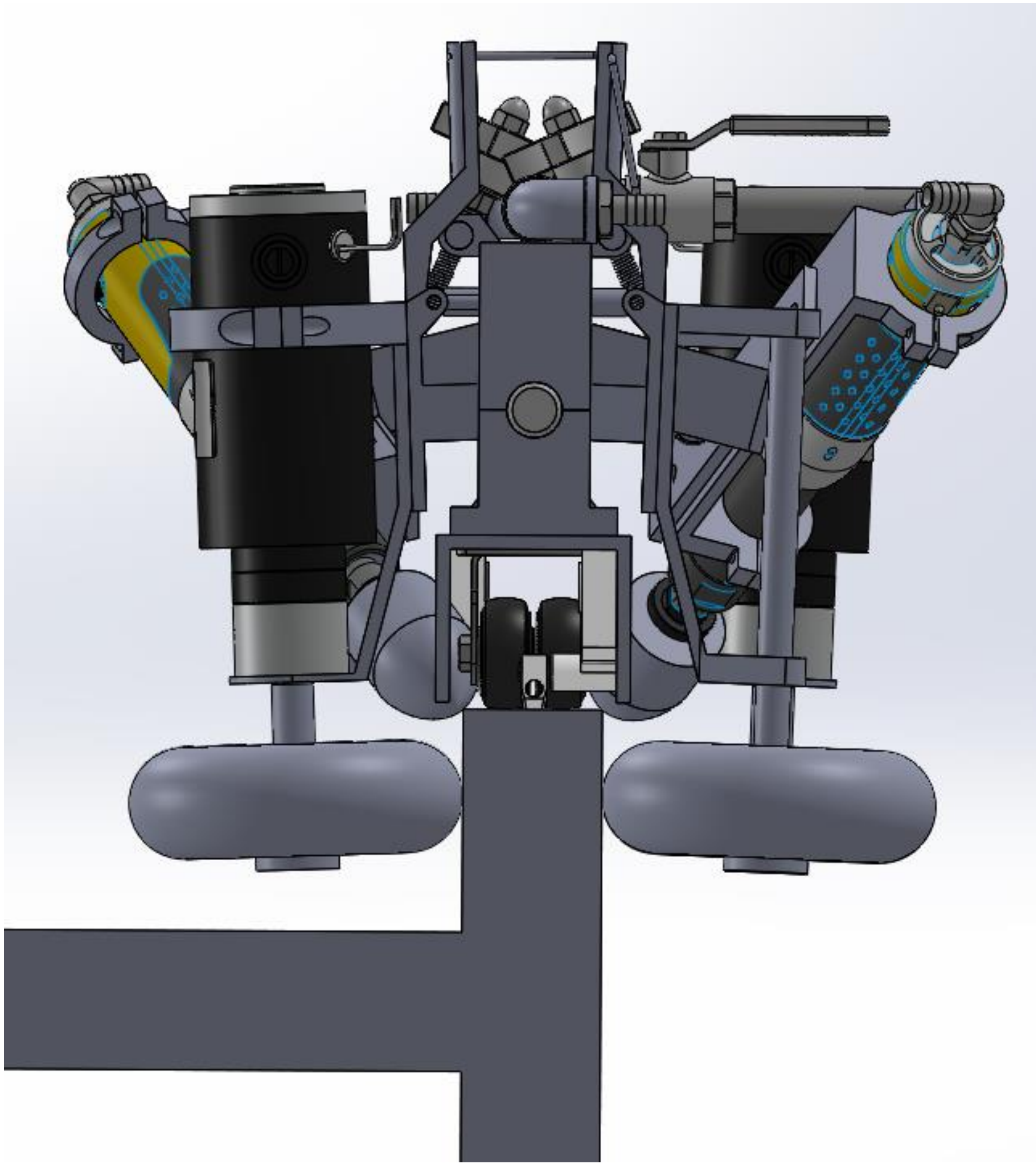

Figure (15) – First Semester Front View on Thick Beam

This shows the device on the thickest flange produced at Stupp Bridge. It can be seen that this maximum size is easily accommodated.

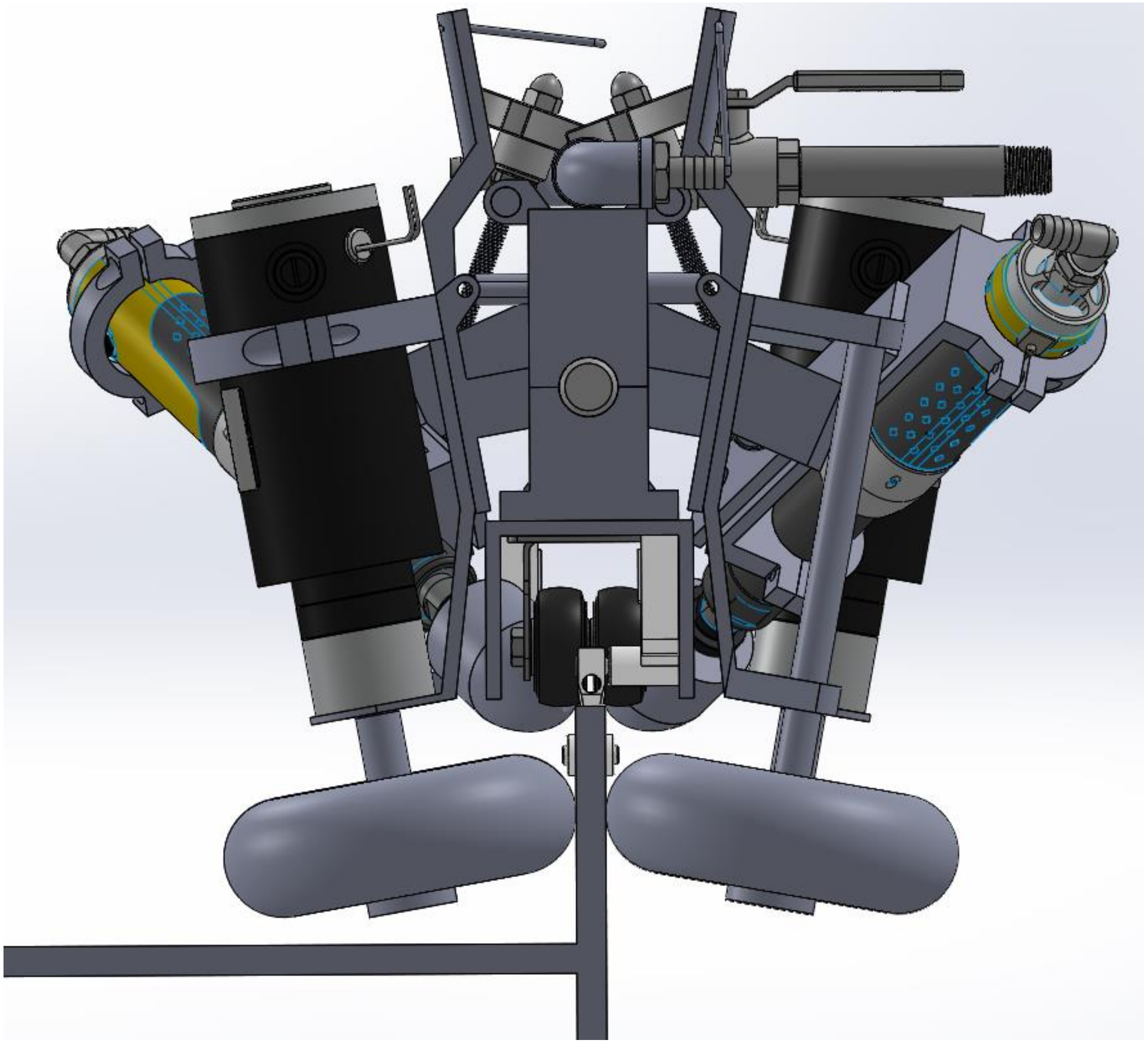

Figure (16) – First Semester Front View on Thin Beam

Similar to Figure (15), this is a view of the device on the thinnest flange at Stupp Bridge. The combination of Figure (15) and Figure (16) show that the device will clearly work for all flange sizes in between 0.5 and 2.5 inches.

APPENDIX C

DETAILED VIEWS OF FINAL DESIGN

Here you will find the final design of the grinding system. All models were produced by me, supplied by a manufacturer, or downloaded from an open source CAD library.

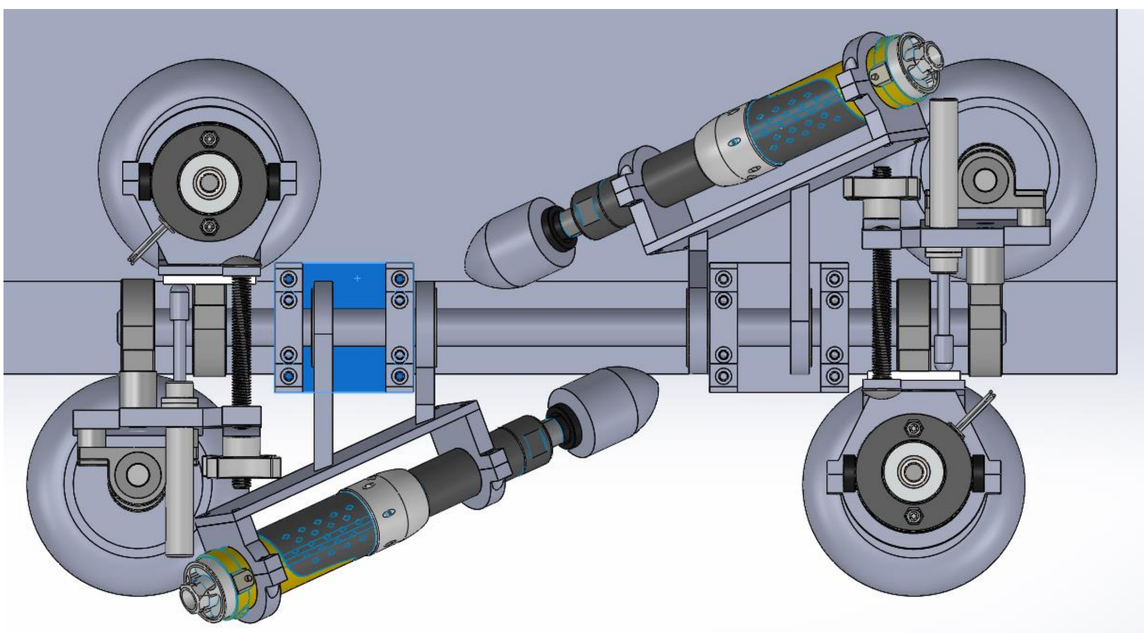

Figure (17) – Second Semester Top View on Thick Beam

This view shows how the grinders, motors, and idler wheels all pivot from the central shaft. You can see how the design was simplified to use more basic components.

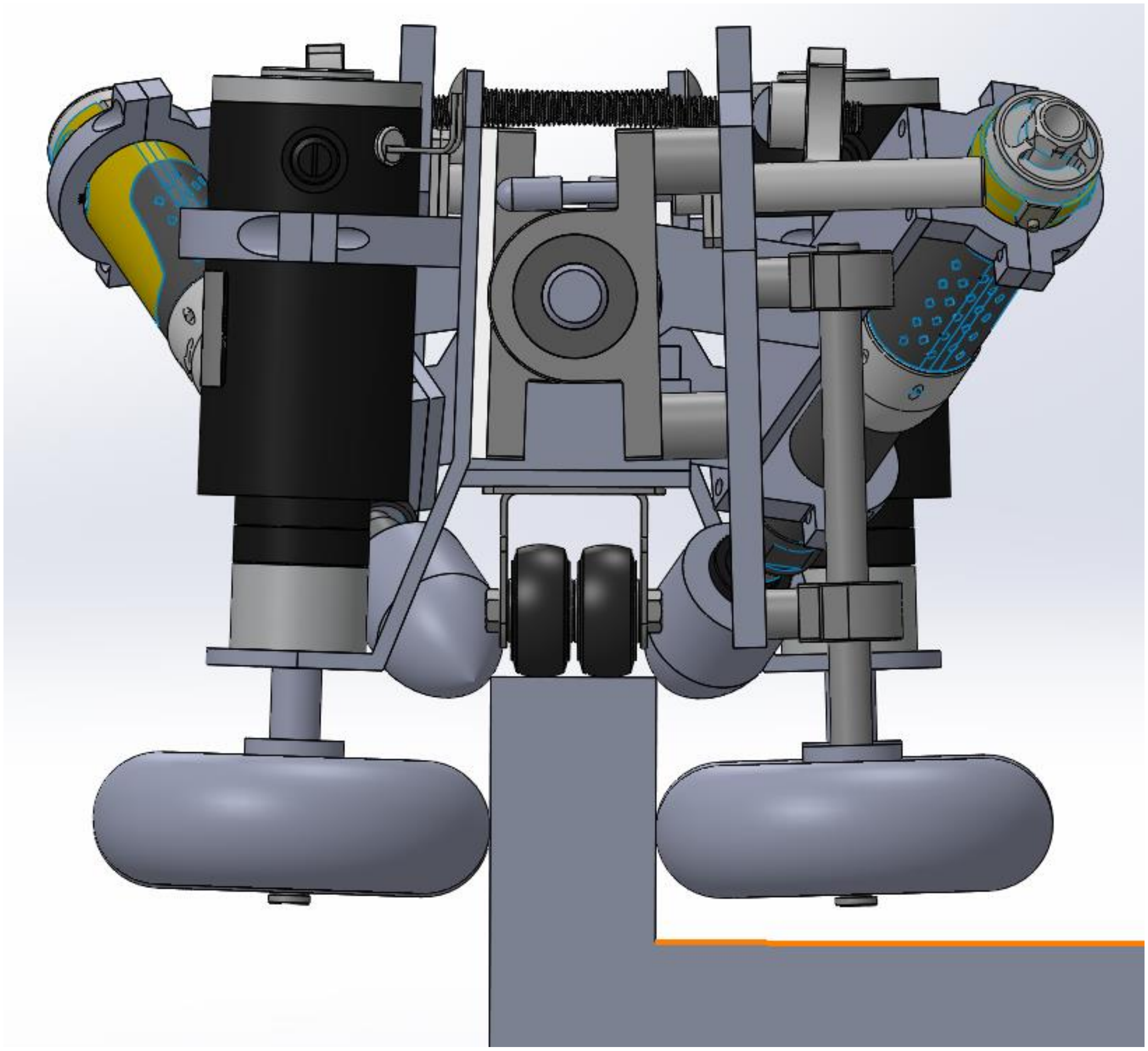

Figure (18) – Second Semester Side View on Thick Beam

This side view is good for seeing how the system attached to the beam. You can see the gas spring that provides the motor clamping force, as well as the screw mechanism that the operator uses to detach the device from the beam.