Massachusetts Institute of Technology

Deterministic design of a T-base lathe

2.770 FUNdaMENTALS of Precision Product Design

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Executive Summary

In this document I describe the design, manufacturing and testing process of the T-base lathe I built for the class 2.77 FUNdaMENTALS of Precision Machine Design.

Key Results

I would like to highlight some points about the design of this T-base lathe:

1. **High performance.** I have tested the lathe to face an aluminum test piece and the surface roughness is 6um.
2. **Low cost.** The total economic investment that I have made on this lathe is $\sim$ 30$.
3. **Completely machined.** I have machined the large majority of the elements of this lathe.
4. **Deterministic design.** Several spreadsheets have been created along with the CAD models to deterministically design the lathe and calculated its expected performances based only on first principles, without FEA.
5. **FUNdaMENTALS.** During the design process I made extensive use of the fundamental principles that have proven to enhance the performance of the machine.

Contents

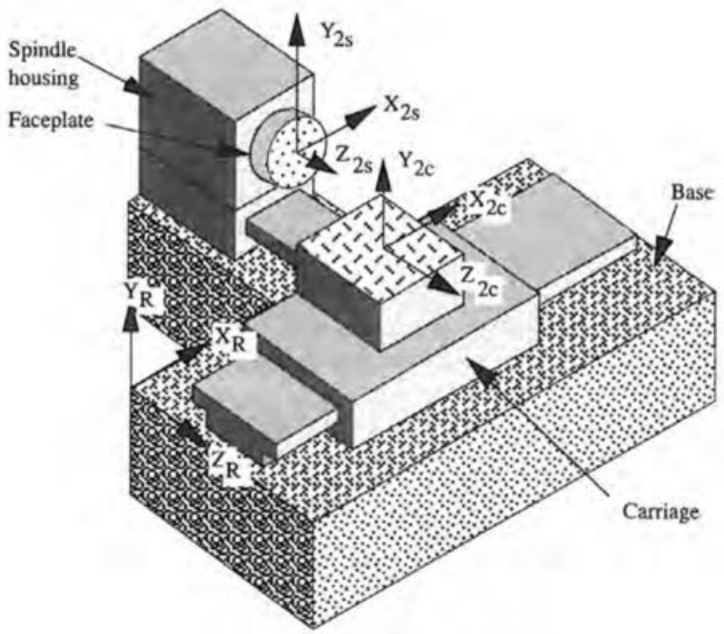
1 Introduction

This project is part of the 2.770 FUNdaMENTALS of Precision Machine Design class at MIT. The goal of the class is to learn deterministic design, selection, and assembly of machine elements to create and manufacture a robust precision machine or system. The class is structured in different blocks in which we design, build, and test a series of modules including kinematic and elastic averaging couplings and linear and rotary motion axes. As a graduate student, I had to make two linear motion axes and combine with the rotary axis to create a T-base lathe.
The goal of this document is to provide a full overview of the main features of the lathe and the design of them rather than an extensive detailed calculation, which can be found on each week's assignment folder.

1.1 What is a T-base lathe

The principle of a T-base lathe, as described in Slocum [1992], is to have two independent axis that intersect to form a "T" shape, as illustrated in Figure 1. The tool is mounted on the carriage of one of these axis and the spindle is mounted on the other. A nice feature about this design is that the axes are not stacked on top of each other so the geometry and associated nesting of mechanical components is greatly simplified.

Figure 1: Illustrative example of a T-base lathe. Source: Slocum [1992]



2 Functional requirements

During the early stage of the design I defined the Functional requirements and did a complete FRDPARRC analysis of them that can be found in each week's assignment. My goal in this report is to summarize the functional requirements that drove some of the design choices that I will describe further.

1. **Lathe for woodworking:** I intend to use this lathe to machine wood pieces.
2. **Adaptive design:** T-base lathes are useful for facing operations (for example for precise optical devices) and not so much for turning. However, I personally would not find a use to such a type of lathe so I want this lathe to be able to be used as a turning lathe.

3. **Easy to machine:** In order to keep the cost in the budget and also as a way to improve my machining skills, I intend to machine the large majority of the elements of this T-base lathe.
4. **Total error** $\sim 0.5mm$**:** With this level of error I expect to get a good surface finish before the sanding process of the wood.
5. **Budget:** The total cost of the lathe has to be within my personal budget for this class, since the only thing provided is the motor. However, I consider the cost differently from an industrial design: I am not taking into account the machining time of each element, since I consider that to be just helpful to improve my machining skills. The only cost considered is the one of the elements that I have to buy.
6. **Compact and easy to disassemble.** For storage purposes I would like to be able to assemble and disassemble the lathe.

These are the main requirements that will make this lathe not only useful as a learning tool but also in a future. However, there are some extra requirements that I will describe at each module that I have introduced just as a learning process.

2.1 Cutting forces

One of the main initial calculations that was made was to estimate the cutting forces that such a lathe would have to withstand. There is not a clear unified way to perform this calculation for wood elements and therefore I performed the calculation the same way it is made for metals, in particular for aluminum. The detailed calculation can be found in the spreadsheet *LM 1.xls*. The result is presented in this document.

Vertical cutting force $\sim 130N$;
Horizontal cutting forces $\sim 40N$;

3 Linear modules design

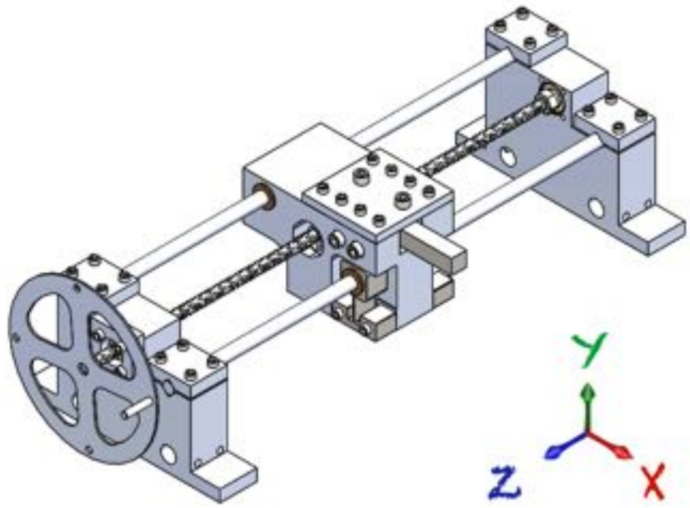
The first modules that I designed were the two linear modules. In class we studied essentially two different types: the boxway and the twin rail Slocum [2008]. After making some sketch models I decided to go for the twin rail design. The main reason that drove this decision is that it is a lighter, easier to store approach and the likelihood of finding some stock rods in any of the machine shops was higher. In addition, most of my class mates seemed to have chosen the boxway design so I was going to be able to learn about its design by peer-reviewing them.
I made two different linear modules, one to accommodate the tool and the other to attach the spindle, as can be seen in Figures 2 and 4.

3.1 Design overview

In this section I intend to provide an overview of the key design features of the linear modules and the motivation for them. Again, for detailed explanations, please refer to the week's assignments.

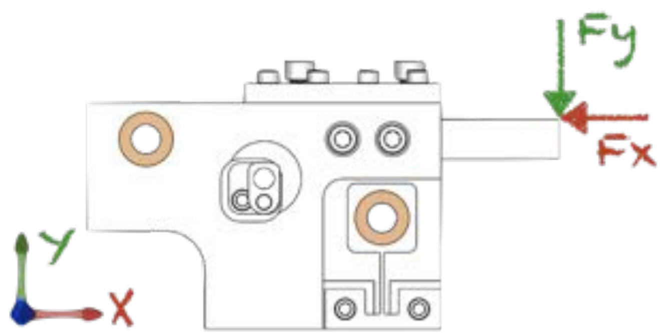
1. **Rails:** Both linear modules use 3/8" rods as rails. This diameter was selected after calculating the stiffness of the assembly to the cutting loads. This calculation can be found in both *LM 1.xls* and *LM 2.xls*. The two rails are clamped between two side elements that care then clamped into the ground. The thickness of the side elements uses St. Venant?s principle Slocum [2008] to guarantee a fixed type clamping and they are machined together to reduce errors in the locating features.

Figure 2: Overview of the tool linear module



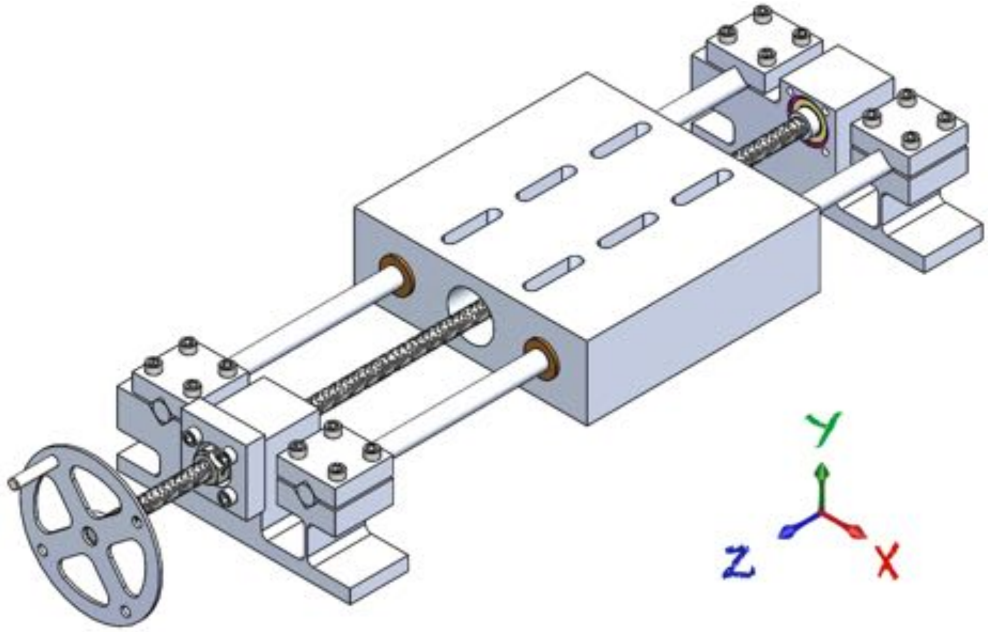
2. **Carriage:** The carriage has press-fitted oil impregnated brass bushings. They are cheaper that other solutions (ball-circulating, etc.) and best suited for dirty environment (we will be producing dust from cutting the wood with the lathe). For the tool holding carriage, there is a vertical offset of the rails. The motivation is to reduce the coupling between Fx and Rz. The flexure in one of the rails reduces the lateral stiffness of that rail. Therefore, I considered that having the line of action of Fx aligned with the most stiff rail would reduce the angular motion in Z (see Figure fig:carr).

Figure 3: Overview of the tool holding carriage



3. **Flexures:** The flexures are designed to allow some misalignment between the rails. This is particularly important in the case of the rails used, since they are not perfectly straight (they are extruded rods and not precision shafts). There is a difference between the two linear modules since in the tool holding the flexure is a separated piece that is bolted into the carriage and in the second case it is part of the side element. The reason is that I wanted to keep the second linear module low profile. For the one in the carriage the flexure is mounted using the principle of self-help to prevent buckling since due to its location, F_Y loads the beam in traction, and not compression.

Figure 4: Overview of the linear module for the spindle



3.2 Design spreadsheets

For both linear modules I have created a design spreadsheet (named *LM 1.xls* and *LM 2.xls* respectively) to assist in the design providing calculations of stiffness, geometrical errors and expected performance of these linear modules. Both spreadsheets can be found in the week assignments and here I am just going to provide an overview of the main calculations performed and summarize the end results.
The first step was to estimate the errors for the axis of the LM. To do so I used one of the spreadsheets provided in Slocum [2008] to apportion errors from different sources (i.e. geometric, thermal, etc.). The result was that thermal loads are not a concern in this case and probably geometric and process errors (i.e. force in the tool) are to be expected the main source of errors.

The next step is to calculate the stiffness of each one of the elements of the module:

1. **Rail stiffness:** The calculation of their stiffness is made using beam theory for a fixed-fixed beam. I use this assumption correct given that I have followed St. Venant's principle at the clamping point. However, for a real application I would run an FEA to contrast the results.
2. **Bushing stiffness:** I have used Hertz contact equations and the diameters of the bushing, rail and bushing hole to estimate the stiffness of this elements
3. **Flexure stiffness:** I used beam theory (also Timoshenko in case the flexure needs to be short) to design the flexure. Another possibility that the spreadsheet offers is to add the extra compliance via a hourglass, in which case a modified version of one of the spreadsheets provided in Slocum [2008] is used.

Using these individual stiffness and the geometry of the carriage the global stiffness is calculated. The result of this calculation is presented in Figure 5.

Figure 5: Stiffness of the linear modules at its center of stiffness

	LM 1	LM 2	
K_X	5.80E+05	1.48E+06	N/m
K_Y	9.85E+05	2.81E+06	N/m
K_Z	1.00E+07	1.00E+07	N/m
K_rz_x	1.10E+06	1.76E+05	N/rad
K_rz_y	2.37E+04	6.65E+04	N/rad
K_ry_z	4.31E+03	5.45E+04	N/rad
K_ry_x	5.17E+04	1.35E+05	N/rad
K_rx_y	7.33E+03	1.04E+05	N/rad
K_rx_z	8.79E+04	2.58E+05	N/rad

Finally using the cutting forces and the clearances of each element I was able to calculate the expected geometric and systematic errors at the center of the carriage, which are shown in Figure 6.

Figure 6: Geometric and systematic errors for the tool's linear module

	GEOMETRIC	SYSTEMATIC	
X	2.29E-01	-6.63E-02	mm
Y	-2.29E-01	-1.30E-01	mm
Z	0.00E+00	3.84E-03	mm
RX	-1.78E-03	-1.70E-02	rad
RY	-1.78E-03	-8.17E-03	rad
RZ	-2.53E-03	-5.37E-03	rad

4 Rotary modules design

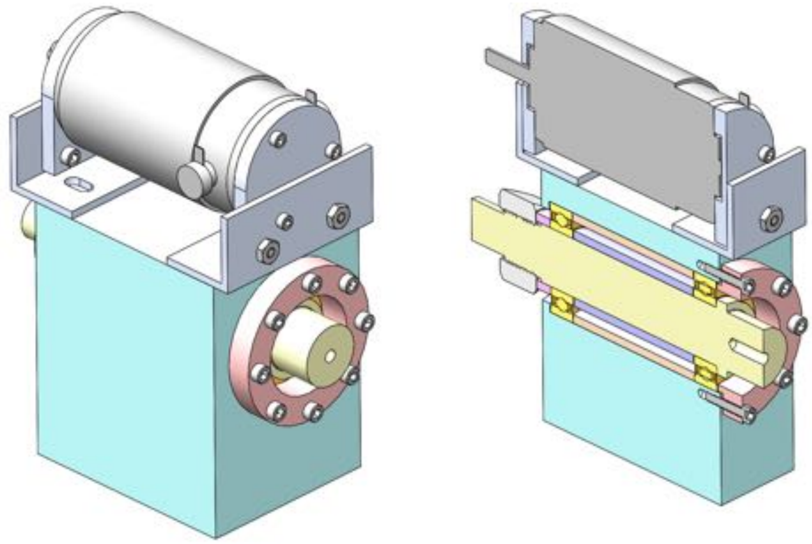
After the design of the linear modules I designed the three rotary modules: two leadscrews to actuate the linear modules and the spindle of the lathe. This time both leadscrews are identical but quite different from the spindle and this is why I am going to describe their design separately.

4.1 Leadscrew design overview

The whole lathe has to be able to be disassembled and assembled back easily and the cost of its components needs to be kept as low as possible. Therefore its design was mainly cost driven. I knew that there are some elements that I would not be able to machine, like the bearings and the leadscrew itself.
Regarding its functional requirements, the leadscrew is the element that provides stiffness in the axial direction to the carriage. On the other hand, it is not expected to have to deal with high thermal loads since the required speed of the translation stage is low (i.e. approx 2m/min). Finally, I decided to introduce an anti-backlash system to learn about the potential ways to prevent this source of error.

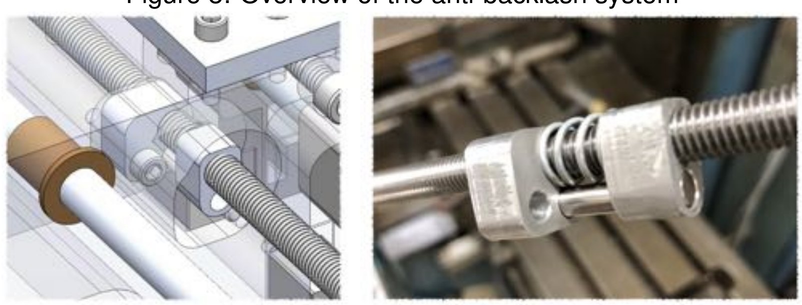
1. **Bearings:** I found out that the cheapest option for the bearing was to buy skateboard ball bearings, Sackorange 608zz Double Shielded,8x22x7 Miniature Ball Bearings. These bearings have high load capacity, even if for our application this is not necessary. Another advantage is that the Conrad bearings are more forgiving of potential misalignment due to manufacturing errors. On the other hand, these bearings are sealed, which is important given that the environment in which they are going to be working is expected to have particles that could damage them and reduce their lifetime.

Figure 7: Overview of the spindle design



2. **Leadscrew:** For the leadscrew, the cheapest option was to buy an M8-1.25 threaded rod (pretty popular since it is used to actuate RepRap 3D printer).

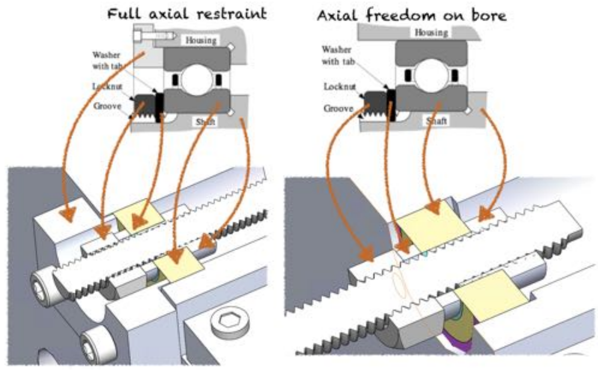
Figure 8: Overview of the anti-backlash system



3. **Anti-backlash system:** The anti backlash system that I have designed (see Figure 8 is essentially a two nut system with a spring preload between them (the preload is easily calculated with $F = Kx$). One of the nuts is attached to the carriage with a screw and a 5mm dowel pin guarantees that the two nuts do not have relative motion.

For the mounting of the bearings I have not used a thermocentric design since, as mentioned above, thermal loads are not expected to be critical. Also, I have decided not to axially preload the bearings, knowing that this will have an impact on the stiffness of my system. The reason is simply that I will apply the preload techniques for the spindle and so I have decided to go with a simpler, less performant system for the leadscrew due to time constraints.
Therefore I decided to mount the bearings leaving one of them free to slide in the bore, as shown in Figure 9. The main (and probably only) advantage is that it is a non expensive mounting solution. In real applications I would use a surface finish for the bore (or more likely just another mounting configuration), as well as use washers with tabs and design the bolted flange so that cones of influence overlap, as recommended in **?**.

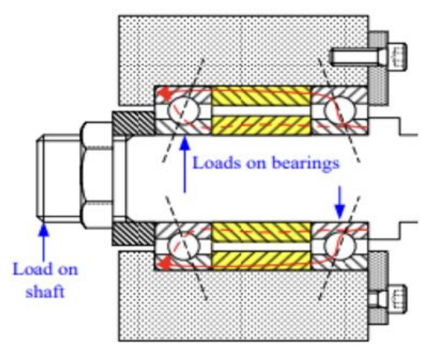
Figure 9: Overview of the leadscrew bearing mounting



4.2 Spindle design overview

The functional requirements of the spindle are different from the leadscrew. One of the most important ones is the the spindle is expected to deal with thermal loads since it is going to be operated at approximately 1500rpm.
For the bearings, I found out that the cheapest option given the general dimensions of the spindle were the Sealing Deep Groove Radial Ball Bearing 6005Rs 25x47x12mm.
The mounting of the bearings is also different from the leadscrew since now I wanted to have a thermocentric assembly as well as axially preload the bearings to increase the axial stiffness of the spindle. Figure **??** and 10 show the final mounting scheme. A critical element of this design is that the spacers need to be ground to the exact length to give the desired preload.

Figure 10: Overview of the spindle thermocentric bearing mounting



The rest of the parameters of the spindle were designed using the excel spreadsheet that I developed to that purpose (*RM.xls* and fundamental principles.

4.3 Design spreadsheets

For both the leadscrew and the spindle I have developed a design spreadsheet (named *Leadscrew.xls* and *RM.xls* respectively) to assist in the design providing calculations of stiffness, geometrical errors and expected performance of these linear modules. Both spreadsheets can be found in the week assignments and here I am just going to provide an overview of the main calculations performed and summarize the end results for the rotary motion, since the leadscrew is a simplified version of it.

The first step, as well as for the linear module, was to estimate the errors for the axis of the rotary module, and it was done using the same principle.

The next step is to calculate the stiffness of the bearings:

1. **Axial bearing stiffness:** the axial stiffness of the bearings is calculated assuming a 1% deformation of the ball at the rated axial capacity and applying Hooke's law
2. **Radial bearing stiffness:** the axial stiffness of the bearings is calculated assuming a 0.5% deformation of the ball at the rated axial capacity and applying Hooke's law

Using these individual stiffness and the geometry of the shaft the global stiffness is calculated. The result of this calculation is presented in Figure 5 and 11.

Figure 11: Stiffness of the spindle at the part

Radial	1.69E+04	N/mm
Axial	5.95E+04	N/mm
Angular	1.27E+05	N/degree

Finally using the cutting forces and the clearances of each element I was able to calculate the expected geometric and systematic errors at the center of the carriage, which are shown in Figure 6.

Figure 12: Geometric and systematic errors for the spindle

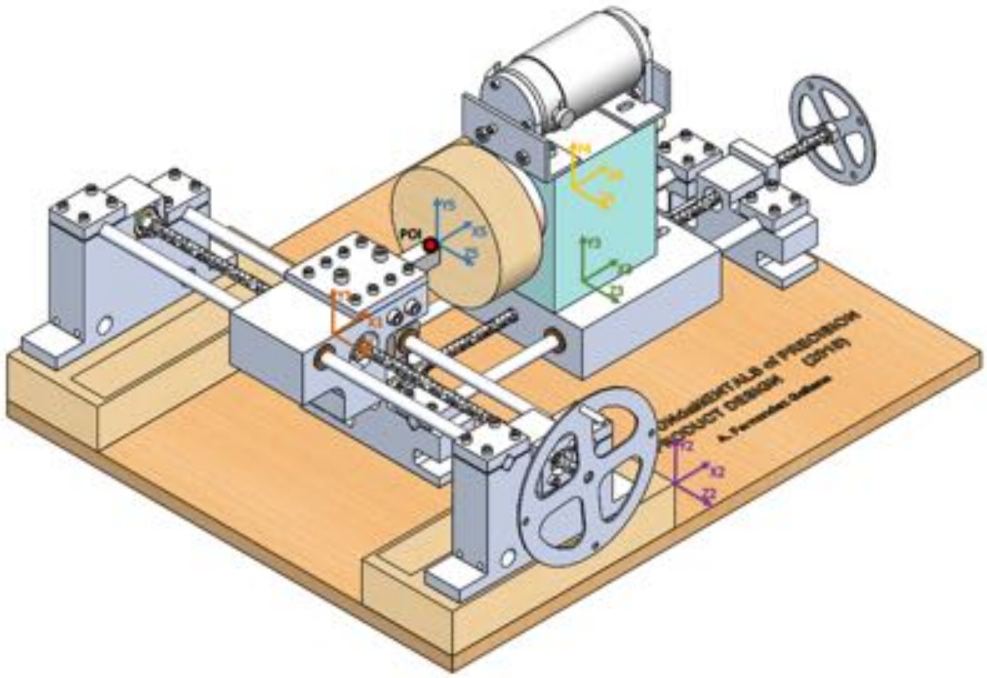
	GEOMETRIC	SYSTEMATIC	
Radial	1.55E-02	7.94E-03	mm
Axial	0.00E+00	6.46E-04	mm
Angular	2.00E-01	1.05E-03	degrees

To motorize the spindle I have used an o-ring as belt to connect it with the motor, which is place on top of the spindle housing. This is not the final solution, since I intend to turn this lathe into a pseudo-classical lathe by turning the spindle 90 degrees so that I can perform turning operations. In this final configuration the motor will seat next to the spindle and not on top of it. However, for the class project a T-base lathe had to be integrated. In addition, right now the shaft diameter is unique and provides a constant reduction factor of $\frac{1}{3}$. In the future I will turn the spindle again to different diameters to provide discrete spindle speed control.

5 T-base lathe integration and HTM

All the presented modules have been designed to be integrated as a T-base lathe. They have actually also been designed so that the spindle can be rotated 90 degrees. During the first weeks of the class we learned about kinematic and elastically averaged couplings. For this lathe I think that an elastically averaged position of the different element is the right approach. However, given that the T-base lathe is not the final use of the modules and the cost of an aluminum plate of the size of the lathe I decided that for the class purpose I would use a wood plate to integrate all the modules, as shown in Figure 13.

Figure 13: T-base lathe and CS for the HTM calculation



Once the full geometry of the lathe has been defined and the CS of each element has been defined we can make an error budget for the full machine using the HTM method Slocum [2008]. This is made in the *HTM.xls* spreadsheet that should be review for further information. The calculation is done at the worst case scenario, which is when both linear modules are at the middle of its travel, i.e. when the structural loop is longest and therefore the whole stiffness of the machine is reduced. The result is presented in Figure 14, using Prof. Slocum Error Budget Spreandsheet.

Figure 14: T-base lathe error budget HTM calculation

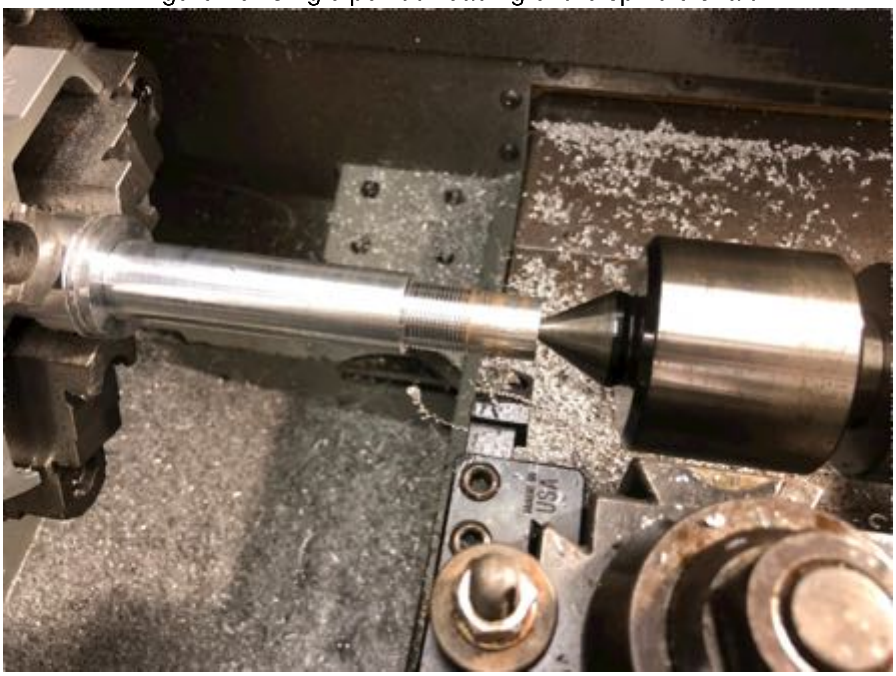
All Axes	For the Entire Machine (in mm)					
	Number of axes	All axes' Geometric Errors				F=kX displacement
	5	Random			Systematic	
		Sum	RSS	Avg(SUM, RSS)	Sum	Sum
	deltaX	6.59E-02	4.33E-02	5.46E-02	6.19E-02	-8.78E-02
	deltaY	6.80E-02	4.40E-02	5.60E-02	6.25E-02	-2.32E-01
	deltaZ	6.86E-02	4.44E-02	5.65E-02	7.06E-02	6.39E-02
	Vector displacement	**1.17E-01**	**7.61E-02**	**9.65E-02**	**1.13E-01**	**2.57E-01**

This error calculation takes into account the geometrical errors and the displacement errors due to loading for the entire machine. In order to calculate what is the surface roughness that can be expected from such a machine we need to take into account only the spring errors (i.e. $F = Kx$). Figure 14 shows that this error is $\sim$260 μm. This value is smaller than the .5 mm that we set as functional requirement (the part are going to be sanded after the cutting process).

6 Manufacturing

I have machined all the elements of the lathe from stock metal parts from different machine shops around, which has had a great impact on keeping the project cost under budget. The manufacturing techniques include essentially CNC lathe and mill. As an example, Figure 15 was taken during the single point threading of the spindle shaft. Most of the work has been done at the Edgerton Student Machine Shop.

Figure 15: Single point threading of the spindle shaft



7 Final assembly

The integration of each one of the modules after manufacturing has been a very smooth process. The same can be said about the overall assembly of the lathe, which was one of the functional requirements. Figure 16 gives an illustration of the parts that compose the lathe, which can be seen fully assembled in Figure 17.

8 Testing

A key part of the learning process is to close the loop of the design via intensive testing of the modules. Again, in this document I will only summarize the testing and introduce the ultimate test, the surface roughness test of a faced test part.

In general one of the main problems that I encountered during the testing was to set-up tests that would reveal the features of my lathe, since it is a rather high stiffness machine. Besides the fact that more force needs to be applied in order to dial some motion, it is in general complicated to find a stiff structural loop between the testing point and the dial indicator to guarantee that the motion measured is exclusively from the testing part. In that respect my first set of tests was not successful at all and all the stiffness seemed to be order(s) of magnitude lower than predicted. Nevertheless, after improving the test set-up I obtained better results. As an example, Figure 18 presents the results of the testing on the tool's linear module. The

Figure 16: Parts of the T-base lathe

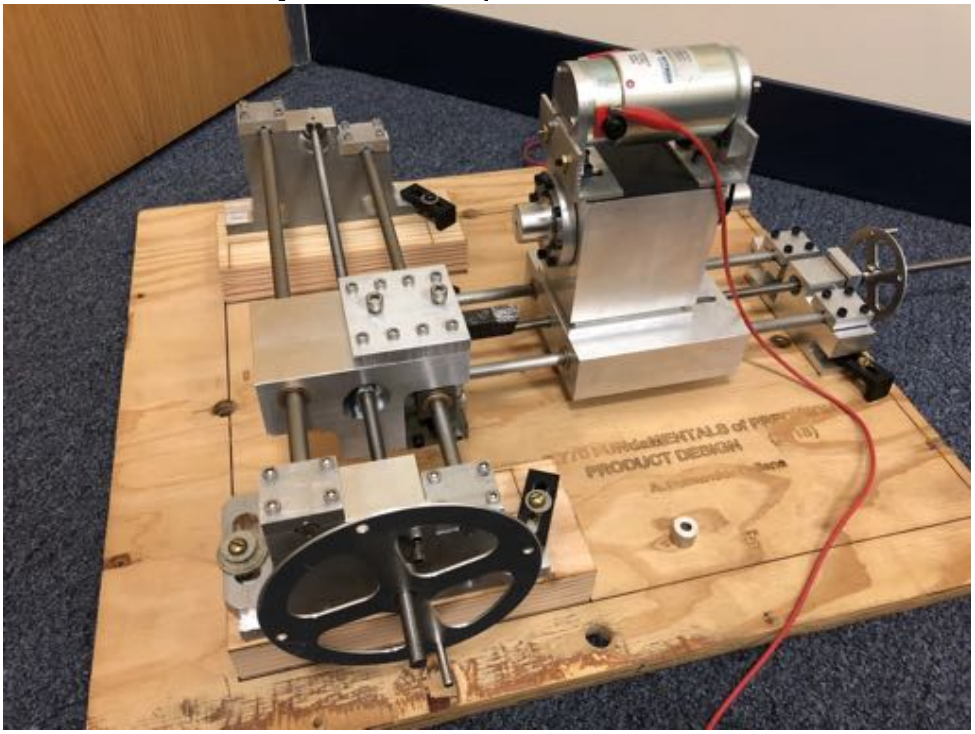
predictions are within a reasonable error given that we are only using first principles to do the calculations (i.e. no FEA or more refined techniques).

Figure 18: Result of the stiffness test on the tool's LM

	Predicted	Measured	
K_X	5.80E+05	1.20E+05	N/m
K_Y	9.85E+05	5.90E+05	N/m
K_Z	1.00E+07	-	N/m
K_rz_x	1.10E+06	-	N/rad
K_rz_y	2.37E+04	1.77E+04	N/rad
K_ry_z	4.31E+03	3.53E+04	N/rad
K_ry_x	5.17E+04	-	N/rad
K_rx_y	7.33E+03	-	N/rad
K_rx_z	8.79E+04	-	N/rad
Structure	4.31E+04	2.10E+04	N/m

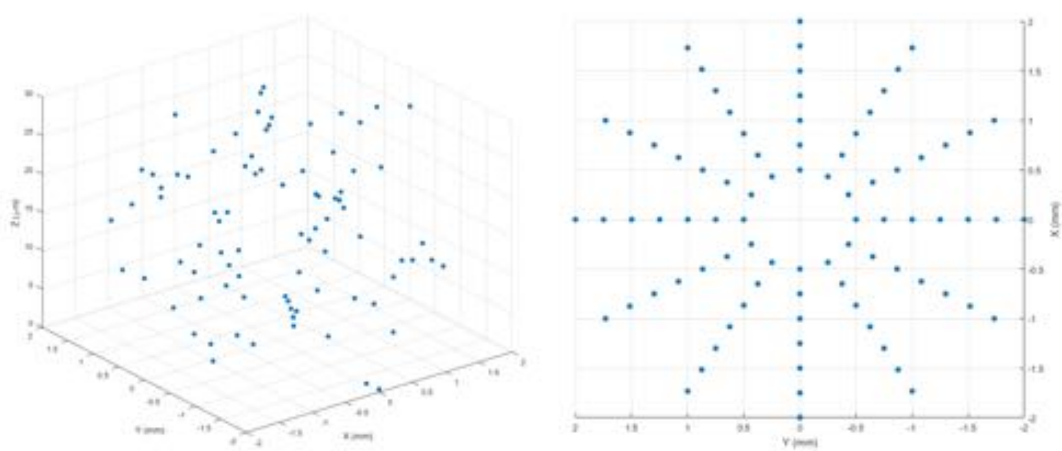
The last part of the testing was actually to perform a facing operating into a test. The result of the test is presented in Figure 19, which is the measurement of a test piece with a dial indicator set on a prototrack mill. We can see that the roughness is approximately 30 μm, which is about an order of magnitude better than the prediction. The explanation for this is probably that the estimation of the cutting force, which had a lot of variety depending on the source used for the calculation. It is surprising because I was expecting a

Figure 17: Assembly of the T-base lathe



worse finish due to the low speed of the spindle but I imagine that this effect might be compensated by the above-mentioned error in the force calculation.

Figure 19: Result of the facing roughness test



9 Cost

One of the main requirements of the design was to make it easy to manufacture in order to keep the economic investment on the lathe as low as possible. The final cost of the lathe is $\sim$ 30$ and it covers the cost of the bearings, bushings and leadscrews, which are the only elements I have not manufactured.

Figure 20:

T-BASE LATHE BUDGET			
Element	**Number**	**Unitary cost**	**Total cost**
	LINEAR MODULE		
Bushings	4	$ 1.41	$ 5.63
	TOTAL PER MODULE		$ 5.63
	LEADSCREW MODULE		
Small bearings	2	$ 0.45	$ 0.90
Leadscrew	1	$ 6.92	$ 6.92
	TOTAL PER MODULE		$ 7.81
	SPINDLE		
Large bearings	2	$ 1.63	$ 3.26
	TOTAL PER MODULE		$ 3.26
TOTAL			**$30.14**

10 Conclusion - Lessons learned

The T-base lathe performed as expected and seems that it will be a useful machine to do small wood or aluminum parts with relatively good accuracy. During the design, fabrication and testing of this lather I have learned the process for a deterministic design, which starts with a clear definition of the functional requirements, risks and countermeasures and ends by closing the loop when performing tests and correcting accordingly.
On the other hand, this project has help me improve my manufacturing skills. At the beginning of the semester I was only comfortable with basic lathe and milling operations and now I feel confident with a wider spectrum of techniques, including CNC; without which I would not have been able to produce this lathe.

11 Future work

My initial goal is to build a full CNC lathe. For this I will have to work on motorizing the two leadscrews and program a controller for them. I also need to improve the current baseplate to improve the stiffness, as well as locating the motor in a more suitable position. In addition, adding some safety to the belt in case it breaks seems also an appropriate upgrade.

12 Acknowledgment

I would first like to thank my peer review peers Nina Petelina and Filippos Sotiropoulos for their useful comments and recommendations in the design. Finally, I would like to thank specially Mark Belanger, head of the Edgerton Machine shop for his help and for making me (hopefully) a better manufacturer.

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