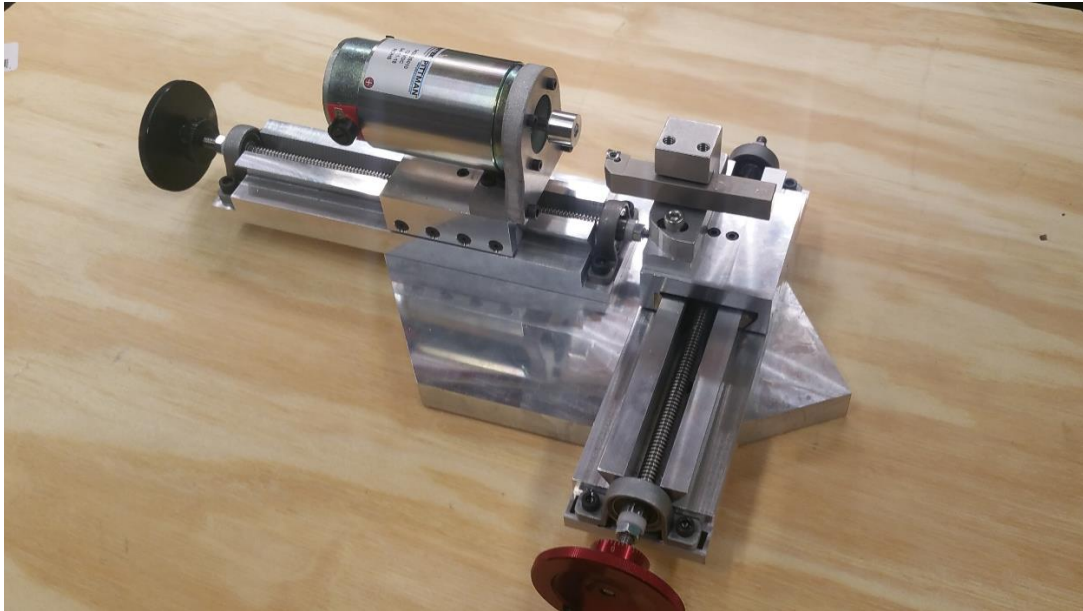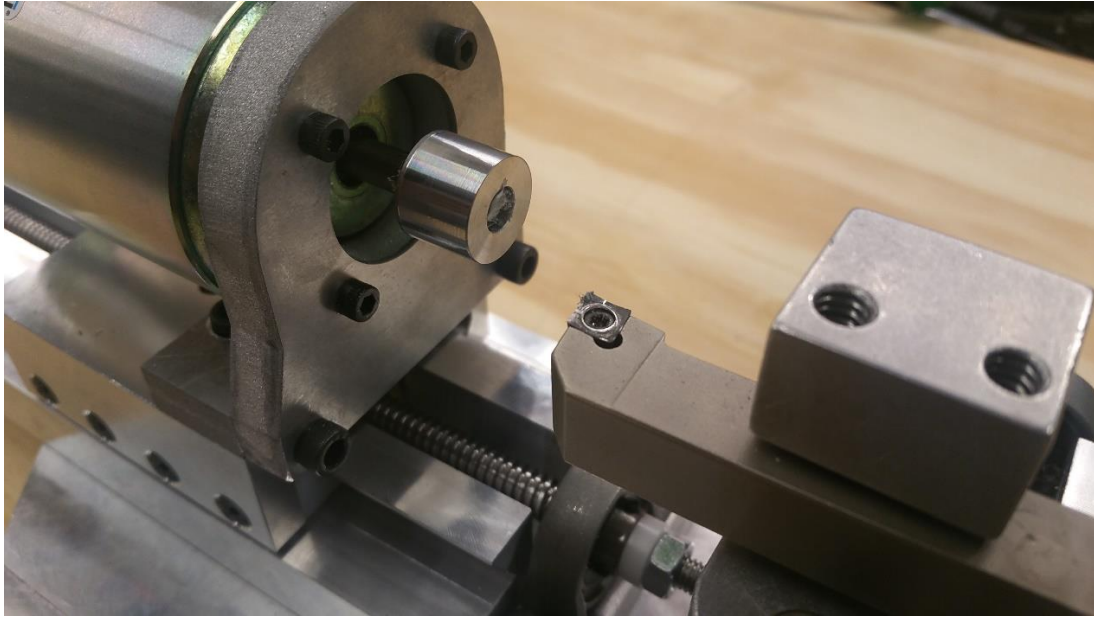# T-based Lathe – PROJECT REPORT

2.77 Precision Machine Design
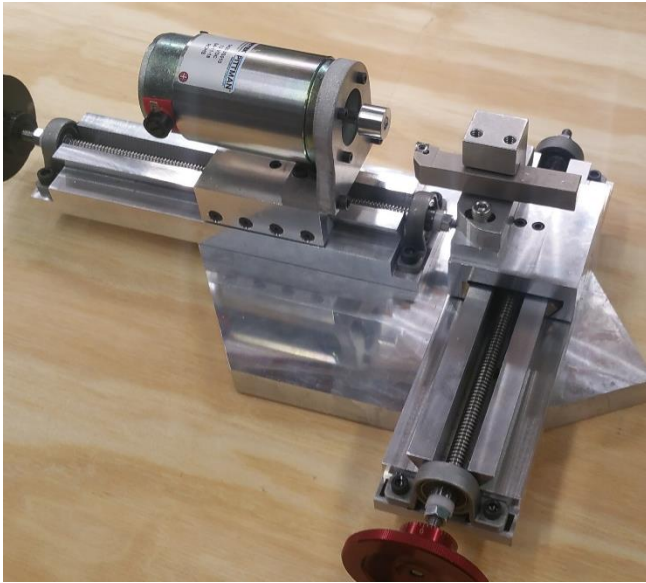
VICTOR PROST – SPRING 2018

T-based Lathe – PROJECT SUMMARY

2.77 Precision Machine Design

VICTOR PROST – SPRING 2018

Project Goal

The goal of this work was to design a T-based lathe to face a 5/8" diameter aluminum part with a maximum flatness error of 20 micron with a depth of pass of 0.1 mm.

	TARGET	PREDICTED	MEASURED
Flatness Error [um]	<20	5.7	3.4
Depth of pass [mm]	>0.1	-	0.1

Figure 1: T-based lathe assembly (left), faced test parts (right)

Design Strategy

From the design requirements for this project, I proceeded the following way:

- Constructed a list of **functional requirements** for the overall machine
- Estimated the **apportionment of the total error** for each axis and each source of error (Geometric and Load-induced)
- Divided the **machine in several modules**: Linear motion slides, Leadscrews, Spindle & Tool post
- From this basic layout of the overall machine, constructed an **error model using HTMs** to define the **stiffness and geometric error requirements** for each module,
- Similarly, for each module:
    - o I constructed a list of functional requirements based on the above calculations
    - o Generated a series of design concepts
    - o Made an analytical model to size the different elements, predict and compare the different concepts' performance to the functional requirements
    - o Built, tested the best concept and closed the loop on the model (measured VS predicted)
- Assembled, tested the entire machine and compare the results to the model.

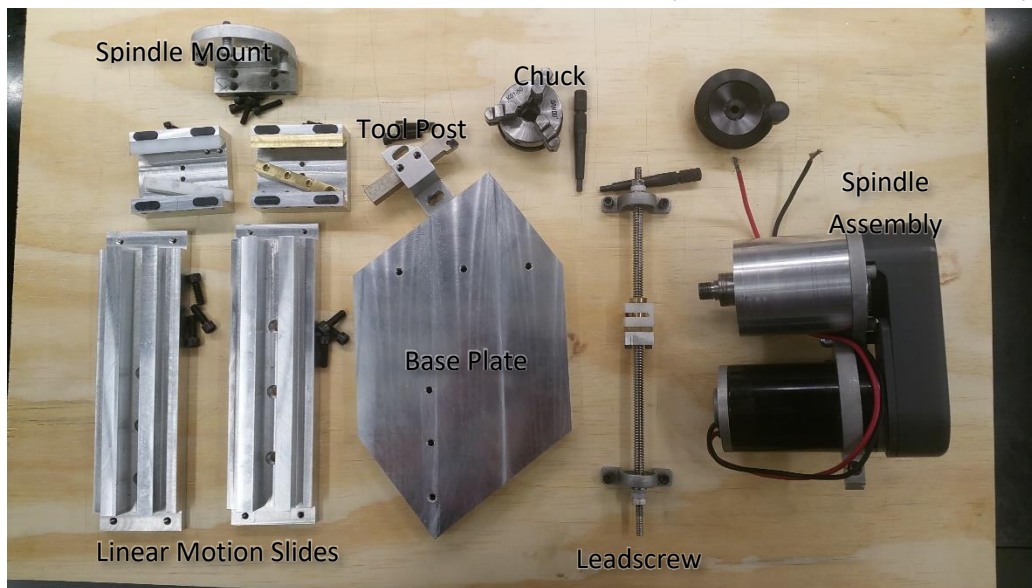# Modules [Linear Motion Slides/Leadscrews/Spindle] and Assembly

Figure 2: T-based Lathe Modules & Elements

Test and results

The flatness of the 0.1mm test cuts were then measured on a ZYGO white light interferometer, exhibiting a maximum height difference on the face of 3.4 ± 0.2 microns, and a misalignment between the two linear motion axes of 0.89 ± 0.05 mrad matching our design requirements and our predictions of 5.7 microns load induced errors.
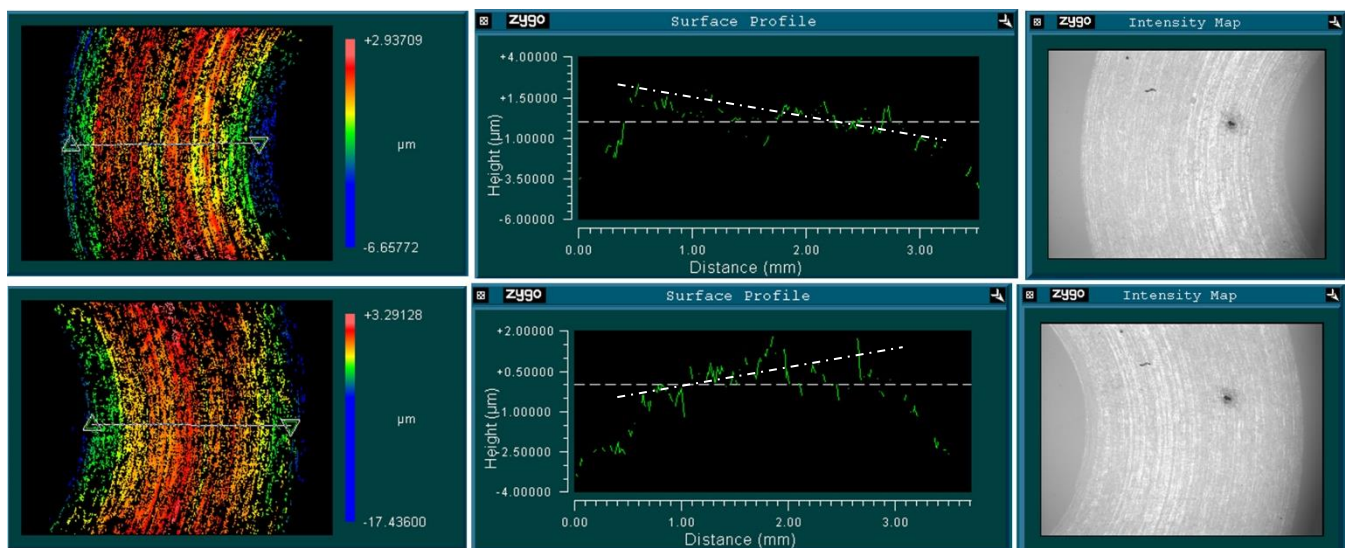

Figure 3: ZYGO White Light Interferometry of test part

Discussion

The T-based lathe performed as expected and this project was valuable in teaching me a deterministic and rigorous design process that includes: defining the functional requirements, concept generation, thorough analysis with a clear statement of the assumptions, testing and reflecting on the model.

Table of Content

Introduction

The work was done in the scope of a Precision Machine Design course at MIT (2.77) where we examined design, selection, and combination of machine elements to produce a robust precision machine or system. It introduced a deterministic physics-based design process considering each machine elements with respect to its physics, mechanics, accuracy, repeatability and resolution.

The final project was to design a T-based lathe to face a 5/8" diameter aluminum part with a maximum flatness error of 20 micron with a depth of pass of 0.1 mm. The following document explains the design process that was used:

- Overall Machine Specification and error apportionment
- Divided the machine into several modules and laid everything out in a structural loop
- Constructed the entire error budget and set each module's functional requirements
- For each module:
  o Generated a list of concepts
  o Made analytical models to size the different elements, predict and compare the different concepts' performance to the functional requirements (for example, stiffness of the module, accuracy and repeatability)
  o Built, tested the best concept and closed the loop (measured VS predicted)
- Assembled, tested the entire machine and compared the results to the model.

Machine Specifications

As a first step, it is important to construct the list of high level requirements, specifications and goals for the new machine and allocate errors base on the total allowable error before jumping into concept generations and analysis. This early exercise is essential to our coarse to fine design process where the overall machine is divided in a subset of modules with functional requirements directly related to these high level requirements.

Functional Requirements

These initial basic requirements are based on the user needs in terms of the performance of the lathe.

Table 1: High Level Functional Requirements for the T-based Lathe

Functional Requirements	Design Parameters	Analysis
Flatness error	20 um[1]	Facing test
Max Part Diameter	20 mm	Travel of the Y-axis
Max Part Length	100 mm	Travel of the X-axis
Small Driving Torque	<250 Nmm[2]	Torque Wrench
Resolution Leadscrew	40 um	Dial Indicator

These initial requirements, and the total allowable error of the point of interest of the machine dictate design decisions and component selection. I then performed an error apportionment, allocating part of the total error to each axis:

---

[1] For a depth of pass of 0.1mm on a 15mm diameter aluminum part
[2] 10 N on a handle of 50mm in diameter

The total error is 20 microns and we have 3 axes; thus, each axis is allowed a 11microns error budget (based on linear and root square sum errors).

Modules layout and structural loop

Before starting the design of any elements in the machine, I divided the machine into several modules:

- 2 Linear motion slides
- 2 Leadscrews
- 1 Spindle Assembly
- 1 Tool Post

From that first division, I started to a stick figure and blocks of the skeleton of the machine and laid each of the module on that drawing. From this drawing I was able to identify the structural loop of the T-based lathe.
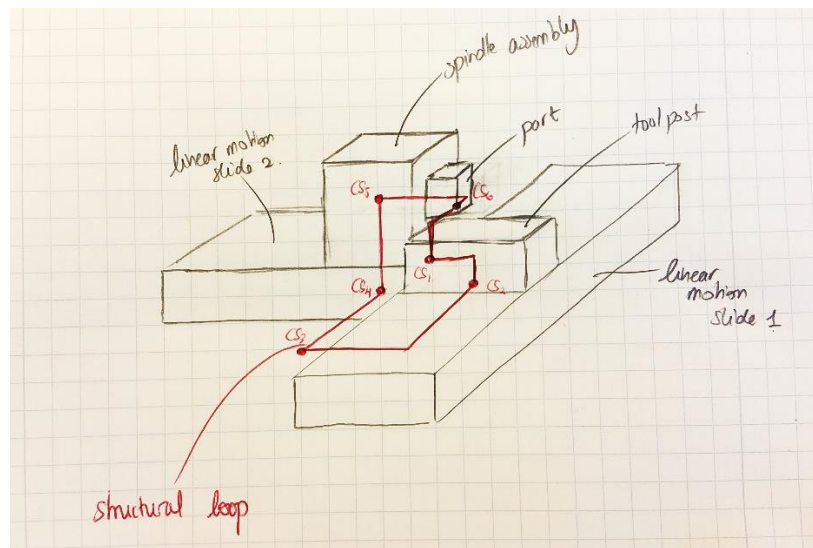

Figure 4: High Level Layout of the Lathe and Structural Loop

Error budgeting

Having this initial structural loop and layout of the modules, I performed an error budget analysis using HTMs (homogenous transformation matrices) and the expected cutting loads on the part (From the Kalpakjian, *Manufacturing Engineering and Technology*) to understand the effects of load-induced and geometric errors of each module to the total error at the point of interest.

This error budgeting enabled me to set more detailed functional requirements to each module in terms of stiffnesses, manufacturing tolerances (geometric errors) to satisfy the high level functional requirements of the machine. Having that in the format of a spreadsheet (developed by Prof. Slocum), allowed me to revise the requirements of each module and layout of the machine during the design process and ensure that the overall functional requirements would still be met.

Figure 5: Cutting Loads for Aluminum

TOOL LOADS		
MMR:	Aluminum	
Cutting Force (Fz), [N]	110	N
Thrust Force (Fx), [N]	83	N
Radial Force (Fy), [N]	83	N
Roughing:	Aluminum	
Cutting Force (Fz), [N]	53	N
Thrust Force (Fx), [N]	41	N
Radial Force (Fy), [N]	41	N
Finishing:	Aluminum	
Cutting Force (Fz), [N]	9	N
Thrust Force (Fx), [N]	7	N
Radial Force (Fy), [N]	7	N

Modules Design

From that high-level analysis, I then focused on each module using the functional requirements set during the error budgeting process. The process for each of these modules was the following:

- From the module specific requirements, generated a series of design concepts
- Made an analytical model to size the different elements, predict and compare the different concepts' performance to the functional requirements
- Built initial sketch models and tested them to verify the validity of these models.
- Selected the best concept, made a CAD model, built, tested it and closed the loop on the model (measured VS predicted)

Linear Motion Slides (LMS)

The detailed functional requirements for the linear motion slides computed from the error budgeting were the following:

|	
Lateral Stiffness Kx [N/um] (leadscrew)	>4
Lateral Stiffness Ky [N/um]	>4
Vertical Stiffness Kz [N/um]	>5
Yaw Stiffness (about Z) K_yaw [Nm/mrad]	>5
Pitch Stiffness (about Y) K_pitch [Nm/mrad]	>1
Roll Stiffness (about X) K_roll [Nm/mrad]	>1
X - Repeatability at Tool Tip [um]	<5
Y - Repeatability at Tool Tip [um]	<5
Z - Repeatabilityat Tool Tip [um]	<10
X - Accuracy at Tool Tip [um] (Geometric)	<10
Y - Accuracy at Tool Tip [um] (Geometric)	<10
Z - Accuracy at Tool Tip [um] (Geometric)	<20
X - Accuracy at Tool Tip [um] (Load induced)	<10
Y - Accuracy at Tool Tip [um] (Load induced)	<10
Z - Accuracy at Tool Tip [um] (Load induced)	<20
Range of Motion	>100
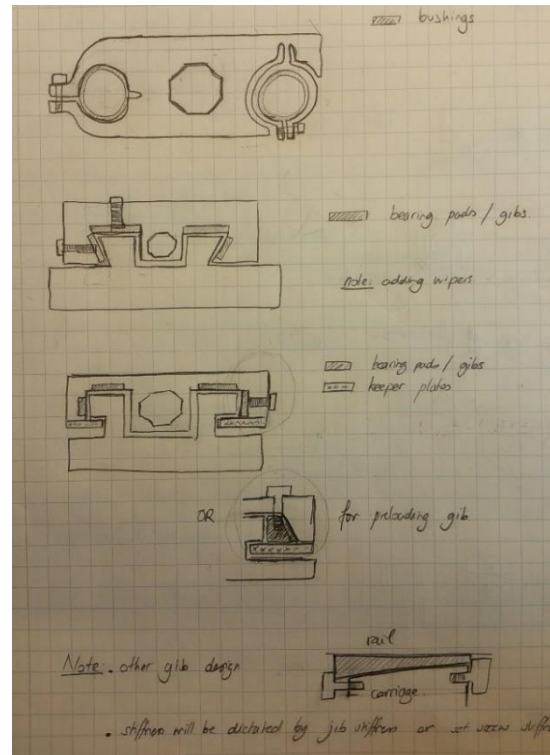
Figure 6: (left) LMS functional requirements, (right) concept sketches for different linear motion slides

I then explored different concepts that would be suitable for this application including, twin rails, dovetails, box ways, roller/rails as well as ball/groove designs. I then selected the dovetail rails and twin rails concept to do the analytic model and predict their performance.

For each of the concept, I applied beam bending theory, beam shearing, and beam axial compression to do the stiffness analysis of the LMS:

> In more depth, for the dovetail system, for the angular stiffnesses, I assumed a linear load distribution on the contact areas, found the pressure magnitude and computed the angle deflection assuming a distributed stiffness on the contact areas.

For the twin rail system, it was mostly beam-bending with pin-pin supports and point loads. I assumed point loads since the bushing's length will be the same as the diameter of the rails. Similarly, I assumed pin-pin supports since the end plates constraining the beams will be the same thickness as the rail diameters (conservative estimate from St Venant Principle).

Next, I did an HTM analysis on the LMS, the carriage and the tool to find the errors gains from the LMS deflection on the tool tip, putting each reference frame at the center of stiffness of each element considered.

From the stiffness, HTM analysis and cutting loads, I looked at the tool deflection under roughing cutting loads, to get an estimate of the accuracy of a cut.

Lastly, I used the wear equation and PV analysis to get a sense of the repeatability of the system.

From the analysis, I was able to understand the trade-offs of each design.

- The dovetail design is very stiff but will be very sensitive to the manufacturing since any "bump" on the rail of the carriage will impose a large normal force, thus the force required to move the linear slide will increase drastically. Also preloading the system with gibs will not be an easy task.
- The twin rail design is much more compliant especially in roll unless the footprint (size) of the LMS is increased significantly (diameter of the rails and spacing between the rails). However, the compliance of the system and the quasi-kinematic design make it easier to manufacture, assemble and less sensitive to geometric errors of the LMS. In addition, having the tool closer to the center of stiffness (and increasing the tool height to give some clearance for cutting the part) will help alleviate the error motions from the roll stiffness of the LMS.

I then went on and made a sketch models of each design and tested them in terms of stiffness and geometric errors to validate my analytical model.
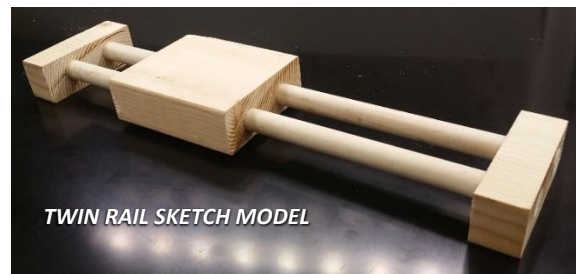

Figure 7: Concept sketch models out of wood and stiffness test example using a digital load scale and a dial indicator

From the analysis and the sketch models, I selected the dovetail LMS design for the ease of preloading the rails, the increased overall stiffness of the system. I was then able to size each component in the dovetail system thanks to my analytical model used as a design spreadsheet: aluminum rail size, aluminum carriage, Teflon bearing pads, brass gibs, preload mechanism (Belleville washers) to fulfill all the functional requirements.
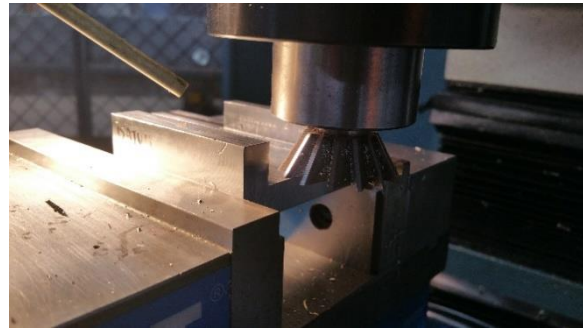
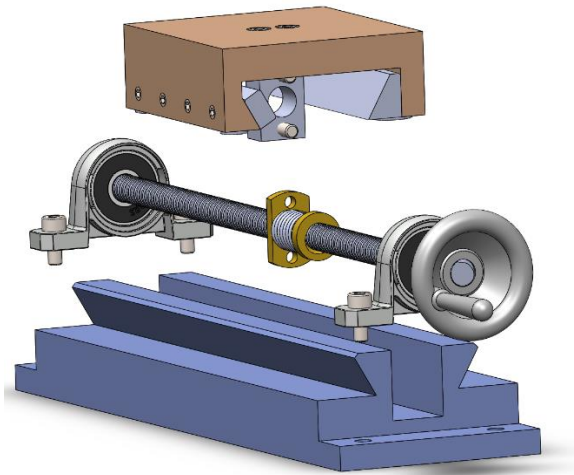From the CAD model, the LMS was then manufactured mainly on the mill

Figure 8: (left) CAD of the LMS with the tentative leadscrew, (right) machining the dovetail on the mill

The resulting parts where then assembled, preloaded and tested to ensure that it performed within the functional requirements:
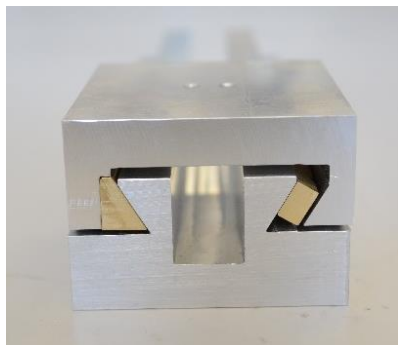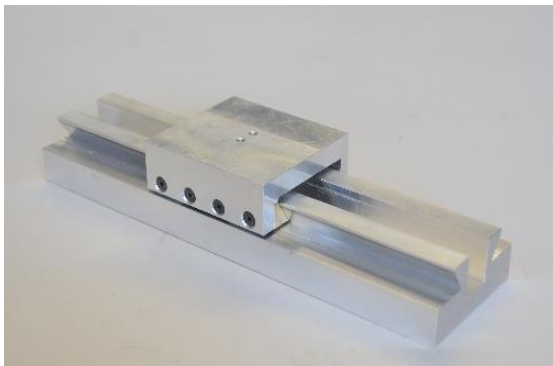

Figure 9: (left) Manufactured LMS with brass bearing surfaces, (right) stiffness test on the mill with using a dial indicator and a digital load scale

Table 2: Summary of the measured characteristics of the LMS compared to the target functional requirements

	Target	Measured
Lateral Stiffness Ky [N/um] (leadscrew)	>4	NA
Lateral Stiffness Kx [N/um]	>4	24.3
Vertical Stiffness Kz [N/um]	>5	7.7
Yaw Stiffness (about Z) K_yaw [Nm/mrad]	>5	13.7
Pitch Stiffness (about X) K_pitch [Nm/mrad]	>1	5.9
Roll Stiffness (about Y) K_roll [Nm/mrad]	>1	4.7
X - Repeatability at Tool Tip [um]	<5	-
Y - Repeatability at Tool Tip [um]	<5	-
Z - Repeatabilityat Tool Tip [um]	<10	-
X - Accuracy at Tool Tip [um] (Geometric)	<10	3.5
Y - Accuracy at Tool Tip [um] (Geometric)	<10	5.5
Z - Accuracy at Tool Tip [um] (Geometric)	<20	3.5
X - Accuracy at Tool Tip [um] (Load induced)	<10	3.8
Y - Accuracy at Tool Tip [um] (Load induced)	<10	4.9
Z - Accuracy at Tool Tip [um] (Load induced)	<20	8.5
Range of Motion	>100	130

The manufactured LMS matched the analytical model and was able to achieve the target requirements. A second similar LMS was then built for the other axis.

Leadscrews

Similarly, the detailed functional requirements for the leadscrews computed from the error budgeting were the following:

Table 3: Leadscrew functional requirements

Driving Torque [Nmm]	< 250
Axial Stiffness [N/um]	> 0.5
Resolution [um]	< 40
Backdriveable [-]	No
Safety Factor - VM Stress under cutting loads [-]	> 2
Safety Factor - Compressive load buckling [-]	> 2
Safety Factor - Fatigue Stress/Endurance limit [-]	> 2
Y - Accuracy at Tool Tip [um] (Alignment & Bowing)	< 20
Z - Accuracy at Tool Tip [um] (Alignment & Bowing)	< 20
X - Accuracy at Tool Tip [um] (Cutting Loads)	< 100
X - Accuracy at Tool Tip [um] (LeadError)	< 5
Thermal Stability	Yes
Range of Motion	>100

During concept generation I mostly focused on layouts and bearing mounts. I mainly considered:

- BUSHING CONCEPT: constrains the leadscrew axially and radially in one mount while the other mount provides additional radial support. The axial constraint is provided by a bushing with a shoulder, a step on the leadscrew, and a nut on custom threads on the leadscrew (see schematic below). This concept doesn't put the leadscrew in tension nor compression while constraining it axially and it is thermally stable. However, having normal loads on bushing and washers will likely result in a high driving torque.

- BALL BEARING CONCEPT: constrains the leadscrew radially using the ball bearings and axially using two nuts and two washers or shoulder bushings. The nuts will also preload the bearings and put the leadscrew under tension. This design is thermally stable, and the stiffness of the leadscrew will be in series with the bearing and bushing stiffnesses. Having these rolling elements will decrease the driving torque.

- THRUST BEARING CONCEPT: this third concept is similar to the first concept but the washer and bushings being replaced by thrust bearings. It has the same characteristics but will require a smaller driving torque thanks to the thrust bearings. (more expensive than bushings though)
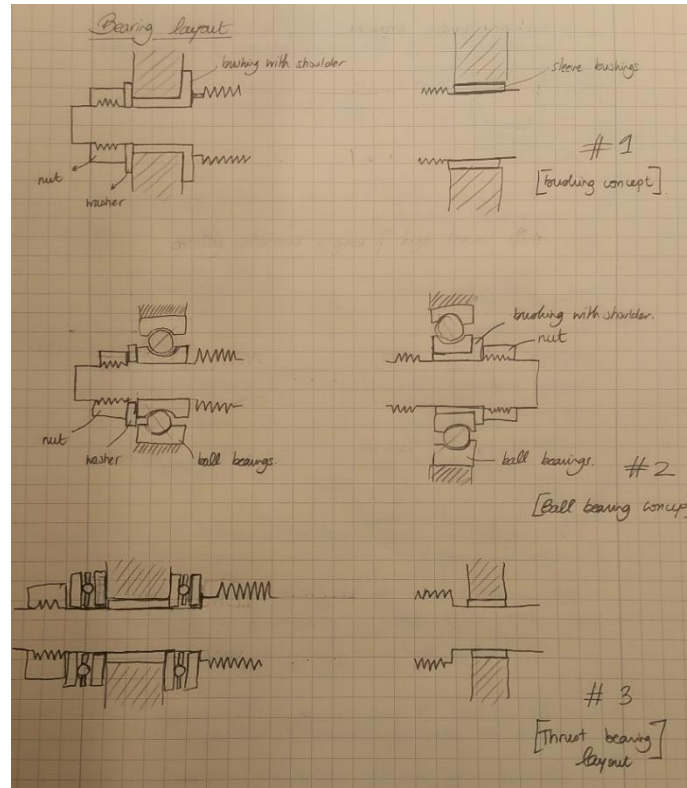
Figure 10: Leadscrew bearing layout concept sketches

From the analysis, I considered the ball bearing concept from the three described to have a low driving torque and ease of assembly and preloading compared to the other two. The analysis included:

- The required driving torque using the screw equation, cutting loads, friction of the carriage on the rails and friction at the bearing blocks on the leadscrew.
- The axial and bending stiffness of the leadscrew (beam under compression and pin-pin beam), as well as the bushings and bearings (hertz contact).
- Look at the back driveability (screw equation condition)
- Stress analysis on the leadscrew from cutting loads and misalignment.
- Geometric errors due to misalignment of the leadscrew (force displacement on a pin-pin beam, resulting in a load on the carriage → displacement at the tool tip)
- Load induced errors at the tool tip due to leadscrew's assembly axial stiffness and cutting loads.

From the CAD model, the leadscrew with a custom anti-backlash nut was then manufactured, mainly on the lathe and the mill, and assembled on the linear motion slide:
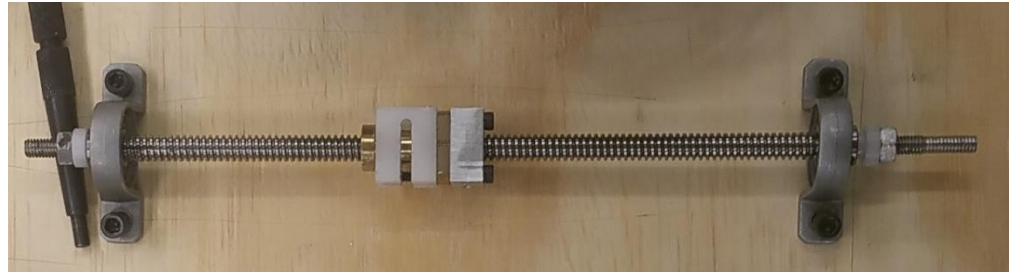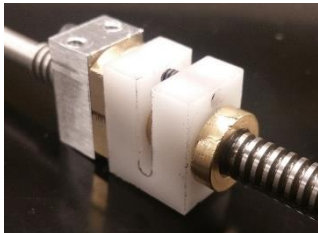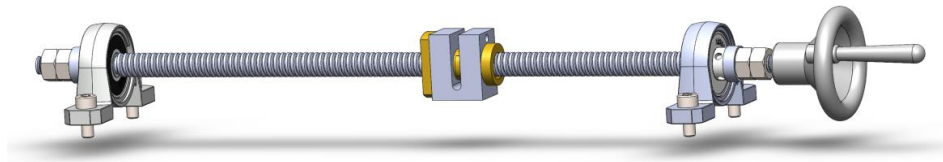

Figure 11: (top) CAD of the leadscrew, (left) close up of the custom anti-backlash nut, (right) leadscrew assembly

The resulting parts where then assembled, preloaded and tested to ensure that it performed within the functional requirements
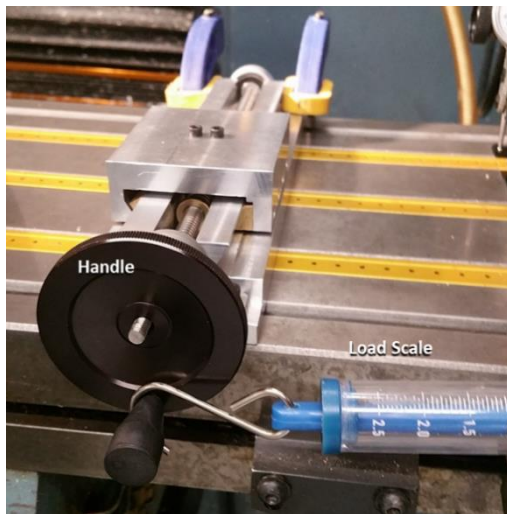

Figure 12: (left) driving torque measurements (right) leadscrew stiffness measurements

The stiffness and driving torque match the requirements and a second leadscrew was fabricated and assembled on the second linear motion slide.

Table 4: Summary of the characteristics of the leadscrew

	Target	Measured
Driving Torque without cutting loads [Nmm]	< 200	128
Axial Stiffness [N/um]	> 0.5	2.6

Spindle Assembly

For the spindle assembly, we were provided an electric motor that required to be mounted on the carriage. From the error budgeting, in order to reduce the errors, the motor shaft had to be located as close to the center of stiffness of the carriage as possible while enabling the tool to reach the part from the second linear axis.

Motor mounts were then designed in conjunction with the tool post mount such that the tool tip would be centered on the motor shaft. In addition, the sizing of the mounts was done such that the load-induced errors from the cutting loads would not deflect the tool tip relative to the part more than 11 microns (from allowable error).

From the CAD model, the motor mounts and the tool post were then manufactured, mainly on the waterjet and the mill, and assembled on the linear motion slides.
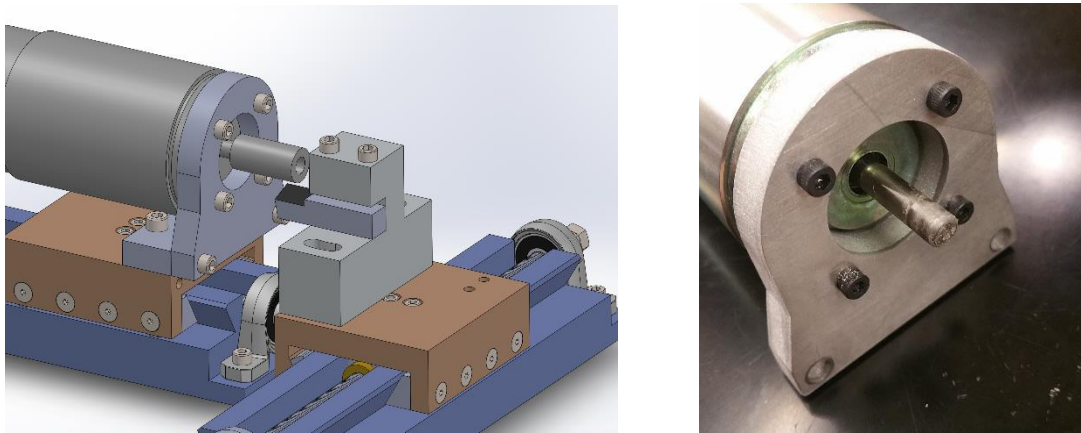

Figure 13: (left) CAD of the electric motor, motor mount, and tool post, (right) manufactured motor mount

Full Lathe Assembly and Tests

Assembly

From all these modules, the error budget model for the entire machine was updated from the CAD and layout of the machine.
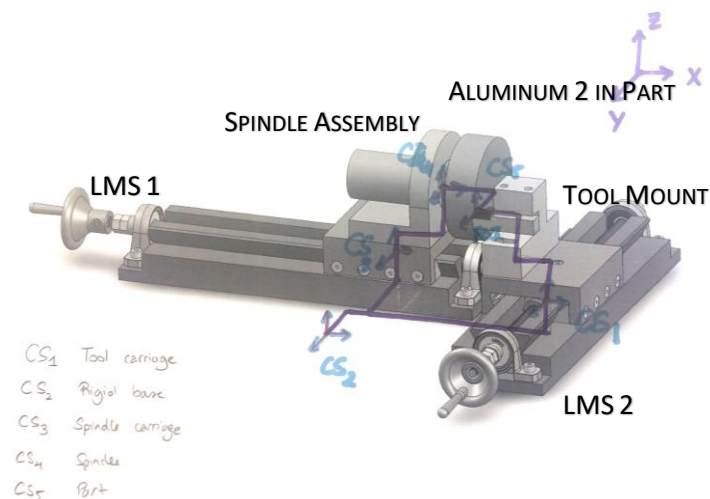

Figure 14: Structural loop and error budget setup

Table 5: Summary table of the errors for finishing pass cutting loads (9N, 7N, 7N), in the worst-case scenario, when the servo positions is at 0,0 (biggest part possible)

		For the Entire Machine				
All Axes	**Number of axes**	**All axes' Geometric Errors**				**F=kX displacement**
	5	**Random**			**Systematic**	
		Sum	**RSS**	**Avg(SUM, RSS)**	**Sum**	**Sum**
	deltaX	0.0262	0.0167	0.0215	0.0085	0.0057
	deltaY	0.1183	0.0869	0.1026	-0.0265	0.0103
	deltaZ	0.0336	0.0212	0.0274	0.0480	-0.0208
	Vector displacement	0.1257	0.0910	0.1083	0.0555	0.0239

From the error modelling for a finishing pass (0.1 mm), the tool tip error is 5.7 um, 10.3um and 20.8 um in X, Y, Z which satisfies the 20 um allowed error in the X direction (flatness error).

A base plate was then made from an aluminum sheet, faced off, drill and tapped such that all of the modules could be aligned on it. To align the two axes, the entire machine was assembled on a mill and a dial indicator was used to align the linear motion slides with the X and Y axes of the mill.
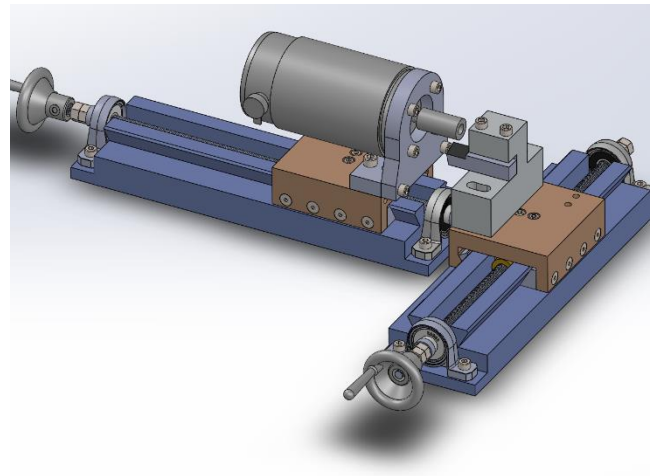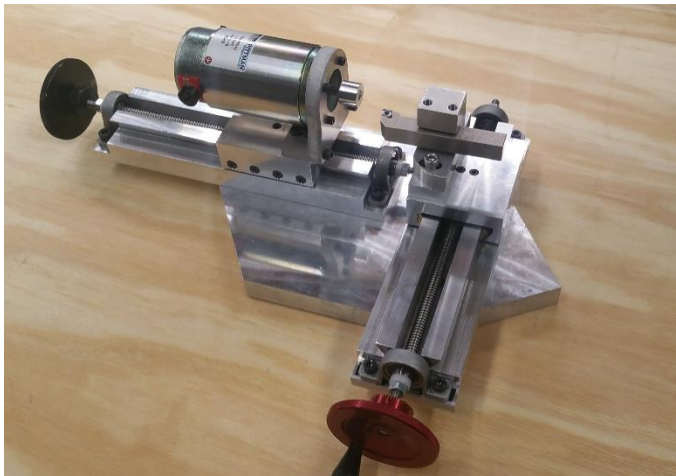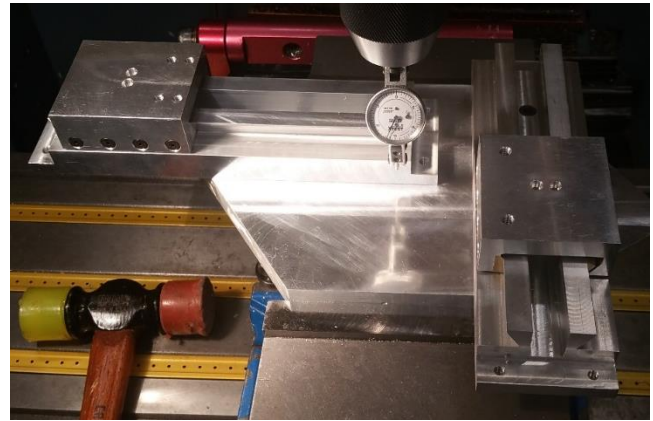

Figure 15: Assembly of the T-based lathe

Test and closing the loop

Several parts were then faced on the T-based lathe to ensure that the two axes were perpendicular. The motor mounts were then shimmed to account for any misalignment. After this alignment procedure, another set of parts were faced with a depth of cut of 0.1 mm to measure the performance of the lathe.
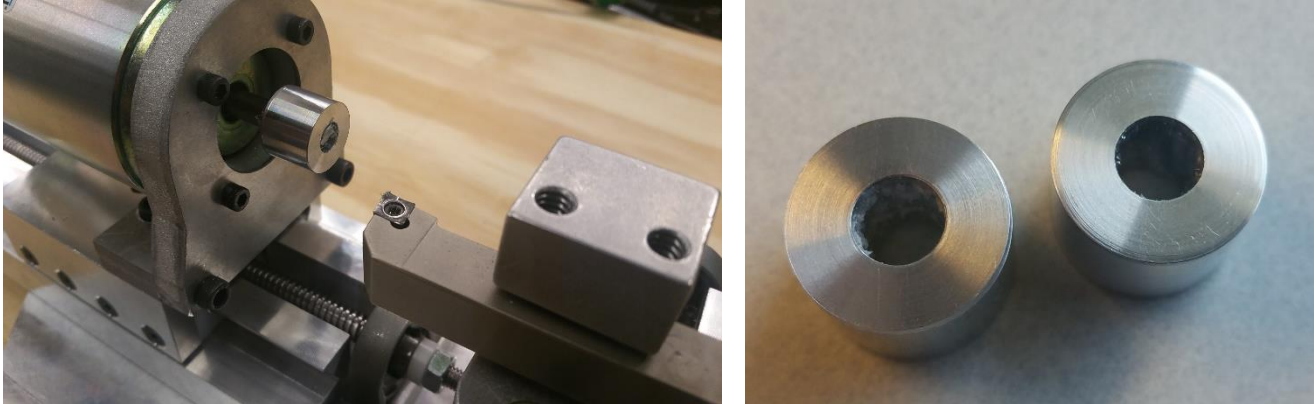

Figure 16: Test cut of an 5/8" diameter aluminum part

The flatness of the 0.1mm test cuts were then measured on a ZYGO white light interferometer, exhibiting a maximum height difference on the face of 3.4 ± 0.2 microns, and a misalignment between the two linear motion axes of 0.89 ± 0.05 mrad matching our design requirements and our predictions of 5.7 microns load induced errors.
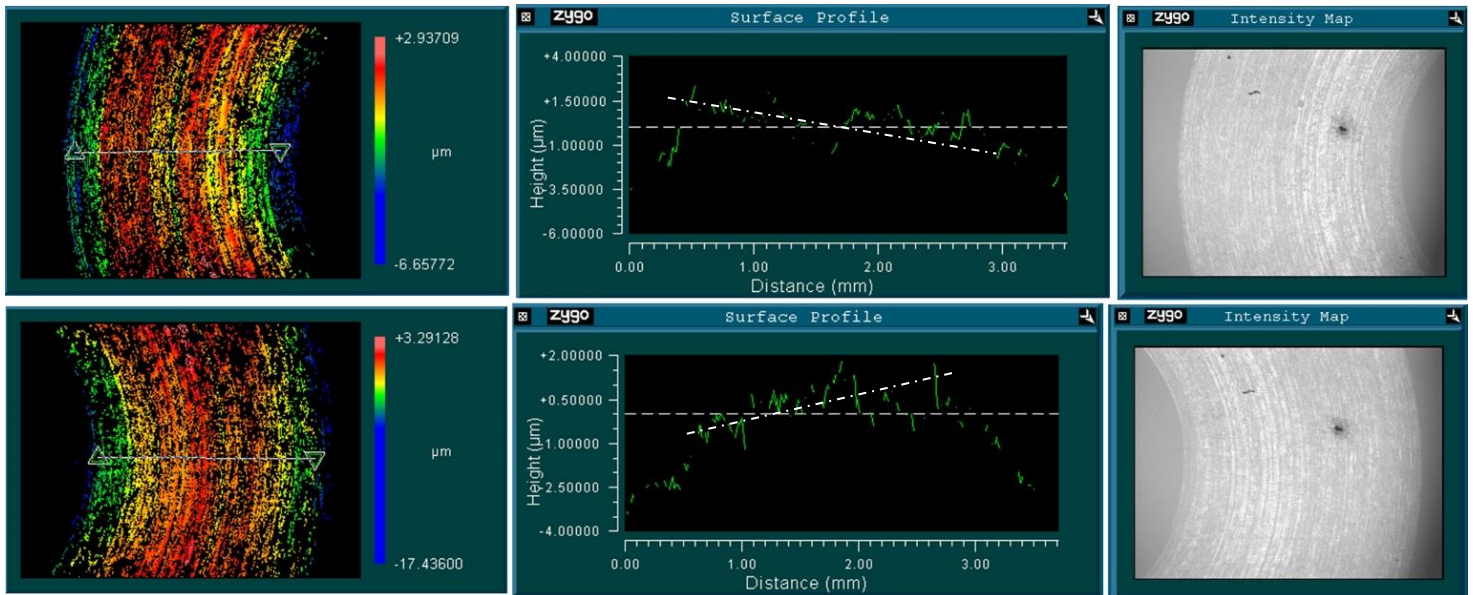

Figure 17: ZYGO White Light Interferometry of test part

	TARGET	PREDICTED	MEASURED
Flatness Error [um]	<20	5.7	3.4
Depth of pass [mm]	>0.1	-	0.1

Table 6: Summary of the T-based Lathe characteristics

Discussions

The T-based lathe performed as expected however the current spindle assembly would need to be replaced by an electric motor running an actual spindle with a chuck or collet to allow us to not only face the part but also turn the diameter.

Overall, I am very pleased with this project that was a valuable experience especially from learning and practicing the design process discussed all along this report. This process definitely enables us to rapidly rigorously and deterministically design precision systems.

I would like to especially thank the teaching assistants, David Taylor and Hilary Johnson for their help, my peer review team Sahil Shah, Rashed Al-Rashed and Guillermo Diaz as well as all my classmates for being great shop buddies.
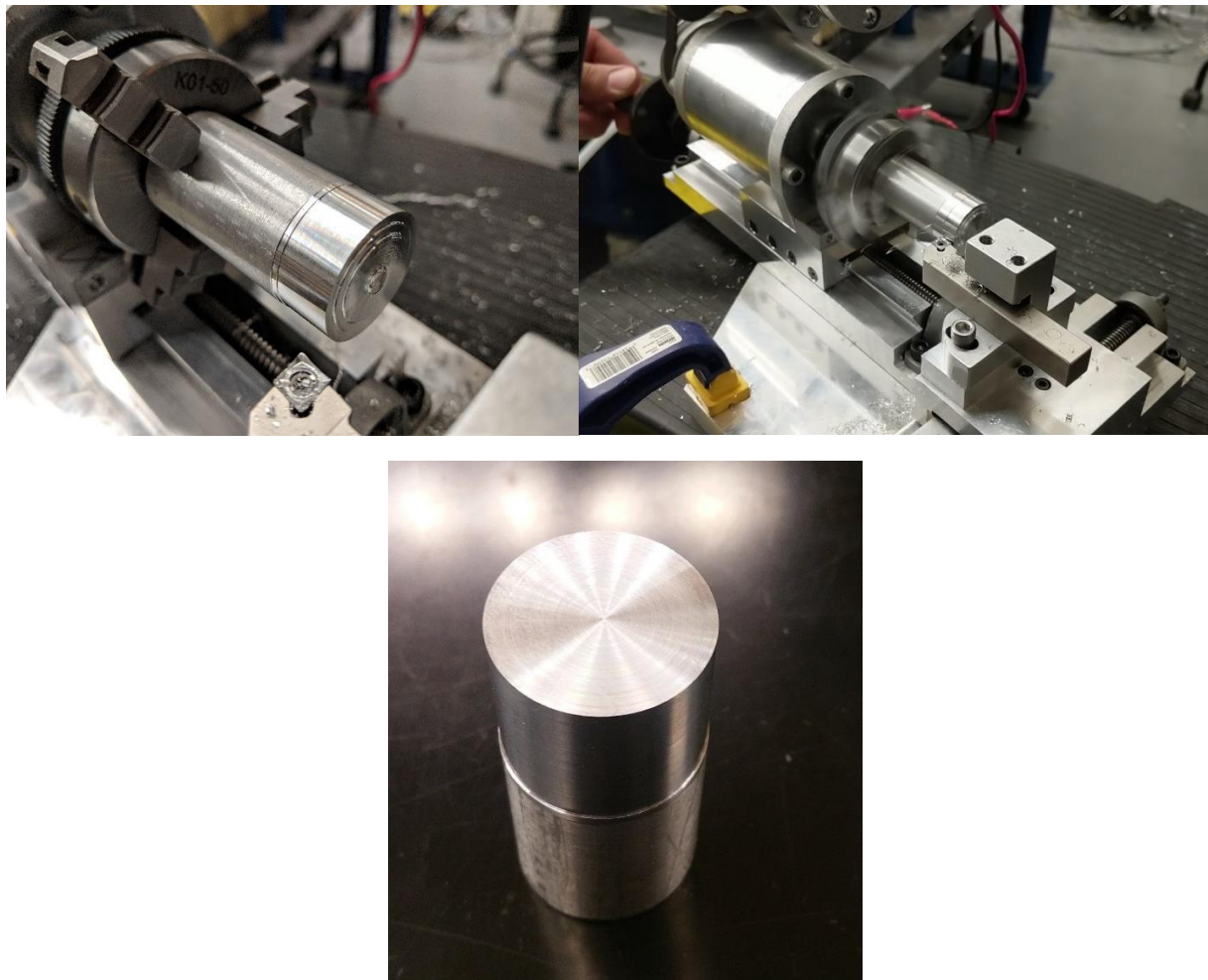

Figure 18: T-based lathe improved after class with an actual spindle assembly (chuck, electric motor, timing belt...), turning down and facing an 1.25" diameter aluminum part