Structural Design of a Desktop CNC Mill

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***Abstract*—CNC (Computer Numerical Controlled) mills are complex devices which experience both static and dynamic loads. Through a deeper understanding of the milling operation through a statics perspective, a first order estimate for the structural design is able to be made. Knowledge of the potential trade offs will allow the design of a new desktop CNC mill that is structurally sound and meets other functional requirements.**

***Index Terms*—CNC; Structural Design; End Milling; Prediction.**

I. INTRODUCTION

Since people have began using metals, they have been concerned with advancing metalworking to shape metals. Machining, a process by which a power-driven tool uses a subtractive cutting process is an effective way to shape a material. Milling, a type of machining, involves a rapidly rotating cutting tool with multiple cutting edges which come into contact with a work piece. The appeal of milling is that it is an effective way to machine flat or curved shapes on both the inside and outside surfaces of a work piece. A common milling machine is the Bridgeport machine found in many machine shops. Milling machines allow repeatability and precision, which make them effective in an industrial setting. Since modern electronics started taking off in the middle of the 20th century, more automation has been integrated into milling machines, resulting in Computer Numerical Controlled (CNC) milling machines. These machines use a combination of sensors, motors and control systems in order to perform the precise milling operations[1], [2].

However, a CNC mill will cost from around $3000 to upwards of $10000, even for those meant for small scale use. In addition, operating a CNC mill is challenging as it requires knowledge of G-code, a language which guides the positioning of the cutting tool during a machining operation. G-code is a difficult programming language to learn, and errors made in G-code could cause damage to both the machine and the work material[3].

This summer, I am attempting to design and fabricate a CNC mill. The idea is to create a low cost mill that could fit compactly on a tabletop in order to bring the CNC mill to more people. I hope that this mill would be utilized instead of using more expensive additive rapid prototyping technologies for custom parts and class projects. Current additive rapid prototyping technologies do not result in parts that are necessarily structural as they print the parts in layers that are weak against shear stresses. Furthermore, current technologies are only able to utilize polymers like ABS, which is not ideal if parts that need to withstand significant loads are to be fabricated[4].

To design the mill, is is necessary to establish some idea about the typical loads experienced by a machine during a milling operation. This will allow one to determine what structures to use for various parts of the mill, as well as what components to purchase. As a first order estimate, one can only look at static loading scenarios, and ignore dynamic loads. This may prove to be an erroneous path as vibrational modes of mills are important. However, if the structure stiff enough such that static displacements are minimal, this estimate could be valid.

Figure 1. Roland MDX-15 is a desktop CNC milling machine produced by Roland DGA. While it does meet most of the requirements for the device being designed, it is unable to machine anything beyond aluminum and has a prohibitive cost of around $4000[5]. Furthermore, it can only machine work pieces that are 152.4 mm x 101.6 mm x 60.5 mm in size, which is smaller than desirable.

In any design project, there are different sets of requirements, usually set by different stakeholders to the project. In the next section, the more specific technical requirements will be detailed. The list below is a subset of the requirements for the device set by the customers, the eventual end users of the machine.

1) The machine must be compact. This is too allow it to fit on a table top or workbench easily.
2) As a safety measure, the machine should be easily encloseable to prevent chips of metal from spraying around. This is again to make it easier to use the machine in a classroom or tabletop setting.
3) Materials that can be machined can be as weak as polystyrene and as strong as mild steel
4) Cutting speeds and feed rates (rates at which the cutting tool is moving) are not too important, as long as most

jobs get completed within a day. A maximum rate is to be set where the milling operation would require coolant as this would add unnecessary complication to the machine.

5) The machine should be able to handle a workpiece that is 20x20x20cm. This is a bound that allows most parts that students need to be fabricated. As a reach goal, the machine should have one axis that is unobstructed, allowing parts that are longer in a single dimension to still be made.
6) The precision of the milling operations should be $1mil$ ($25\mu m$). This translates to a maximum error in any direction of 0.5mil if one uses a cube where the length of the diagonal is $2mil$ (therefore length from center to a vertex is 1 mil). The shortest distance from the center of the cube to any face is then $0.5mil$.
7) The machine should cost around $500 in parts. This would mean using standard components where possible and restricting the structure to common materials like steel and aluminum.

Although this list is not complete, it is able to constrain the design enough and provide guidance in design decisions.

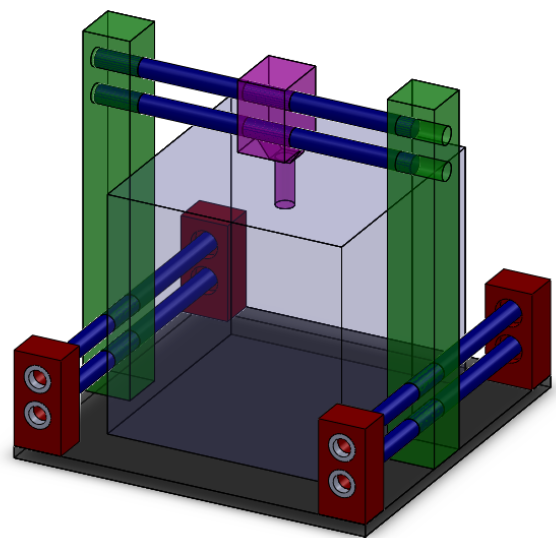

Figure 2. Basic structural design. In purple is the cutter and the carriage used to support it. The shafts used to move along two axes are shown in blue, while their pillow blocks are in red. The green columns are the ones supporting the x axis. The third axis is not pictured, and will probably be integrated into the carriage of the cutter. The cube in the middle is the 200 by 200 by 200mm space that a workpiece could occupy.

II. Basic design and requirements

Based on those functional requirements, a basic design and a list of functional requirements are needed. The basic design will allow calculations to be made whether the functional requirements are realistic, and what is necessary for them to be met.

A major initial design decision is where to position the actuation of the system. One choice is to mount the cutting head on a gantry that can move in the horizontal plane, with a vertical degree of freedom. Another design would be to mimic a Bridgeport mill, where a table can move in the x,y and z directions, and the cutting head can be lowered and raised as well as rotated. Although the Bridgeport design is more flexible as it could allow the cutting of diagonal faces, it is less compact than the gantry design. The gantry design allows the work piece to be enclosed in an area easily, which is preferable for a device meant for a classroom setting. Therefore, the design for the mill would follow a gantry style. This is very similar to the Roland MDX-15 machine shown in Figure 1.

The actuation of the cutting tool has to be done by some sort of linear actuator. Different options that are available are a rack and pinion, a belt drive and a leadscrew. The rack and pinion actuation is inferior to both in two crucial ways. If chips fall onto the rack, it may cause the pinion to roll over the chips, leading to misalignment and maybe jumping of teeth. Another concern is that the driving motor will need to be geared down so that one turn of the motor does not cause too much linear movement, making the linear movement not very sensitive. This is the same for the belt driven system, where the pulley has to be geared down. Using a leadscrew results in a significant speed reduction dependent on the pitch angle of the leadscrew.

With the limitations discussed above and the requirement for workpiece size, the basic structural design shown in Figure 2 was developed. Two axes of the machine would be actuated using leadscrews while the third would be actuated by a rack and pinion. Most linear slides consist of a leadscrew actuator in the middle with two shafts running parallel to provide support[6]. Choosing which linear slide to use is important in order to make sure the precision desired is achieved and the slides do not get damaged in normal loading conditions. From the design in Figure 2, the following technical concerns come up:

- What are the forces experienced by the end mills in different operations?
- What is the required geometry and design for a beam to support the cutting tool at the required stiffness?
- What are the loads experienced by the bearings, and what kind of bearings would be appropriate?
- What is the required torque on the leadscrew to withstand the loads applied to it? This question would help determine the torque of the motors required.

Answering these questions will enable the design of the basic structure of the machine that could meet the functional requirements prescribed previously.

III. Cutting forces

To formalize a discussion on cutting forces experienced in milling operations, a deeper knowledge of milling is required. There are many different operations in milling, which is shown in Figure 3. As expected, the different forces all cause different loads on the cutter. In addition, the different operations mean that there are many different cutters available, from end mills to face cutting cutters. Although this does mean that the different operations cause and experience different loads, there are a few basic factors that span most operations.

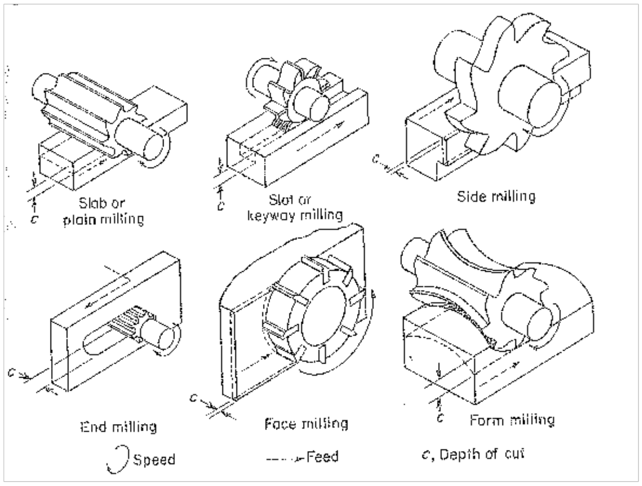

Figure 3. Some milling operations. Important details to note is the rotation of the cutter, direction of feed and depth of cut[2]. It is expected for the machine being designed to carry out the first five operations(all other than form milling).

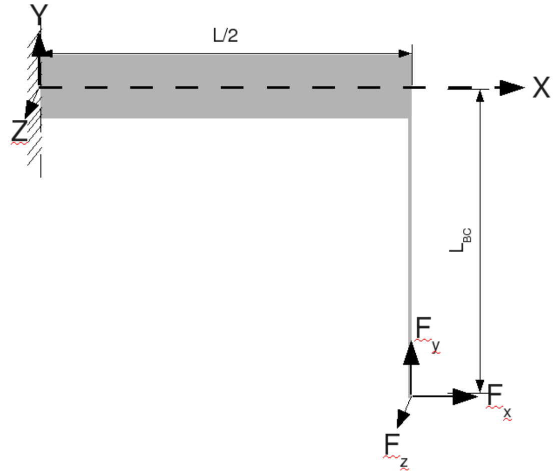

Figure 4. Forces on the main support beam from the cutter. F_x,F_y and F_z are 2500N, 625N and 1250N respectively. L(25cm) is the length of the beam and L_{BC}(25cm) is the length from the centroid of the contact area of the cutter to the centroid of the beam cross section. Both L and L_{BC} are estimates based on the required dimensions for the work area.

Firstly, the actual load experienced by the cutter depends on the contact area between the cutter and the work piece. The greater the contact area, the more actual cutting of material occurs, therefore more load. The next factor is the feed rate, the speed of which a workpiece is being moved towards a cutter. The feed rate proportionally increases the forces experienced, especially tangential to the cutter. At the same time, the spindle rate of the cutter matters, as the greater the spindle rate, the greater the force. Finally, the type of material being worked on matters. Machining guides provide tables with the machinability factors of different materials, which could be used as a linear quantity to scale feed rates and cutting speeds[2], [7].

A survey of published papers rarely revealed exact numerical data. One tabulated the forces in the radial, tangential and axial directions for a few different operations[8]. The maximum forces were in the tangential direction, followed by the radial and finally the axial, in a consistent $4:2:1$ pattern. In all the operations, the maximum forces in those three directions was about 2500N, 1250N and 625N. These will be used as the load for all subsequent calculations. It is important to understand the condition for this loading. The workpiece was ASSAB760 plain carbon steel and the cutter was a high speed steel end mill running at a feed rate of 4mm/min and a spindle speed of 600rpm. These settings are on the high side, and the machine being designed does not need to meet these standards. However, using this as a baseline comparison will aid in defining what is needed.

IV. Main Support Beam

Based on the previous section on cutting forces, the geometry of main support beam needs to be determined with the goal to minimize deflections of the cutting tool. To get a conservative estimate, the beam will be considered to be rigidly supported at both ends. This assumption would result in a smaller deflection if the ends could also deflect, but should be good enough for an estimate. The support columns for the beam can made much more rigid than the beam without changing the design, thus this assumption should be valid.

Figure 4 shows the forces experienced by the cutting tool and the resulting forces and moments on the beam. These will cause five effects which will cause a deflection of the cutting tool[9]. They are:

1) Torsion of the beam will cause a deflection in both y and z and is one of the biggest effects. Multiplying by $L_{BC}$is required to translate the angle of twist of the beam to a deflection at the tip.

$$\delta z_1 = \frac{M_x(L/2)}{E \cdot I} \cdot L_{BC}$$

$$\delta y_1 = \frac{M_x(L/2)}{E \cdot I} \cdot L_{BC}$$

2) The moment from $F_x$will cause a deflection in y.

$$\delta y_2 = \frac{M_y \cdot (L/2)^3}{3 \cdot E \cdot I_{zz}}$$

3) The force F_z will cause a deflection in z.

$$\delta z_2 = \frac{F_z \cdot (L/2)^3}{3 \cdot E \cdot I_{yy}}$$

4) Similarly, the force F_y will cause a deflection in y.

$$\delta y_3 = \frac{F_z \cdot (L/2)^3}{3 \cdot E \cdot I_{zz}}$$

5) Finally, the force F_x will cause a stretching in x

$$\delta x = \frac{F_x}{A}$$

To meet the desired tolerance of $25\mu m(1mil)$, the sum of each of of the deflections needs to be at most $12.5\mu m$. This is to set the deflection to a sphere of $25\mu m$ around the desired point. This is again another approximation, but is still accurate.

Geometry	A	I_{yy}	I_{zz}	J
Circular with radius r [mm]	$r = 10$	$r = 10$	$r = 10$	$r = 10$
Rectangle height $3b$ and base b [mm]	$b = 1$	$b = 25.1$	$b = 30.2$	$b = 29.8$
Two circles of radius r[mm] and distance $50mm$ apart	$r = 0.691$	$r = 4.99$	$r = 20.2$	$r = 15.1$
Two circles of radius r[mm] and distance $80mm$ apart	$r = 0.691$	$r = 3.13$	$r = 13.5$	$r = 9.85$

Table I
GEOMETRIC PARAMETERS NEEDED TO ACHIEVE NEEDED VALUES FOR CROSS SECTIONAL AREA, AREA MOMENT OF INERTIA AND POLAR MOMENT OF INERTIA. IT IS CLEAR THAT I_{zz} IS THE MAIN CONSTRAINT, AND IF THE BEAM IS DESIGNED SUCH THAT IT MEETS THE VALUE FOR I_{zz}, ALL THE OTHER CONSTRAINTS WOULD BE MET. THE NEXT OBSERVATION IS THAT THE FOURTH CASE GIVES THE BEST BALANCE OF STRENGTH TO WEIGHT AS THE MATERIAL NEEDED IS MINIMAL.

For the subsequent calculations, I have chosen, structural steel, which has an elastic modulus of 200GPa and Poisson's ratio of 0.3. This is based on observations of what is conventionally used for linear shafts in linear guides[6].

Solving for all these equations gave a set of values for A, I_{yy}, I_{zz} and J.It was quickly apparent that using the forces as found above would require a much larger beam than would be reasonable based on the space constraints the machine had to meet. To mitigate this problem, the forces were reduced by a factor of 10. This resulted in the following values. Achieving the factor of 10 reduction in speed is easily achieved by reducing feed rates and reducing the contact area of the cutter and workpiece. Although this will result in a longer machining time, this is acceptable for a machine meant for student use in a non production environment.

$$A = 3 \times 10^{-6} m^2$$

$$I_{yy} = 99 \times 10^{-9} m^4$$

$$I_{zz} = 1.88 \times 10^{-6} m^4$$

$$J = 1.98 \times 10^{-6} m^4$$

There are a few ways to try determining the appropriate geometry for the beam. The Table 1 shows the some of the possible permutations:

From the table, it is clear that the best choice is using the two shafts of radius 13.5mm and 80.0mm apart as it uses the least amount of material. This is easy to implement as most leadscrew designs consist of a leadscrew in the middle and two linear shafts to support it. Furthermore, the mass of the beam would be minimal using the two tube solution, as the cross sectional area of the beam would be the least. In designing the beam, a low mass and high stiffness help move the natural frequency of vibration up to higher frequencies. This means that any excitation to the beam at lower frequencies will not cause as much vibrations, making the beam more vibrationally stable.

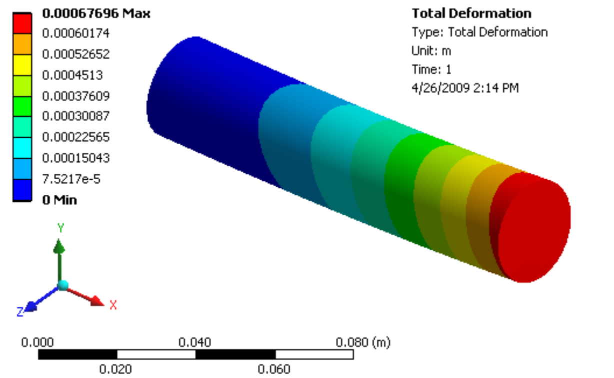

Figure 5. Deformation of the beam under the load. The maximum deformation is $67.5 \times 10^{-5} m$ which is above what desired, but expected since this is an exaggerated scenario. Since it is well within an order of magnitude of the tolerance, the simulation does validate the design.

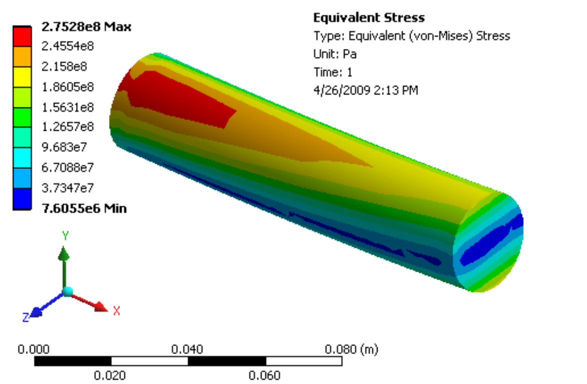

Figure 6. Stresses on the beam. The maximum von-Mises stress on the beam, 275MPa is still below the tensile stress of steel which is usually around 450MPa[9].

As a final check, the single beam of with radius $13.5mm$ is put under the loading condition to make sure the beam does not yield. This is an exaggerated scenario, as with two beams of this size, the stresses experienced would be less. The results of the simulation are shown in Figure 5 and Figure 6. Both results validate the analysis above, as the total deformation experienced by the beam is below the limit, and the von Mises stress of the beam is below the elastic yield strength of the material.

Through this analysis, it is apparent that it is possible to construct a suitable beam to bear the loads of the cutter. However, the feed rate of the cutter would need to be reduced when cutting strong materials like steel, in order to limit the forces on the cutter. If the cutter has to experience a tangential load of $250N$ and the corresponding radial and axial loads, using two cylindrical shafts of diameter of radius 13.5mm and 80.0mm apart would be sufficient.

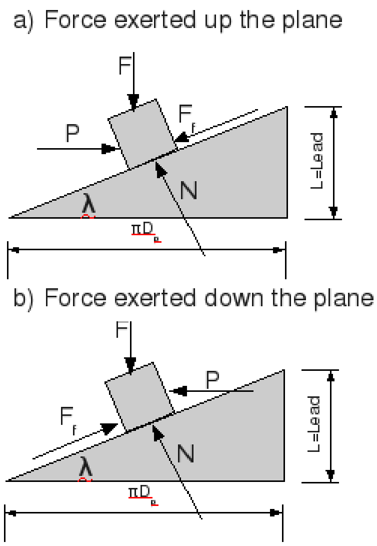

Figure 7. Free body diagram of an "unwrapped" lead screw for both cases of the force being applied up and down the plane of the screw thread.

V. Leadscrew choice

The leadscrew would be the primary means of actuating most axes in the machine. The forces provided by the lead screw can be understood if a screw is unwrapped, forming a triangle as shown in Figure 7. The load on the leadscrew, F, is the main parameter to be concerned with. Other parameters to is the size of the screw, indicated by the pitch diameter, D_p, the lead of the screw, L, and the coefficient of friction, f. Note that the lead of the screw is defined as the axial distance that the screw would move in one complete revolution[10].

The forces acting on each triangle are the load, the driving force, normal contact force and friction. There are two different ways the forces can act, when the driving force is exerted up the plane and when the driving force is exerted down the plane. For the first case, setting the forces in the horizontal and vertical directions to zero yields,

$$\sum F_x = 0 \Rightarrow P = F_f \cos\lambda + N \sin\lambda$$

$$\sum F_y = 0 \Rightarrow F + F_f \sin\lambda = N \cos\lambda$$

Since $F_f = f \cdot N$, where f is the coefficient of dynamic friction,

$$P = F \frac{f \cdot \cos\lambda + \sin\lambda}{\cos\lambda - f \cdot \sin\lambda}$$

Since the torque, T, exerted on the leadscrew can be expressed as $T = P \cdot D_p/2$,

$$T_u = \frac{FD_p}{2}\left[\frac{\tan\lambda + f}{1 - f \cdot \tan\lambda}\right]$$

where the subscript on the torque is to denote the direction of the force relative to the plane. Similarly, when the force is being exerted down the plane,

$$T_d = \frac{FD_p}{2}\left[\frac{f - \tan\lambda}{1 + f \cdot \tan\lambda}\right]$$

These two equations show that the relative magnitudes of f and $\tan f$ are important. When the screw thread is very steep, the friction force may not be able to overcome the tendency of the load to slide down the plane and the load will slide. Since lead angles are usually small, the friction force is usually large enough to oppose the load and keep it from sliding down the plane. This means that the screw is self locking, which is desirable, as it limits movement without active control.

The load F that needs to be transmitted through the leadscrew varies, as it has to both accelerate and decelerate during cutting operations. Therefore, only once the rest of the system has been designed can the load be determined accurately. In addition, other constraints need to be set. Therefore, the following are the steps needed to utilize the equations above for torque.

1) Calculate the tensile stress area based on the load and the maximum allowed tensile stress
$$A_t = \frac{F}{\sigma_a}$$
2) Using tables for lead screws, determine the leadscrew with the closest acceptable A_t and find its corresponding A_s, shear stress area[10].
3) Calculate the A_{s1} based on the load and the acceptable shear stress on the threads of the leadscrew
$$A_{s1} = \frac{F}{\tau_a}$$
4) Calculate the length of the of the nut on the screw
$$h = A_{s1}\frac{1}{A_s}$$
5) Using the pitch, p ($p = L$), and minimum pitch diameter, D_p, of the screw with the A_t found above, find the lead angle.
$$\lambda = \tan^{-1}\frac{L}{\pi D_p}$$
6) If we assume a coefficient of friction, we can now use the equation found above to determine the torque required to move the load.

Therefore, if appropriate constraints for shear stress and tensile stress can be determined, the problem would be constrained sufficiently to select a suitable leadscrew.

It must be noted that lead screws are not the most efficient method of transmitting torque as they experience a lot of friction. A more efficient method is using ball-screws which have a layer of ball bearings between the screw and the nut. However, based on the expected duty and loads on the leadscrew, its lack of efficiency may be suitably compensated by its lower cost.

VI. Forces on bearings

Bearings are to be used at the interface to all rotating parts. They will be under significant mechanical load, which must be accounted for in choosing which bearing to use. Bearings can handle both radial and thrust loads but different types of bearings have different abilities to withstand either. In addition, since bearings are going to be rotating during use, there is

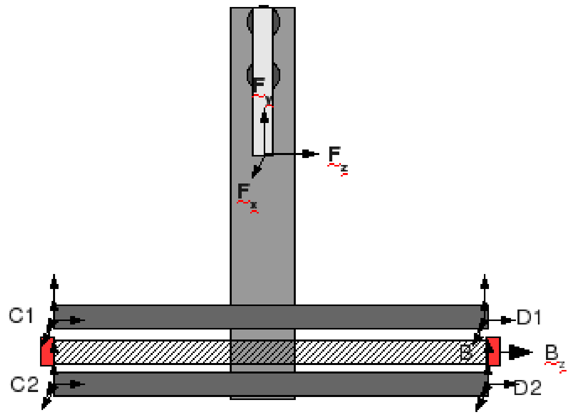

Figure 8. Loads on the bearings. The bearings (in red) will only have to support F_z as the linear shafts should help support the forces and moments in the other directions.

inherent wear to the system due to friction. Different bearings have different service lives, and choosing one with the right service life for the expected loads and usage is important.

Going forward with the design using a leadscrew to propel the the cutting head, we know that the shafts running parallel to the leadscrew will bear most of the load from the cutting head (Figure 8). However, these shafts will not be able to sustain any load along their axis as they are designed to allow free moment along them. Therefore, the forces in the z direction(Figure 8) will have to be borne primarily by the bearings on the lead screws. This is assuming the shafts are perfectly smooth and that all shafts and column are as depicted in Figure 8 and not deformed significantly due to loading. Significant deformation or misalignment of the shafts or column would cause the bearings to have to bear a significant radial load in addition to their axial load.

Taking the maximum force again, the bearing will have to withstand an axial force of about $250N$. This is well within the loads that normal bearings have to bear. To mitigate the assumptions made previously, that the bearings only have to bear axial loads, we also have to consider how the bearing bears moment or radial loads. The basis for the previous assumptions is that the leadscrew in the bearing is perfectly lined up, which is not necessarily true. Slight misalignment would cause some of the axial load to be applied in a radial direction. This means that using a pure thrust bearing would not be ideal.

Consequently, it is recommended to use tapered roller bearings(Figure 9). These bearings have a taper where the greater the angle, the greater the axial force that the bearing can withstand. Furthermore, the taper distributes the load to a bigger surface area, which allows the bearing to support greater loads compared to spherical (ball) bearings. The geometry also means that the tangential speed of the surface of each of the rollers are the same throughout the contact area, preventing differential friction which helps reduce rolling friction and wear. In addition, the rollers are guided by a flange on the inner ring which stops the rollers from sliding out at high speed due to their momentum[10].

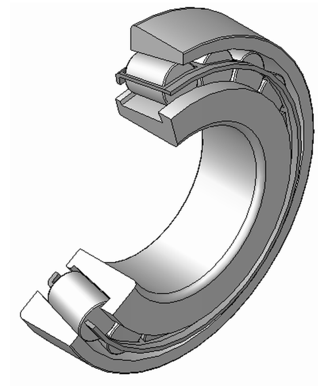

Figure 9. Tapered roller bearings. The greater the angle of the taper, the greater the axial load that the bearing can withstand. The taper also allows a greater contact area between the rollers and the rings, enabling greater loads to be experienced by the bearings.

Since it is possible to find appropriate bearings, the basic design of this subsection is valid. Once the columns and the cutting tool and carriage are designed, a more accurate value for the load on the bearings will available. The same bearings could be used for the leadscrew for actuating the carriage. Although this would be overkill, this will minimize the number of unique parts in machine, which will help reduce costs.

VII. Diagnosis and Reflection

This project was my first experience in carrying out any analysis before designing a system completely. It required me to take a different approach. First, define and prioritize the functional requirements of the system. Based on those requirements, come up with a basic design that would meet the requirements. Then, use the basic design and knowledge about the loading of the device to identify key design constraints. Perform the necessary analysis on those key constraints in order to quantify them. In performing the analysis, it may be desirable to simplify the system somewhat. With the quantified design constraints, it is possible to proceed to a more fleshed out design. Once that design is made, the design should be verified to ascertain if the design constraints were not exceeded. This should be an iterative process that will probably take a few cycles before a design that suitably achieves its requirements is achieved. At each iteration, the models could be made more complex, to account for not just the statics, but also the dynamics of the system and concerns about manufacturing.

The main problems that I faced in this project were in identifying the key design decisions to make. This is where I feel knowledge learned in my statics came to bear the most. To paraphrase the professor, the point of the class is not just to be able to crunch some numbers, but to develop an ability to judge systems qualitatively as well. In executing the analysis on this system, I felt that if I had to perform a very thorough analysis, I would have to spend a lot more time and not yield too much. It would be more effective for me to simplify the model, and get values that were not as precise. Making the decision about the complexities of the models was where a lot of my qualitative understanding came through.

This paper does not sufficiently or completely describe or show the difficulties I faced in carrying out the project. When I started working on the problem, I was not too certain about where the crucial design decisions needed to be made. It seemed that a lot of the system would need to be analyzed then designed appropriately. Only after consultation with others was I able to scope the design decisions I needed to make sufficiently. I was persistently caught in the conflict between doing too little and too much analysis, and found it difficult keeping a middle ground. My belief throughout this time was that even if I performed too much analysis, all models would have limitations, the results would not be that much more useful. Therefore, I made the analysis easier by using simple models but making my assumptions clear. This would allow me to follow my logic in the future, and improve upon the models.

VIII. Conclusion and Future work

The requirements set by the user for the desktop CNC milling machine are capable of being met. For steel to be worked on, the feed rate and spindle speed has to be decreased to reduce the forces experienced by the cutter. The structure is currently designed for a maximum tangential load from the cutter of 250N, and corresponding axial and radial load . Using linear slides with a leadscrew drive would be an effective way to actuate the system and move everything. Analysis through free body diagrams and load displacement calculations demonstrate that meeting the requirements for workpiece size is possible. Moreover, since steel is sufficient to provide rigidity to the structure, the machine meets its requirement for using common materials.

This could be suitable for a class project in mechanical design. Since trying to determine how to make the models for loading took up most of my time, it could be part of a series of exercises in identifying design points. This is a useful skill to nurture, as the ability to come up with good models is important in order to be able to use the analytical tools we have developed.

As for the machine itself, it will be designed based on these calculations this summer, and will be tested to determine the validity of the analysis. Hopefully it will show that most of the assumptions made were valid and justified. If it fails to do so, it will be a good learning experience. Since most analysis and assumptions has been documented, it will be possible to trace where the analysis was faulty, which will allow the creation of superior models in subsequent iterations.

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