FEYZIYE SCHOOLS FOUNDATION

IŞIK UNIVERSITY

FACULTY OF ENGINEERING

MECHANICAL ENGINEERING DEPARTMENT

MECH/MECT/MAKİ4902

GRADUATION DESIGN PROJECT

DESIGN OF A TABLETOP LATHE

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1. ABSTRACT

The object for this project to design a tabletop lathe. Lathes are used to shape materials to desired geometry. Commonly machined materials are wood or metal. However, primary reason of using lathe machine is removing unwanted piece from material. Design constraints are given at the very beginning of the project which are maximum diameter of machined workpiece is 50mm, maximum length of workpiece will be 300mm and material of machined workpiece will be aluminium due to machining with steel will require much more struggling and wood would be in amateur status for project. Aluminium is more moderate material to work with. Feed rate will be 0,2mm/rev. Rake angle will be based as 7° and lathe must machine class of Aluminium which must have maximum 150MPa shear strength.

Calculations to find a tangential and feed force due to these parameters that had been said in previous sentence. After calculating feed and tangential forces, calculations about motor and spindle shaft were made by following that parameters and 3D drawing is done in SolidWorks and static analysis of them were made in ANSYS Workbench. Calculations in under part that contains bed, slides, power screw were made by taking weights of selected material. Also, all fatigue and dynamic calculations were made.

Differently from traditional lathes, tool post won’t have hand wheel because it will be moved by programmable controller. Chuck is selected to be as 3-jaw because of being more suitable-cylindrical workpiece- to clamp with 3 jaws. Budget was determined to be not exceeding 7500USD after benchmarking analysis.

2. INTRODUCTION

2.1 Requirements

Materials must be selected correctly, and requirements list are given below.

- Maximum clamping range 50mm 3-jaw chuck
- 2- AISI1045 shafts for motor and spindle shaft
- Stepped Motor
- 6- bolts and nuts to assembling under the holes of motor
- Headstock
- Bed & Slides
- Power screw & motor of power screw
- V- belt and pulley system
- 4 – bearings
- Medium carbon steel sheet for chassis of lathe body
- Left-handed HSS cutting tool
- Tool Post & Compound
- Programmable Controller
- 1- MT2 Tailstock & MT2 Live Center
- 2- Feather keys to assembling intro pulleys to prevent sliding
- Bolts to make tailstock stand only move forwards and backwards for centering.

2.2 Design Constraints

Maximum Dimension of the CNC Mini Lathe (mm)	1100 x 450 x 700 (length x height x depth)
Processing Material	Aluminium (Al)
Maximum Dimension of the Processing Material (mm)	300 x 50 (length x diameter)
Maximum Shear Strength of the Processing Material (MPa)	150
The Maximum Value of the Depth of Cut (mm)	2
Feed Rate (mm/rev)	0.2
Budget ($)	7500

Table 1. Design Constraint Table

3.WORK PLAN

Tasks	Responsible Person	1	2	3	4	5	6	7	8	9	10	11	12	13	14
1) Literature Researches	Everyone														
2) Concept Design	Everyone														
3) Cutting Process	Caner														
4) Motor Calculation and Selection	Caner & Furkan														
4.1) AC Motor Selection and Calculation	Caner														
4.2) V-Belt and Pulley Calculation for AC Motor	Caner														
4.3) Stepper Motor Selection	Furkan														
5) 3D Design	Caner														
6) Material Selection	Ege & Furkan														
6.1) All Materials Selection without P.S System	Ege														
6.2) Power Screw Systems' Materials	Furkan														
7) Static and Strength Calculations	Everyone														
7.1) Motor and Spindle Shaft Calculations	Caner														
7.2) Bolt and Nut Calculations	Furkan														
7.3) Power Screw Systems Static & Strenght Calc.	Furkan														
7.4) Welding and Key Calculations	Ege														
8) Power Screw Systems	Furkan														
9) Ansys Analysis	Ege														
10) Control	Caner														
11) Cost Analysis	Ege														
12) Technical Drawing	Furkan														
13) Documentation	Ege														

Table 2: Work Plan

4.MATERIAL SELECTION

4.1. Headstock

It’s located left side of machine. Main function of headstock is housing the spindle which turns the chuck for machining processes. In our design it will be used to house spindle and, it will create resistive force by means of support which comes from point contact areas after assembly of spindle shaft.[1]

				38mm flange diameter Headstock[2]	42mm flange diameter Headstock[3]
Category	Criteria	Weight	Basic Solution	Alternative 1	Alternative 2
Properties	Diameter of shaft flange	5	0	-1	1
	Dimensions of headstock	4	0	-1	1
	Cost	3	0	1	-1
	Height	2	0	-1	1
	Shape of covering body	1	0	1	1
	Total			-7	9

Table 3: **Pugh Matrix of Headstock**

Initial criteria is diameter of shaft flange because chuck will be fit to there firstly Then, housing diameter should be enough to house enough length. Our lathe stock will house 12mm and 22mm will be exceed from headstock. To minimize possibility of bending of failure, bearings will also minimize this possibility and alternative 2 is way cheaper than alternative 1 and alternative 2 is higher. Results show that selecting alternative 2 is the best selection.

4.2. Selected Headstock

- 160mm(height)*150mm(length)*120mm(depth)

Tailstock [3]

- Cost 2,026.64 TRY

Figure 4.2.1: Selected Headstock

4.3. Chuck

Chuck is mounted to headstock of lathe. Jaws hold the part/rotating workpiece. Also, it can hold non-symmetric parts too In this project, chuck is required to hold the workpiece or machining.[4] As its've said in initial sentence. We will use the chuck to hold a symmetric, cylindrical workpiece. In Pugh matrix below, comparison of 4 different chucks were done.

				80mm×3 Chuck [5]	63mm×3 Chuck [6]	80mm×4 Chuck [7]	63mm×3 Manually Remoted Chuck [8]
Category	Criteria	Weight	Basic Solution	Alternative 1	Alternative 2	Alternative 3	Alternative 4
Properties	Clamping Range	5	0	-1	1	1	-1
	Strength Resistance	4	0	1	1	1	1
	Price	3	0	0	0	0	0
	Height	2	0	-1	0	0	0
	Number of Jaws	1	0	1	1	-1	1
	Total			-2	10	8	0

Table 4: Pugh Matrix of the Selection of the Chuck

Calculations were made due to having maximum 50mm of clamping range for chuck. Alternative 2 and 4 got directly -1 because of having clamping range more than 50mm. Prices are nearly same for both alternatives, so they got 0. Both have 1 point due to having high yield strength due to being made of steel. For cylindric geometry, it should be cylindrical due to design constraints which leads to select 3-jaw chuck which is more suitable to hold cylindrical workpieces. And, if the height increases, bore diameter of spindle shaft will be increase due to geometry. By conclusion, alternative 2 got highest point which equals to 10 is the best selection for the design.

4.3.1. Properties of the Chuck

- 63mm*3 Chuck. 63 mm diameter and 3 jaw chuck [6]
- Cost: 1,176TRY
- Clamping Range: 50mm maximum.
- Dimensions: 63mm × 63mm × 42mm.

Figure 4.3.1.1: 63mm M14x1 Mini 3-Jaw Metal Lathe Chuck

4.4. Tailstock

Function of tailstock is supporting free end of the workpiece while it's machining. Tailstock will be used to center operations in case of needing in our design. Live center is inserted to tip of tailstock and from handwheel [9], it may move towards or backwards.

				MT4 Tailstock [10]	MT3 Tailstock [11]	MT2 Tailstock [12]	MT2 Tailstock [13]	MT2 Tailstock [14]
Category	Criteria	Weight	Basic Solution	Alternate 1	Alternate 2	Alternate 3	Alternate 4	Alternate 5
Properties	Dimensions of each part	5	0	-1	-1	1	1	1
	Diameter of Morse Taper System	4	0	1	1	-1	-1	1
	Height	3	0	-1	1	-1	1	1
	Price	2	0	-1	1	-1	1	1
	Shape	1	0	1	-1	1	1	-1
	Total			-5	3	-3	7	13

Table 5: Pugh Matrix of the Selection of the Tailstock

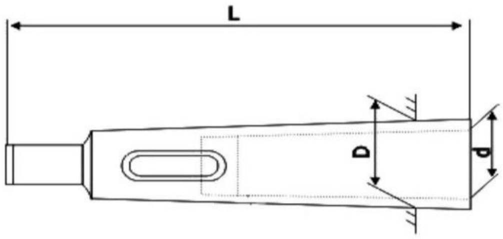
Due to being proportional for our lathe machine, alternative 1 and 2 got -1 because, for example dimension of handwheel is a little bit much for design. Length of alternative 2 is 210mm which will restrict more than the other alternatives movement area of tool post. By basing Morse Taper System, alternative 3 and 4 have even little or high shank or centre diameter dimensions. Morse Taper System will be explained in next part. Shape of alternative 1 is not good for visuality which makes it to have -1. By conclusion, Alternative 5 is selected for the design.

4.4.1. Morse Taper System

Morse taper is a spindle that is set tip of the tailstock body for mounting the live centre tool. Morse taper in tailstocks have hollow and cylindrical structure (female part). Male part (live centre) is put into it. Which means these two parts are produced for fitting to each other. It prevents slippage and most beneficial things for morse taper design are installing and uninstalling the lathe centre are simple so do centering the live centre is.[15]

Types of Morse Tapers are given below:

- ➢ Morse Taper 1 (MT1)
- ➢ Morse Taper 2 (MT2)
- ➢ Morse Taper 3 (MT3)
- ➢ Morse Taper 4 (MT4)



Item No	Morse taper	D	L	d	Accuracy (mm)	Price
105255-12	2/1	17.780	92	12.065	0.003	7usd
105255-13	3/1	23.825	99	12.065	0.003	10usd
105255-14	3/2	23.825	112	17.780	0.003	10usd
105255-15	4/1	31.267	124	12.065	0.003	23usd
105255-23	4/2	31.267	124	17.780	0.003	23usd
105255-24	4/3	31.267	140	23.825	0.003	23usd
105255-25	5/1	44.399	156	12.065	0.003	49usd
105255-34	5/2	44.399	156	17.780	0.003	45usd
105255-35	5/3	44.399	156	23.825	0.003	39usd
105255-45	5/4	44.399	171	31.267	0.003	35usd

Figure 4.4.1.1: Metric dimensions of Morse Taper Types

Where,

D: Taper diameter(mm), d: Centre diameter (mm), L: Length of Live Centre (mm)

4.4.2. Selected Tailstock

Figure 4.4.2.1: Selected Tailstock and Dimensions

- ➢ Myers Store [14]
- ➢ Cost:603,89TRY

4.5. Live Centre

Live Centre (As known as lathe centre) is used for supporting the machine whether it's fixed or rotating end while machining. Working principle of it is occurring of friction farthest point on it between workpiece and centre due to rotation of the workpiece.[16] Live centre will be inserted into tailstock to do centering process in case of any needing.

				MT2 Live Center/Standard Tip [16]	MT2 Live Center/ Bullnose Tip [17]	MT3 Live Center/ Conic Tip [18]	MT3 Live Center / Bullnose Tip [19]
Category	Criteria	Weight	Basic Solution	Alternate 1	Alternate 2	Alternate 3	Alter-nate 4
Properties	Tip Of Live Center	5	0	1	-1	1	-1
	Type Of Morse Taper (MT2, MT3…)	4	0	1	1	-1	-1
	Maximum Working Load	3	0	-1	-1	1	1
	Price	2	0	1	-1	1	-1
	Material	1	0	1	1	1	1
	Total			9	-3	7	-7

Table 6: Pugh Matrix of Live centre

The reason to put type of tip at the first criteria because every type of live centre has different attributes. Alternative 1 is standard pointed live centre and it has not specific tip or hardening. Since our design is about machining by cutting tool standard pointed live centre selected due to do nothing with him at the first case. Second criteria are having the diameters of MT2 systems because tailstock is selected as MT2. MT3 live centres got -1 no matter what their attributes are. Since maximum loads of alternative 3 and 4 are high due to their catalogue variables, alternative 1 and 2 got -1. All of them are made of steel. By conclusion, alternative 1 is selected.

4.5.1. Selected Live Centre

- MT2 Live Center/Standard Tip.[16]
- Material: Hardened Steel
- High Accuracy
- Maximum Load Capacity: 2500lb
- Cost: $19,99

Figure 4.5.1.1: Shape of Live Centre

4.6. Tool Post & Compound

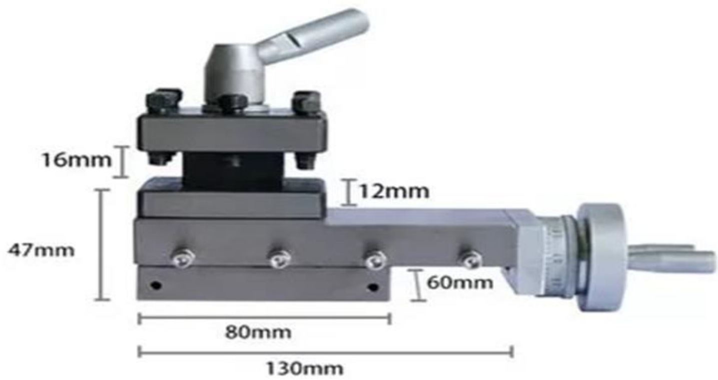
Tool post is the part that holds to cutting tool during the cutting operation. Tool post designs are only for one cutter or multiple cutters. This tool slots may be accepting the cutters which are designed to be the tip of the shape as square, round or dovetailed. Security of cutter is provided by locked bolts. Number locked bolts may be two or more than two. In this design, as always, tool post will be used to tighten the cutting tool in it while machining.[20] Compound slide is a material to prevent a possibility of turning tool post through clockwise or anti-clockwise. Maximum turning angle may be differ between designs. In this design, tool post and compound will be assembled to each other and advantage of compound which will create a possibility to make unique and nicer machining.[21]

				167mm length Tool Post & Compound [22]	150mm length Tool Post & Compound [23]	130mm length Tool Post & Compound [24]
Category	Criteria	Weight	Basic Solution	Alternate 1	Alternate 2	Alternate 3
Properties	Dimensions	5	0	-1	1	1
	Max. Turning Angle	4	0	-1	1	1
	Weight	3	0	1	-1	1
	Price	2	0	0	0	0
	Design	1	0	1	1	1
	Total			-6	7	13

Table 7: Pugh Matrix of the Selection of Tool Post & Compound

Dimensions of alternative 1 may restrain the allowable moving area because there must be produced bigger platform to put them. Maximum turning angle is an important parameter due to making more unique designs by having it more. Alternative 2 is a little bit heavier than the other two selections. Although, tool posts in Pugh Matrix aren't heavy it will reduce the life of slides faster than the other two and prices are so close to each other which leads them to get 0. Designs are almost same which is advantage for all. But if we conclude these parameters. Alternative 3 is selected. All of the three tool posts are AXA tool posts because dimensions of BXA tool posts would not be preferable for our design. [25]

4.6.1. Selected Tool Post & Compound



- 130mm length Tool Post & Compound [24]
- Weight: 3kg
- Material: Cast Iron

Price: 1,050.75TRY

Figure 4.6.1.1: Selected Tool Post & Compound

4.7. V- belt and Pulley

Power must be transferred from one axle to another to work the system accurately. Belt and pulley system is used for this transfer. Function of belt and pulley will be the same in our system. It will transfer the torque from motor shaft to spindle shaft which will rotate chuck to machine. Diameters of two pulleys may differ from each other two which may require increasing or decreasing applied torque. [26]

				A section	B section
Category	Criteria	Weight	Basic Solution	Alternate 1	Alternate 2
Properties	Dimensions/Suitability for Design	5	0	1	-1
	Efficiency	4	0	-1	-1
	Maintenance Costs	3	0	1	1
	Tolerance	2	0	1	1
	Price	1	0	0	0
	Total			6	-4

Table 8: Pugh Matrix of the Selection of Type of V-Belt

Pugh Matrix is done between section A and B because these two sections have most close dimensions to our design. But, dimensions of A is more suitable to our design and since our required power and torque values[?] are low, section a would be enough. However, efficiency of belt drives is low which got them -1. Section A would be enough to use. Gear drive may be used in this application also. But, after the Pugh Matrix below, reasons will be explained in **Table 8**.

Gear drive may be used in this application also. But, after the Pugh Matrix below, reasons will be explained in **Table 8.**

				Oleostatic	Gold Label COG Belt	Linea - X	Power Wedge
Category	Criteria	Weight	Basic Solution	Alternate 1	Alternate 2	Alternate 3	Alternate 4
Properties	Speed Transmission under Friction	5	0	1	-1	-1	-1
	Efficiency	4	0	-1	1	1	1
	Suitability for Design	4	0	1	1	-1	-1
	Failure Costs	3	0	1	-1	-1	-1
	Price	1	0	0	0	0	0
	Total			8	0	-8	-8

Table 9: Pugh Matrix of the Selection of Oleostatic Type of V-Belt

Category	Criteria	Weight	Basic Solution	Gear Drive Alternative 1	Belt drive Alternative 2
Properties	Toughness to dislocation	5	0	-1	1
	Efficiency	4	0	1	-1
	Performance stability in weather conditions	3	0	1	-1
	Maintenance Costs	2	0	-1	1
	Cost	1	0	-1	1
	Total			-1	1

Table 10: Pugh Matrix of the Selection of Power Transmission System

Belt drives got more tough about little dislocation and misalignment. However, gear drive may affect from even little misalignments. However, loss of efficiency is more in belt drive due to friction which created from pulleys so, torque gets lost in translation. Performance of pulleys may affect negatively when environment temperature increases which leads pulleys to slip undesirably. Lubrication is needed for gear drive that makes maintenance cost of it is higher, so do initial costs. So, using section A V belt and pulley drive would be more preferrable

4.8. Material Selection for Lathe Body

Shaft bodies are drawn in SolidWorks. However, it would be better to explain the concept of welding. Welded parts will be the sheet that lies under the bed and perpendicular shaft which the chuck will be assembled inside of it. But main welding will be done with ribs. To select an appropriate body for shaft, main criterias should be easy to weldment and not requiring pre-weld or post-weld operation to avoid any welding failure. Three of most common materials for welding is given below:

- Low Carbon (Mild) Steel
- Aluminium
- Stainless Steel

Initially, by comparing aluminium to two types of steels, 1XXX classes of aluminium may be required less effort. For example, 6XXX and 2XXX classes some special procedures should be done like welding in using proper material to avoid crack prevention or proper material should be used for welding without post failure. Most importantly, aluminium does not have much strength as steel. Dust and spatter occur on surface of aluminium which leads to requirement to cleaning surface to remove surface scale. By mixing these considerations, aluminium will not be selected for body sheet.

Proper design consideration for stainless steel is intergranular corrosion which happens when stainless steels is subjected to high temperatures. Combining chromium and carbon inside of the sheet metal is best solution for prohibiting corrosion. Consideration of adding material to metal to failure free welding may not be the good selection therefore, stainless steel is not going to be the material of sheet.

Mild (low carbon) steel is the best solution due to being more ductile by comparing them with other two types because of having less carbon atoms inside. Main selection reason in low-carbon steels, pre and post heating applications were not required for welding failures. With this selection, some negative effects like martensite occurring and hydrogen cracking would be removed. Second reason is having high amount of usage in very wide application areas and research and development projects are made throughout years to make it more weldable. In medium and high carbon steel, pre and post heating applications are required to avoid from struggles. [27]

4.9. Ribs

Ribs are shaped triangular or different geometry and they used for supporting two perpendicular sheets. They are thinner than will be welded wall and they provide rigidity also. Ribs will be welded to shaft body by corner joint. [28] Corner joint will be made by fillet joint. Rib calculations will be explained in part.

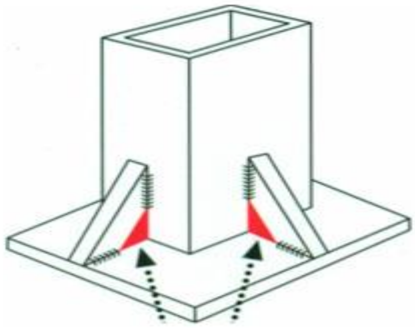


Figure 4.9.1: Rib Welding (Example)

5.DESIGN AND CALCULATIONS

5.1. Tabletop Lathe Machine

Tabletop lathes are called also mini lathes. They are machines that created to process and design small and precision parts. Lathe is a machine tool that shapes cylindrical materials into products by using cutting process. Mini lathe has almost same properties as universal lathe. However, they have some advantages against universal lathe such as mobility, storable and less power consuming.

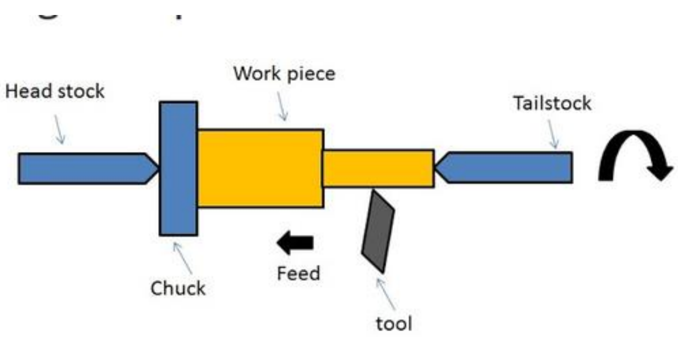


Figure 5.1.1: Working Principle of the Lathes

The cutting tool must be moved over rigid linear bed in order to perform cutting process. This movement can be done with two different ways. First way is manual movement required to operator and power transmission parts. The other way is automation which requires microprocessor, motor, and some drivers.

According to the knowledges above, there are two types of movement. These are determining to the types of lathes: Manual mini lathes and CNC mini lathes.

5.2. Manual Mini Lathes

They are machines that move linearly over the materials rotating around its axis and perform chip removal with the help of cutting tools. The most basic difference from CNC lathe to manual lathe is that manual lathe is operated by hand and CNC lathe is operated by microprocessors. This operation is performed with the help of an operator. That’s why, operator must be control the machine during the time the material is to be processed. The cutting process starts when the carriage moves on a power screw. There is a gearbox system to be able to move the carriage more easily while processing the hard materials. The purpose of this is reducing the force applied according to the rate of the gears. That is, the operator deals with to the hard materials with less force thanks to gearbox system.

Figure 5.1.2: Manual Mini Lathe

5.3. CNC Mini Lathes

Explanation of CNC is Computer Numerical Control. CNC based lathe can also performs the same operations as well as manual mini lathes. However, there is difference compared to the manual mini lathe. They are working depending on a specific microprocessor such as PLC, Arduino etc. If the working principle explained simply, motors are required for the x-axis and z-axis movements. Motor drivers are needed to operate stepper motors. Thanks to these drivers and microprocessor we can drive motors synchronously with each other. Therefore, operator can control x-axis and z-axis movement on their control screen. However, operator does not have to wait at the machine until the process is finished.

Figure 5.3.1: CNC Mini Lathe

5.4. Selection of the Mini Lathe Types

		CNC Mini Lathe	Manual Mini Lathe
Criteria	Efficiency	✓	X
	Production Cost	X	✓
	Working With Less Experience	✓	X
	Production Variety	✓	X
	Safety	✓	X

Table 11: Comparison of CNC Mini Lathe and Manual Mini Lathe

According to table above, working with CNC Machining have advantages every criteria except of producing them more expensive than Manual types. However, enough budget is given to work with which gives no problem about exceeding budget. More accurate machining applications, more product variety rather than Manual Mini Lathes and almost no touching is required during machining. Process can be done by remoting far away since no injuries due to spilling of tool or it will be resolved being there when there is any fire starts to happen also

5.5. Conceptual Design

In this part, conceptual design, and detailed explanation of it will be done.

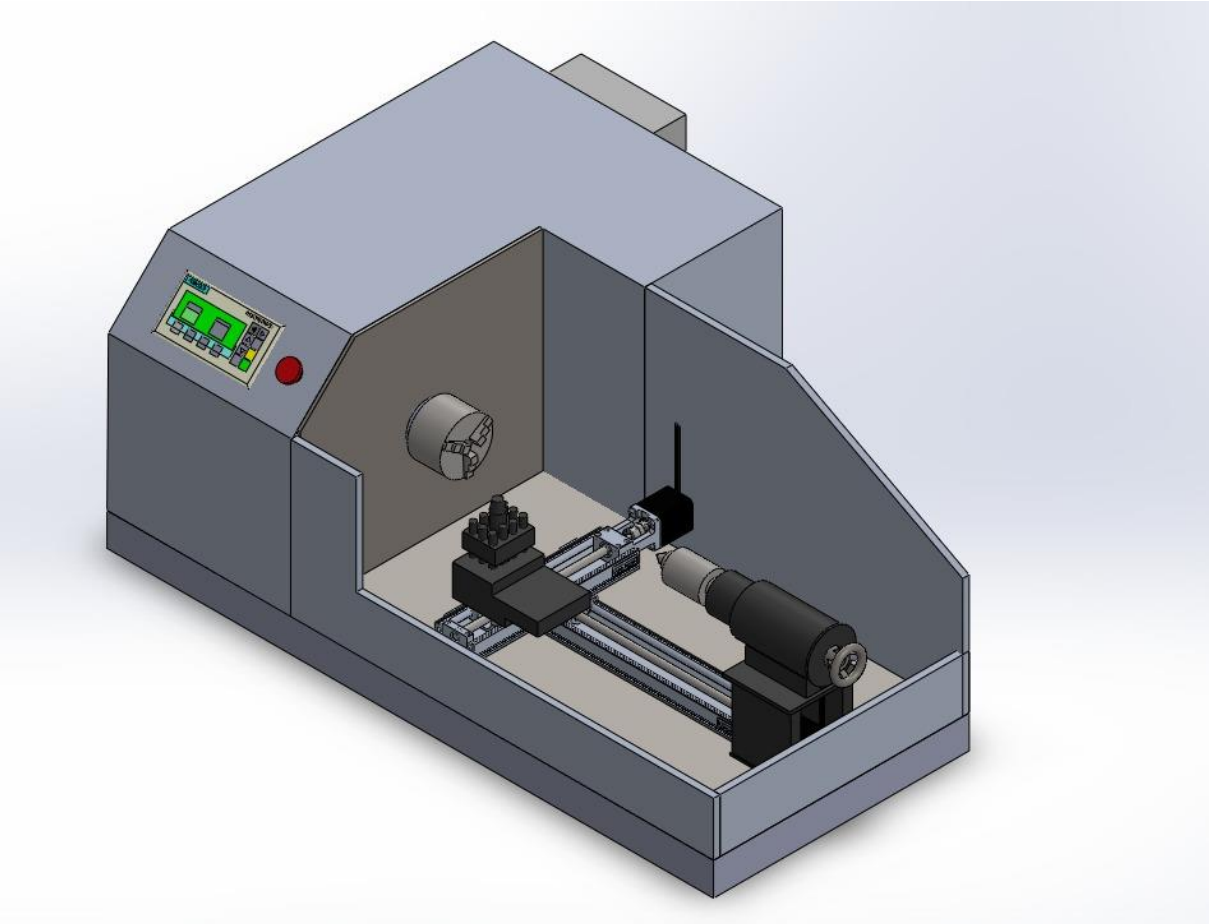


Figure 5.5.1: Conceptual Design of Mini Lathe

In the figure above, we can see the tabletop mini lathe machine that uses for machining. This machining process occurs when a material attached between a chuck and tailstock. This material turns in its axis with the help of chuck, meanwhile cutting tool is moving in x-axis and z-axis, this movement execute the machining process. This process requires these ingredients;

- Headstock
- Tailstock
- Chuck
- Control Panel
- Live Center
- Tool Post & Compound
- Power Screw System

- V- Belt and Pulley
- Motor and Spindle Shafts
- AC Motor
- 2- Stepper Motor
- Automation Panel
- Emergency Stop Button

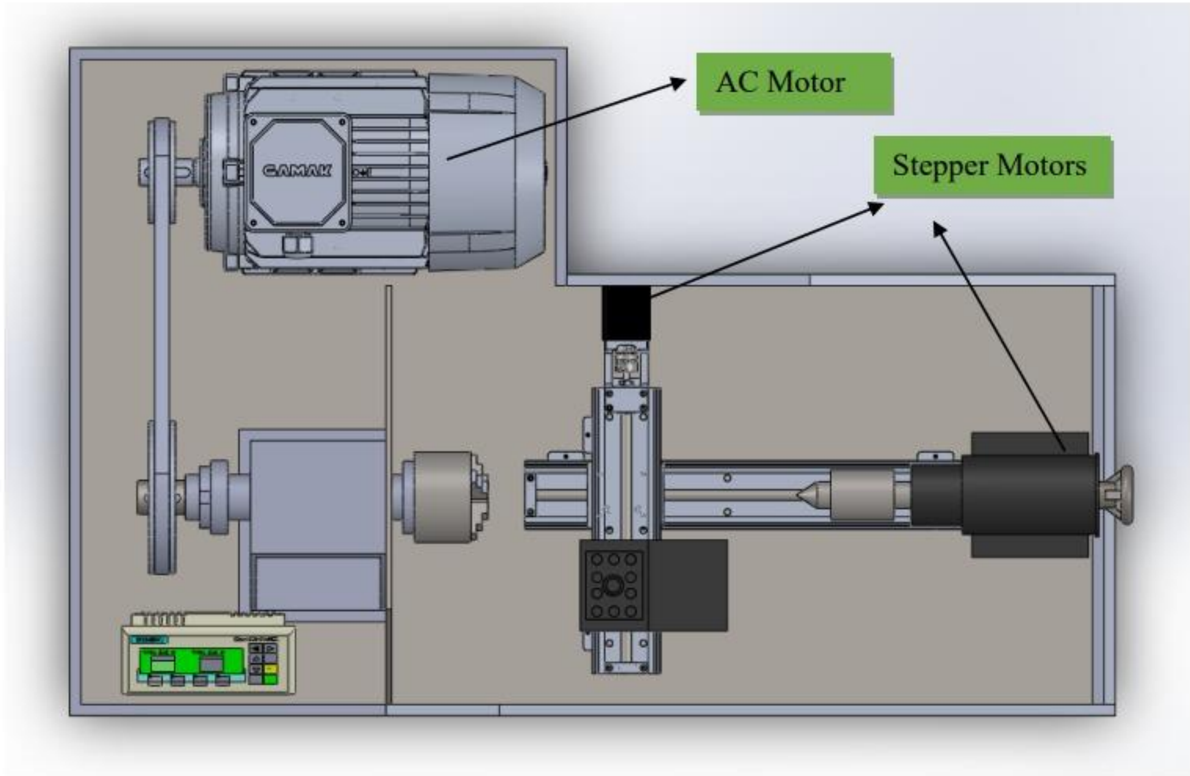


Figure 5.5.2: Top View of Design

As mentioned above, this machining process requires movement in three axes. On above figure, movement axes can be seen as chuck (rotating), x-axis and z-axis (linear). Each of these axes requires its own motor. As seen, that on the above figure, in the linear axes movements stepper motors has been used and in the rotating axis AC motor has been used. The reason that using AC motor is chuck requires high torque. In that time, alternatives are either too expensive, such as servo motors, or too slow, such as stepper motors. The power transmission is used to increase amount of torque since the AC motor which provide required amount of torque is too heavy that its own weight is more than machines' total weight.

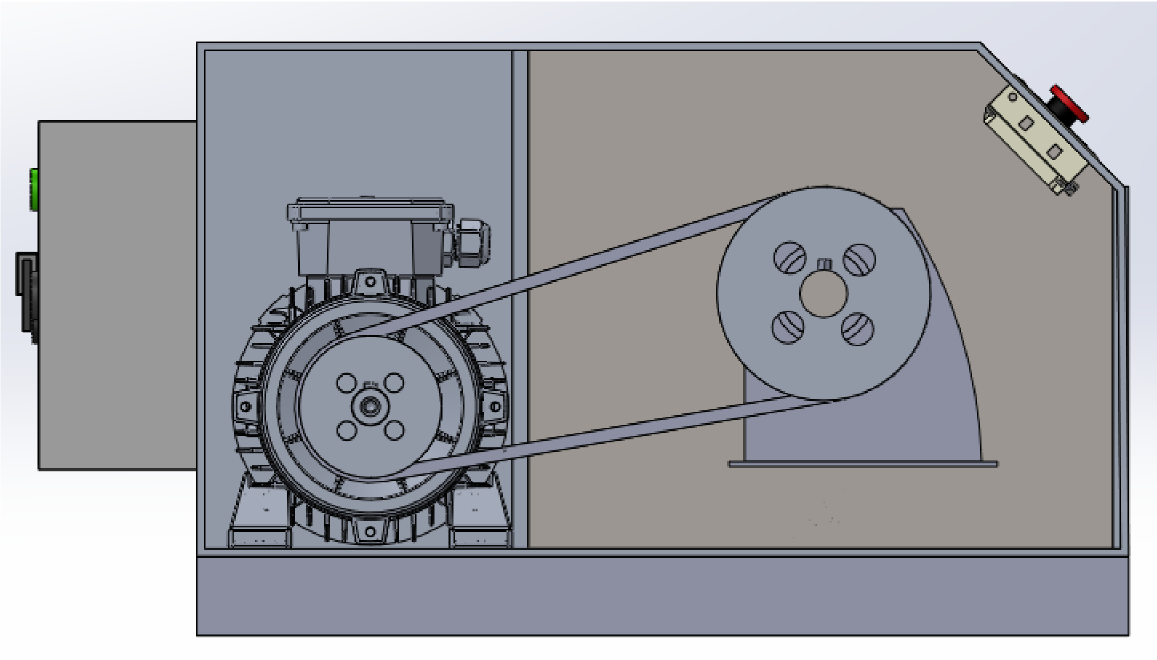


Figure 5.5.3: V-Belt and Pulley in SolidWorks

According to the above figure, V-Belt and pulley system used for the tabletop CNC mini lathe project. The main purpose of the using V-Belt and pulley system is that increasing the amount of torque which supplied from the AC motor. There are many different types of transmission systems. The two of them is generally used in the lathe machines; V-Belt & Pulley System and Gearbox System. In this project V-Belt & Pulley System is used since maintenance hardness, cost / maintenance cost is better than Gearbox System.

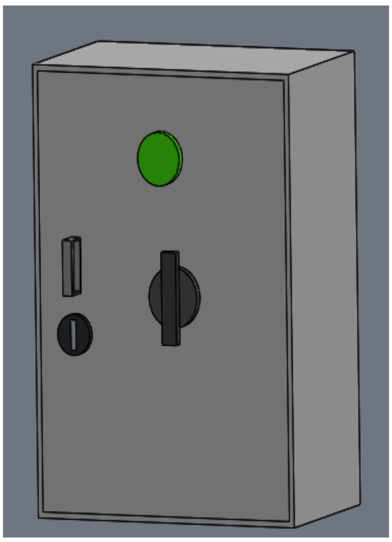


Figure 5.5.4 Control Panel

Automation Panel(left) is used to store PLC, motor drivers and other components safely. Its location consciously selected its current position to be able to easy wiring and maintenance of AC motor

5.6. Cutting Process

The most important part of the project is the cutting process. As it is known that this machine will only work on aluminium materials. The table below contains information about the cutting process constraints.

Maximum Dimension of the CNC Mini Lathe (mm)	1100 x 450 x 700 (length x height x depth)
Processing Material	Aluminium (Al)
Maximum Dimension of the Processing Material (mm)	300 x 50 (length x diameter)
Maximum Shear Strength of the Processing Material (MPa)	150
The Maximum Value of the Depth of Cut (mm)	2
Feed Rate (mm/rev)	0.2

Table 12: Constraints of the Cutting Process

where β_a is the average friction angle between the tool's rake face and the moving chip, and α_r is the rake angle of the tool.

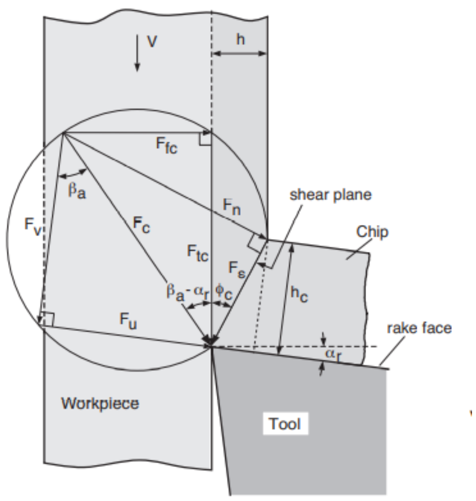


Figure 5.6.1: Cutting Force Diagram

The friction angle (β_a) values change between 15° to 30°. However, there are not any constant data (Orthogonal Cutting Database) so the maximum friction angle value accepted 30° in this project.

The cutting processes generally have 5° to 7° rake angle values. Therefore, the rake angle (a_r) values accepted between 5° to 7° in this project.

The founded shear angle (φ_c) formula below,

$$\varphi_c = \frac{\pi}{4} - (\beta_a - a_r) \text{ [29]}$$

As it is known in the project, most calculations were made according to the maximum case. Therefore, in order to find the maximum shear angle value, the friction angle value is taken as 30° and the rake angle value is taken as 7°.

►**Note:** In the formula below, the degrees must be in radians from.

$$\frac{\pi}{4} = 45° \cong 0.7853981634\,rad \quad 30° \cong 0.5235987756\,rad \quad 7° \cong 0.12217304764\,rad$$

$$\varphi_c = \frac{\pi}{4} - (\beta_a - a_r) = 0.7853981634\,rad - (0.5235987756\,rad - 0.12217304764\,rad)$$

$$= 0.383973\,rad \cong 22°$$

The tabletop lathe will process aluminium. It has been decided that this lathe will process materials have a maximum of **150 MPa** Shear Strength. In more detail, this lathe can work with the following materials.

AA 1000 Series	Shear Strength
1050 Series	52 to 81 MPa
1050A Series	44 to 97 MPa
1060 Series	42 to 75 MPa
1070 Series	48 to 79 MPa
1080 Series	49 to 78 MPa
1080A Series	49 to 81 MPa
1085 Series	48 to 79 MPa
1100 Series	54 to 95 MPa
1100A Series	59 to 99 MPa
1200 Series	54 to 100 MPa
1230A Series	59 to 99 MPa
1235 Series	52 to 56 MPa
1350 Series	44 to 110 MPa
1435 Series	54 to 87 MPa

AA 2000 Series	Shear Strength
2014 (H111, O)	130 to 150 MPa
2014A (H111, O)	130 to 150 MPa
2017 (O)	130 MPa
2017A (H111, O)	120 to 150MPa
2024 (O)	130 MPa
2219 (O)	110 MPa

AA 3000 Series	Shear Strength
3003 Series	68 to 130 MPa
3004 Series	100 to 150 MPa
3005 Series	84 to 150 MPa
3102 (H112, O)	58 to 65 MPa
3103 Series	68 to 130 Mpa
3104 Series	110 to 150 MPa
3105 Series	77 to 140 MPa
3203 Series	72 to 120 MPa
3303 (O)	69 MPa

AA 4000 Series	Shear Strength
4004	63 MPa
4006 Series	70 to 91 MPa
4007 Series	80 to 90 MPa
4015 Series	82 to 120 MPa
4045	69 Mpa
4047	69 MPa
4104	63 MPa
4115 Series	71 to 130 MPa
4145	69 MPa
4147	63 MPa
4343	64 MPa

AA 5000 Series	Shear Strength
5005 Series	70 to 130 MPa
5005A Series	71 to 130 MPa
5010 Series	64 to 120 MPa
5026 (H111, 0)	150 MPa
5040 Series	140 to 150 MPa
5049 (H111,-112,-12,-14,-16,-18,-22,-24)	130 to 150 MPa
5050 Series	91 to 140 MPa
5110A	110 MPa
5110A Series	66 to 110 MPa
5252 Series	120 to 150 MPa
5457 Series	85 to 130 MPa
5652 (H112, - 22,-24,-32,-34	110 to 150 MPa
5657 Series	92 to 110 MPa
5754 (F, H12,-14,-16,-18,-22,-24)	120 to 150 MPa

AA 6000 Series	Shear Strength
6005 (T1-T4)	120 MPa
6008(T4)	120 MPa
6012(T4)	120 MPa
6014 Series	96 to 150 MPa
6016 (T4)	130 MPa
6025 Series	110 to 140 MPa
6060 Series	86 to 130 MPa
6061(T1, O)	84 to 130 MPa
6063(0, T1,T4,T42,T5,T52,T6)	70 to 150 MPa
6066(O)	90 MPa
6082(H111,O,T4,T42,T452)	86 to 140 MPa
6105(T1)	120 MPa
6351 Series	84 to 150 MPa

Table 13: AA Series and Their Shear Strength

►**Note:** According to the **Table 10**, CNC mini lathe will operate different type and series of the Aliminium 1000, 3000, 4000, 5000 and 6000 series.

As we remember from design constraints, this lathe will operate maximum 2 mm depth of cut (b) and uncut chip thicknesses value accepted between 0.05mm to 0.2 mm.

According to the above all knowledge about the lathe, we can find the measured main cutting forces as functions of tool geometry and the cutting conditions.

Tangential Force:

$$F_{tc} = bh[\tau_s \frac{\cos(\beta a - a_r)}{\sin\varphi_c \cos(\varphi_c + \beta a - a_r)}]$$ [30]

Feed Force:

$$F_{fc} = bh\ [\tau_s \frac{\sin(\beta a - a_r)}{\sin\varphi_c \cos(\varphi_c + \beta a - a_r)}]$$ [30]

Where,

b: depth of cut	⍰ Maximum 2 mm
h: uncut chip thicknesses	⍰ Maximum 0.2 mm
βa : friction angle	⍰ Maximum 30°
a_r : rake angle	⍰ Maximum 7°
τ_s : shear strees	⍰ Maximum 150 MPa

Tangential Force:

$$F_{tc} = 2\times0.2[150\times \frac{\cos(30° - 7°)}{\sin22° \cos(22° + 30° - 7°)}] = 208.5\ N$$

Feed Force:

$$F_{fc} = 2\times0.2\ [150\times \frac{\sin(30° - 7°)}{\sin22° \cos(22° + 30° - 7°)}] = 88.5\ N$$

According to the design constraints, maximum workpiece of the diameter for used aluminium is **50 mm**.

The torque required depending on these conditions is as follows,

Torque = Tangential Force (F_{tc}) × Workpiece radius (R) × Factor of Safety (FoS)

$$Torque = 208.5 \times 0.025 \times 2$$

Maximum Required Torque is **10.43 Nm.**

The power depending on the spindle speed. Therefore, the maximum speed of the spindle 2000 rpm in this project.

The required power depending on this speed is as follows,

Power = $\frac{Torque \times Spindle\ Speed\ (N) \times 2\pi}{60}$

Power = $\frac{10.43 \times 2000 \times 2\pi}{60} = 2183.4\ W$

Maximum Power is **2.1834kW.** According to the above calculations, the most important criteria's are Maximum Power and Maximum Torque.

Required Torque	10.43 Nm
Required Power	2.1834 kW

Table 14. Required Torque and Power Values

►**Note:** The power and torque values found are added to the factor of safety values.

5.7. Motor Selection for Spindle

According to the required power and spindle speed, the motor have at least 2.19 kW power and 2000 rpm.

Anma Gücü	Tip	Anma Gücünde							Kalkışta				Devrilme Moment Oranı	Eylemsizlik Momenti	Yaklaşık Ağırlık
		Hız	Akım	Moment	Güç Katsayısı Cos φ	Verim η			Akım Oranı		Moment Oranı				
kW		d/dak	A	Nm		4/4	3/4	1/2	Doğrudan	Y/Δ	Doğrudan	Y/Δ			
0,75	AGM2E 80 M 2a	2860	1,7	2,5	0,82	77,4	77,4	74,6	6,2	-	2,5	-	3	0,00053	8
1,1	AGM2E 80 M 2b	2880	2,3	3,6	0,87	79,6	79,6	77,9	6,3	-	2,7	-	3	0,00066	8,8
1,5	AGM2E 90 S 2a	2880	3,3	5	0,81	81,3	81,3	79,8	6,3	-	2,3	-	3	0,0011	11,5
2,2	AGM2E 90 L 2b	2870	4,5	7,3	0,85	83,2	83,2	81,9	6,6	-	2,6	-	3,1	0,0014	13,9
3	AGM2E 100 L 2a	2880	5,9	9,9	0,89	84,6	84,6	84,1	6	-	2,5	-	3	0,0025	20
4	AGM2E 112 M 2a	2880	7,9	13,3	0,85	85,8	85,8	85	7,2	2,4	2,8	0,9	3,5	0,0039	21,5
5,5	AGM2E 132 S 2a	2900	10,3	18,1	0,89	87	87	86,5	6,6	2,2	2,5	0,8	3,1	0,0108	37
7,5	AGM2E 132 S 2b	2910	13,6	24,6	0,90	88,1	88,1	87,9	7,2	2,4	3	1	3,4	0,014	44
11	AGM2E 160 M 2a	2945	19,5	35,7	0,91	89,4	89,4	88,6	7,7	2,6	3,4	1,1	3,6	0,030	67
15	AGM2E 160 M 2b	2945	28,3	48,6	0,85	90,3	90,3	89,7	7,5	2,5	3	1	3,5	0,041	81
18,5	AGM2E 160 H 2c	2950	32,3	59,9	0,91	90,9	90,8	90,1	7,7	2,6	2,5	0,8	3	0,048	102
22	AGM2E 180 M 2a	2950	38,3	71,2	0,91	91,3	91,3	90,8	8,2	2,7	3	1	3,5	0,066	135
30	AGM2E 200 L 2a	2970	52	96,5	0,91	92	92	91,2	8,3	2,8	2,7	0,9	3	0,13	160
37	AGM2E 200 L 2b	2970	65	119	0,89	92,5	92,5	91,7	8,3	2,8	2,7	0,9	3	0,15	190

Table 15: GAMAK Motor Catalogue

According to the Table 11, Gamak AGM2E 90 L 2b series motor is sufficient for spindle. The rated torque of this motor is 7.3 Nm but we need to 10.43Nm. The difference is that the V-belt pulley power transmission system will be used when connecting the engine to spindle.

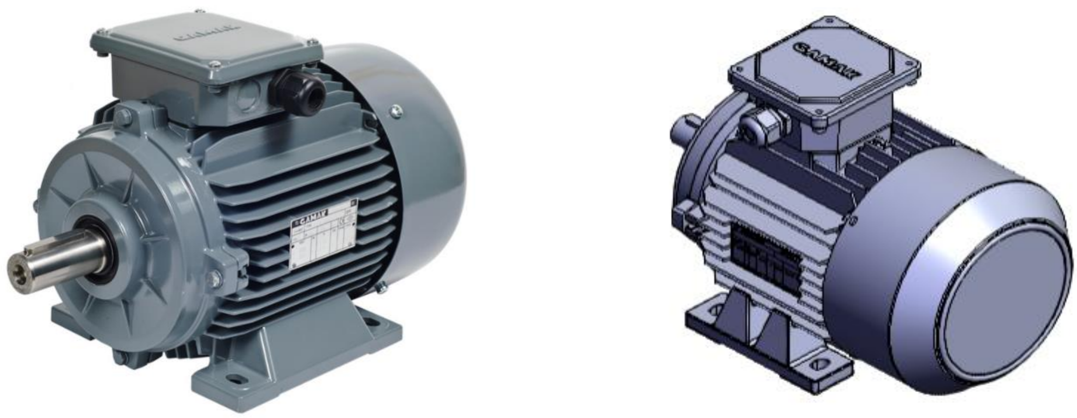


Figure 5.7.1: GAMAK AGM2E 90 L 2b Motor and Their Solid Drawing Version [31]

5.8. Bolt Calculations [32]

$$M_T = F_{pre} \times \left(\frac{d_2}{2} \times tan\ (\varphi + p')\right) \pm\ \mu \times \frac{D_k}{2}$$

$$\varphi = \frac{P}{\pi \times d_2}$$

$$p' = \frac{\mu}{cos \times \frac{\alpha}{2}}$$

$$friction\ of\ coefficient\ (\mu) = 0.20$$

$$\frac{\alpha}{2} = 29°$$

$$M_T = 10.3\ Nm$$

(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
M 7	1	7.000	6.350	5.773	5.918	0.613	28.9
M 8	1.25	8.000	7.188	6.466	6.647	0.767	36.6
M 10	1.5	10.000	9.026	8.160	8.876	0.920	58.3
M 12	1.75	12.000	10.863	9.858	10.106	1.074	84.0
M 14	2	14.000	12.701	11.546	11.835	1.227	115
M 16	2	16.000	14.701	13.546	13.835	1.227	157
M 18	2.5	18.000	16.376	14.933	15.294	1.534	192
M 20	2.5	20.000	18.376	16.933	17.294	1.534	245
M 22	2.5	22.000	20.376	18.933	19.294	1.534	303
M 24	3	24.000	22.051	20.320	20.752	1.840	353
M 27	3	27.000	25.051	23.320	23.752	1.840	459
M 30	3.5	30.000	27.727	25.706	26.211	2.147	561
M 33	3.5	33.000	30.727	28.706	29.211	2.147	694
M 36	4	36.000	33.402	31.093	31.670	2.454	817
M 39	4	39.000	36.402	34.093	34.670	2.454	976
M 42	4.5	42.000	39.077	36.416	37.129	2.760	1104
M 45	4.5	45.000	42.077	39.416	40.129	2.760	1300
M 48	5	48.000	44.752	41.795	42.587	3.067	1465
M 52	5	52.000	48.752	45.795	46.587	3.067	1755
M 56	5.5	56.000	52.428	49.177	50.046	3.067	2022
M 60	5.5	60.000	56.428	53.177	54.046	3.374	2360

Table 16. Metric Standards [33]

$$d_1 = 10\ mm$$

$$d_2 = 9.026\ mm$$

$$d_3 = 8.16\ mm$$

$$pitch = 1.5\ mm$$

$$D_k = 1.4 \times 10\ mm = 14mm$$

$$\varphi = 0.052$$

$$p' = 0.23$$

$$F_{pre} = 3840\ N$$

There are 4 bolts on system so,

$$F_{pre}\ for\ 1\ bolt = 960\ N$$

$$F_{pre}\ for\ 1\ bolt\ with\ \left(8.5\ safety^{[34]}\right) = 8160\ N$$

$$\tau_{max} = \frac{F_{pre} \times \frac{d_2}{2} \times tan\ (\varphi\ + p')}{J} \times C$$

$$J = \frac{\pi \times {d_3}^4}{32}$$

$$\tau\ (\frac{N}{mm^2}) = \frac{8160\ N \times 4.51mm \times (0.052 + 0.23)}{0.09818 \times (8mm)^4} \times 4mm$$

$$\tau = 115{,}26\ Mpa$$

$$\sigma = \frac{F_{pre}}{A}$$

$$\sigma = \frac{8160\ N}{0.785 \times (8mm)^2}$$

$$\sigma = 162.42\ MPa$$

$$\sigma(yield) = \sqrt{\sigma^2 + 3 * \tau^2}$$

$$\sigma\ (yield) = 257.3\ MPa$$

4.8 bolt grade was estimated due to calculated yield strength value. So,

- $\sigma(yield)$ =4*8*10= $320\ MPa$
- $\sigma(uts) = 4 * 100 = 400\ MPa$

$257.3\ MPa < 320\ MPa$ it is appropriate

Figure 5.8.1: 4.8 grade bolt.

5.9. Design of V-Belt and Pulley

The transmission ratio can be calculated as follows:

$$i = \frac{n_{sp}}{n_{lp}} = \frac{3000rpm}{2000rpm} = 1.5$$

where, n_{sp} : small pulley's speed (rpm) , n_{lp} : large pulley's speed (rpm)

According to the literature research about the V-belt, they are generally used in belt-pulley power transmission mechanisms are as follows:

Z section, A section, B section, C section, D section, E section, 20 section, 25 section, 45 section, 50 section, AX section, BX section, CX section, SPZ section, SPA section, SPB section, SPC section, 19 section, XPZ section, XPA section, XPB section, XPC section, 3V section, 5V section, 8V section, 3VX section, 5VX section, 8VX section, 13X6 section, 17X5 section, 21X6,5 section, 26X8 section, 28X8 section, 30X10 section, 33X10 section, 36X12 section, 37X10 section, 42X13 section, 47X13 section, 52X16 section, 55X16 section, 65X20 section, 70X20 section, AA section, BB section, CC section, XDV2-38 section, XDV2-48 section, XDV2-58 section.

As seen above, there are many types of v-belts. However, these are only suitable for our systems is that Z section, A section, B section, C section, D section, E section, SPZ section, SPA section, SPB section, SPC section, XPZ section, XPA section, XPB section, XPC section.

The most important criterion taken into consideration when choosing between these is the design conditions. **According to our design conditions, the type of A section V-belt is most useful V-belt for our mini lathe.**

Belt Characteristics [35]

section	Z	A	B	C	D	E	20	25	45	50
a (mm)	10	13	17	22	32	40	20	25	45	50
s (mm)	6	8	11	14	19	25	12,5	16	20	20
pitch length - internal length Δ_i (mm)	25	33	43	62	76	105	48	61	91	85
external length - pitch length Δ_e (mm)	13	17	26	26	43	52	31	39	35	41
weigth (gr/m)	60	100	175	300	610	930	240	400	1200	1365
min. pulley diam. (mm)	60	90	125	200	300	500	160	250	320	320
working temperature	-30°C ÷ +80°C									
relevant standards	RMA/MPTA IP20 - DIN 2215 - ISO 4184									
relevant antistatic standard	ISO 1813									
materials	CR blend - polyester cord - cotton/polyester fabric									

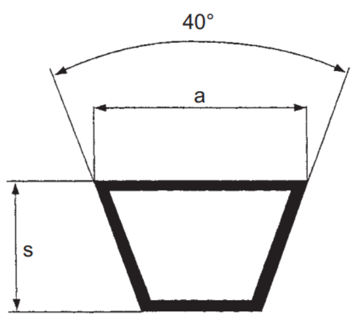


Figure 5.9.1: V-Belts Standards and Characteristics

According to the Figure 5.9.1 data's, the diameter of the small pulley have to be 90 mm.

Considering diameter d_{sp} = 90 mm for the smaller pulley, the pitch diameter of the larger pulley is :

$d_{lp} = i \times d_{sp} = 1.5 \times 90 = 135\ mm$

Peripheral speed of the belts is determined by

$V_B = \frac{\pi \times d_{sp} \times n_1}{60 \times 1000} = \frac{\pi \times 90 \times 3000}{60 \times 1000} = 14.137$ m/s

Belt Pitch Length and Correct Centre Distance

For i = 1.5 (1<i<3) the centre distance is given by;

centre distance, $C \geq \frac{(i+1) \times d_{sp}}{4} + d_{sp}$

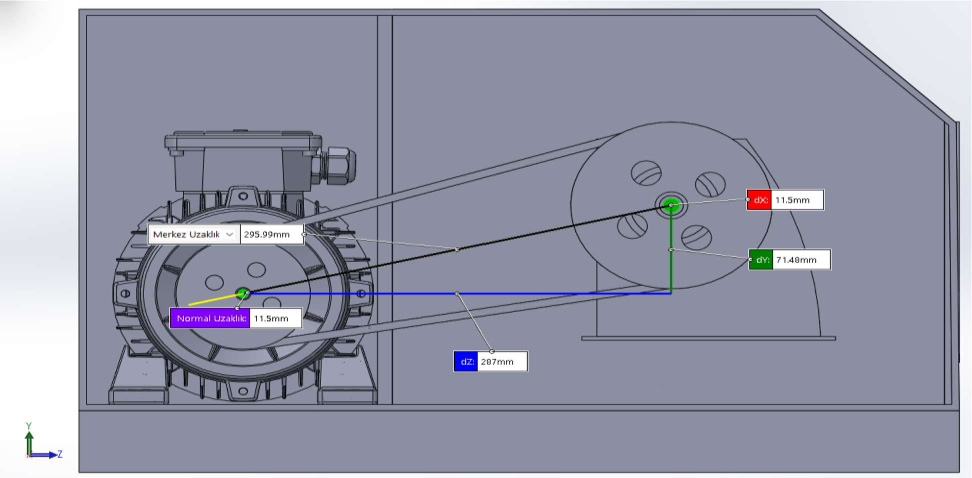


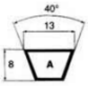
Figure 5.9.2: Centre Distance between Pulley's in SolidWorks

$C = 296\ mm \geq \frac{(1.5+1) \times 90}{4} + 90 = 296\ mm \geq 146.25\ mm$

The pitch length of the belt is determined by:

$$L' = \frac{\pi}{2}(d_{sp} + d_{lp}) + 2C + \frac{1}{4C}(d_{lp} - d_{sp})^2$$

$$L' = \frac{\pi}{2}(90 + 135) + (2 \times 296) + \frac{1}{4 \times 296}(135 - 90)^2 = 947.139\ mm$$



A SECTION

Code	Internal length L_i (mm)	Code	Internal length L_i (mm)	Code	Internal length L_i (mm)	Code	Internal length L_i (mm)	Code	Internal length L_i (mm)	Code	Internal length L_i (mm)	Code	Internal length L_i (mm)	Code	Internal length L_i (mm)	Code	Internal length L_i (mm)
A 15	382	A 28	710	A 36 1/2	925	A 43 3/4	1111	A 53 1/4	1355	A 70	1775	A 86	2187	A 107	2725	A 147	3737
A 18	457	A 28 1/2	724	A 37	942	A 44	1120	A 54	1372	A 71	1800	A 87	2212	A 108	2743	A 148	3750
A 19	480	A 29	737	A 37 1/4	946	A 44 1/2	1132	A 55	1400	A 72	1825	A 88	2240	A 110	2800	A 155	3937
A 20	508	A 29 1/2	750	A 37 1/2	950	A 45	1143	A 56	1422	A 73	1854	A 89	2267	A 112	2845	A 158	4000
A 21	535	A 30	767	A 38	965	A 45 1/2	1150	A 57	1450	A 74	1880	A 90	2286	A 113	2870	A 162	4115
A 22	560	A 30 1/2	775	A 38 1/2	975	A 46	1168	A 58	1475	A 75	1900	A 91	2311	A 114	2896	A 167	4250
A 23	587	A 31	787	A 39	992	A 46 1/2	1180	A 59	1500	A 76	1930	A 92	2337	A 116	2946	A 173	4394
A 23 1/2	600	A 31 1/2	800	A 39 1/2	1000	A 47	1200	A 60	1525	A 77	1956	A 93	2360	A 118	3000	A 180	4572
A 24	610	A 32	813	A 40	1016	A 48	1220	A 61	1550	A 78	1980	A 94	2388	A 120	3048	A 187	4750
A 24 1/2	620	A 32 1/2	825	A 40 1/2	1030	A 48 1/4	1225	A 62	1575	A 79	2000	A 95	2413	A 124	3150	A 197	5000
A 24 3/4	630	A 33	838	A 41	1041	A 49	1250	A 63	1600	A 80	2032	A 96	2438	A 128	3250	A 217	5477
A 25	637	A 33 1/4	847	A 41 1/2	1050	A 50	1270	A 64	1625	A 81	2060	A 97	2464	A 130	3302		
A 25 1/2	647	A 33 1/2	850	A 41 3/4	1060	A 51	1300	A 65	1650	A 82	2083	A 98	2500	A 132	3350		
A 26	660	A 34	867	A 42	1067	A 51 1/2	1307	A 66	1676	A 83	2100	A 100	2540	A 134	3404		
A 26 1/2	670	A 34 1/2	875	A 42 1/2	1075	A 52	1320	A 67	1700	A 83 1/2	2120	A 102	2591	A 136	3454		
A 27	686	A 35	900	A 43	1100	A 52 1/2	1337	A 68	1725	A 84	2134	A 104	2650	A 140	3550		
A 27 1/2	700	A 36	914	A 43 1/2	1105	A 53	1346	A 69	1750	A 85	2160	A 105	2667	A 144	3658		

TABLE 4 - P_b (kW) referred to d (mm)

RPM \ mm	71	80	90	100	112	125	132	150	170	190	200	212
100	0,13	0,16	0,21	0,25	0,30	0,35	0,38	0,46	0,54	0,62	0,66	0,71
200	0,22	0,29	0,37	0,45	0,54	0,65	0,70	0,84	0,99	1,14	1,22	1,30
500	0,42	0,59	0,78	0,96	1,18	1,41	1,53	1,85	2,19	2,53	2,70	2,90
700	0,53	0,76	1,01	1,26	1,55	1,86	2,03	2,45	2,91	3,37	3,59	3,86
900	0,63	0,91	1,22	1,53	1,89	2,28	2,49	3,02	3,59	4,15	4,43	4,75
1000	0,67	0,98	1,32	1,66	2,06	2,49	2,71	3,29	3,91	4,52	4,82	5,18
1400	0,81	1,23	1,69	2,14	2,67	3,23	3,53	4,29	5,10	5,88	6,27	6,72
1500	0,84	1,28	1,77	2,25	2,81	3,41	3,72	4,52	5,37	6,19	6,59	7,06
1700	0,89	1,38	1,93	2,46	3,08	3,74	4,08	4,96	5,89	6,77	7,20	7,70
1800	0,91	1,43	2,00	2,55	3,21	3,89	4,26	5,16	6,13	7,04	7,48	7,99
2500	1,02	1,70	2,44	3,15	3,98	4,84	5,29	6,38	7,50	8,52	8,98	9,50
2900	1,04	1,81	2,63	3,42	4,32	5,25	5,73	6,88	8,02	9,00	9,42	9,87*
3000	1,05	1,83	2,67	3,47	4,40	5,34	5,83	6,98	8,12	9,07	9,48*	9,91*
3500	1,03	1,90	2,82	3,70	4,69	5,69	6,18	7,33	8,38	9,15*		
3600	1,02	1,91	2,85	3,74	4,74	5,74	6,23	7,37	8,38*			
4000	0,97	1,91	2,90	3,84	4,86	5,86	6,34	7,40*				
5000	0,70	1,75	2,82	3,78	4,77	5,62*	5,99*					
6000	0,24	1,33	2,38	3,24*								

P_d (kW) referred to i

RPM \ i	1,00÷1,01	1,02÷1,03	1,04÷1,06	1,07÷1,08	1,09÷1,12	1,13÷1,16	1,17÷1,22	1,23÷1,32	1,33÷1,50	over 1,51
100	0,00	0,00	0,00	0,01	0,01	0,01	0,01	0,01	0,02	0,02
200	0,00	0,00	0,01	0,01	0,02	0,02	0,02	0,03	0,03	0,04
500	0,00	0,01	0,02	0,03	0,04	0,05	0,06	0,07	0,08	0,09
700	0,00	0,01	0,03	0,04	0,06	0,07	0,08	0,10	0,11	0,13
900	0,00	0,02	0,04	0,05	0,07	0,09	0,11	0,13	0,14	0,16
1000	0,00	0,02	0,04	0,06	0,08	0,10	0,12	0,14	0,16	0,18
1400	0,00	0,03	0,06	0,08	0,11	0,14	0,17	0,20	0,22	0,25
1500	0,00	0,03	0,06	0,09	0,12	0,15	0,18	0,21	0,24	0,27
1700	0,00	0,03	0,07	0,10	0,14	0,17	0,20	0,24	0,27	0,30
1800	0,00	0,04	0,07	0,11	0,14	0,18	0,22	0,25	0,29	0,32
2500	0,00	0,05	0,10	0,15	0,20	0,25	0,30	0,35	0,40	0,45
2900	0,00	0,06	0,12	0,17	0,23	0,29	0,35	0,41	0,46	0,52
3000	0,00	0,06	0,12	0,18	0,24	0,30	0,36	0,42	0,48	0,54
3500	0,00	0,07	0,14	0,21	0,28	0,35	0,42	0,49	0,56	0,63
3600	0,00	0,07	0,14	0,22	0,29	0,36	0,43	0,50	0,57	0,64
4000	0,00	0,08	0,16	0,24	0,32	0,40	0,48	0,56	0,64	0,72
5000	0,00	0,10	0,20	0,30	0,40	0,50	0,60	0,70	0,80	0,89
6000	0,00	0,12	0,24	0,36	0,48	0,60	0,72	0,84	0,96	1,07

Table 17. A section V- Belt Properties

A37 1/4 V – belt selected.

Belt Tensioning [36]

From Table 13 of A section (d = 90 mm, 3000 rpm, i = 1.5)

P_b = 2.67 kW

P_d = 0.48 kW

The arc oc contact ϒ of the belt on smaller pulley is determined by:

$$ϒ = 180^\circ - 57\times \frac{d_{lp} - d_{sp}}{C} = 180^\circ - 57\times \frac{135-90}{296} \cong 174.3^\circ$$

From **Table 13** for ϒ = 171.33°

$C_ϒ \cong 0.9748$

Arc corretcion factor:

β	180	174	169	163	157	151	145	139	133	127	120	113	106	99	91	83
$C_β$	1,00	0,98	0,97	0,96	0,94	0,93	0,91	0,89	0,87	0,85	0,82	0,80	0,77	0,73	0,70	0,65

Table 18: Arc Correction Factor

From **Table 18** for A37 1/4 belt

C_L = 0.87 (947.139 mm equals 37.28 inch)

inches	9½	16	22	24	28	32	35	48	53	75	81	90	128	144	180	210	285	330	420	540	720	780
Z	0,69	0,77	0,82	0,84	0,87	0,89	0,91	0,98	1,00													
A		0,73	0,79	0,80	0,83	0,85	0,87	0,93	0,95	1,03	1,05	1,07	1,16	1,19	1,25	1,29						
B			0,73	0,75	0,77	0,80	0,81	0,87	0,89	0,96	0,98	1,00	1,08	1,11	1,16	1,20	1,29	1,33	1,40			
C						0,72	0,73	0,79	0,80	0,87	0,88	0,90	0,97	1,00	1,05	1,09	1,16	1,20	1,27			
D												0,80	0,87	0,89	0,94	0,97	1,04	1,07	1,13	1,20	1,27	
E															0,90	0,94	1,00	1,03	1,09	1,15	1,23	1,25
20										0,91	0,93	0,95	1,02	1,05	1,10	1,14	1,22					
25										0,82	0,83	0,85	0,92	0,95	1,00	1,03	1,10	1,13				

Table 19: Correction Factor C_L

The correction coefficient is 1.1

(Table of the right)

The corrected power is

$P_c = 2.2 \times 1.1 = 2.42$ kW

$P_a = (P_b + P_d) \times C_y \times C_L$

$P_a = (2.67 + 0.48\ ) \times 0.9748 \times 0.87$

$P_a = 2.67$ kW

Number of the belts Q necessary for transmission

of the power Pc is established by:

$Q = \frac{P_c}{P_a} = \frac{2.42}{2.67} = 0.9063$

Therefore, 1 piece of A37 ¼ V belt is sufficient for the system.

Drivers (1) AC electric motors: high slip, squirrel cage, synchronous: DC electric motors: parallel excitation; multi-cylinder internal combustion engines; gas or steam turbines.	(2) AC electric motors: high torque, high slip, single phase, wound rotor, commutator; DC electric motors: series and compound excitation; single-cylinder internal combustion engines with direct coupling or with countershaft; steam engines.					
Applications	Daily operating hours					
	0-8(1)	8-16(1)	16-24(1)	0-8(2)	8-16(2)	16-24(2)
Light use Centrifugal pumps and compressors, belt conveyors, (light materials) fans and pumps up to 7,5 kW.	1,1	1,1	1,2	1,1	1,2	1,3
Normal use Shears for steel sheet presses, belt and chain conveyors, (heavy material) sifters, generator sets, machine tools, kneading machines, industrial washing machines, printing presses, fans and pumps over 7,5 kW.	1,1	1,2	1,3	1,2	1,3	1,4
Heavy use Hammer mills, piston compressors, belt conveyors for heavy loads, lifters, textile machines, continuous paper machines, piston and dredging pumps, ripping saws.	1,2	1,3	1,4	1,4	1,5	1,6
Extra heavy use High power mills, stone crushers, calendars, mixer, cranes, diggers, dredgers.	1,3	1,4	1,5	1,5	1,6	1,8

5.10. Design of Motor Shaft [38]

Mass of the Pulley;

According to the SolidWorks, 90 mm pulley's volume is 118886.42 mm^3

The pulley material is Aluminum so its density $\rho = 2700$ kg/m^3

90 mm Pulley mass is 118886.42×10^{-9} m^3 $\times$ 2700 kg/m^3 = 0.3209 kg

According to above parameters,

Weight of the 90 mm pulley is $0.3209 \times 9.81 = 3.1489$ N

Determining Tight Side and Slack Side of the V-Belt [37]

Motor Torque is 7.3 Nm

$\frac{T_1}{T_2} = e^{\mu\vartheta}$, $\mu = 0.25$, $\theta = 180 - \frac{180D}{\pi d} = 94.05^{\circ}$

$T_1 = 1.5T_2$

$Torque = (T_1 - T_2) \times R\ ,\quad 7.3\ Nm = (\ 1.5T_2 - T_2) \times 0.045$

$$T_2 = 324.44\ N\ so,$$

$$T_1 = 486.66\ N$$

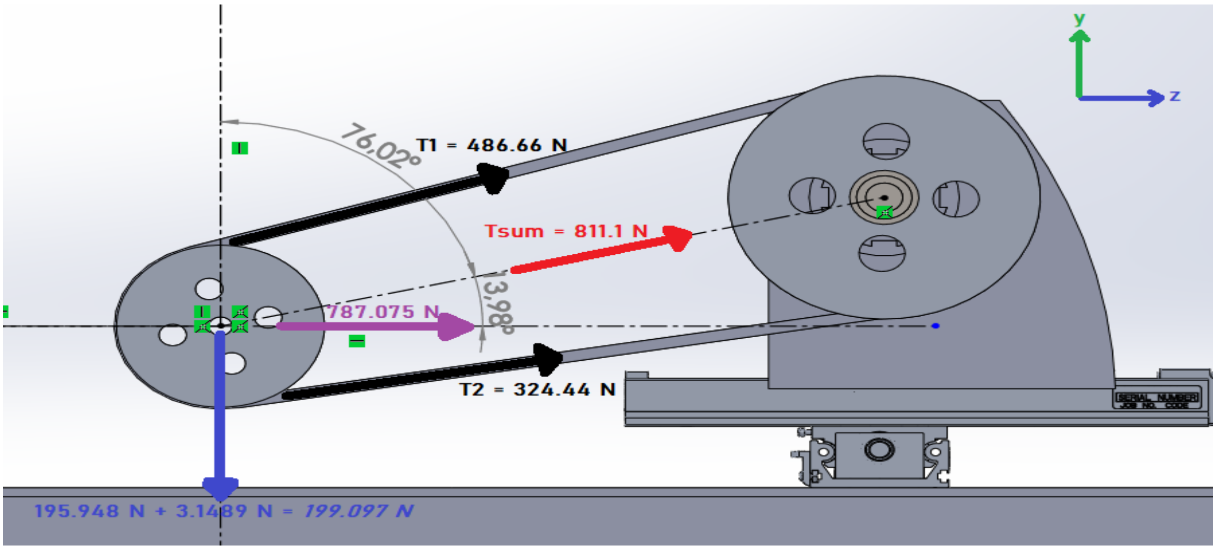


Figure 5.10.1: Determination Radial Forces

$T_{z\text{-axis}} = T_{sum} \times \cos 13.98° = 787.075$ N , $T_{y\text{-axis}} = T_{sum} \times \cos(76.02° + 13.98°) = -195.948$ N

According to the Figure x.x, there are two forces in y-axis. These are $T_{y\text{-axis}}$ and weight of the pulley.

►Note: 195.948 N is total tension of the y-axis, 3.1489N is weight of the pulley. Therefore, because these forces are in the same direction, both are combined.

X-Y axis Bending and Shear Diagrams;

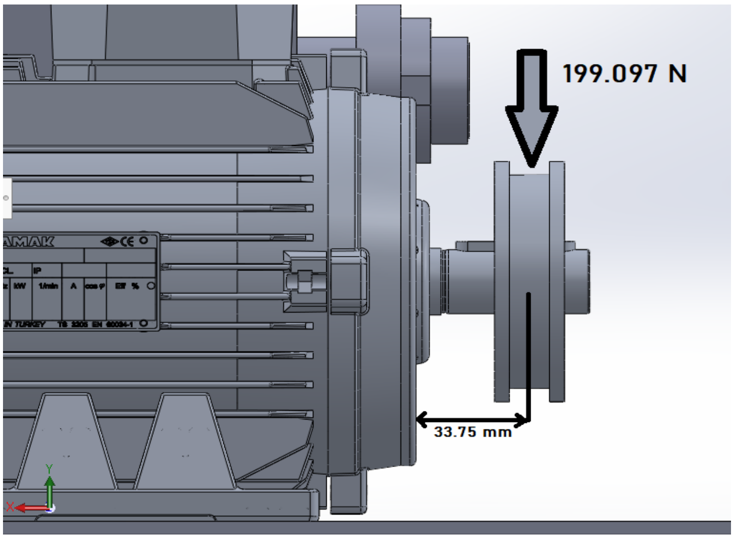


Figure 5.10.2: Side View of Motor, Motor Shaft and Pulley Configuration

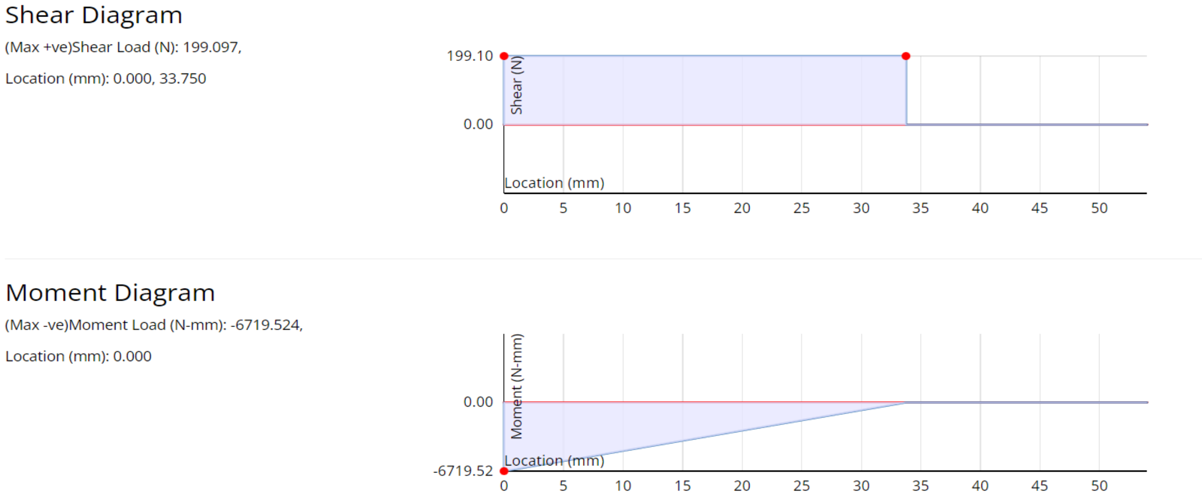


Figure 5.10.3: Shear and Moment Diagrams

$$\Sigma M_{MOTOR} = RM - 199.097 \times 33.75,$$

$$RM_{XY} = 6719.523 \text{ Nmm}$$

X-Z axis Bending and Shear Diagrams:

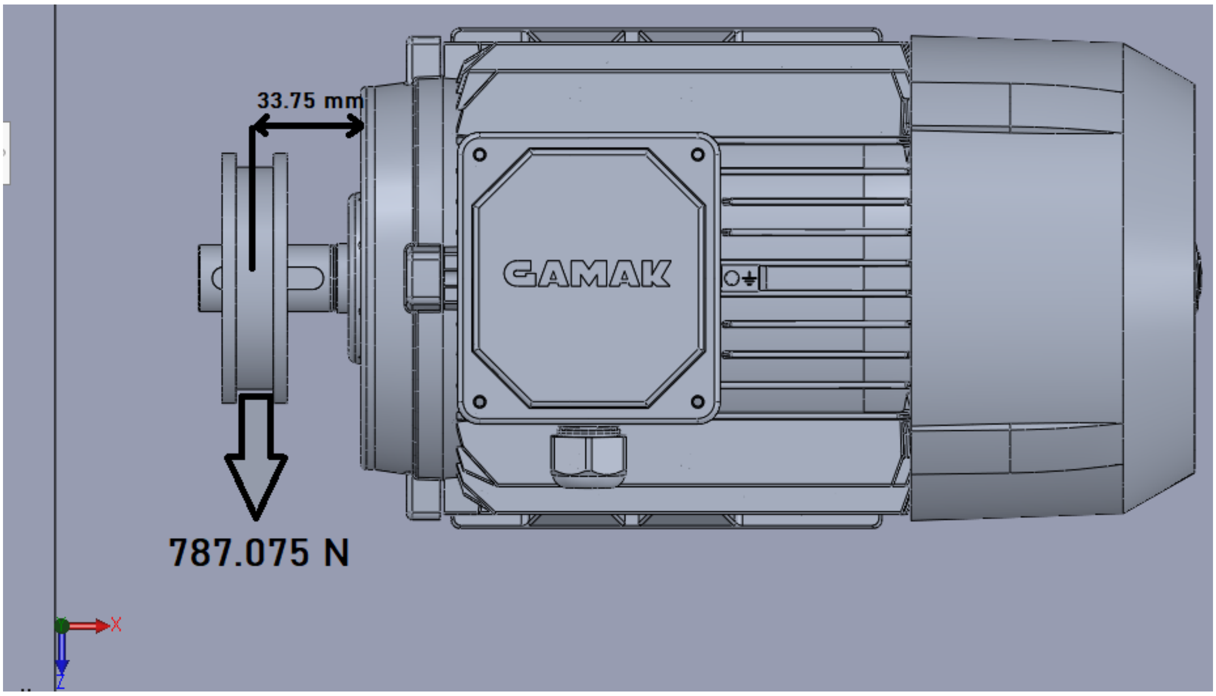
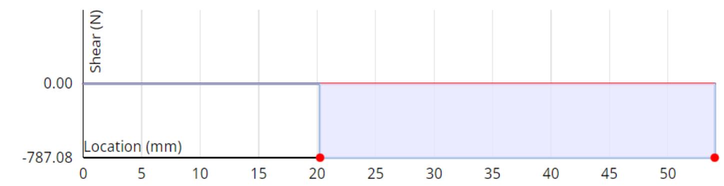


Figure 5.10.4: Top View of Motor and Motor Shaft/Pulley Configuration

Shear Diagram

(Max -ve)Shear Load (N): -787.075,

Location (mm): 20.250, 54.000



Moment Diagram

(Max -ve)Moment Load (N-mm): -26563.781,

Location (mm): 54.000

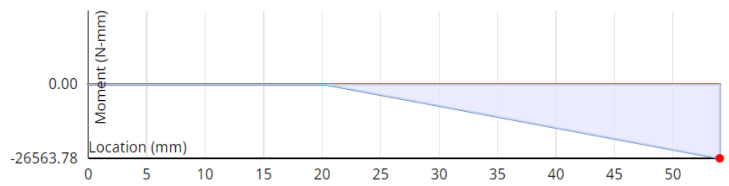


Table 20: Shear and Moment Diagrams

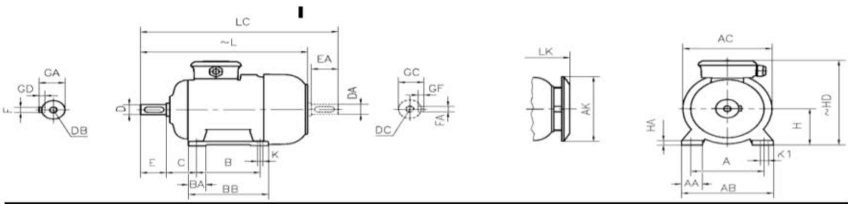
$$\Sigma M_{MOTOR} = RM_{XY} - 787.075 \times 33.75,$$

$$RM_{XZ} = 26563.781 \text{ Nmm}$$

Equivalent Moment Formula is $\sqrt{RM_{XZ}^2 + RM_{XY}^2}$

Point / Directions	RM_{XZ}	RM_{XY}	Equivalent Moment
Motor	26563.781	6719.523	27374.3
Pulley	0	0	0

Table 21: Equivalent Moment



BOYUTLAR / DIMENSIONS

Ayaklı motor boyutları : B3, B6, B7, B8, B15, V5, V6 Kurulma Düzenlerinde / Dimensions of foot mounted motors for mounting arrangement : B3, B6, B7, B8, B15, V5, V6																							
H	HD ~	HA	A	AA	AB	ØAC	ØAK	K	K1	B	B'	BA	BA'	BB	L ~	LC	LK ~	C	E EA	DB DC	ØD ØDA	GA GC	FxGD FAXGF
90	222	12	140	40	180	176	151	10	15	100	125	35	60	155	361	416	390	56	50	M8	24	27	8X7

*Verim değerleri IEC 60034-2-1 : 2014 standardına uygun olarak endirekt ölçüm metodu ile hesaplanmıştır. Ek kayıplar, değişken yük değerlerinde yapılmış olan test sonuçlarına göre belirlenir. /
Efficiencies are calculated according to indirect method where the additional load losses are determined from exact measurements at different load points.

Table 22. AC Motor's Dimensions

According to the Table 18, Motor Shaft diameter must be <u>maximum</u> 24 mm.

5.11 Shear Stress on Motor Shaft

$$\tau_{shear} = \frac{F}{A}$$

Where,

F: Force Acting on Pulley,

A: Cross Section Area of the Shaft,

$$F_{Res} = \sqrt{787.075^2 + 199.097^2} = 811.866\ N$$

$$\tau_{shear} = \frac{F}{\frac{\pi d^2}{4}} = \frac{811.866 \times 4}{\pi \times 24^2} = 1.7946\ MPa$$

5.12 Torsional Stress on Motor Shaft

$$\tau_{torsion} = \frac{T \times \frac{d}{2}}{J}$$

Where,

T: Torque Acting on Motor Shaft,

d: Motor Shaft Diameter,

J: Polar Moment of Inertia

$$\tau_{torsion} = \frac{T \times r}{J} = \frac{7300 \times 12}{\frac{\pi}{32} d^4} = 2.6894\ MPa$$

5.13 Bending Stress on Motor Shaft

$$\sigma_{bending} = \frac{32M}{\pi d^3}$$

Where,

M: Bending Moment,

d: Motor Shaft Diameter,

$$\sigma_{bending} = \frac{32M}{\pi d^3} = \frac{32 \times 27374.3}{\pi \times d^3} = 20.17\ MPa$$

5.14 Equivalent Stress on Motor Shaft

Motor shaft be exposed to bending, shear and torsion in this section. Therefore, Von- Misses is used to find the equivalent of these stresses.

$$\sigma_{equivalent} = \sqrt{\left(\sigma_{bending}\right)^2 + 3 \times (\tau)^2}$$

$$\sigma_{equivalent} = \sqrt{(20.17)^2 + 3 \times (2.6894 + 1.7946)^2}$$

$$\sigma_{equivalent} = 21.613\ MPa$$

$$\frac{\sigma_{yield}}{\sigma_{equivalent}} = FoS$$

$$\frac{\sigma_{yield}}{21.613} = 2.5$$

$$\sigma_{yield} = 54.034\ MPa \cong 55\ MPa$$

5.15 Fatigue Calculation for Motor Shaft [39]

Surface Factor (k_a):

$$k_a = a \times S_{ut}^b$$

Where,

S_{ut}: Ultimate Tensile Stress of the Shaft Material

According to the Figure 5.6.7, a and b are founded

Surface Finish	Factor a S_{ut}, kpsi	Factor a S_{ut}, MPa	Exponent b
Ground	1.34	1.58	−0.085
Machined or cold-drawn	2.70	4.51	−0.265
Hot-rolled	14.4	57.7	−0.718
As-forged	39.9	272.	−0.995

Figure 5.15.1: Factor "a" According to Different Surface Finishes

$$k_a = 4.51 \times 620^{-0.265} = 0.82$$

5.15.1 Size Factor (kb):

$$k_b = \begin{cases} (d/0.3)^{-0.107} = 0.879d^{-0.107} & 0.11 \le d \le 2 \text{ in} \\ 0.91d^{-0.157} & 2 < d \le 10 \text{ in} \\ (d/7.62)^{-0.107} = 1.24d^{-0.107} & 2.79 \le d \le 51 \text{ mm} \\ 1.51d^{-0.157} & 51 < d \le 254 \text{ mm} \end{cases}$$

Figure 5.15.1.1: Assumptions of Size Factor

d: Diameter of the shaft,

$$\left(\frac{24}{7.62}\right)^{-0.107} = 0.8844$$

5.15.2 Loading Factor (k_c):

$$k_c = \begin{cases} 1 & \text{bending} \\ 0.85 & \text{axial} \\ 0.59 & \text{torsion}^{17} \end{cases}$$

$k_c = 0.59$

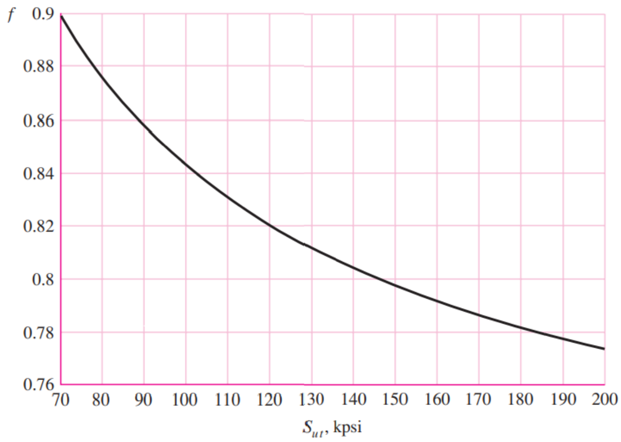
5.15.3 Temperature Factor (k_d):

The mini lathe designed according to room temperature. Therefore, the temperature factor will be 1.

Temperature, °C	S_T/S_{RT}	Temperature, °F	S_T/S_{RT}
20	1.000	70	1.000
50	1.010	100	1.008
100	1.020	200	1.020
150	1.025	300	1.024
200	1.020	400	1.018

$$k_d = 1$$

5.15.4 Fatigue Strength Fraction (k_f):



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5.15.5 Endurance Limit (S_e):

Endurance Limit for Cold Drawn 1045 Carbon Steel is 370 MPa. The new endurance limit calculated is below;

$$S_e' = \frac{S_e \times k_a \times k_b \times k_c}{k_d}$$

$$S_e' = \frac{370 \times 0.82 \times 0.8844 \times 0.59}{1} = 158.313\ MPa$$

The maximum and minimum stress on the spindle shaft was calculated.

$$\sigma_{max} = 21.613\ MPa$$

$$\sigma_{min} = 0\ MPa$$

Mean stress and amplitude stress will be calculated as,

$$\sigma_{mean} = \frac{\sigma_{max} + \sigma_{min}}{2} = 10.8065\ MPa$$

$$\sigma_{amplitude} = \frac{\sigma_{max} - \sigma_{min}}{2} = 10.8065\ MPa$$

Fatigue factor of safety for Soderberg Line calculated below,

$$n_{fs} = \frac{1}{\frac{\sigma_{amplitude}}{S_e{}'} + \frac{\sigma_{mean}}{S_y}} = 11.508$$

5.16. Design of Spindle Shaft [38]

5.16.1 Mass of the Pulley

According to the SolidWorks, 90 mm pulley's volume is 257356.13mm^3

The pulley material is Aluminum so its density ρ = 2700 kg/m^3

90 mm Pulley mass is 257356.13× 10^{-9} m^3 × 2700 kg/m3 = 0.6948 kg

According to above parameters,

Weight of the 90 mm pulley is 0.6948 × 9.81 = 6.8159 N

5.16.2 Mass of Chuck;

Chuck is made of Carbon Alloy, so its density is =7.850 kg/m^3

Size of Chuck: 63mm × 63mm × 42mm=166698mm^3=16698 × 10^{-9} m^3

Mass of Chuck: 16698*10^{-9}m^3 × 7850kg/m^3=1.3085kg

Weight of the chuck is 1.3085 × 9.81=12.8371 N

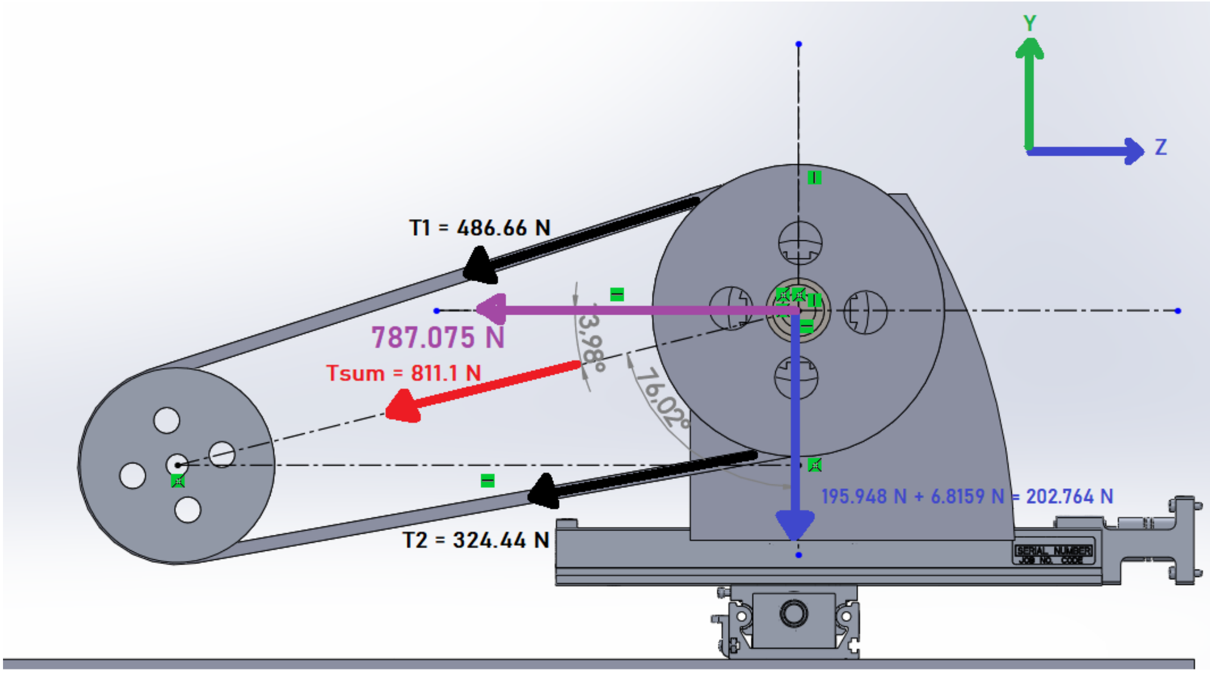


Figure 5.16.2.1. Tension Forces in Pulley

$T_{z\text{-axis}} = T_{sum} \times \cos 13.98° = 787.075$ N , $T_{y\text{-axis}} = T_{sum} \times \cos 76.02° = 195.948$ N

According to the Figure 5.16.2.1 there are two forces in y-axis. These are $T_{y\text{-axis}}$ and weight of the pulley.

►Note: 195.948 N is total tension of the y-axis, 6.8159 N is weight of the pulley. Therefore, because these forces are in the same direction, both are combined.

5.16.2.1 Y-Z axis Bending and Shear Diagrams;

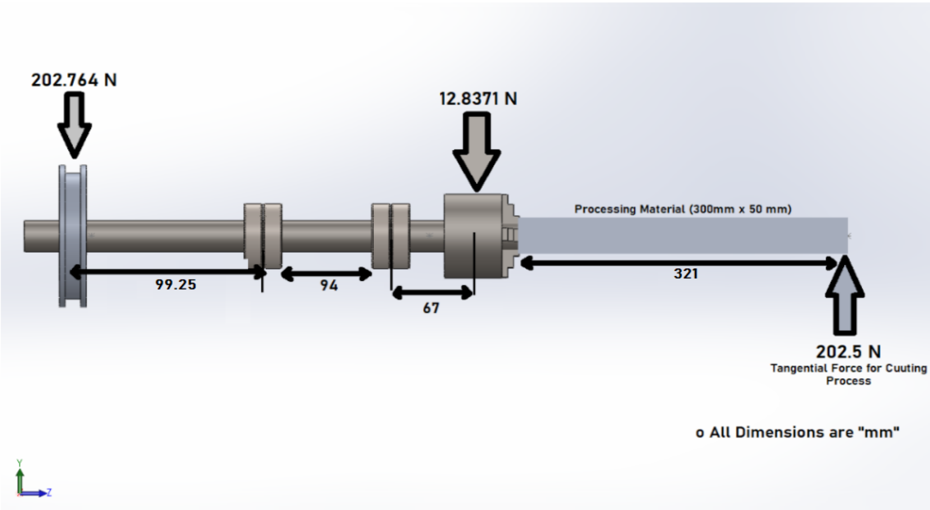


Figure 5.16.2.1.1: Y-Z axis Acting Forces

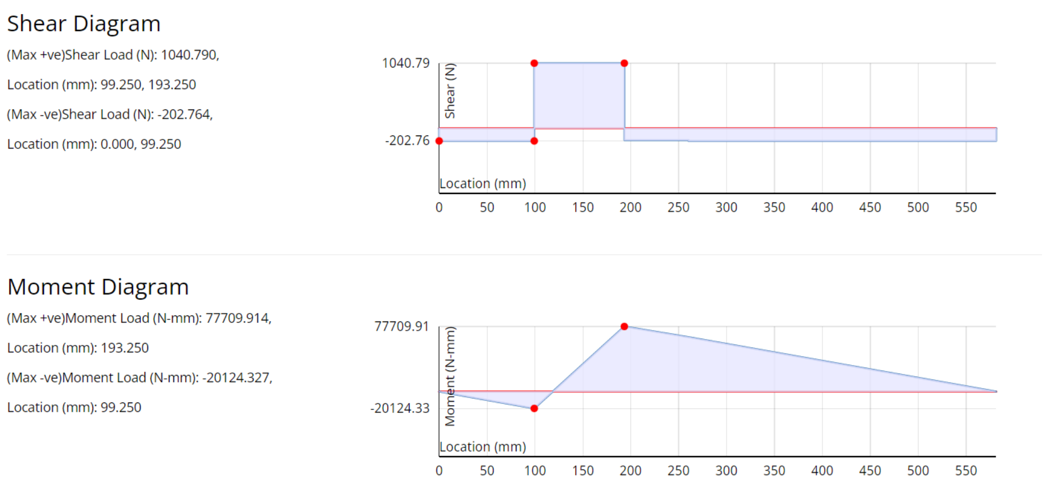


Figure 5.16.2.1.2: Y-Z axis Bending and Shear Diagrams

If taking a moment point A bearings, the force acting at point B is found.

$\Sigma M_{PointA} = 0;$

$(-202.764 \times 99.25) \mp (B\times94) + (12.8371 \times (67+94)) + (-202.5 \times (321+67+94)) = 0$

$B \cong \mp1230.45$ N

If taking a moment point B bearing, the force acting at point A is found.

$\Sigma M_{PointB} = 0;$

$(-202.764 \times (99.25+94)) \mp (A\times94) + (12.8371 \times 67) + (-202.5 \times (321 + 67)) = 0$

$A \cong \mp1243.55$ N

When we check point A and B direction:

$-202.764 - 1230.45 - 12.8371 + 1243.55 + 202.5 = 0$

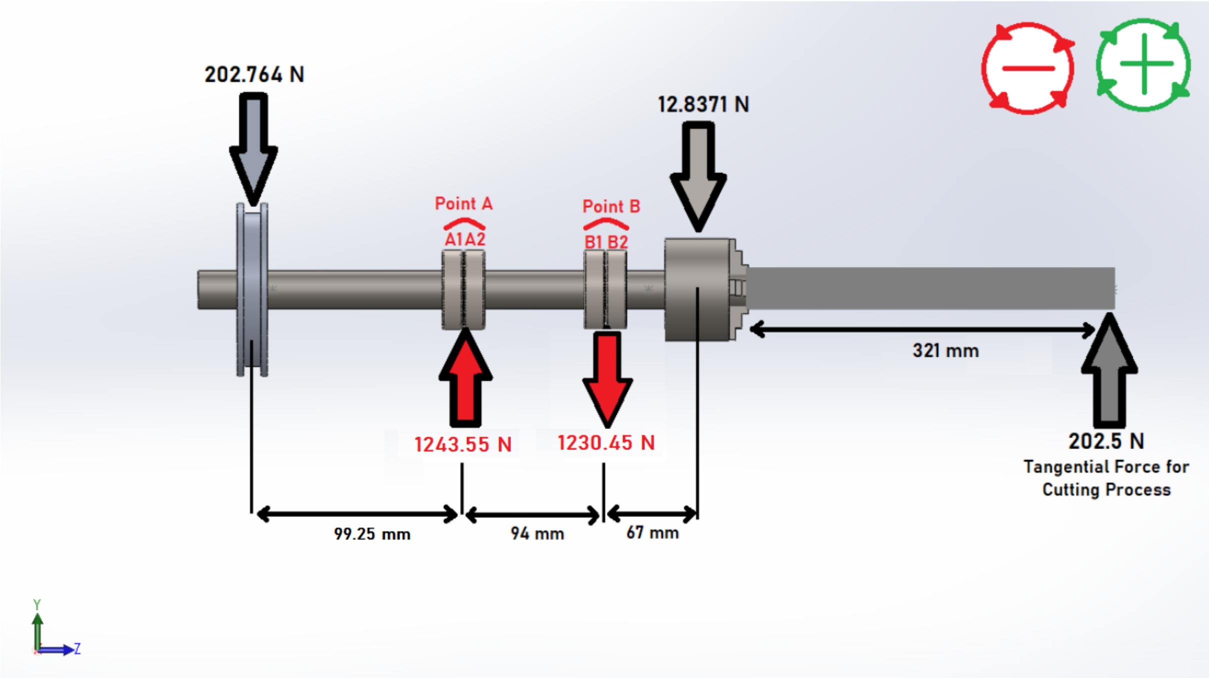


Figure 5.16.2.1.3: Y-Z axis Diagram with Reaction Forces on Bearings

5.16.2.2 X-Z axis Bending and Shear Diagrams:

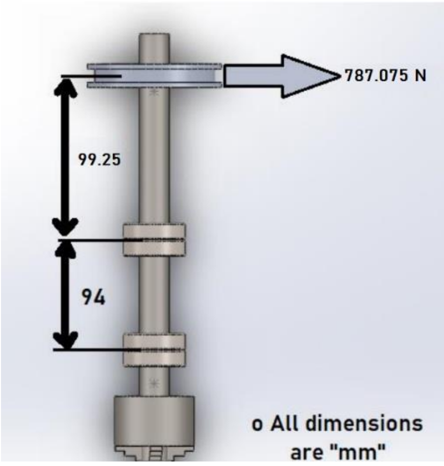


Figure 5.16.2.2.1: X-Z axis Acting Forces

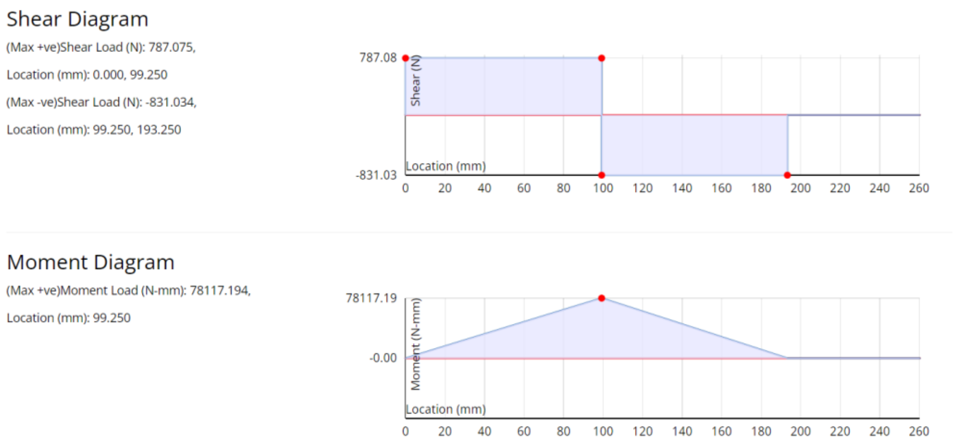
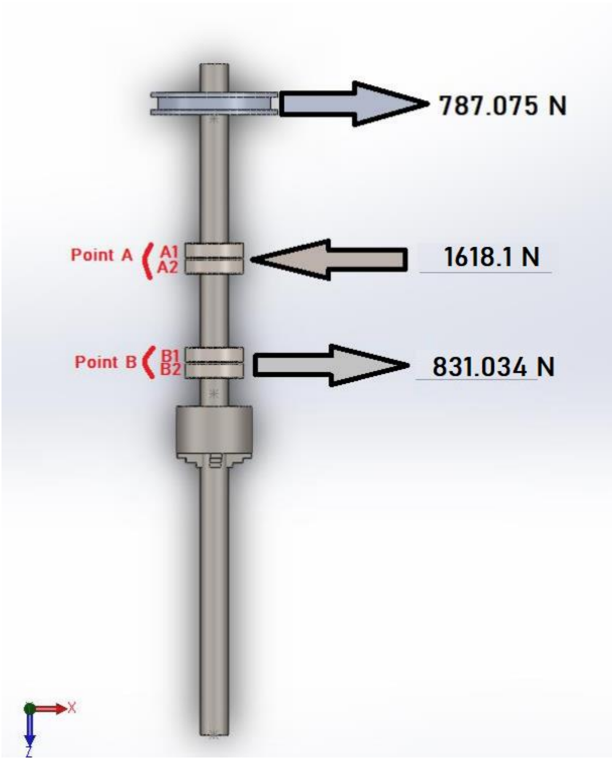


Figure 5.16.2.2.2: X-Z axis Bending and Shear Diagrams

If taking a moment point A bearings,

the force acting at point B is found.



$\Sigma M_{PointA} = 0;$

$(787.075 \times 99.25) \mp (B \times 94) = 0$

$B \cong \mp 831.034$ N

$B \cong +831.034$ N

If taking a moment point B bearings,

the force acting at point A is found.

$\Sigma M_{PointB} = 0;$

$(787.075 \times (99.25+94)) \mp (A \times 94) = 0$

$A \cong \mp 1618.1$ N

$A \cong -1618.1$ N

When we check point A and B direction;

$787.075 - 1618.1 + 831.034 = 0$

5.17 Shear Stress on Pulley

$$\tau_{shear} = \frac{F}{A}$$

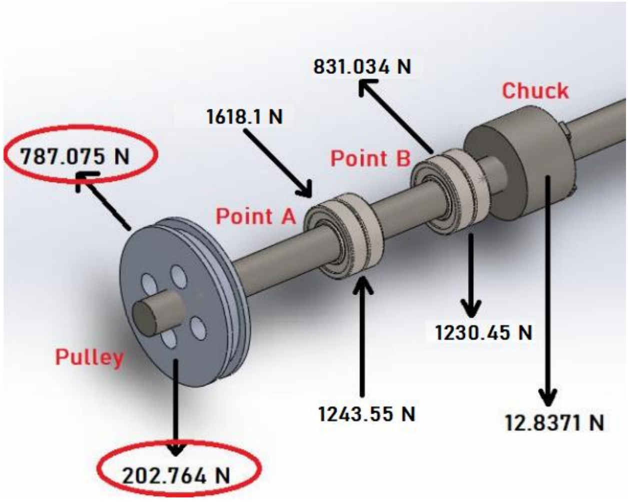
Where,

F: Force Acting on Pulley,

A: Cross Section Area of the Shaft,

$F_{Res} = \sqrt{787.075^2 + 202.764^2} = 812.77$ N

$$\tau_{shear\_pulley} = \frac{F}{\frac{\pi d^2}{4}} = \frac{812.77 \times 4}{\pi \times 30^2} = 1.1498\ MPa$$



5.17.1 Shear Stress on Point A

$$\tau_{shear} = \frac{F}{A}$$

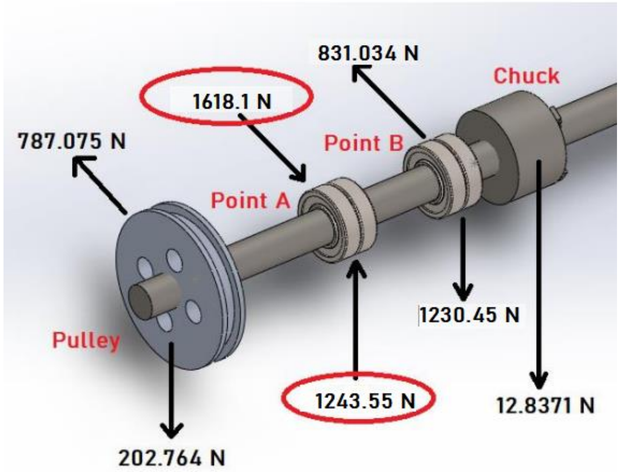
Where,

F: Force Acting on Point A,

A: Cross Section Area of the Shaft,

$F_{Res} = \sqrt{1243.55^2 + 1618.1^2} = 2040.75$ N

$$\tau_{shear\_pointA} = \frac{F}{\frac{\pi d^2}{4}} = \frac{2040.75 \times 4}{\pi \times 30^2} = 2.887\ MPa$$



5.17.2Shear Stress on Point B

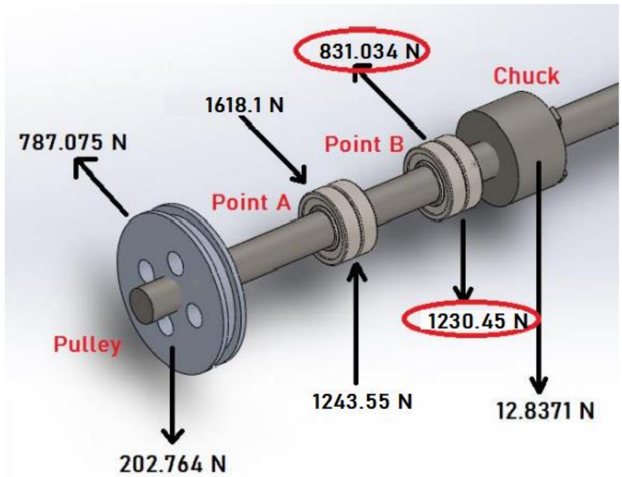
$$\tau_{shear} = \frac{F}{A}$$

Where,

F: Force Acting on Point B,

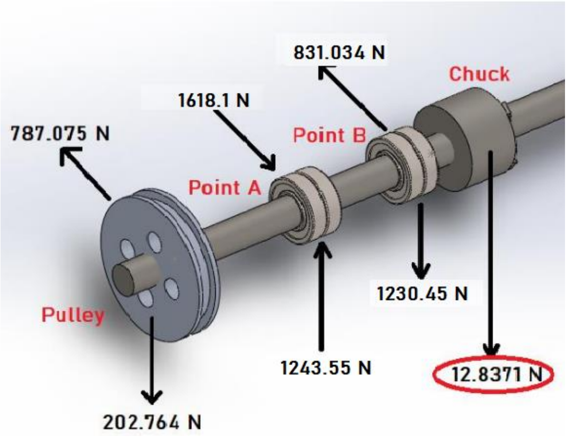
A: Cross Section Area of the Shaft,

$F_{Res} = \sqrt{1230.45^2 + 831.034^2} = 1484.797$ N



$$\tau_{shear\_pointB} = \frac{F}{\frac{\pi d^2}{4}} = \frac{1484.797 \times 4}{\pi \times 30^2} = 2.1\ MPa$$

5.17.2.1 Shear Stress on Chuck



$$\tau_{shear} = \frac{F}{A}$$

Where,

F: Force Acting on Chuck,

A: Cross Section Area of the Shaft,

$$\tau_{shear\_chuck} = \frac{F}{\frac{\pi d^2}{4}} = \frac{12.8371 \times 4}{\pi \times 30^2} = 0.0181\ MPa$$

When investigating to all parts, clearly seen that Point A have the most shear stress so it is critical part of the system.

$\tau_{shear\_pulley} = 1.1498\ MPa$, $\tau_{shear\_pointA} = 2.887\ MPa$,

$\tau_{shear_{pointB}} = 2.1\ MPa$, $\tau_{shear_{chuck}} = 0.0181\ MPa$

5.18 Moment Calculations on Point A and Point B

<u>For Point A on Y-Z axis</u>

$$-202.764 \times 99.25 = A_{YZ}$$

$$A_{YZ} = -20124.327\ Nmm$$

<u>For Point A on X-Z axis</u>

$$-787.075 \times 99.25 = A_{XZ}$$

$$A_{XZ} = -78117.19\ Nmm$$

<u>For Point B on Y-Z axis</u>

$$-(12.8371 \times 67) + \left(202.5 \times (321 + 67)\right) = B_{YZ}$$

$$B_{YZ} = 77709.91\ Nm$$

5.18.1 Bending Stress on Point A

$$\sigma_{bendingPointA} = \frac{32M_A}{\pi d^3}$$

Where,

M_A: Equivalent Bending Moment on Point A,

d: Motor Shaft Diameter,

Equivalent Moment on Point A is below,

$\sqrt{A_{YZ}^2 + A_{XZ}^2} = \sqrt{(-20124.327)^2 + (-78117.19)^2} = 80667.737\ Nmm$

$$\sigma_{bendingPointA} = \frac{32M}{\pi d^3} = \frac{32 \times 80667.737}{\pi \times d^3} = 30.432\ MPa$$

5.18.2 Bending Stress on Point B

$$\sigma_{bendingPointA} = \frac{32M_B}{\pi d^3}$$

Where,

M_B: Equivalent Bending Moment on Point B,

d: Motor Shaft Diameter,

Equivalent Moment on Point B is below,

$$\sigma_{bendingPointB} = \frac{32M}{\pi d^3} = \frac{32 \times 77709.91}{\pi \times d^3} = 29.3165\ MPa$$

When investigating to Point A and Point B, clearly seen that Point A have the most bending stress so it is critical part of the system.

$\sigma_{bendingPointA} = 30.432\ MPa$, $\sigma_{bendingPointB} = 29.3165\ MPa$

5.19 Torsional Stress on Spindle Shaft

$$\tau_{torsion} = \frac{T \times \frac{d}{2}}{J}$$

Where,

T: Torque Acting on Spindle Shaft,

d:Spindle Shaft Diameter,

J: Polar Moment of Inertia

Torque Acting on Spindle Shaft is below,

T = 7.3 $\times 1.5 = 10.95\ Nm = 10950\ Nmm$

$$\tau_{torsion} = \frac{T \times r}{J} = \frac{10950 \times 15}{\frac{\pi}{32} 30^4} = 2.065\ MPa$$

5.20 Equivalent Stress on Spindle Shaft

Motor shaft be exposed to bending, shear and torsion in this section. Therefore, Von- Misses is used to find the equivalent of these stresses.

$$\sigma_{equivalent} = \sqrt{\left(\sigma_{bending}\right)^2 + 3 \times (\tau)^2}$$

$$\sigma_{equivalent} = \sqrt{(30.432)^2 + 3 \times (2.887 + 2.065)^2}$$

$$\sigma_{equivalent} = 31.617\ MPa$$

$$\frac{\sigma_{yield}}{\sigma_{equivalent}} = FoS$$

$$\frac{\sigma_{yield}}{31.617} = 2.5$$

$$\sigma_{yield} = 79.044\ MPa$$

5.21 Bearing Life Calculations

Cylindrical roller bearings are used in the spindle shaft. Axial load on the shaft is less than radial loads on the shaft. In other words, cylindrical roller bearings are used in this system due to the higher radial load capacity.

The type of bearing used is NUP 206 ECP for only B2 bearings because of axial load. Its features are shown in the figure below.

Basic dynamic load rating	C	44 kN
Basic static load rating	C_0	36.5 kN
Fatigue load limit	P_u	4.55 kN
Reference speed		13 000 r/min
Limiting speed		14 000 r/min
Minimum load factor	k_r	0.15
Limiting value	e	0.2
Axial load factor	Y	0.6

Table 23: NUP 206 ECP Properties

The type of bearing used is NU 206 ECP for A1, A2 and B1 bearings for radial load. Its features are shown in the figure below.

Basic dynamic load rating	C	44 kN
Basic static load rating	C_0	36.5 kN
Fatigue load limit	P_u	4.5 kN
Reference speed		13 000 r/min
Limiting speed		14 000 r/min
Minimum load factor	k_r	0.15
Limiting value	e	0.2
Axial load factor	Y	0.6

Table 24: NU 206 ECP Properties

5.21.1 Life Calculation for Bearings A and B;

For A1 and A2;

- Spindle Speed: 2000 rpm
- Force acting on Bearing A1 and A2 is below,

$\sqrt{1618.1^2 + 1243.55^2} = 2040.75\ N$

$$\frac{2040.75}{2} = 1020.375\ N\ action\ on\ A1\ and\ A2$$

$$1020.375\ \times FoS = 1020.375\ \times 2.5 =\ 2550.938\ N$$

- Life will be calculated below,

$$L = \left(\frac{C}{F}\right)^{\frac{10}{3}}$$

$$L = \left(\frac{44000}{2550.938}\right)^{\frac{10}{3}} = 13258.942\ mrev \rightarrow L_h = \frac{L\times 10^6}{60\times rpm} = \frac{13258.942\ \times 10^6}{60\ \times 2000} = 110491.189\ hour$$

For B1 and B2;

- Spindle Speed: 2000 rpm
- Force acting on Bearing B1 and B2 is below,

$\sqrt{831.034^2 + 1230.45^2} = 1484.797\ N$

$$\frac{1484.797}{2} = 742.399\ N\ action\ on\ A1\ and\ A2$$

- For the axial load, clearly seen that $\frac{F_{axial}}{F_{radial}} = \frac{88.5}{742.399} = 0.1192$ so, using only Radial Load.

$$742.399\ \times FoS = 742.399\ \times 2.5 =\ 1856\ N$$

- Life will be calculated below,

$$L = \left(\frac{C}{F}\right)^{\frac{10}{3}}$$

$$L = \left(\frac{44000}{1856}\right)^{\frac{10}{3}} = 38275.22\ mrev \rightarrow L_h = \frac{L\times 10^6}{60\times rpm} = \frac{38275.22\ \times 10^6}{60\ \times 2000} =$$
$$318960.213\ hour$$

5.22 Fatigue Calculation for Spindle Shaft [39]

Surface Factor (k_a):

$$k_a = a \times S_{ut}^{b}$$

Where,

S_{ut}: Ultimate Tensile Stress of the Shaft Material

According to the Table 18, a and b are founded

Surface Finish	Factor a		Exponent b
	S_{ut}, kpsi	S_{ut}, MPa	
Ground	1.34	1.58	−0.085
Machined or cold-drawn	2.70	4.51	−0.265
Hot-rolled	14.4	57.7	−0.718
As-forged	39.9	272.	−0.995

Table 25: Surface Factor Table

$$k_a = 4.51 \times 620^{-0.265} = 0.82$$

5.22.1 Size Factor (k_b):

$$k_b = \begin{cases} (d/0.3)^{-0.107} = 0.879d^{-0.107} & 0.11 \le d \le 2 \text{ in} \\ 0.91d^{-0.157} & 2 < d \le 10 \text{ in} \\ (d/7.62)^{-0.107} = 1.24d^{-0.107} & 2.79 \le d \le 51 \text{ mm} \\ 1.51d^{-0.157} & 51 < d \le 254 \text{ mm} \end{cases}$$

$$\left(\frac{d}{7.62}\right)^{-0.107}$$

Where,

d: Diameter of the shaft,

$$\left(\frac{30}{7.62}\right)^{-0.107} = 0.863$$

5.22.2 Loading Factor (k_c):

$$k_c = \begin{cases} 1 & \text{bending} \\ 0.85 & \text{axial} \\ 0.59 & \text{torsion}^{17} \end{cases}$$

$k_c = 0.59$

5.22.3 Temperature Factor (k_d):

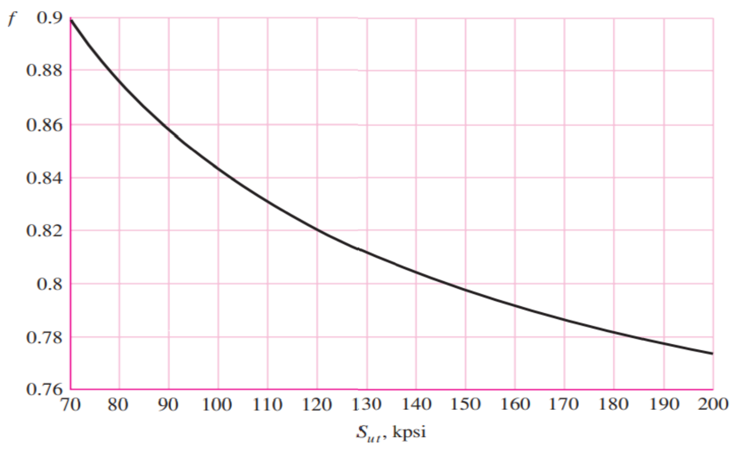
The mini lathe designed according to room temperature. Therefore the temperature factor will be 1.

Temperature, °C	S_T/S_{RT}	Temperature, °F	S_T/S_{RT}
20	1.000	70	1.000
50	1.010	100	1.008
100	1.020	200	1.020
150	1.025	300	1.024
200	1.020	400	1.018

Table 26: Temperature Factor Table

$$k_d = 1$$

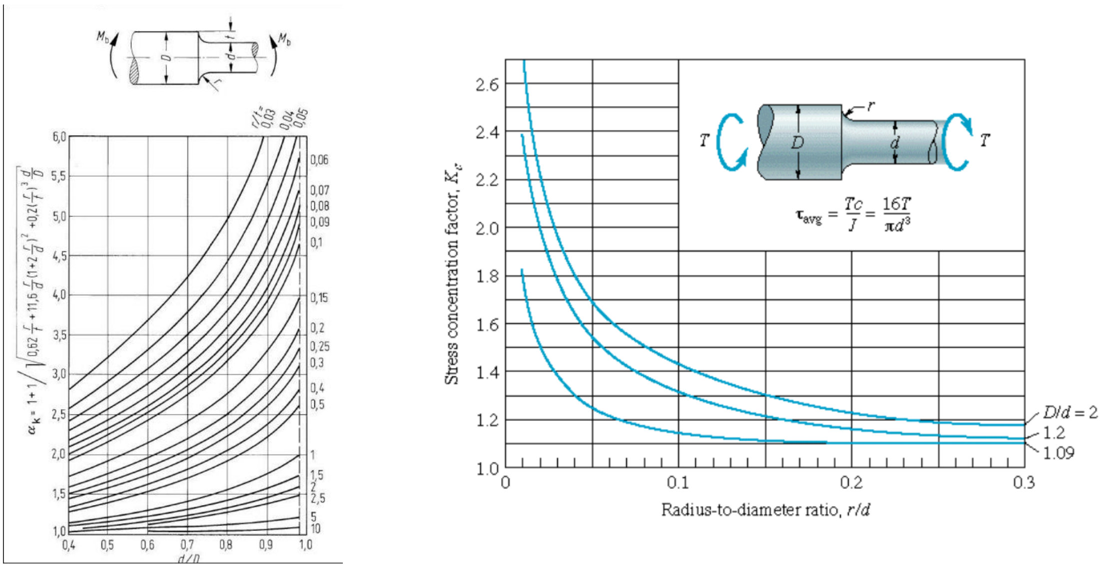
5.22.4 Fatigue Strength Fraction (k_f):



$$S_{ut} = 680\ MPa \cong 98.62$$

$$k_f \cong 0.85$$

5.22.5 Stress Concentration Factor (k_t):



Tables 27 & 28: Stress Concentration Factor Table

According to the Tables above, there is no stress concentration. Therefore, stress concentration factor equals 1.

5.22.6 Endurance Limit (S_e):

Endurance Limit for Cold Drawn 1045 Carbon Steel is 370 MPa. The new endurance limit calculated is below;

$$S_e' = \frac{S_e \times k_a \times k_b \times k_c}{k_d \times k_t}$$

$$S_e' = \frac{370 \times 0.82 \times 0.863 \times 0.59}{1 \times 1} = 154.482\ MPa$$

The maximum and minimum stress on the spindle shaft was calculated.

$$\sigma_{max} = 31.617\ MPa$$

$$\sigma_{min} = 0\ MPa$$

Mean stress and amplitude stress will be calculated as,

$$\sigma_{mean} = \frac{\sigma_{max} + \sigma_{min}}{2} = 15.8085\ MPa$$

$$\sigma_{amplitude} = \frac{\sigma_{max} - \sigma_{min}}{2} = 15.8085\ MPa$$

Fatigue factor of safety for Soderberg Line calculated below,

$$n_{fs} = \frac{1}{\frac{\sigma_{amplitude}}{S_e{}'} + \frac{\sigma_{mean}}{S_y}} = 7.7161$$

In the final part, all the calculations are made and all remains is material selection. Pugh Matrix is given below:

				AISI1045 - Hot Rolled	AISI 1045 - Cold Rolled
Category	Criteria	Weight	Basic Solution	Alternative 1	Alternative 2
Properties	Yield Strength	5	0	-1	1
	Precision	4	0	-1	1
	Price	3	0	1	-1
	Surface Finish	2	0	-1	1
	Internal Stress	1	0	1	-1
	Total			-7	7

Table 29: Pugh Matrix for two types of AISI1045

AISI1045 is the material for both motor and spindle shaft due their high yield strength. AISI1045 has two types of it which are Hot Rolled and Cold Rolled. Pugh Matrix is done to compare these two types. Yield strength of Cold Rolled is higher than Hot Rolled. Hot Rolled has advantages only in price and internal stress. Pulley will create a friction on contact surface which means us to select to have a better surface finish because if friction is applied, deformations may start to occur in contact surface so cold rolled has advantage over hot rolled. By comparing these, AISI1045-Cold Rolled got higher points than hot rolled and is selected to be as material of shafts [40].

5.23. Key and Key Calculations for Spindle and Motor Shaft

A key is a piece of mild steel inserted between the shaft and hub or bore of the pulley to connect these together to prevent relative motion between them. It is always inserted parallel to the axis of the shaft. Key will insert into the pulley of motor shaft. A rectangular extension that's inside of the hub is called as "Keyway" Shaft or mill (doesn't matter which cylindrical material will be assembled and switched for locking inside of it. Keyway and hub will be bore of pulley. In our design, key will prevent a relative motion and power transmission between parts.[41]

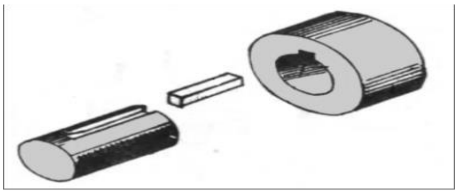
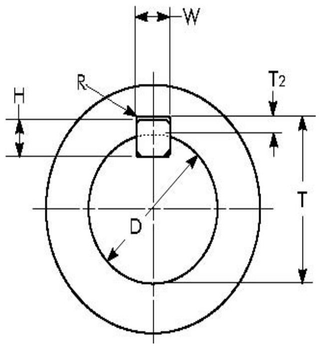


Figure 5.23.1: Key assembly (Example)



Key & Keyway Dimensions - Millimeters										
Shaft Diameter "D"		Key Size Nominal		Keyway Width Hub "W"			Keyway Depth Hub "T2"		Keyway Radius "R"	
Over	Thru	Width "W"	Height "H"	Nominal	Min	Max	Min	Max	Min	Max
6	8	2	2	2	-.0125	+.0125	1.0	1.1	0.08	0.16
8	10	3	3	3	-.0125	+.0125	1.4	1.5	0.08	0.16
10	12	4	4	4	-.0150	+.0150	1.8	1.9	0.08	0.16
12	17	5	5	5	-.0150	+.0150	2.3	2.4	0.16	0.25
17	22	6	6	6	-.0150	+.0150	2.8	2.9	0.16	0.25
22	30	8	7	8	-.0180	+0180	3.3	3.5	0.16	0.25

Table 30 [42]: Key and Keyway dimensions according to ISO/R773 – Js9 Width Tolerance. Width and Height values will be based on this table.

5.23.1 Key Calculations for Motor Shaft

Dimensions of our keyway and hub according to letters in the **Table 5.11.1**

Design Parameters

- Shaft Diameter D: 24mm which is found in motor shaft calculations
- Shaft Torque(T): 7.3Nm which is torque transmitted by shaft
- Length of Key(L): minimum 5.67≅6mm is calculated
- Key Width(b): 8mm (**key width and height is selected as 8mm and 7mm due to Table 5.11.1)**
- Key Height(h): 7mm
- Keyway depth shaft(t1) – Keyway depth hub(t2): 4mm and 3.3mm
- Factor of safety was calculated as 2.5 in motor shaft calculations, so it's based on there.
- Shaft Material is AISI1045-Cold Rolled which has 410MPa shear strength.
- For bearing of the shaft, impact values of steel are based for the key.
- Belly material is pulley of the configuration. Material of pulley(hub) is AL2017-T351 due to preferred due to having not that much shear and bending won't occur due to calculations. So, selecting a intermediate type would not cause trouble and aluminums have good corrosion resistance. mechanical properties by comparing them with other types of AA2000 class which has 250MPa yield strength. Allowable shear stress (allowable surface pressure) of pulley is 250/ (2,5) =100MPa. It's also used in compression of hub calculation below.

- Torque Transmitted by Connection

  $Md = 9550\frac{P}{n} = 9550\,\frac{2{,}2}{3000} = 7.003Nm$

From Figure 6, d=24mm b*h=8*7mm t1=4mm and t2=3.3mm

- Tangential force acting on the key

$$Ft = \frac{Mb}{\left(\frac{d}{2}\right)}\ \frac{7003\left(\frac{N}{mm}\right)}{12mm} = 583.583N$$

➢ Shear of the key

$k = \frac{Ft}{b*l}$ $l \geq \frac{Ft}{etem*b} = \frac{7003}{164*8} = 5{,}67mm$ minimum 5,67 mm for length to overcome of shear. Key should be selected based up to results. Main consideration is that width of pulley is 23mm and key will pass through the bore of pulley. Combining these considerations and dimension standards of manufacturer company. Length of 32mm would enough for the design.

- Bearing of the shaft

$Pm = \frac{2*Md}{d*t1*l} \leq Pem$ $\frac{2*7003}{24*4*32} = 4{,}56(\frac{N}{mm^2}) \leq 100MPa$

In calculations both the values are lower than the shear strength and severe impact dynamic allowable surface pressure values. It's clear to say that material selection is correct, and length of key should be minimum 5,67mm to overcome shearing. According to calculations key, 4 different types of key is selected and compared with each other in Pugh Matrix.

- Bearing of the hub

$$Pg = \frac{2 \times Md}{d \times t2 \times l} \leq Pgem \ \frac{2 \times 7003}{30 \times (3{,}3) \times 32} = 4.421MPa \leq 100MPa$$

				Parallel sunk key	Feather Key	Saddle Key	Square and Round Edged Key	Tangential Key
Category	Criteria	Weight	Basic Solution	Alternate 1	Alternate 2	Alternate 3	Alternate 4	Alternate 5
Properties	Material	5	0	1	1	1	1	1
	Dimensions	4	0	-1	1	1	1	-1
	Torque Transmissibility	3	0	1	1	-1	-1	1
	Load Range	2	0	-1	-1	-1	-1	1
	Simplicity of Assembling	1	0	1	1	1	1	1
	Total			3	11	5	5	7

Table 31: Pugh Matrix between 4 types of Keys

Material of whole keys have high yield strength which means that both keys. Dimensions of alternative 1 is too much for our motor design. Width of pulley is 23.5mm and length of tangential key is little lower so dimensions may not be fit to our system. Saddle and round keys are used in lighter load applications which makes them have a lower torque transmissibility. Tangential keys are used in heavy load applications so, it has more load range. Parallel sunk key has perpendicular edges which makes different assembling-disassembling sequence by comparing it to alternative 2. Tangential keys should be used in two pieces because, each one of them withdraw torsion in only one axis. Feather keys provide axial rotating while transmitting torque.[43] So, alternative 2 got highest point

5.23.1.1. Selected Key

- DIN 6885-8-7-32-A Type Feather Key
- DIN 6885(Medium-carbon steel)

Figure 5.23.1.1.1 Picture of Key

5.23.2. Key Calculations for Motor Shaft

Design Parameters

- Shaft Diameter D: 30 which is found in spindle shaft calculations
- Shaft Torque(T): 7.3*1.5=10.95Nm. Multiplying by 1.5 due to ratio between two pulleys.
- Length of Key(L): minimum 8,006mm≅8mm is calculated to overcome shearing
- Key Width(b): 8mm
- Key Height(h): 7mm
- Keyway depth shaft(t1) – Keyway depth hub(t2): 4mm and 3.3mm
- Factor of safety was calculated as 2.5 in motor shaft calculations
- Shaft Material is AISI1045-Cold Rolled which has 410MPa shear strength.

- For bearing of the shaft, impact values of steel are based for the key.
- Belly material is pulley of the configuration**.** Material of pulley is AL2017-T351 due to average mechanical properties by comparing them with other types of AA2000 class which has 250MPa yield strength. Allowable surface pressure of pulley is 250/(2,5)=100MPa. It's also used in compression of hub calculation below.
- Torque Transmitted by Connection

$Md = 9550\frac{P}{n} = 9550\frac{2,2}{2000} = 10.505Nm$

- Shaft diameter is calculated as 30mm in spindle shaft calculations

From **Table 30**, d=30mm b*h=8*7mm t1=4mm and t2=3,3mm

- Tangential force acting on the key

$$Ft = \frac{M \times b}{\left(\frac{d}{2}\right)} \frac{10505\left(\frac{N}{mm}\right)}{15} = 700.333MPa$$

- Shear of the key

$$k = \frac{Ft}{b \times l}\ l \geq \frac{Ft}{etem \times b} = \frac{10505}{164 \times 8} = 8.006mm$$

- Bearing of the shaft

$$Pm = \frac{2 \times Md}{d \times t1 \times l} \leq Pem\ \frac{2 \times 10505}{304 \times 32} = 5.47MPa(\frac{N}{mm^2}) \leq 100MPa$$

- Bearing of the hub

$$Pg = \frac{2 \times Md}{d \times t2 \times l} \leq Pgem\ \frac{2 \times 10505}{24 \times (3,3) \times 32} = 14.736MPa \leq 100MPa$$

In calculations both the values are lower than the shear strength and severe impact dynamic allowable surface pressure values. It's clear to say that material selection is correct, and length of key should be minimum 8mm to overcome shearing. Bore diameters are equal each other in motor and spindle dimensions According to calculations, key is selected the same which may be seen in **5.23.1.1 Selected Key** part.

5.24. Choosing a Lead Screw and Lead Screw Calculations

5.24.1 Types Of lead Screw

Since the ball screw is used in heavy systems and our CNC machine cuts aluminium (max cutting force = 150 MPa), the lead screw was chosen, so there is no need to bear too much heavy load and the ball screw is expensive and noisy. The lead screw is quiet, cost-effective and low vibration, but the system operates at low efficiency because the lead screw has higher friction. Then accurate types of lead screws were selected.

Acme lead screw was chosen because this shape is easier to machine (faster cutting, longer tool life) than square thread. The thread shape also has a wider base, meaning it's stronger than a similarly sized square thread (so the screw can carry more load).

Acme thread for power screws is not as efficient as a square thread due to the additional friction from the wedge effect but is generally preferred as it is easier to machine and allows the use of a splinter nut. adjusted to withstand wear.[44]

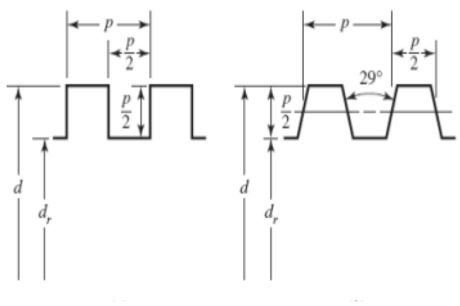


Figure 5.24.1: Square thread and Acme thread

To find the maximum speed and required torque, 3 inputs need to be selected.

My Card =		**pulses/rev**
Time To Distance =		**seconds**
My Screw Pitch=		**mm**

Table 32: Inputs that must be selected

5.24.2 Choosing Pitch [44]

9.5 mm is assumed diameter for power screw.

$$tigtening\ efficiency = \frac{tan\ (\varphi)}{tan\ (\varphi + p')}$$

$$untigtening\ efficiency = \frac{tan(\varphi - p')}{tan\ (\varphi)}$$

$$tan\ (p') = \frac{\mu}{cos \times \frac{\alpha}{2}}$$

$$friction\, of\ coefficient\ (\mu) = 0.20$$

$$\frac{\alpha}{2} = 29°$$

$$tan\ (p') = \frac{0.20}{cos\ 29}$$

$$(p^{'}) = 13°$$

The best choice was found to be 5mm pitch because as the pitch increased, although the efficiency increased, the speed began to decrease, so the optimum efficiency and speed were found at 5mm pitch

					tigtening efficiency	untigtening efficiency			
p' (degree)	φ	tan (φ)	tan(φ+p')	tan (φ-p')	tan (φ) /tan(φ+p')	tan(φ-p')/tan (φ)	efficieny mean	pitch(mm)	Rpm
13,0	0	0,00	0,23		0,00		0,00	0,00	0
	3	0,05	0,29		0,18		0,09	1,20	333
	6	0,11	0,34		0,31		0,15	2,01	200
	12	0,21	0,47		0,46		0,23	3,00	133
	15	0,27	0,53	0,03	0,50	0,13	0,32	4,17	100
	18	0,32	0,60	0,09	0,54	0,27	0,41	5,33	80
	22	0,40	0,70	0,16	0,58	0,39	0,48	6,38	62
	26	0,49	0,81	0,23	0,60	0,47	0,54	7,08	56
	36	0,73	1,15	0,42	0,63	0,58	0,61	8,00	50

Table 33: Efficiency and Rpm

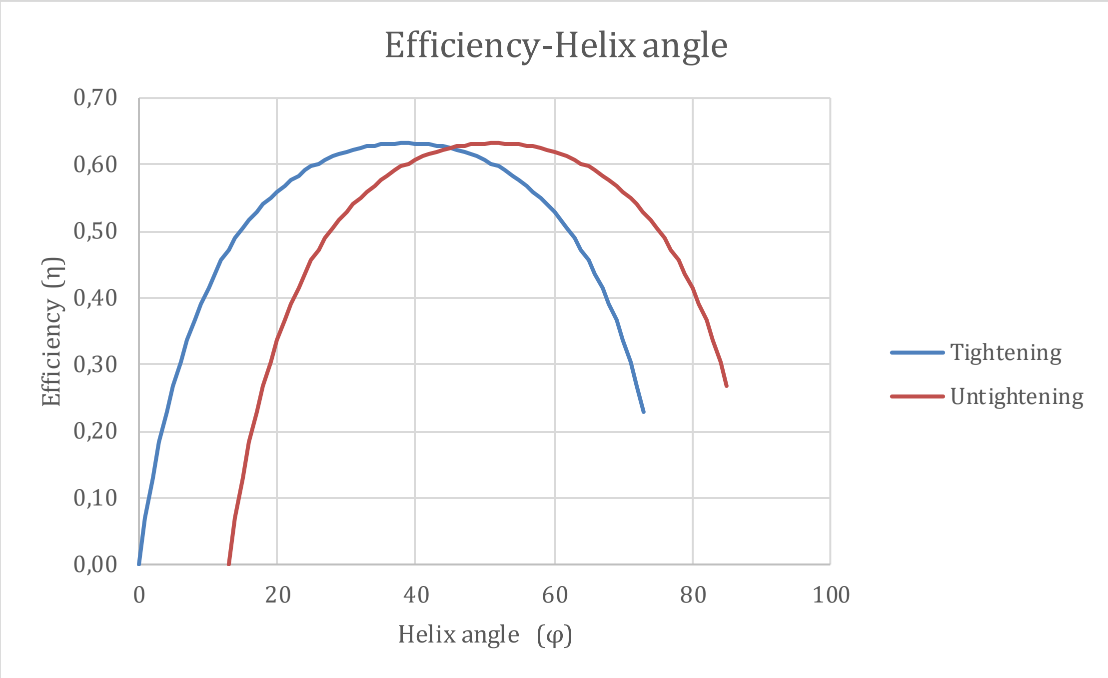


Table 34: Efficiency Graph

5.25 Choosing Drive [45]

Four switch selectable step resolutions. With 1.8 i motor

Half step (400 steps/rev)

1/5 step (1000 s/r)

1/10 step (2000 s/r)

1/64 step (12,800 s/r)

4035: 40V, 3.5A

4850: 48V, 5.0A

6575: 65V, 7.5A

80100: 80V, 10.0A

The best driver was selected by testing motor calculations on excel and looking at motor compatibility and cost calculation. In addition, the most suitable card for system was selected by evaluating the voltage and current values.

Figure 5.25.1: STP-DRV-4850 [46]

5.25.1 Calculation of Time

CNC maximum rpm	2000 rpm
Desired Feed per revolution (fn)	0,2 mm/rev
Maximum feed speed	400 mm/min
	6.66 mm/sn
Time to 10 mm distance (10mm/6.66)	1.5 seconds (maximum)

Table 35: Design Constraints

my card = 400 pulses/rev	**400**	**pulses/rev**
Time to 10 mm distance = 1.5 seconds	**1.5**	**seconds**
my screw pitch= 5 mm	**5**	**mm**

Table 36: Inputs

5.26 Lead Screw Calculations [45]

The maximum speed that the system can reach under maximum load is calculated here. Here, the maximum Rpm and progress coming from the CNC machine stand as an important criterion.

$$10\,mm \times \frac{1\,rev}{5\,mm} \times \frac{400\,pulses}{rev} = 800\,pulses$$

$$\frac{800\,pulses}{1.5\,seconds} = 533\,pulses/sec$$

$$\frac{5\,mm}{1\,rev} \times \frac{1\,rev}{400\,pulses} = 0.0125\,(accuracy)$$

$$10\ mm \times \frac{1\ rev}{5\ mm} = 2\ rev$$

$$\frac{2\ rev}{1.5\ sec} = 1.33\ rev/sec$$

$$\frac{1.33\ rev}{sec} \times \frac{60\ sec}{min} = 80\ rpm$$

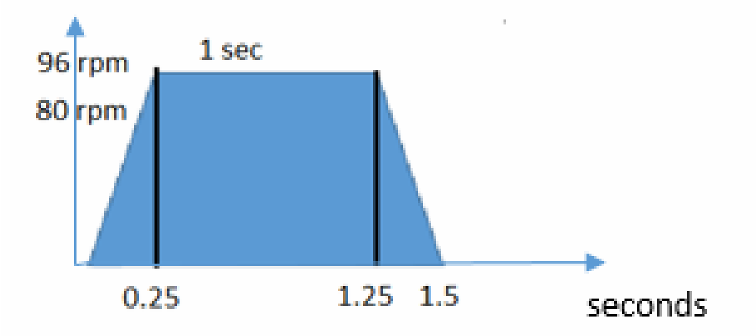


Figure 5.26.1: Area of Trapezoid

$$\frac{1 + 1.5}{2} sec \times h = 800\ pulses$$

h= 640 pulses/sec

$$\frac{640\ pulses}{sec} \times \frac{1\ rev}{400\ pulses} = 1.6\ rev/sec$$

$$\frac{1.6\ rev}{sec} \times \frac{60\ sec}{min} = 96\ rpm$$

Velocity under maximum load was found. Necessary torque calculation will be made and the motor will be selected according to the required torque. In the torque calculation, the forces coming from the machine are added as a resultant force and the weights are collected below as a whole.

X - axis

F tangential	208.5	N
F feed force	88.5	N
Resultant force	226.5	N
Safety Factor	1,54	
Resultant force with safety	350	N
Weight	35,67	kg
Extra Unexpected Weight	2,12	kg
Neva 17048 Motor	0,7	kg
Above bed weight (z axis)	1	kg
Component	3	kg
Cutting Tools	0,5	kg
Total Weight	43	kg

Table 37: Weight of x axis

Torque calculations		
Weight is 100 lb (45kg)	**94**	**lb**
1 g (gravity) = 386.4 in/s^2	**386,4**	**in**/s^2

Table 38: Torque Calculations

$$Tmotor = Trun + Taccel$$

$$Taccel = Jtotal * \left(\frac{\Delta speed}{\Delta time}\right)$$

$$Jtotal = Jmotor + (Jcoupler + Jlead\ screw + Jcarriage)$$

$$Jcarriage\ (Jw) = \frac{W}{g \times e} \times \left(\frac{1}{2 \times \pi \times P}\right)^2$$

Pitch =	5 mm (0.2 inch)
Lead screw length (L)=	417 mm=16.41 inch
Threads per inch (P)= 1/pitch diameter	5 rev/inch
Weight (W)	94 lb (43 kg)
Efficiency (e)	0.43
1 g (gravity) =	386.4 in/s^2
Steel ρ	0.28 lb/in^3

$$Jcarriage\,(Jw) = 0.00057\ lb * in * s^2$$

$$Jscrew = \frac{\pi \times L \times \rho \times r^4}{2 \times g}$$

$$Jscrew = 0.000019\ lb \times in \times s^2$$

$$Jtotal = 0.000593\ lb \times in \times s^2$$

$$Taccel = Jtotal \times (\frac{\Delta speed}{\Delta time})$$

$$\Delta speed = 96\ rpm$$

$$\Delta time = 0.25\ sec$$

$$Taccel = 0.023\ lb \times in$$

$$Ftotal = Fext + Ffriction + Fgravity\,)$$

$$Ftotal = 0 + \ \mu \times W \times \cos\theta\ + 0$$

$$Ftotal = 18.8\ lb$$

$$Trun = \left(\frac{Ftotal}{2 \times \pi \times P}\right)$$

$$Trun = 0.60\ lb \times in$$

$$Tmotor = Trun + Taccel$$

$$Tmotor = 0.60\ lb \times in\ or\ 10.56\ oz \times in\ or\ 0.074\ Nm$$

$\frac{Jtotal}{Jmotor} < 10$ so, it is tried by different motors for j total and the 17048 motor is best choice for system

Specifications

SureStep™ Series Specifications – Connectorized Bipolar Stepping Motors		High Torque Motors					
Bipolar Stepping Motors		STP-MTR-17040x	STP-MTR-17048x	STP-MTR-17060x	STP-MTR-23055x	STP-MTR-23079x	STP-MTR-34066x
NEMA Frame Size		17	17	17	23	23	34
Optional Encoder		Y	Y	Y	Y	Y	N
* Max Holding Torque	(lb·in)	3.81	5.19	7.19	10.37	17.25	27.12
	(oz·in)	61	83	115	166	276	434
	(N·m)	0.43	0.59	0.81	1.17	1.95	3.06
Rotor Inertia	(oz·in²)	0.28	0.37	0.56	1.46	2.60	7.66
	(kg·cm²)	0.05	0.07	0.10	0.27	0.48	1.40
Rated Current (A/phase)		1.7	2.0	2.0	2.8	2.8	2.8
Resistance (Ω/phase)		1.6	1.4	2.0	0.8	1.1	1.1
Inductance (mH/phase)		3.0	2.7	3.3	2.4	3.8	6.6
Insulation Class		130°C [266°F] Class B; 300V rms					
Basic Step Angle		1.8°					
Shaft Runout		0.002 in [0.051 mm]					
Max Shaft Radial Play @ 1lb load		0.001 in [0.025 mm]					
Perpendicularity		0.003 in [0.076 mm]					
Concentricity		0.002 in [0.051 mm]					
* Max Radial Load (lb [kg])		6.0 [2.7]			15.0 [6.8]		39.0 [17.7]
* Max Thrust Load (lb [kg])		6.0 [2.7]			13.0 [5.9]		25.0 [11.3]
Storage Temperature		-20°C to 100°C [-4°F to 212°F]					
Operating Temperature		-20°C to 50°C [-4°F to 122°F] (motor case temperature should be kept below 100°C [212°F])					
Operating Humidity		55% to 85% non-condensing					
Product Material		steel motor case; stainless steel shaft(s)					
Environmental Rating		IP40					
Weight (lb [kg])		0.6 [0.3]	0.7 [0.3]	0.9 [0.4]	1.5 [0.7]	2.2 [1.0]	3.9 [1.7]
Agency Approval		CE (complies with EN55014-1 (1993) and EN60034-1.5.11)					
Accessory Extension Cable		STP-EXT-020					

** For dual-shaft motors (STP-MTR-xxxxxD): The sum of the front and rear Torque Loads, Radial Loads, and Thrust Loads must not exceed the applicable Torque, Radial, and Thrust load ratings of the motor.*

Table 39: Properties of Motors[45]

Motor Rotor Inertia (Nema17048)	**0,37**	**0z-in^2**
Motor Rotor Inertia (Nema17048)	0,00006	lb·in·sec'2

$$\frac{0.00059}{0.00006} = 9.89 \; and \; it \; is \; below \; to \; 10$$

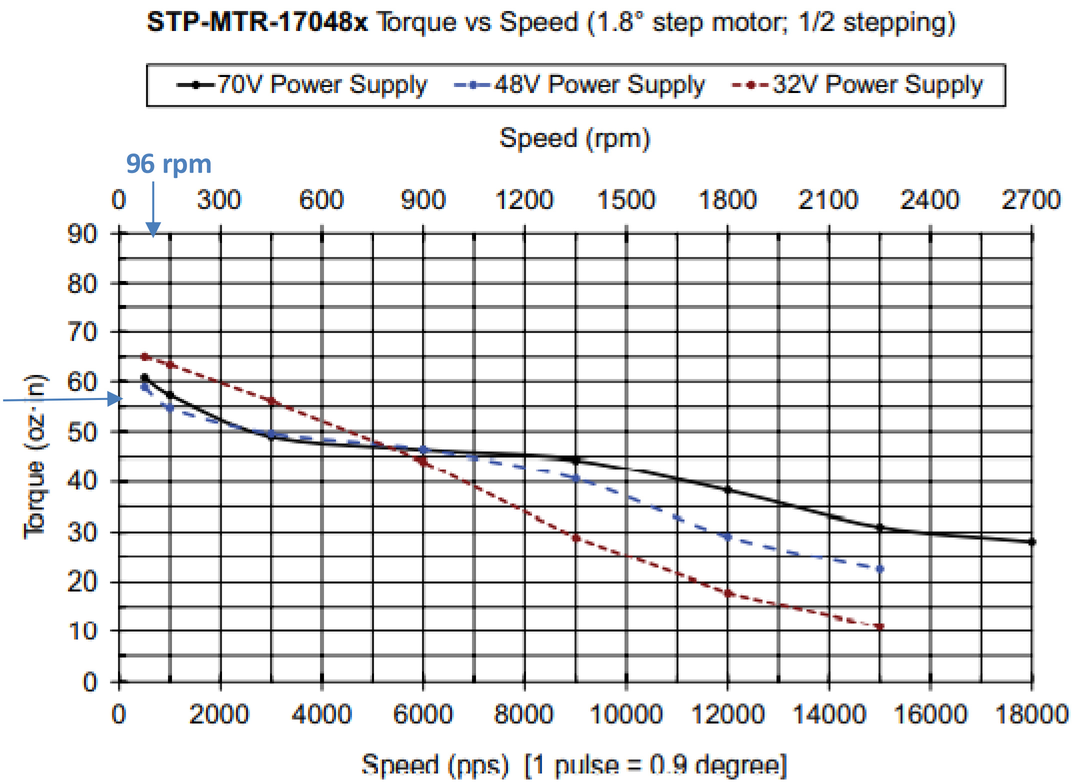


Table 40: Motor specifications [45]

Margin max torque (58 oz x in) =**0.40 Nm**

Holding max torque (Nema 17048) = **0,59 Nm**

STP-MTR-17048[47]

Z – Axis

F tangential	208.5N	N
F feed force	88.5N	N
Resultant force	226.78	N
Safety	1.54	
Resultant force with safety	350	N
Weight	35,67	kg
Component	3	kg
Cutting Tools	0,5	kg
Total Weight	39,17	kg

Table 41: Weight of z axis

Torque calculations		
Weight Is 85 Lb (39kg)	**85**	**lb**
1 G (Gravity) = 386.4 In/s^2	**386,4**	**in/s^2**

$$Tmotor = Trun + Taccel$$

$$Taccel = Jtotal \times (\frac{\Delta speed}{\Delta time})$$

$$Jtotal = Jmotor + (Jcoupler + Jlead\ screw + Jcarriage)$$

$$Jcarriage\ (Jw) = \frac{W}{g \times e} \times (\frac{1}{2 \times \pi \times P})^2$$

Pitch =	5 mm (0.2 inch)
Lead screw length (L)=	265 mm=10.43 inch
Threads per inch (P)= 1/pitch diameter	5 rev/inch
Weight (W)	85 lb (39 kg)
Efficiency (e)	0.43
1 g (gravity) =	386.4 in/s^2
Steel ρ	0.28 lb/in^3

Table 42: Properties Z-axis

$$Jcarriage\ (Jw) = \ 0.00051\ lb \times in \times s^2$$

$$Jscrew = \frac{\pi \times L \times \rho \times r^4}{2 \times g}$$

$$Jscrew \ = \ 0.000012\ lb \times in \times s^2$$

$$Jtotal = 0.000531\ lb\ \times in \times s^2$$

$$Taccel = Jtotal \times (\frac{\Delta speed}{\Delta time})$$

$$\Delta speed = 96\ rpm$$

$$\Delta time = 0.25\ sec$$

$$Taccel = 0.021\ lb \times in$$

$$Ftotal = Fext + Ffriction + Fgravity\ )$$

$$Ftotal = 0 + \ \mu \times W \ \times cos\ \theta \ + 0$$

$$Ftotal = 17\ lb$$

$$Trun = \left(\frac{Ftotal}{2 \times \pi \times P}\right)$$

$$Trun = 0.54\ lb \times in$$

$$Tmotor \ = \ Trun + Taccel$$

$$Tmotor = 0.56\ lb \times in\ or\ 8.96\ oz \times in\ or\ 0.063\ Nm$$

Motor Rotor Inertia(Nema17048)	**0,37**	**0z-In^2**
Motor Rotor Inertia(Nema17048)	0,00006	Lb·In·Sec'2

Table 43: Inertia of Nema17048

$$\frac{0.00053}{0.00006} = 8.33\ and\ it\ is\ below\ to\ 10$$

$\frac{Jtotal}{Jmotor} < 10$ so tried different motors for j total and 17048 engine was the best choice.

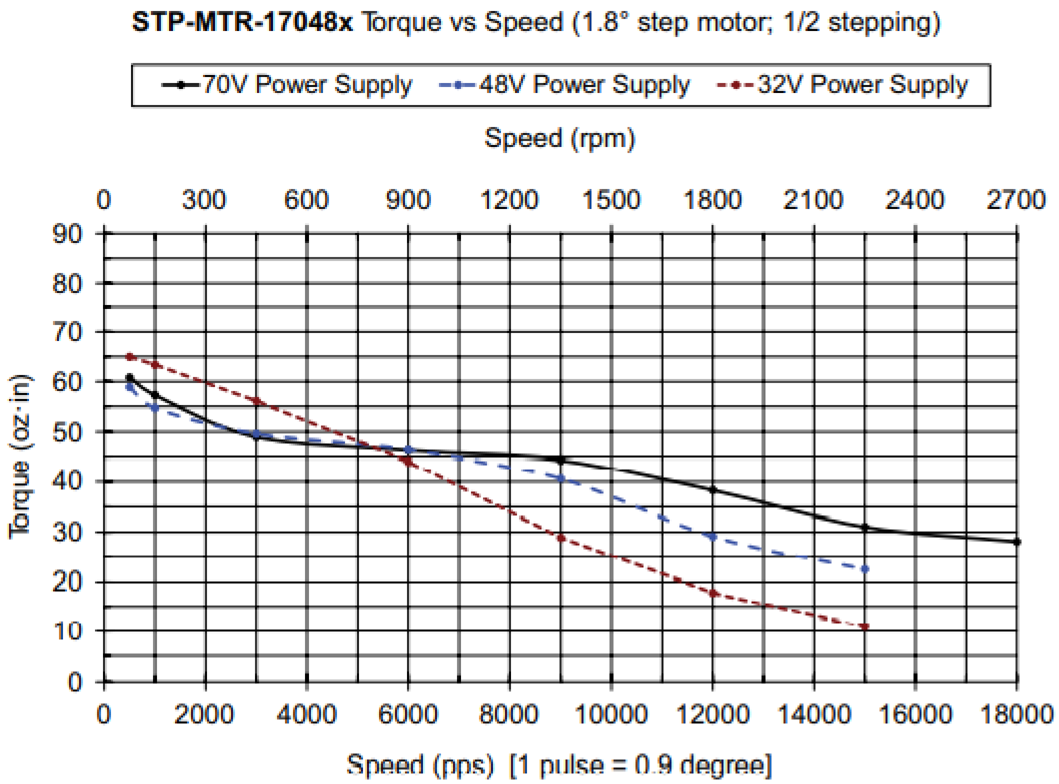


Table 44: Torque vs Speed Graph of STP-MTR-17048x

Motor: 17048 [47]

Drive: 4850,[46]

Figure 5.15.2: Power Supply

Power Supply [48]:

The power supply suitable for the selected voltage and motor has been selected. After the necessary conditions have been met, the cost comparison has been made. Power supply options are shown below. Here, the most suitable power source was selected by applying the power source selection formula, that is, the current in our motor, as well as the card suitability

PWR 3204, PWR 4805 , PWR4810, PWR7005

Power supply should be 0.66 x 2=1.32 <5

Max Rpm with no load

Current (Imax) =	2	A
Inductance (L)=	2,7	Mh
Voltage (V)=	48	V
steps/rev (spr)=	400	steps

$$Max\ speed\ = \left(\frac{V}{2\times L\times Imax\times spr}\right)$$

$$Max\ speed\ = 0.011\times 10^{-3}$$

$$Max\ speed\ = 11.1\ rev/sec\ = 666rpm$$

$$allowable\ Max\ speed\ (1.33\ safety)\ with\ no\ load = 500\ rpm$$

5.26. Choosing a Coupling

Motor Maximum Torque is 0.59 Nm

The use of flexible coupling was found appropriate. Torque from the selected engine is the main criterion. In addition, the most suitable coupling selection has been made for maximum speed, material and cost.

Coupling	For Shaft Diameter	Maximum RPM	Material	Maximum Torque	Cost	Links	
Servomotor Precision Flexible Shaft Coupling	6.35mm × 5mm	10,000 rpm	2017 Aluminum and 304 Stainless Steel	0.90 n*m	$64.79	www1.mcmaster.com/2764K121/	
Servomotor Precision Flexible Shaft Coupling	6.35mm × 5mm	10,000 rpm	2017 Aluminum and 304 Stainless Steel	2.03 n*m	$75.45	www1.mcmaster.com/2764K224/	
Clamping Precision Flexible Shaft Coupling	6.35mm × 5mm	6,000 rpm	7075 Aluminum	1.12 n*m	$50.85	www1.mcmaster.com/2464K17/	
Clamping Precision Flexible Shaft Coupling	5mm × 6mm	6,000 rpm	303 Stainless Steel	1.69 n*m	$110.26	www1.mcmaster.com/2463K125/	

Table 45: Choosing Coupling

Figure 5.26.1: Coupling

Preload Force Calculation

$$M_T = F_{pre} \times \left(\frac{d_2}{2} \times tan\,(\varphi + p')\right)$$

$$\varphi\ = \frac{P}{\pi \times d_2}$$

$$p'\ = \frac{\mu}{cos \times \frac{\alpha}{2}}$$

$$friction\ of\ coefficient\ (\mu) = 0.20$$

$$\frac{\alpha}{2} = 29°$$

$$M_T = 0.59\ Nm$$

$$d_1 = 9.53\ mm$$

$$d_2 = 9\ mm$$

$$d_3 = 8\ mm$$

$$\varphi\ = 0.17$$

$$p' = 0.23$$

$$F_{pre} = 321.53\ N\ \ also\ F_{pre} = F_{axial}$$

Figure 5.26.2: Lead Screw (Example)

$$\tau_{max} = \frac{T}{J} \times C$$

$$J = \frac{\pi \times {d_3}^4}{32}$$

$$\tau\ (\frac{N}{mm^2}) = \frac{590\ N \times mm}{0.09818 \times (8mm)^4} \times 4mm$$

$$\tau = 5.86\ Mpa$$

$$\sigma = \frac{F_{pre}}{A}$$

$$\sigma = \frac{321.53\ N}{0.785 \times (8mm)^2}$$

$$\sigma = 6.4\ Mpa$$

$$\sigma(yield) = \sqrt{\sigma^2 + 3 \times \tau^2}$$

$$\sigma\ (yield) = 12.01\ Mpa$$

$$so\ it\ can\ select\ an\ applicable\ material\ for\ 12.01\ Mpa$$

lead screw converted to 6.35mm from 9.53mm				
Lead screw	metarial	tensile strength	yield strength	cost
9.53mm* 265mm	1018 Carbon Steel	441 Mpa	370 Mpa	$8.32
9.53mm* 265mm	4140 Alloy Steel	655 Mpa	415 Mpa	$12.94
9.53mm* 265mm	304 Stainless Steel	505 Mpa	215 Mpa	$18.65
9.53mm* 265mm	303 Stainless Steel	690 Mpa	500 Mpa	$70.37

Table 46: Lead Screws

So $it\ is\ selected\ a\ AISI\ Type$ 304 $Stainless\ Steel$ for lead screw materials

$\sigma\ (yield) = 215\ Mpa\ for\ AISI\ Type\ 304\ Stainless\ Stell$ so $safety\ = \frac{215\ Mpa}{9.01\ Mpa}$

so $safety = 23.8$

$$Efficiency(\eta) = \frac{F_{pre} \times P}{2 \times \pi \times M_T}$$

$$F_{pre} = 321.53\ N$$

$$P = 5\ mm$$

$$M_T = 0.59\ Nm$$

$$Efficiency(\eta) = 0.43 = \%43$$

5.27. Buckling

5.27.1 For Z-axis

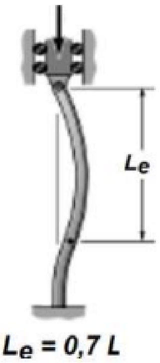


Figure 5.27.1: Visual Expression of Buckling

$$Pcr\ = \pi^2 \frac{E \times I}{L^2}$$

$$Le = 0.7 * 221.52 = 155.06\ mm$$

$$d = 9.5\ mm$$

$$I = \frac{\pi \times d^4}{64}$$

Unsupported distance (L) =	221,52	mm
Modules of elasticity for steel (E) =	200000	N/mm^2
Le=	155,06	mm
least moment of inertia (I)=	404,89	mm^4

$$maximum\ axial\ load\ (Pcr)\ =\ 33239\ N$$

$$my\ Pcr = 321.5\ N$$

$$safety = \frac{33239\ N}{321.5\ N}$$

$Safety = 103.3$ it is safe for buckling.

5.27.2 For X- axis

$$Pcr\ = \pi^2 \frac{E \times I}{L^2}$$

$$Le = 0.7 * 373.91\ = 261.7\ mm$$

$$d = 9.5\ mm$$

$$I = \frac{\pi \times d^4}{64}$$

Unsupported distance (L) =	373.91	mm
Modules of elasticity for steel (E) =	200000	N/mm^2
Le=	261.7	mm
least moment of inertia (I)=	404.89	mm^4

$$maximum\ axial\ load\ (Pcr)\ =\ 11666.6\ N$$

$$my\ Pcr\ = 321.5\ N$$

$$safety = \frac{11666.6\ N}{321.5\ N}$$

$safety = 36.2$ it is safe for buckling.

5.28 Nuts

Nuts			
Thread size	**Metarial**	Dynamic load	price
9.53 mm-5 mm	PET Plastic[49]	1550 N	26.7 $
9.53 mm-5 mm	Manganese Bronze[50]	2980 N	30 $

Selected the steel lead screw and it is known from shingley's machine design book that bronze is the best nut for steel lead screw. In addition, Manganese bronze nuts have good strength and wear resistance.

Type of load	Lower limit of f_s
For a static load less frequently used	1 to 2
For an ordinary single-directional load	2 to 3
For a load accompanied by vibrations/impact	4 or greater

Table 47: Safety Factor

So safety factor should be above to 4 on nut.

$$P = \frac{P_F}{F} \times 9.8 \left(\frac{N}{mm^2}\right)$$

p : Contact surface pressure on the tooth from an axial load (P_F) (N/mm^2)

F : Dynamic permissible thrust (N)

P_F : Axial load (N)

Screw tooth surface is 9.8 N/mm^2

$$P_F = 321.5\ N$$

$$F = 2980\ N$$

$$P = 1.05\ Mpa$$

Sliding speed

$$V = \frac{\pi \times D \times n}{cos\ \alpha \times 10^3}$$

V: Sliding speed (m/min)

D: diameter (mm)

n: Revolutions per minute

α: pitch angle

$$D = 9.5\ mm$$

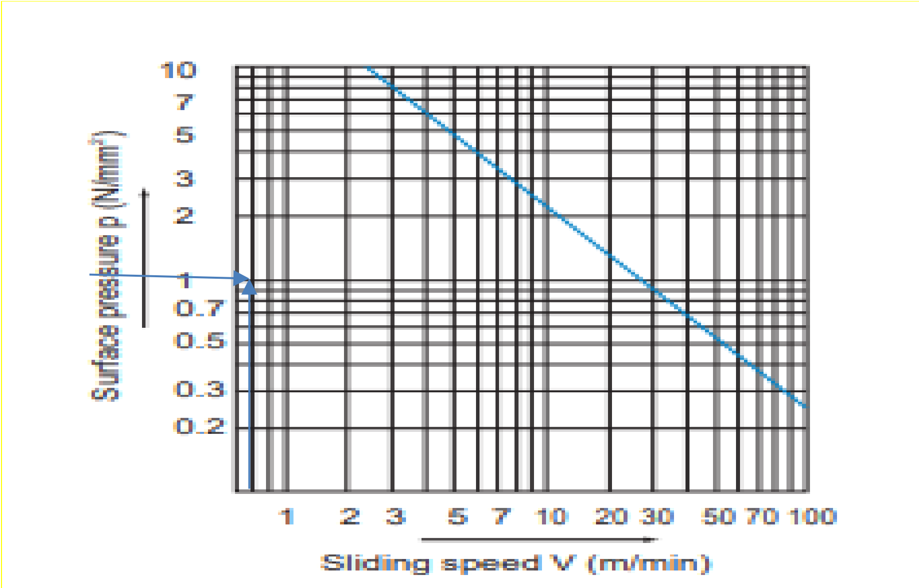
$$n = 96\ rpm$$

$$\alpha = 29°$$

$$V = 3.31\ m/min$$

$$P = 1.05\ Mpa$$

$$V = 3.31\ m/min$$



Results founded values are below the chart so manganese nut can be selected.

5.29. Bearings

5.29.1 For Z - axis

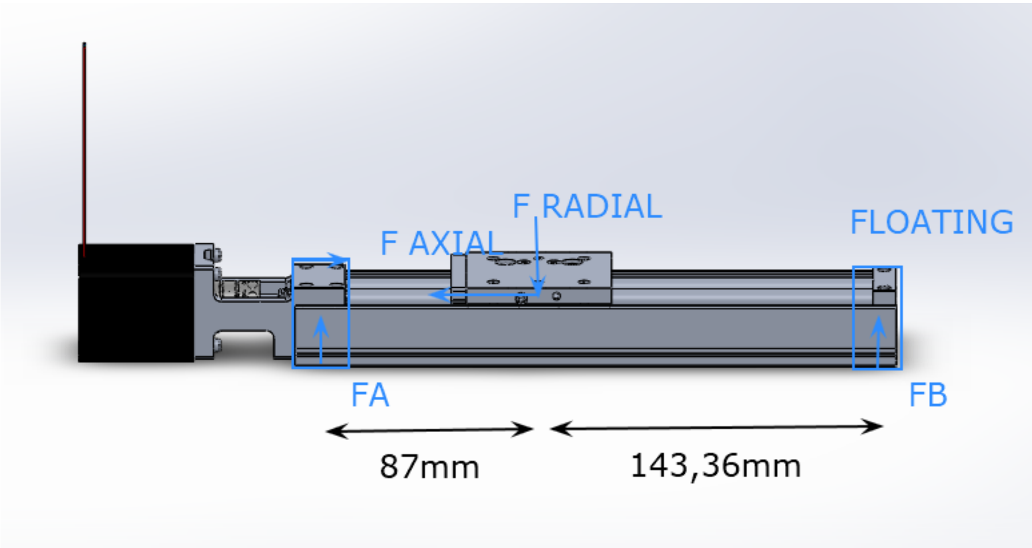


Figure 5.29.1.1: Z axis Forces for Bearing

Floating Bearing Distance	**143,36**	mm
Fixed Bearing Distance	**87,00**	mm
F_{Axial} (FA)=	**321,53**	N
F_{Radial} (FR)=	**382,59**	N

$$F_{Radial} = 39\ kg \times 9.81\ m/sec^2 = 382.59\ N$$

$$MA = 0; \quad 382.59\ N \times 87\ mm - Fb \times 230.36\ mm = 0$$

$$F_{Radial}\ on\ b\ point(Fb) = 382.59\ N \times (\frac{87\ mm}{230.36})$$

$$F_{Radial}\ on\ b\ point = 144.48\ N$$

$$Fa = FR - Fb$$

$$F_{Radial}\ on\ a\ point\ (Fa) = 238.1\ N$$

First, the ball bearing was chosen, the lead screw diameter was focused on the inner diameter of the bearing, then the ball bearing was chosen because the system was not overloaded. The first part was chosen as fixed and the other part as floating.

Deep groove ball bearing preferred and purchased from SKF

Figure 5.29.2: Deep grove ball bearing [51]

Ball Bearing		
method 2 Dynamic loading from SKF (C)=	3320	N
method 2 STATİC loading from SKF (Co)=	1400	N

$$e = \frac{F_{Axial}\ (FA)}{(Co)}$$

$$e = 0.23\ then\ it\ can\ see\ at\ table\ on\ machine\ design$$

F_a/C_0	e	$F_a/(VF_r) \le e$ X_1	Y_1	$F_a/(VF_r) > e$ X_2	Y_2
0.014*	0.19	1.00	0	0.56	2.30
0.021	0.21	1.00	0	0.56	2.15
0.028	0.22	1.00	0	0.56	1.99
0.042	0.24	1.00	0	0.56	1.85
0.056	0.26	1.00	0	0.56	1.71
0.070	0.27	1.00	0	0.56	1.63
0.084	0.28	1.00	0	0.56	1.55
0.110	0.30	1.00	0	0.56	1.45
0.17	0.34	1.00	0	0.56	1.31
0.28	0.38	1.00	0	0.56	1.15
0.42	0.42	1.00	0	0.56	1.04
0.56	0.44	1.00	0	0.56	1.00

*Use 0.014 if $F_a/C_0 < 0.014$.

Table 48: Equivalent Radial Loads Factors for Ball Bearings [52]

$$e = 0.36\ on\ table$$

$$Bearing\ on\ a = \frac{F_{Axial}\ (FA)}{F_{Radial}\ (FR)}$$

$$Bearing\ on\ a = 1.35\ should\ be > 0.36\ so\ that\ is\ appropriate$$

$$x = 0.56\ and\ y = 1.23$$

So, equivalent force is equal:

$$F(eq) = x \times Fr + y \times FAxial$$

$$F(eq) = 0.56 \times 238.1\ N + 1.23 \times 321.53\ N$$

$$F(eq) = 528.82\ N$$

$$L = (\frac{C}{F})^3$$

$$L = (\frac{3320\ N}{528.82\ N})^3$$

$$La = 247.45\ million\ rev$$

$$Lh = \frac{La \times 10^6}{60 \times RPM}$$

$$RPM = 96\ (load\ andrpm)$$

$$Lh = 42960.13\ hrs$$

$$Lh = 4.97\ years$$

For B point

$$Lb = \left(\frac{3320\ N}{382.59\ N}\right)^3$$

$$Lb = 653.45\ million\ rev$$

$$Lh = \frac{Lb \times 10^6}{60 \times RPM}$$

$$RPM = 96\ (load\ andrpm)$$

$$Lh = 113446.5\ hrs$$

$$Lh = 13.13\ years(underload\ andrpm)$$

5.29.2 For x axis

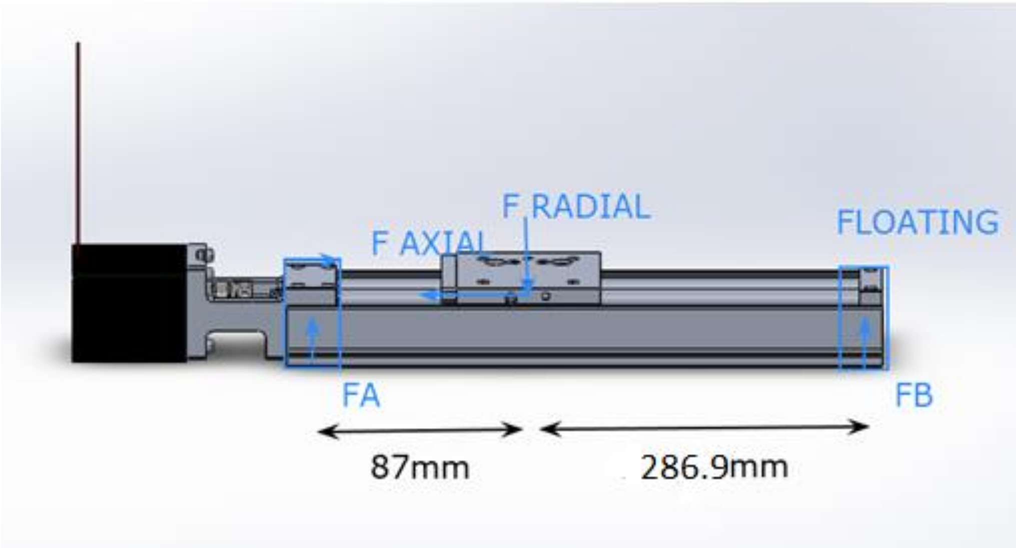


Figure 5.29.2.1: X axis forces for bearing

Floating Bearing Distance	**286.9**	Mm
Fixed Bearing Distance	**87,00**	Mm
F_{Axial} (Fa)=	**321,53**	N
F_{Radial} (Fr)=	**421.83**	N

$$F_{Radial} = 43\ kg \times 9.81\ m/sec^2 = 421.83\ N$$

$$MA = 0; \quad 421.83\ N \times 87\ mm - Fb \times 373.9\ mm = 0$$

$$F_{Radial}\ on\ b\ point(Fb) = 382.59\ N \times (\frac{87\ mm}{230.36})$$

$$F_{Radial}\ on\ b\ point = 159.3\ N$$

$$Fa = FR - Fb$$

$$F_{Radial}\ on\ a\ point\ (Fa) = 262.52\ N$$

First, the ball bearing was chosen, the lead screw diameter was focused on the inner diameter of the bearing, then the ball bearing was chosen because the system was not overloaded. The first part was chosen as fixed and the other part as floating.

Deep groove ball bearing preferred and purchased from SKF

Figure 5.29.2.1: Deep Groove Ball bearing [52]

Ball Bearing		
Method 2 Dynamic Loading from SKF (C)=	3320	N
Method 2 Static Loading from SKF (Co)=	1400	N

$$e = \frac{F_{Axial}\ (FA)}{(Co)}$$

$$e = 0.23\ then\ \imath\ looked\ at\ table\ on\ machine\ design$$

F_a/C_0	e	$F_a/(VF_r) \le e$ X_1	Y_1	$F_a/(VF_r) > e$ X_2	Y_2
0.014*	0.19	1.00	0	0.56	2.30
0.021	0.21	1.00	0	0.56	2.15
0.028	0.22	1.00	0	0.56	1.99
0.042	0.24	1.00	0	0.56	1.85
0.056	0.26	1.00	0	0.56	1.71
0.070	0.27	1.00	0	0.56	1.63
0.084	0.28	1.00	0	0.56	1.55
0.110	0.30	1.00	0	0.56	1.45
0.17	0.34	1.00	0	0.56	1.31
0.28	0.38	1.00	0	0.56	1.15
0.42	0.42	1.00	0	0.56	1.04
0.56	0.44	1.00	0	0.56	1.00

*Use 0.014 if $F_a/C_0 < 0.014$.

Table 49: Equivalent Radial Loads Factors for Ball Bearings [52]

$$e = 0.36\ on\ table$$

$$Bearing\ on\ a = \frac{F_{Axial}\ (FA)}{F_{Radial}\ (FR)}$$

$$Bearing\ on\ a = 1.35\ should\ be > 0.36\ so\ that\ is\ appropriate$$

$$x = 0.56\ and\ y = 1.23$$

So equivalent force is equal ;

$$F(eq) = x \times Fr + y \times FAxial$$

$$F(eq) = 0.56 \times 262.52\ N + 1.23 \times 321.53\ N$$

$$F(eq) = 542.49\ N$$

$$L = (\frac{C}{F})^3$$

$$L = (\frac{3320\ N}{542.49\ N})^3$$

$$La = 229.2\ million\ rev$$

$$Lh = \frac{La \times 10^6}{60 \times RPM}$$

$$RPM = 96\ (load\ andrpm)$$

$$Lh = 39792.4\ hrs$$

$$Lh = 4.6\ years$$

5.29.2.2 for point B

$$Lb = (\frac{3320\ N}{421.83\ N})^3$$

$$Lb = 487.44\ million\ rev$$

$$Lh = \frac{Lb \times 10^6}{60 \times RPM}$$

$$RPM = 96\ (load\ andrpm)$$

$$Lh = 84625\ hrs$$

$$Lh = 9.79\ years(underload\ andrpm)$$

5.30. Rails

252 mm * 9.5 mm	Rails					
	AISI 1566		AISI 6061-T6 Aluminum Alloy base		440C Stainless Steel	
	Tensile strength (MPa)	Yield strength (MPa)	Tensile strength (MPa)	Yield strength (MPa)	Tensile strength (MPa)	Yield strength (MPa)
	980	785	310 MPa	276 MPa	760-1970 Mpa	450-1900 Mpa
	60 $		53 $		109.4 $	
	/www.mcmaster.com/59585K83		https://www.mcmaster.com/1049K15/		https://www.mcmaster.com/6557K21/	

Table 50: Different Types of Rails [53]

Bearings on rails (lineer ball bearing)			
materails	Dynamic load	Static load	Price
6061 Aluminum	400 N	4159	21 $

four linear balls bearing selected.

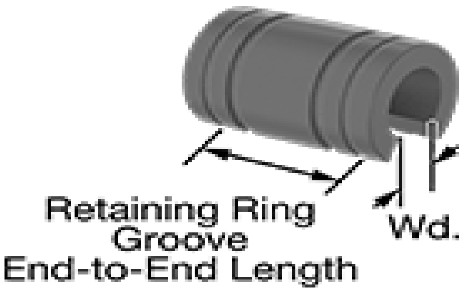


Figure 5.30.1: Bearing on rails [54]

5.31. Calculations for Weldment of Ribs

At the last part, ribs will be joint welded in three locations which are left part of the lathe body, right and middle parts. Theoretical explanation of rib was made on **Part 4.9**.

Length of the rib shouldn't be longer than 3 times of thickness of primary sheet so from SolidWorks, thickness of steel sheets is 5mm. So, height of the rib should not be more than 15mm. 10mm is selected due to design constraints. Material is selected as ASTM A36 mild(low) carbon steel to be as the same material with sheet bodies and it has high yield strength which as 250MPa. Taking factor of safety as 2.5 due to being more stable in long term. Failing time is going to be longer. Allowable normal stress is 250/2.5=148MPa

- $Throat\ Thickness: t = s \times sin45° = 0.707s = 0.707 \times 10 = 7.07mm$
- $Minimum\ area\ of\ the\ weld\ or\ throat\ area: A = 0.707 \times 10 \times 10 = 70.7mm^2$
- $Allowable\ force\ for\ double\ fillet\ weld: F = 1.414 \times \sigma t \times s \times l = 1.414 \times 148 \times 10 \times 10 = 20927.2N$ this is the force of welding that will be applied to sheet while it' done.

Next step is calculating the strength of the weld seam: $\sigma u, s = v1 \times v2 \times v3 \times \sigma s$

- V_1: Weld Seam Factor
- V_2: Weld Quality Factor
- V_3: Impact Factor
- $\sigma_{S:}$ Allowable stress for structural element(MPa)

5.31.1 Weld Seam Factor(V1) Selection

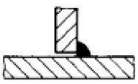
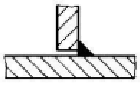
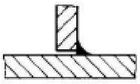
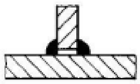
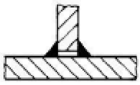
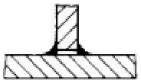
Köşe dikişler		Tek taraflı			İki taraflı		
		Dışbükey	Düz	İçbükey	Dışbükey	Düz	İçbükey
Dikiş türü							
Kaynak alanı		a · l			2a · l		
V_1	$\sigma_{\perp}$ 1), 5)	0,28	0,3	0,33	0,38	0,42	0,5
	$\sigma_{\perp}$ 2), 4)	0,5	0,54	0,6	0,54	0,6	0,7
	$\tau_{\parallel}$ 3), 6)	0,28	0,3	0,33	0,38	0,42	0,5

Table 51: Weld Seam Factor Selection for Fillet Welds [55]

Ribs will be welded via fillet weld which shape is concave and its double sided. So according to figure above. It's selected as to be **V_1=0,38**

Weld Quality Factor(V2) Selection

Various weld grades;

1. Grade weld V_1=1

2. Grade weld V_2=0,8

3. Grade weld V_3=0,5

From the variables right above, it's selected as to be V_2=0,8

Impact factor (V3) Selection

- **Weak impacts v3=1…0,9**
- **Moderate impacts v3=0,8…0,7**
- **Strong impacts v3=0,5**
- **Very strong impacts v3=0,3**

Bed in lathe and rotation of chuck with workpiece inserted in and tangential force applied will apply a force to lathe chassis. These are neither strong nor weak which leads us to take **V_3=0.8**

$$\sigma u, s = v1 \times v2 \times v3 \times \sigma s = 0.38 \times 0.8 \times 0.8 \times 148 = 35.9936MPa$$

Strength of welding is found 35.9936MPa, and allowable normal stress is 100MPa. 35,9936<100 which means that weld strength is sufficient and won't overcome to yield strength of low carbon(mild) steel.

6. ANSYS WORKBENCH ANALYSIS FOR MOTOR AND SPINDLE SHAFT

Materials selected and calculations are made by hand so, analysing them in ANSYS Workbench must be done to determine comparing systems and obtaining where the maximum stresses occur. Two critical parts are determined which are spindle and motor shaft because, without them, lathe machine won't work. In detailed way, torque transmission wouldn't be possible.

6.1. ANSYS Analysis of Motor Shaft

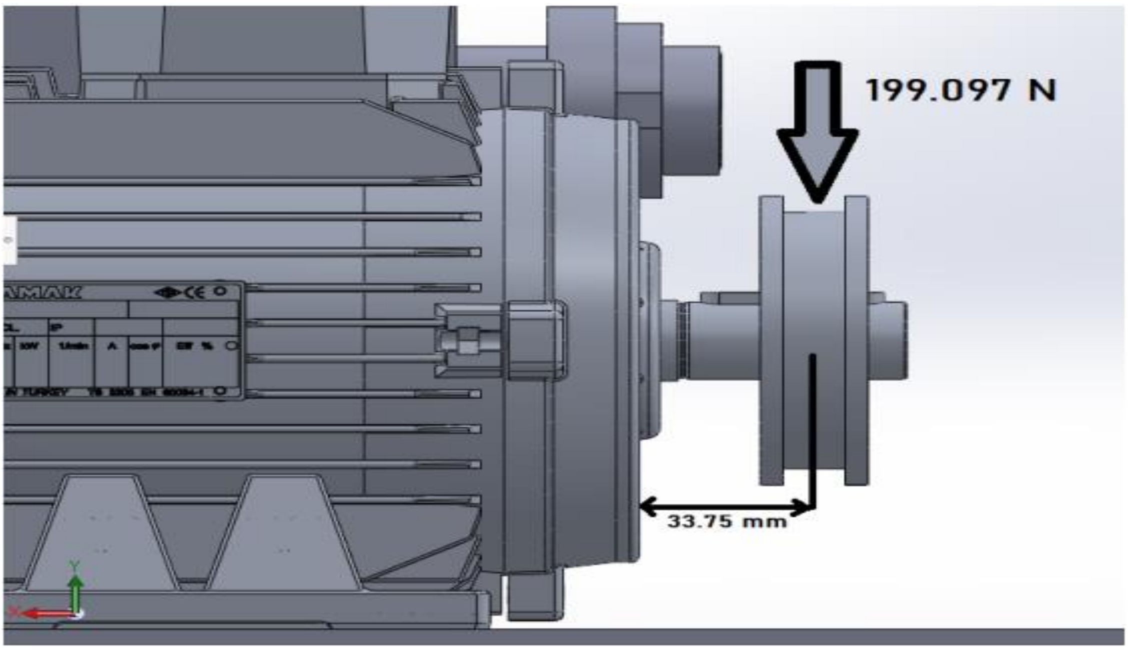


Figure 6.1.1: X-Y axis (Side) view of motor and motor shaft

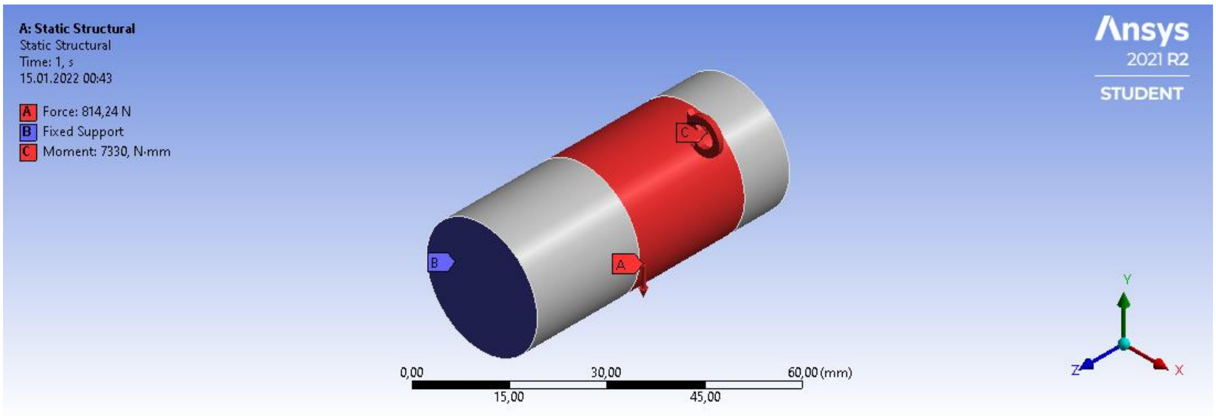


Figure 6.1.2: Setup of Motor Shaft

AISI1045-Cold Rolled is created from the start and mechanical properties are added. Length of shaft is xx mm taken from SolidWorks. B stands for fixed support because it fitted to motor from left side. Red area is width of pulley which equals 23mm. and 814,24N is equivalent tension and it's applied through negative y-axis. Torque is calculated as 7.3Nm and it converted to mm that is 7300Nmm that can be seen in **Figure 6.1.2**. Which applied to opposite circle face of point B. Mass of shaft may be negligible due to being so light as 0.3209 kg. In ANSYS, there is no direct selection in Static Structural(A5) like torque or bending. They can be added as "Moment" and that gives no problem in solution.

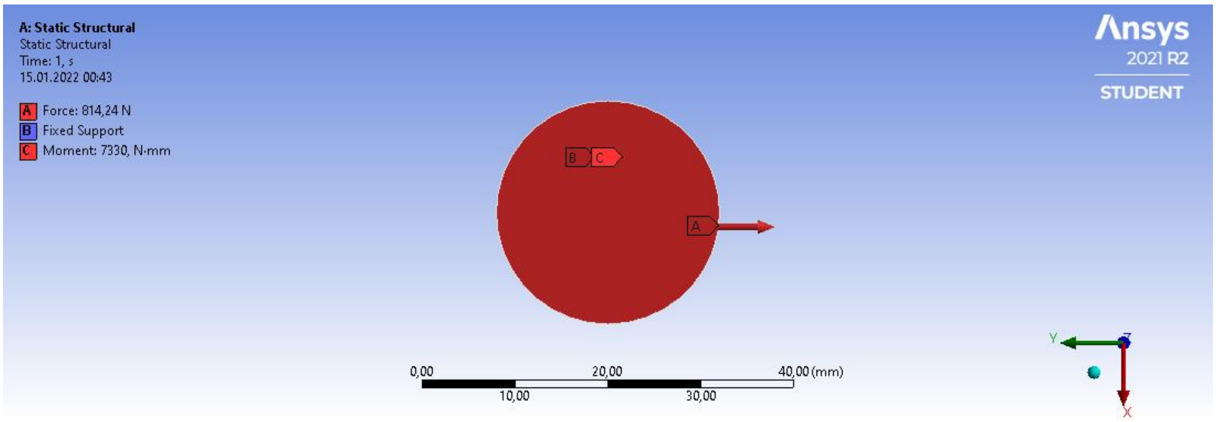


Figure 6.1.3: Applied Torque

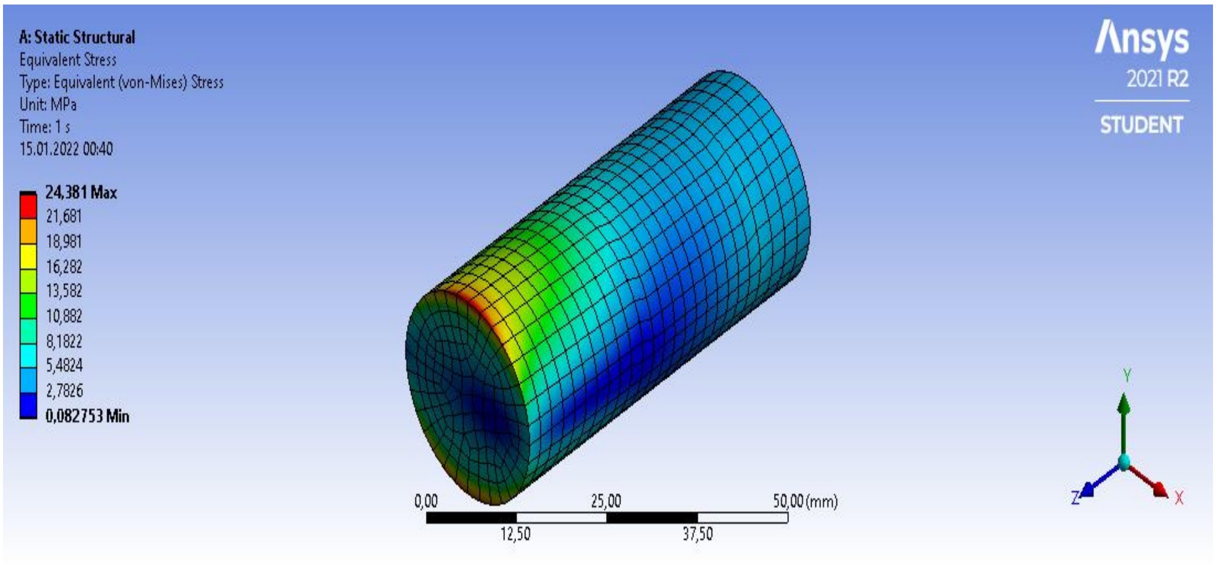


Figure 6.1.4: Equivalent Stress

Default mesh quality is given as 2 in meshing options. Increasing the mesh quality will increase number of elements in mesh which will increase accuracy of results. So, mesh quality is increased by 1 and inserted as 3 to take more accurate results. Equivalent(von-Mises) stress is found in 24,381MPa in ANSYS. Motor shaft is calculated as 21,613MPa and two results are so close to each other. That is done for providing of paper-made calculations and observing stress distribution in shaft surface. These result show that selecting AISI1045-Cold Rolled is suitable correction for our motor shaft

6.2 ANSYS Analysis of Spindle Shaft

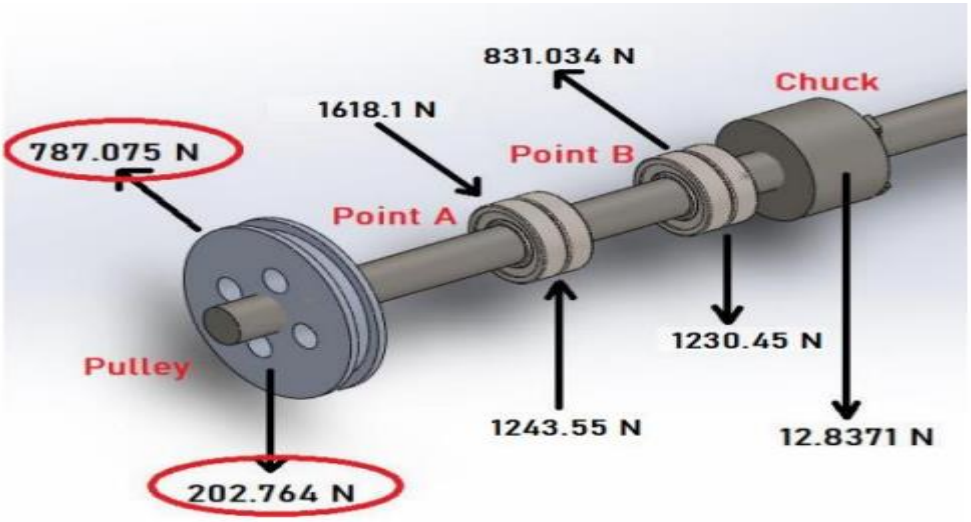


Figure 6.2.1: Configuration of Spindle Shaft

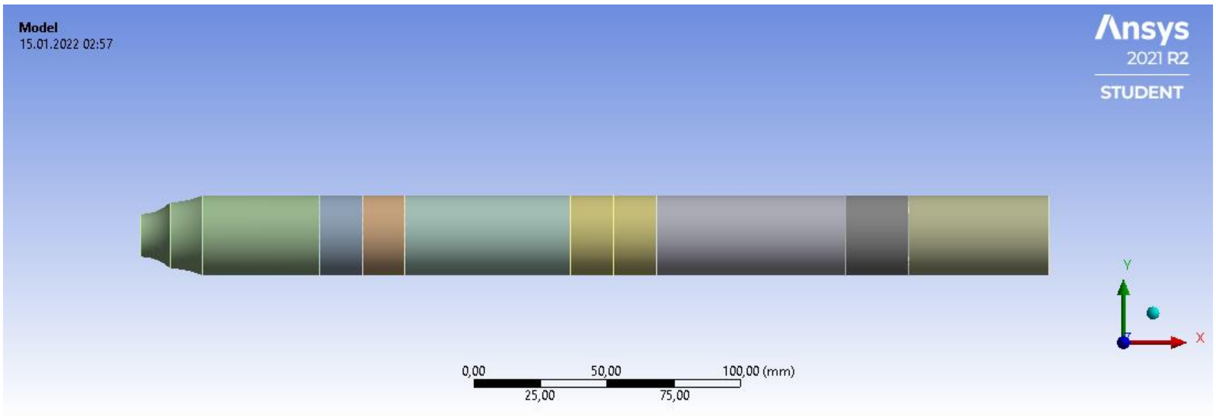


Figure 6.2.2: Geometry of Spindle Shaft

Likewise in Motor Shaft Analysis, mechanical properties of AISI1045-Cold Rolled are inserted. Shaft are inserted. is not flat as it's defined on geometry. Shaft is stepped and diameter is increased by arc radius and has 16mm diameter in left side due to design constraint of chuck that that 16mm shaft diameter backwards. Chuck will be fitted from left side. Afterwards, diameter has increased by shape of arc (not by perpendicular lines). First two areas in different colours stand for bearings and other yellow areas are too. width is 16mm in every one of these. Dark grey area in right side symbolizes pulley which have 23.5mm width. Dimensions between centres of bearings and pulley are inserted also. Setup of spindle shaft is given below. Length of shaft is 340mm taken from SolidWorks.

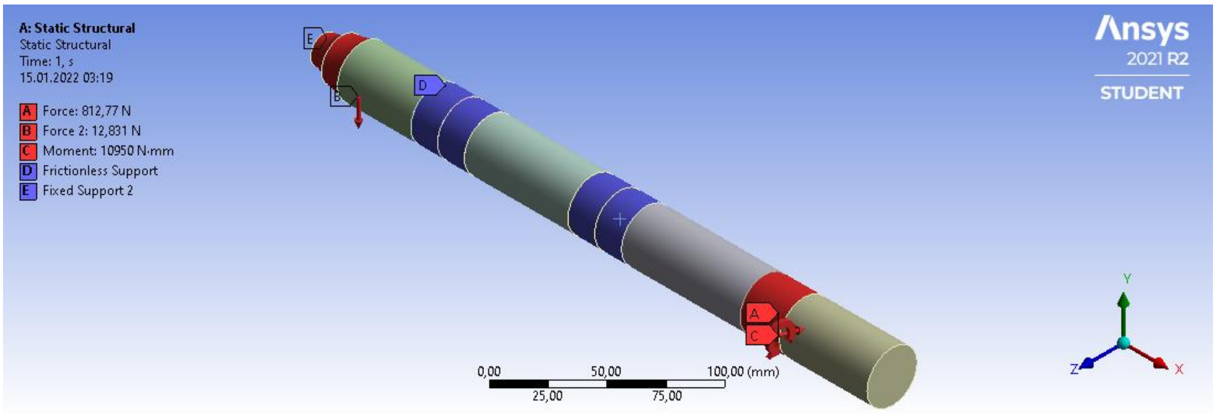


Figure 6.2.3: Mechanical Setup of Spindle Shaft

4 purple pieces stand for bearings and their contact surface of areas are defined as frictionless contact. Forces in A that are forces that applied by pulley are inserted by axially. But ANSYS give equivalent force. Force C is applied by chuck through negative y-axis and fixed support must be applied to calculate it correctly. So, left tip is selected as to be fixed support because will carry the tangential force and should be stay moveless and results are given in 3 different figures to determine the elements which has got highest stress with these parameters.

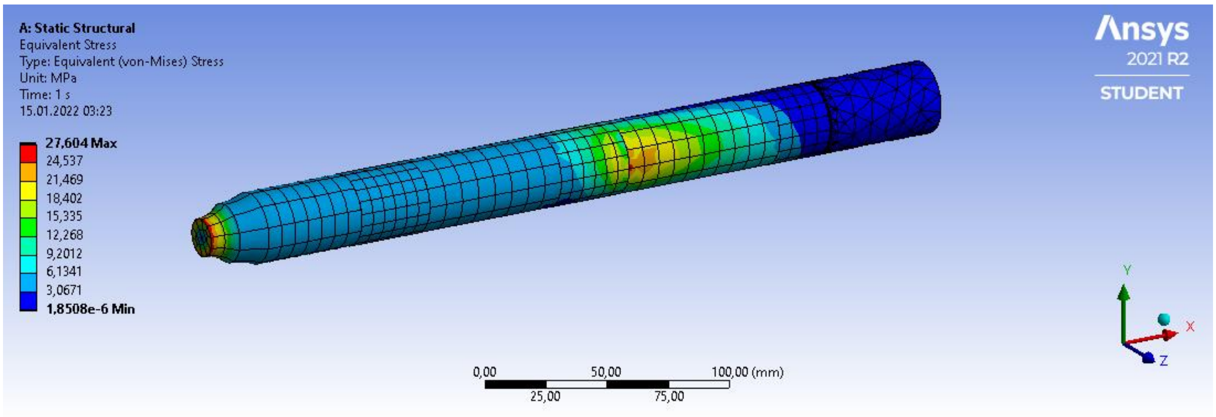


Figure 6.2.4: View from middle of shaft

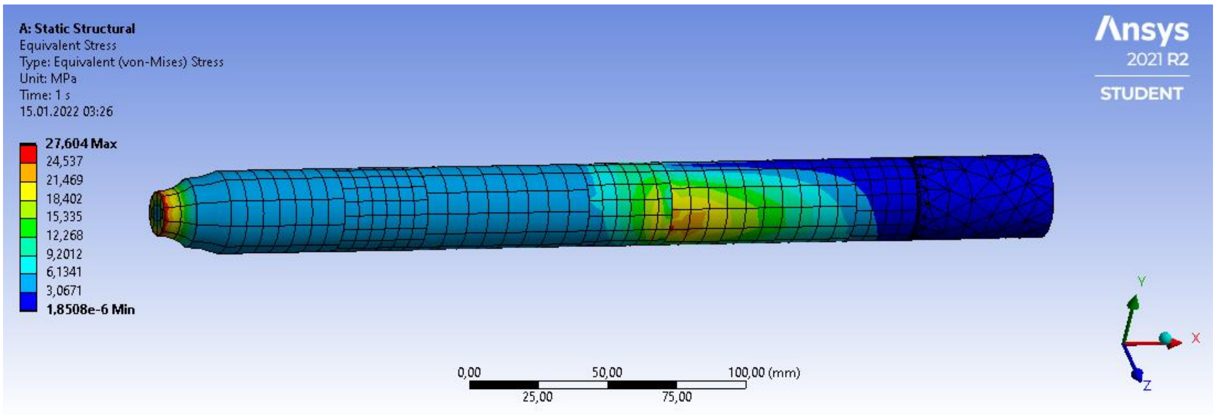


Figure 6.2.5: View from angle which shows maximum stress occurred

Mesh quality is taken 2. Bearings will carry the loads and light blue area provides it by having around 3 or 8MPa stress values. Right part has almost 0MPa stress because it’s free end of the shaft and no forces or moments were applied that area. Because of bearings, stress values are increased between pulley and them. Left size will also affected by bending which caused to have elements and maximum stress values 27,604MPa are occurred from there. In paper-made calculations, equivalent bending stress is found as 31,617MPa. Results in two different applications were so close and 3 or 4MPa difference may be neglected. This analysis showed that by selecting AISI1045-Cold Rolled were good decision and not very high stresses which may break the shafts doesn't exist for these values.

7. CONTROL SYSTEM

CNC Mini Lathe's control system responsible are below,

- When unexpected situations occur, there should be an avoiding system.
- Operator should control spindle speed and distance x-axis and z-axis motion.
- AC Motor must be adjusted from controller.
- Stepper Motors' working distance must be adjusted from controller.

7.1 Components of the Control Systems:

7.1.1 Emergency Stop Button [56]

When unexpected situations occur, operator must push stop button to ensure safety.

Figure 7.1.1.1: Emergency Stop Button

7.1.2 Controller [57]

PLC is the most used controller for the automation systems. Motors which we used in this project are connected to the controller. This controller is a PLC so it is responsible for all control of the system. Also, this unit has an HMI integrated and include their software for CNC Mini Lathe.

Figure 7.1.2.1: Controller

7.1.3 AC Motor Driver [58]

AC Motor is connected to its driver. Then, driver will transmit the data from PLC to AC Motor for adjusting spindle speed.

Figure 7.1.3.1: AC Motor Driver

7.1.4 Stepper Motors Driver [59]

Stepper motor with x-axis and Stepper motor with z-axis are connected to their drivers. Then, Then, driver will transmit the data from PLC to the stepper motors for adjusting working distance from x-axis and z-axis.

Figure 7.1.4.1: Stepper Motor Driver

7.1.5 Limit Switches [60]

Limit switches checks that the power screw system is calibrated with respect to the zero point.

Figure 7.1.5.1: Limit Switches

7.1.6 Power Supply

Power supply will be used to meet the power needs of PLC and Stepper Motor Drivers.

Figure 7.1.6.1: Power Supply

7.1.7 Control System Design

Control system inputs are operator input, program and stop button as seen in the Figure 7.1.1.1. All these data will be processed by the PLC and executed by program. Then, they are performed by the controller. Another of the inputs is the stop button. The main purpose of this button is to shut down the system if it is pressed in unexpected situations. However, it is recommended to be used only in emergencies. Finally, operator input is adjusted spindle speed and power screw systems' speed. As it is known that inputs which used in flowchart entered are connected to PLC. The PLC comes directly with its software. The purpose of the HMI (Human Machine Interface) is to be the monitor that shows the operations performed on the PLC to the user. Then, operations are performed to ensure that all motors operate under the conditions desired by the user. These operations will be carried out with drivers suitable for each motor. The main purpose here is to adjust the spindle speed. On the other hand, two stepper motor drivers are bought for the x-axis and z-axis stepper motors on the power screw system. The purpose here is similar to the AC motor driver. That is, it is used to determine the cutting speed and regulate the position of the screw. Also, the values used will depend on the initial design constraints. Finally, there are limit switches. Their main purpose is to learn the position of x-axis and z-axis. If we explain in more detail, the place where the limit switch is located is the starting point of axis.

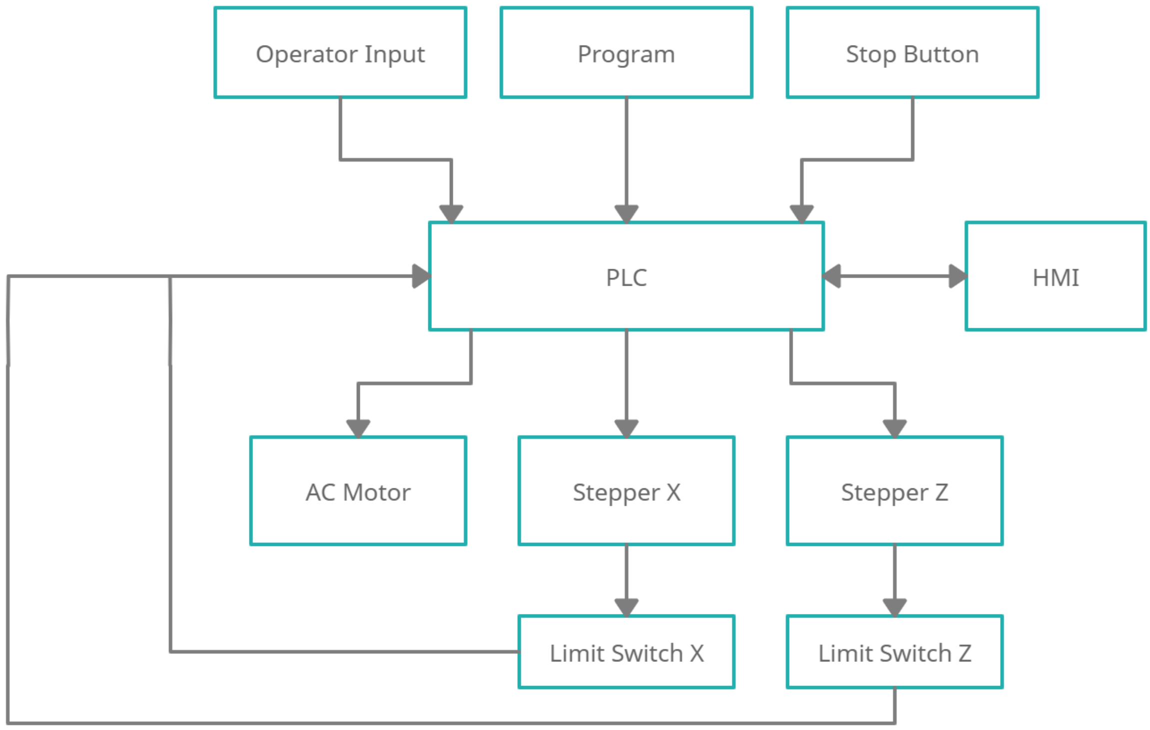


Figure 7.1.7.1: Control System Scheme

8. COST ANALYSIS

Quantity or cost per kilogram and cost of each material and finally, total cost is given in table below

1USD=12TL		
Material	Quantity(piece)/Kilogram	Total Cost(USD)
Shaunseng Trade WM210 38mm mil kutusu	1 piece	231,18
Talos 63mm M14*1 3-Jaw Mini Chuck Lathe	1 piece	95
MT2 torna teleskopik punta	1 piece	50,32
Harbor Freight/Cen-Tech MT2 Live Center	1 piece	19,99
Thsgrt Store/ wm180V Tool Holder	1 piece	87,86
Standard V-Belt, A Type	1 piece	13,61
GAMAK AGM2E 90 L 2b Motor	1 piece	67,66666667
4PCS 90mm Aluminium Alloy Pulley	1 piece	55,3
MPA Spares 135mm Compressor Motor Pulley Wheel	1 piece	34,9
Gates A37 Hi-Power 1/2" Width	1 piece	12,05
STP-DRV-4850 Drive Card	2 pieces	498
STP-MTR-17048 Motor	2 pieces	49
PWR4805- Power Supply	1 piece	169
2764K121- Coupling	2 pieces	129,58
Lead Screw	1 piece	18,65
Lead Screw	1 piece	22
M5*25mm length Bolt (HSS)	12 pieces	13,975
M8*10mm length Bolt (HSS)	4 pieces	8
M5*10mm length Bolt8Carbon Steel)	8 pieces	12
M10 Grade 4.6 bolts & 7549K88- Nuts	2 pieces	60
SKF EEB 3-2Z Deep Groove Ball Bearings	4 pieces	34,96
M5 Hexagon Nuts (DIN934)-Stainless Steel(A2)	26 pieces	25,48
SKF NUP 206 ECP Cylindrical Roller Bearings, Single Row	1 piece	31,85
SKF NU 206 Cylindrical Roller Bearing	3 pieces	102,39
1049K-15 Rails (x and y axis)	2 pieces	212
Linear Ball Bearings	8 pieces	168
AISI1045-Cold Rolled Shaft for Spindle	0,3209Kg	0,23
AISI1045-Cold Rolled Shaft for Spindle	0,6948Kg	0,5
Bed for x-axis	1 piece	50
Bed for z-axis	1 piece	40
	Total Cost (USD)	2313,491667

Table 52: Cost Analysis

9. 3D and Technical Drawing of Mini Lathe Machine

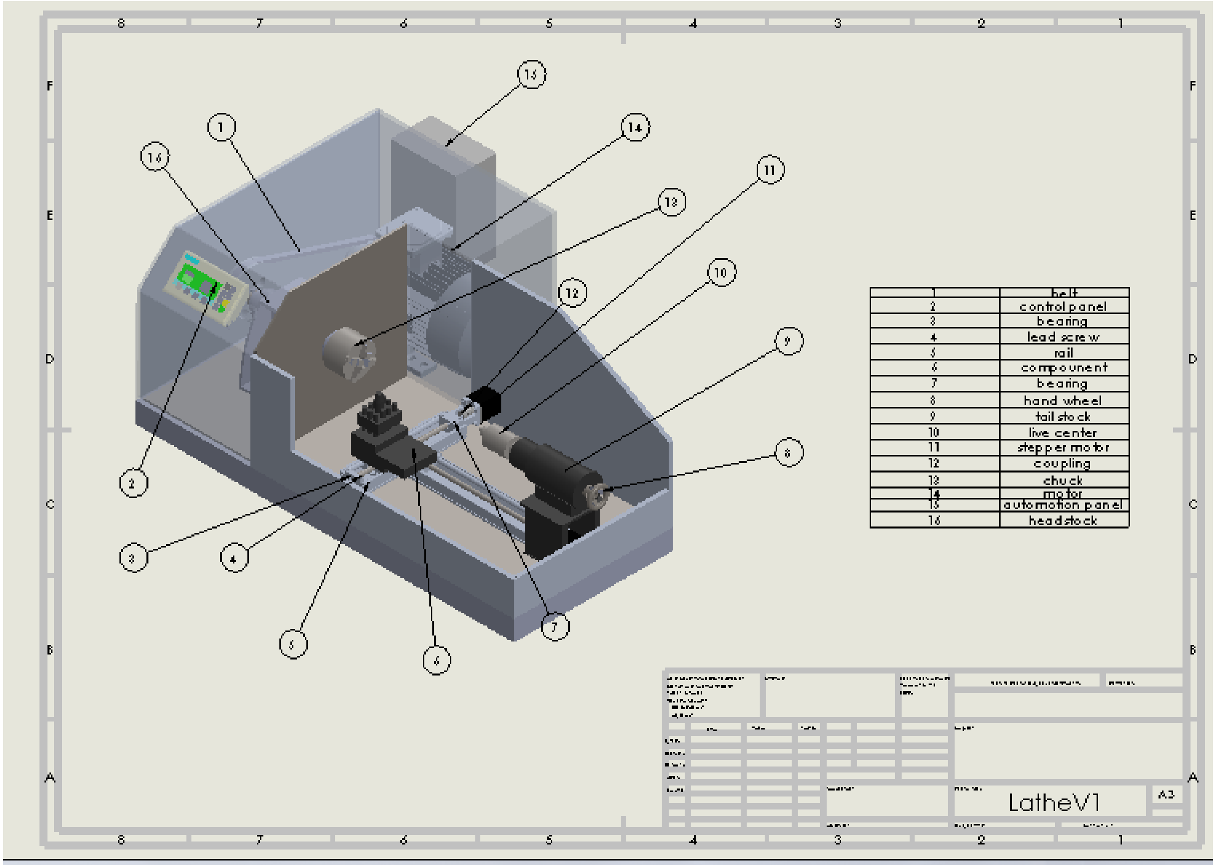


Figure 9.1: Materials of Mini Lathe Machine

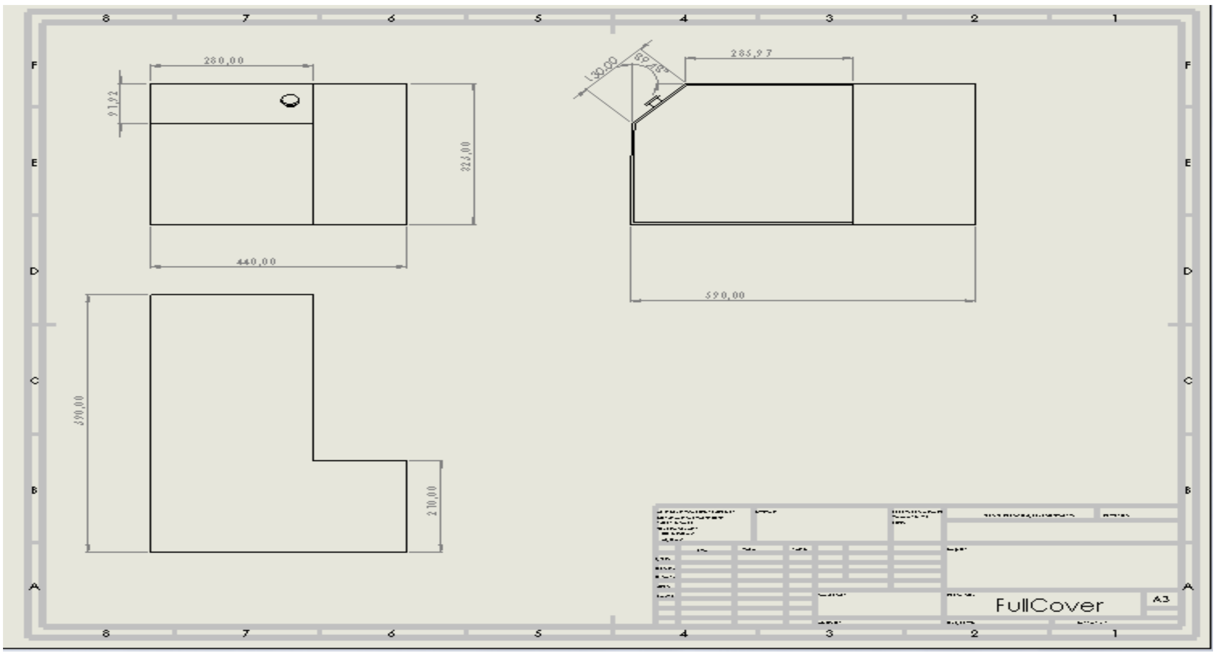


Figure 9.2: Full Cover

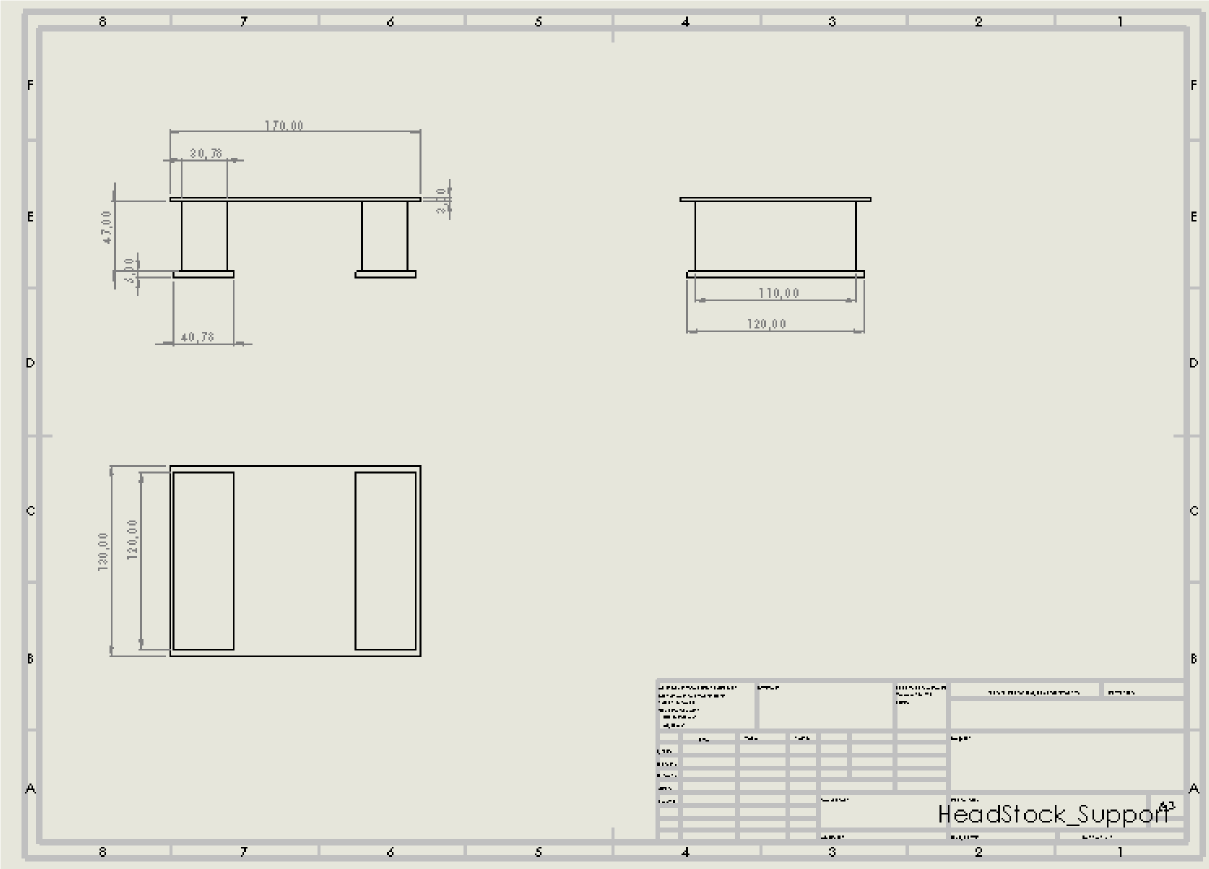


Figure 9.3: Headstock Support

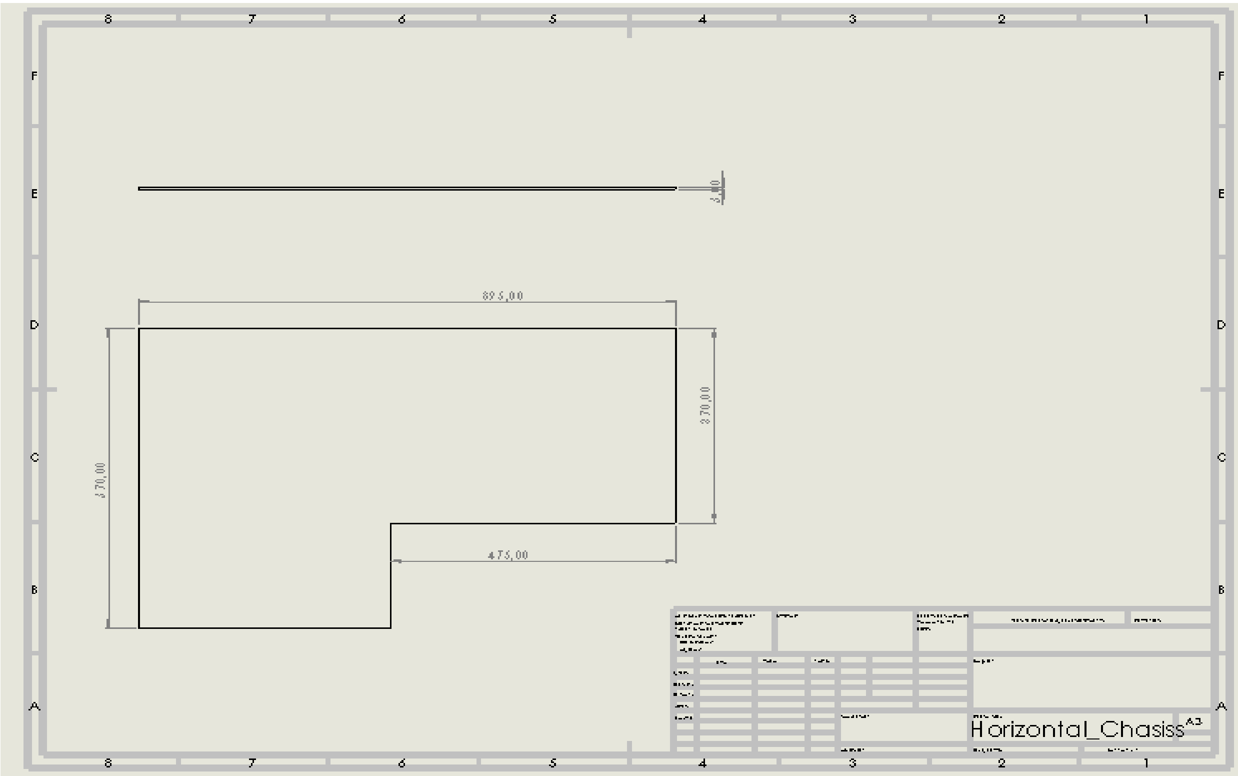


Figure 9.4: Horizontal Chassis

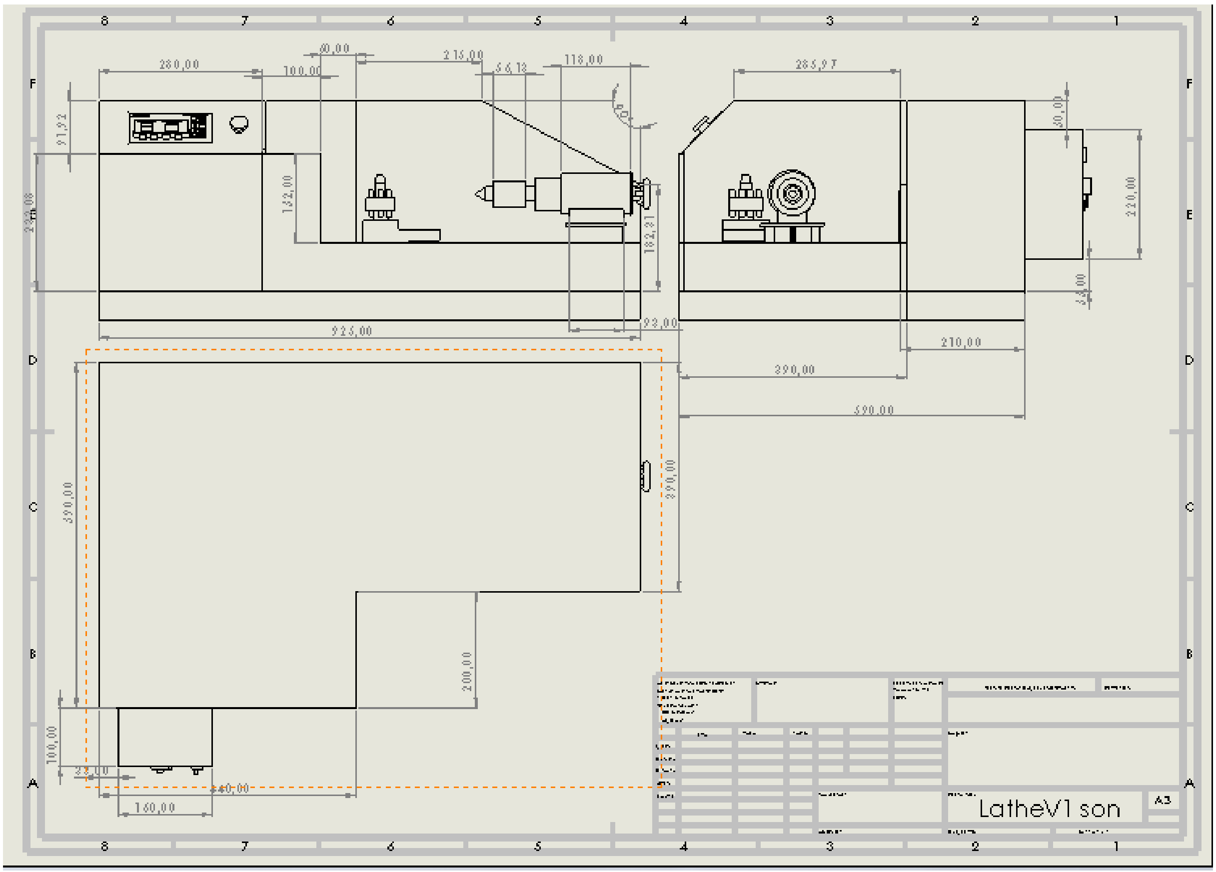


Figure 9.5: Final Version of Mini Lathe Machine

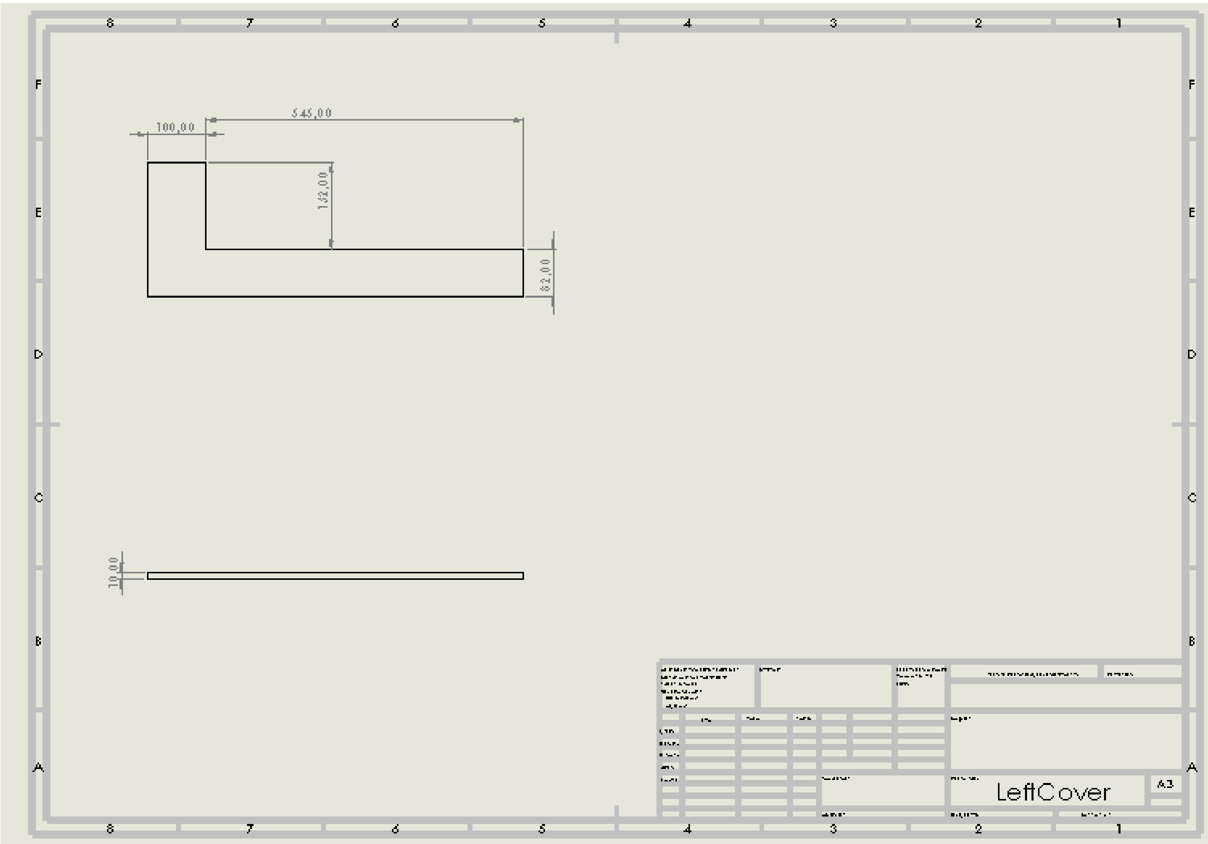


Figure 9.6: Left Cover

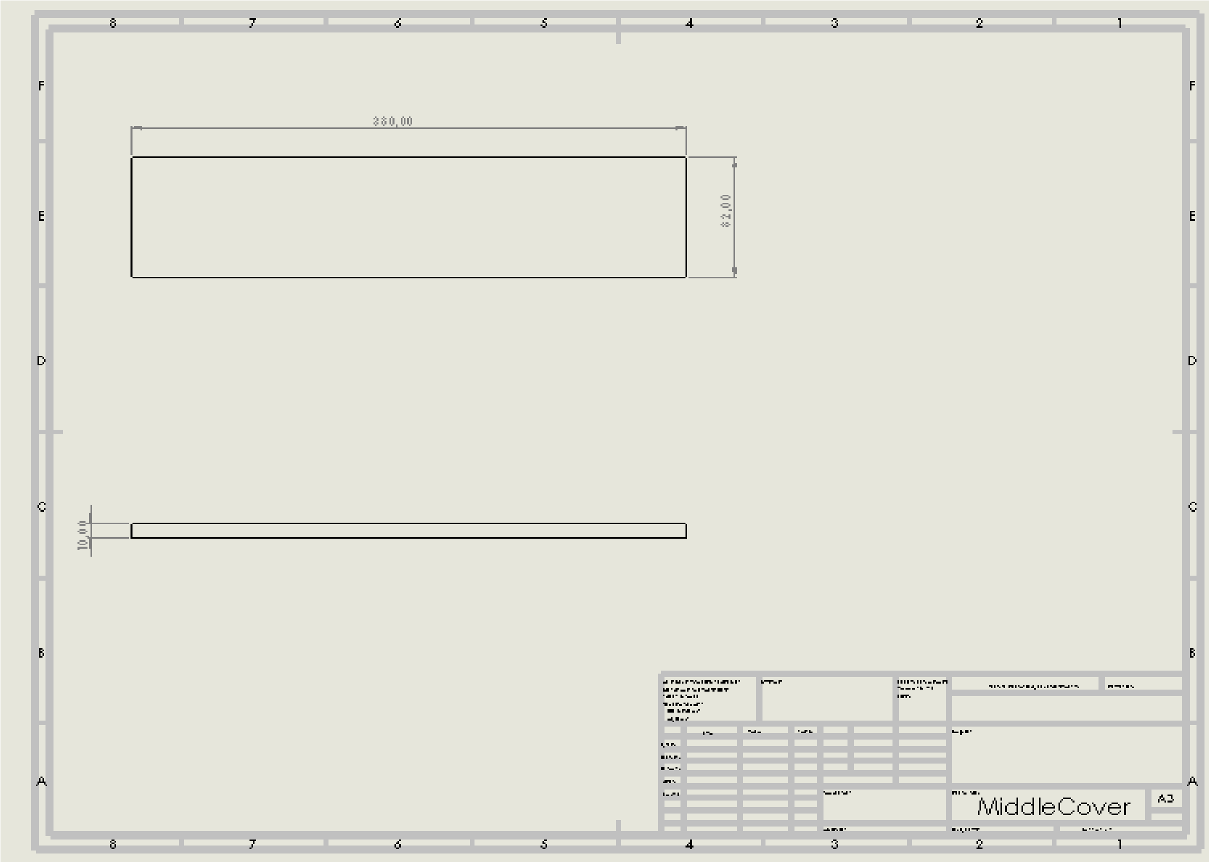


Figure 9.7: Middle Cover

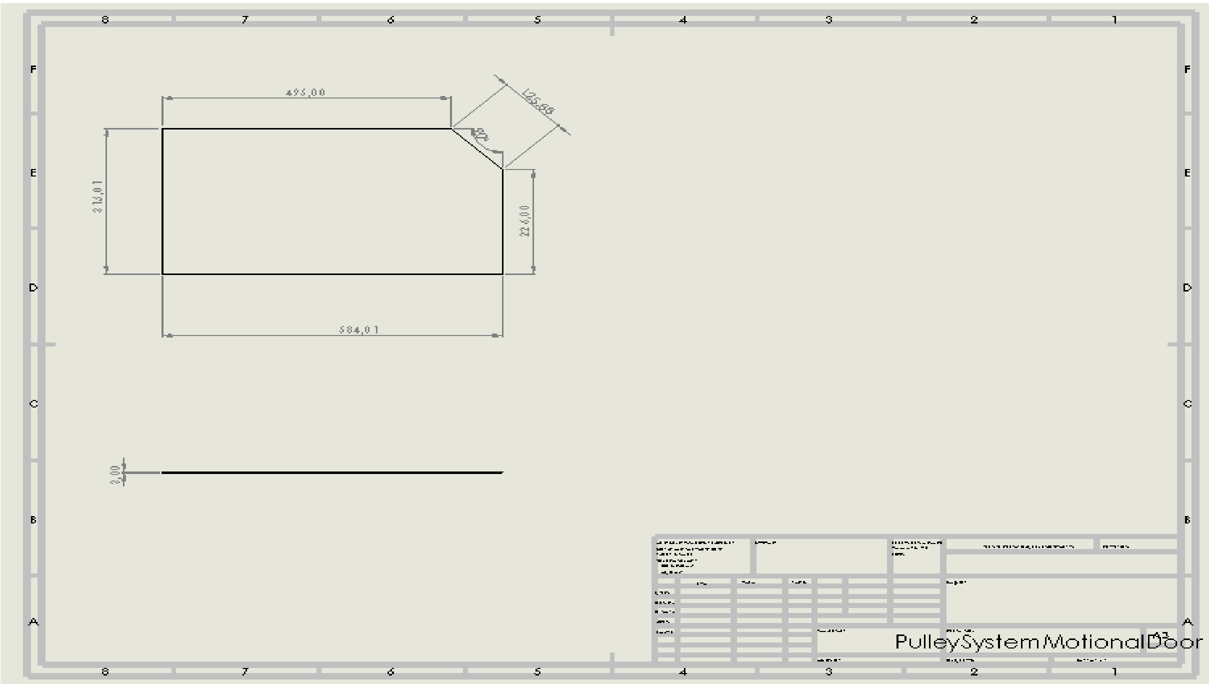


Figure 9.8: Pulley System Motion

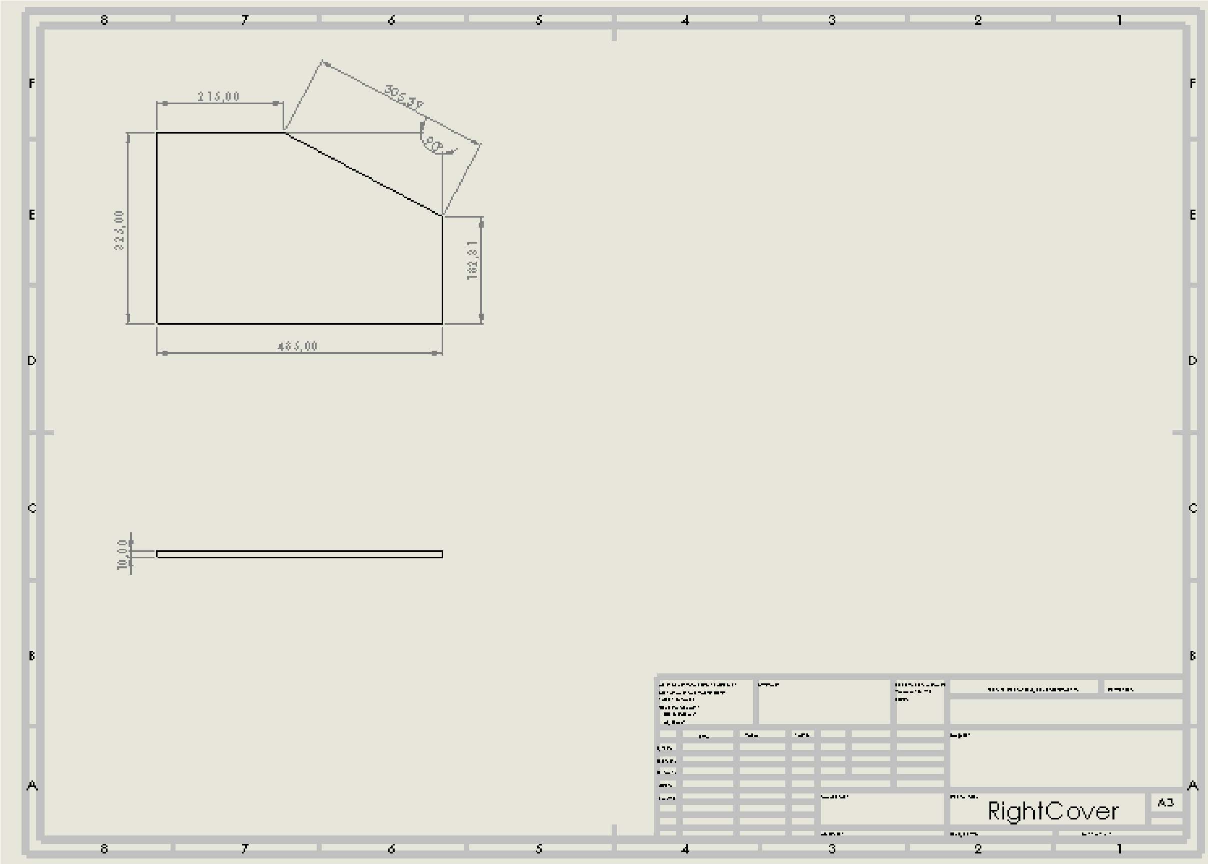


Figure 9.9: Right Cover

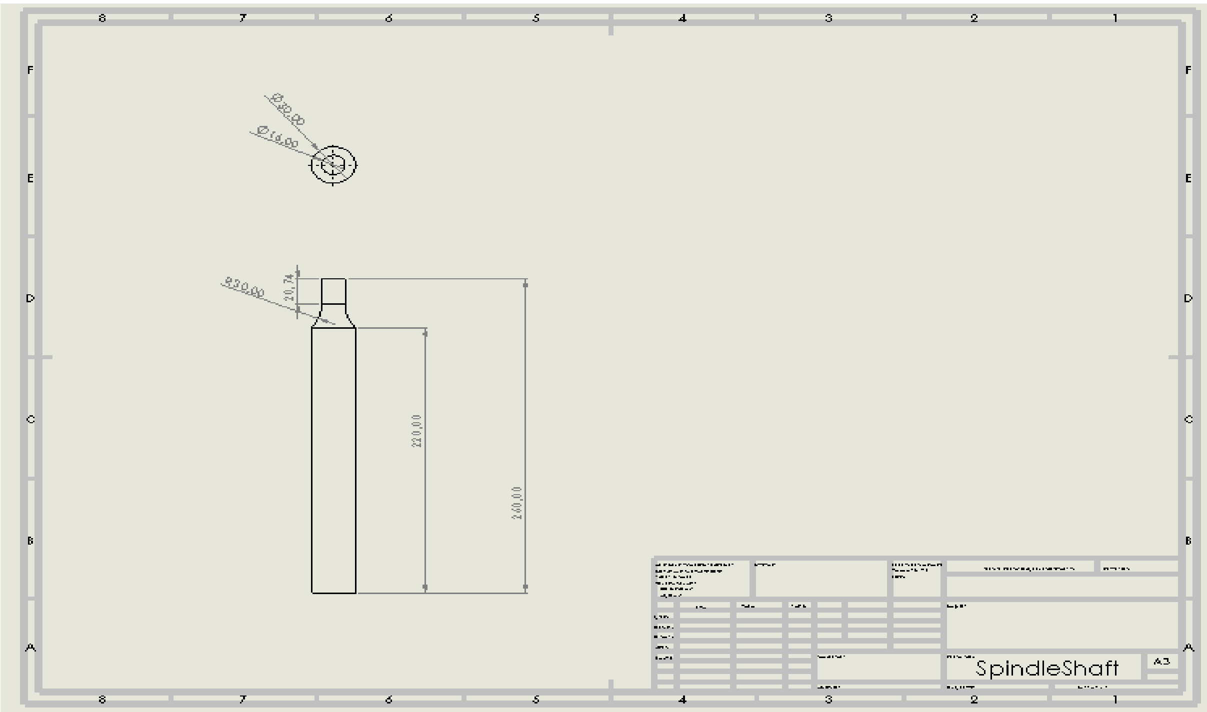


Figure 9.10: Top View of Spindle Shaft

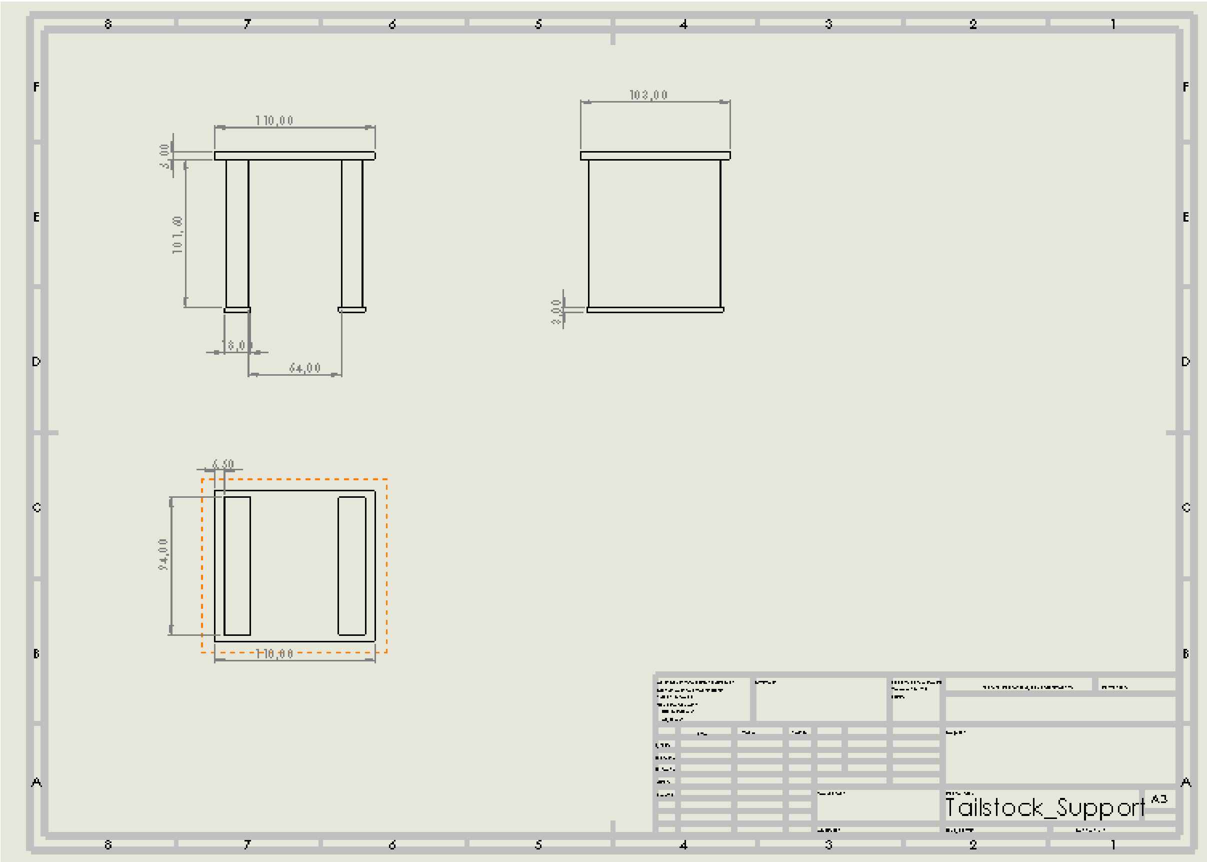


Figure 9.11: Tailstock Support

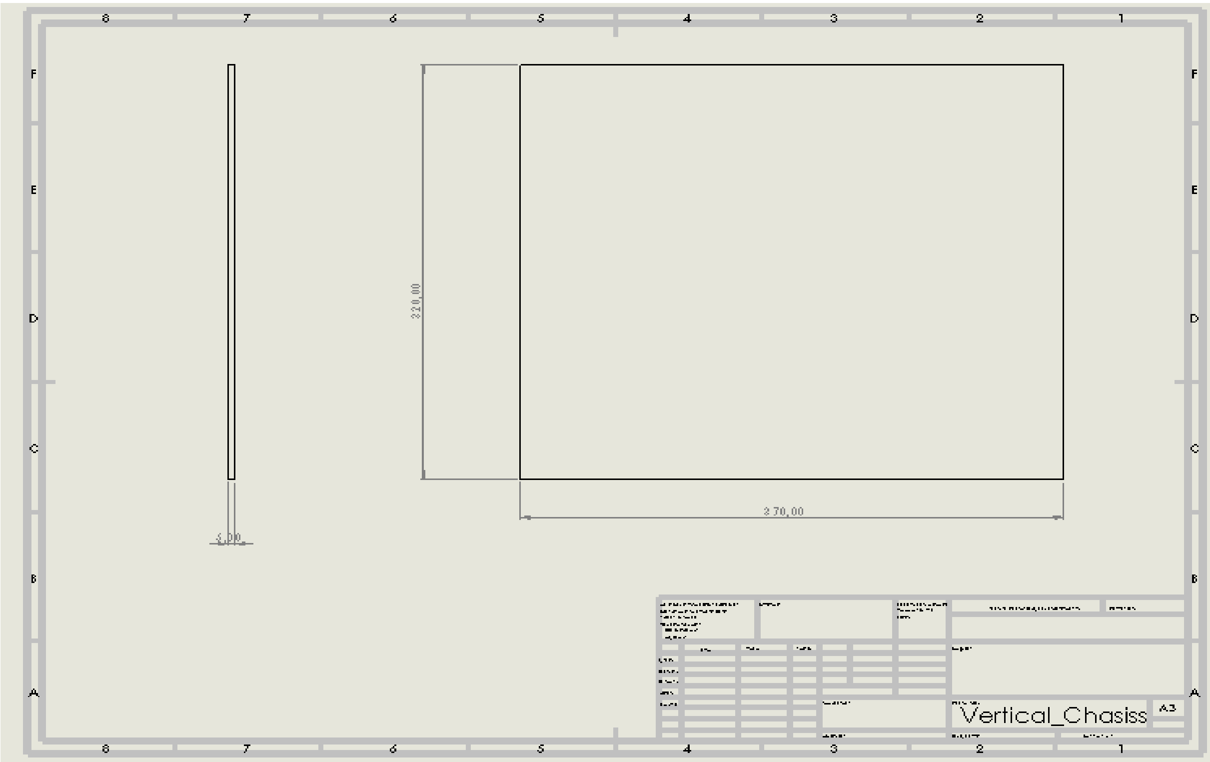


Figure 9.12: Vertical Chassis

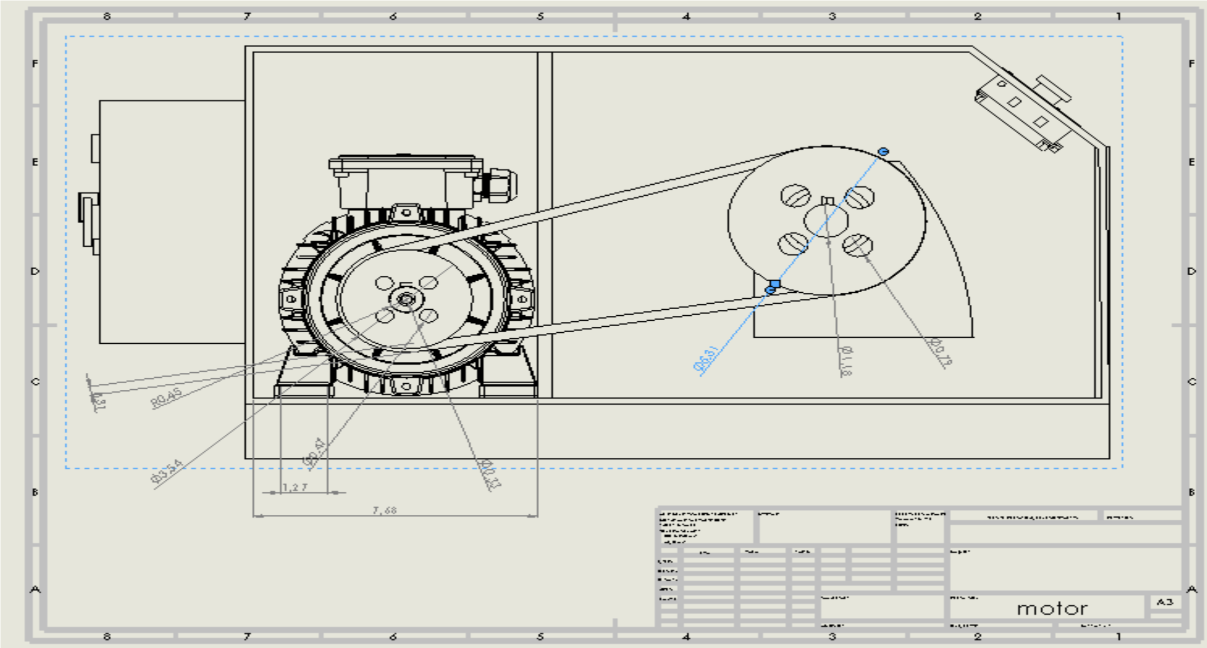


Figure 9.13: V- Belt and Pulley Mechanism

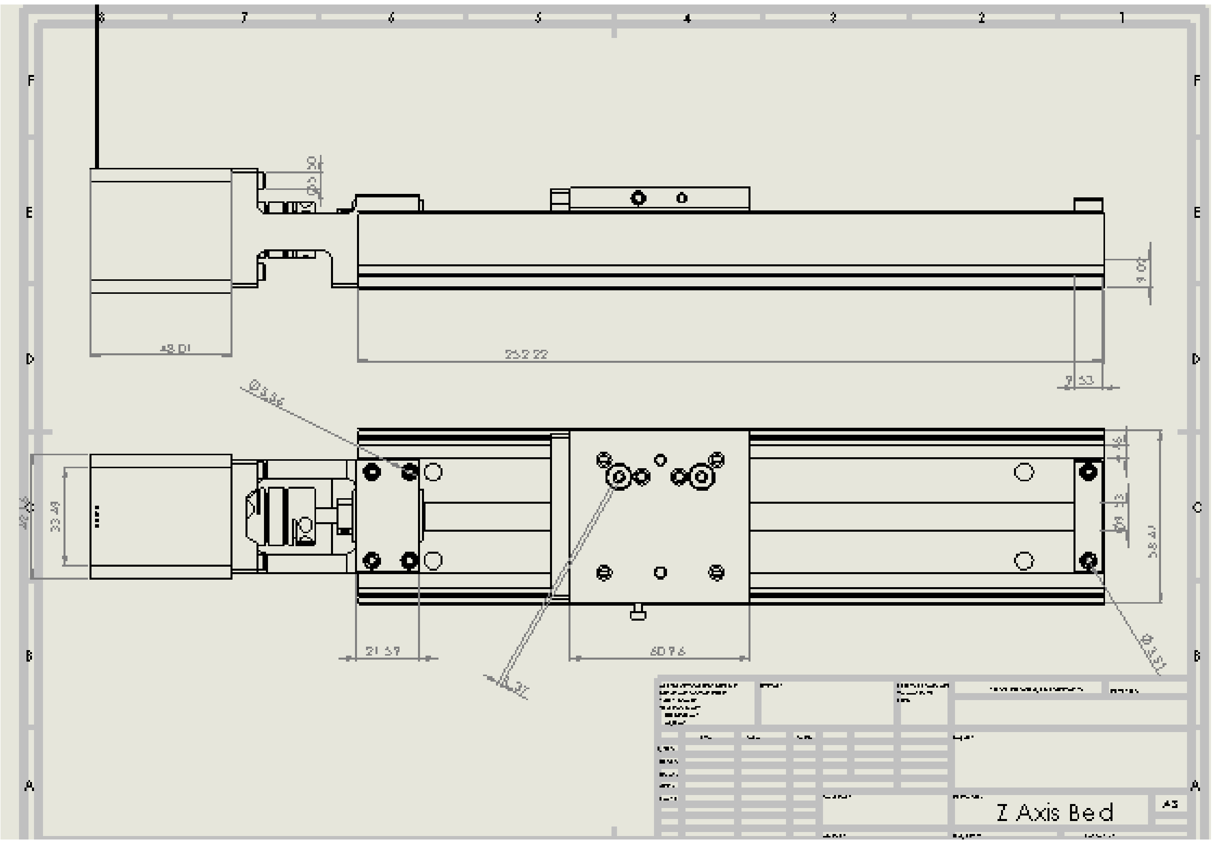


Figure 9.14: Top View of Bed

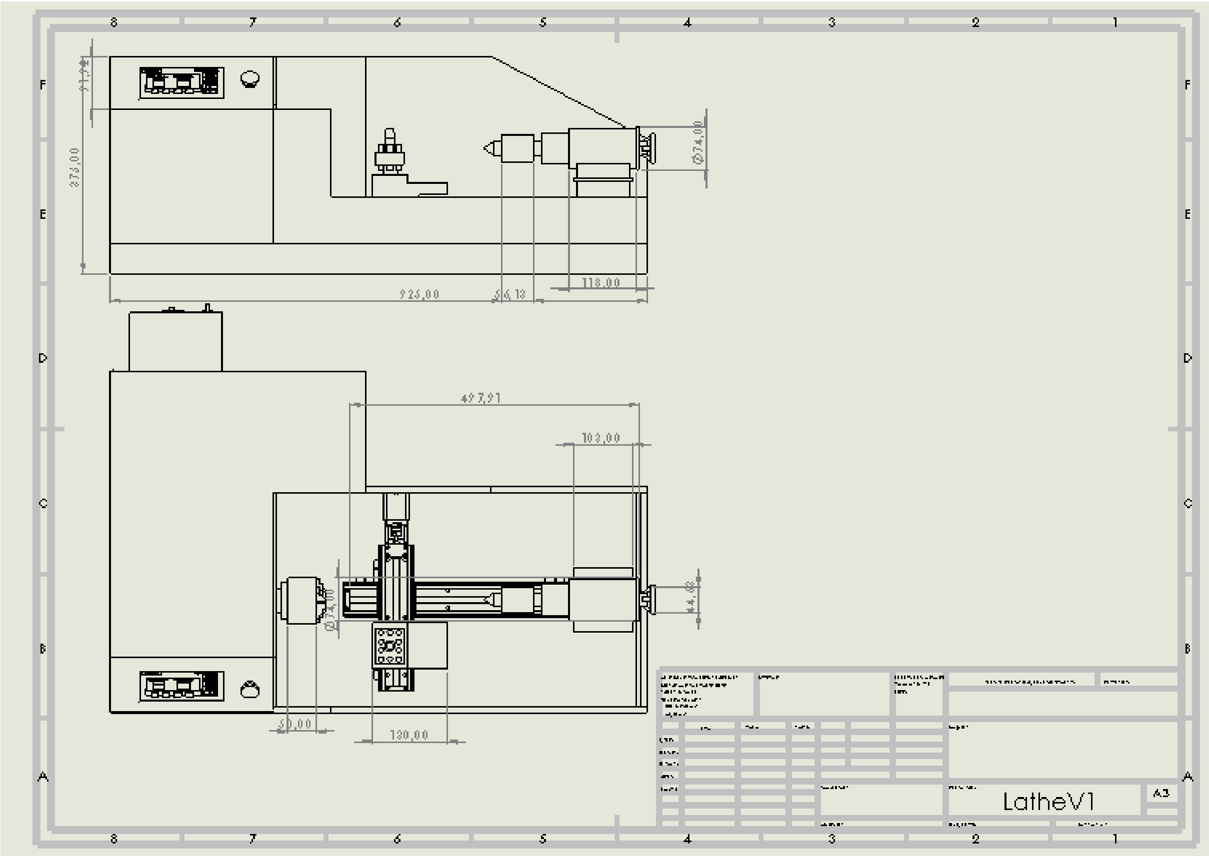


Figure 9.15: Front (Top) & Top (Under) Views of Lathe Machine

10. IMPACT STATEMENT

Used Safety of Work standards were given and explained detailed.

TS EN ISO 23125: Machine tools - Safety - Turning machines

This International Standard specifies the requirements and/or measures to eliminate the hazards or reduce the risks on the following groups of turning machines and turning centres which are defined in 3.1 and designed primarily to shape cold metal by cutting: Group 1: Manually controlled turning machines without numerical control; Group 2: Manually controlled turning machines with limited numerically controlled capability; Group 3: Numerically controlled turning machines and turning centres; Group 4: Single- or multi-spindle automatic turning machines. This International Standard covers the significant hazards listed in clause 4 and applies to ancillary devices (e. g. for workpieces, tools and work clamping devices, handling devices and chip handling equipment, etc.) which are integral to the machine. This International Standard also applies to machines which are integrated into an automatic production line or turning cell in as much as the hazards and risks arising are comparable to those of machines working separately [61]

TS ISO 447: Machine tools;

Direction of operation of controls This second edition, which cancals and replaces ISO 447-1973, incorporates draft Amendment 1. Specifies rules for the direction of operation of controls whose function is to produce movement of controlled machine tool components in one or other of two opposing directions. Its scope does not include controls for components which rotate continuously in the same direction during the normal functioning of the machine. [62]

TS ISO 447: Machine tools; Direction of operation of controls

This second edition, which cancals and replaces ISO 447-1973, incorporates draft Amendment 1. Specifies rules for the direction of operation of controls whose function is to produce movement of controlled machine tool components in one or other of two opposing directions. Its scope does not include controls for components which rotate continuously in the same direction during the normal functioning of the machine [63]

TS EN ISO 6385: Ergonomics principles in the design of work systems

ISO 6385:2016 establishes the fundamental principles of ergonomics as basic guidelines for the design of work systems and defines relevant basic terms. It describes an integrated approach to the design of work systems, where ergonomists will cooperate with others involved in the design, with attention to the human, the social and the technical requirements in a balanced manner during the design process. Users of this International Standard will include executives, managers, workers (and their representatives, when appropriate) and professionals, such as ergonomists, project managers and designers who are involved in the design or redesign of work systems.[64]

TS EN ISO 12100:2010: Safety of machinery - General principles for design - Risk assessment and risk reduction [65]

TS EN 547-1+A1: Safety of machinery; human body measurements; part 1: Principles for determining the dimensions required for openings for whole body access into machinery [66].

TS EN 1005-1+A1: Safety of machinery - Human physical performance - Part 1: Terms and definitions [67]

TS EN 1005-4+A1: Safety of machinery - Human physical performance - Part 4: Evaluation of working postures and movements in relation to machinery [68]

10.1 Standards

Part	Standard or Code	Definition of Code
Spindle Shafts	AISI1045	Meaning of 1045 is that a steel which has low hardenability and intermediate level tensile strength.
Headstock	ISO2727:1973	This standard expresses the dimensions and interchangeability of headstocks that are used in machine for purposes that's special
Chuck	ISO702-4	ISO 702 defines the certain dimensions for interchangeability and corresponding value by basing on these dimensions
Tool Post – Compound	? XA(AXA,BXA etc.) ranking codes	A and B letters in left shows the dimensions of swing over lathe that is dimension from center of chuck to tip of cutting tool
Bearings	ISO76:1987 ISO281:1990	ISO 76:1987 Defines static load of a bearing as surface strength limit when a maximum loas is acting in the middle of the ring ISO281:1990: Standard for dynamic loading conditions where no failure occurs after 1 million revolutions.

Bolt- Nut Connection	**ISO68(Bolt)-ISO1032(Nut)**	**ISO 68 is basics for screw thread dimensions** **ISO 1032: Dimensions of hexagonal nuts**
Feather Key	**DIN 6885**	**Dimensions of hub, key and keyway and detailed explanation of them**
Keyway & Hub	**ISO/R773 – P9 Width Tolerance**	**It explains the tolerance values of keyway width**
V- Belt & Pulley	**ISO 1813** **ASTM D1418**	**Standard about electrical resistance of wide-widthness and hexagonal belts**
Welding	**AWS E6010**	**This welding code must be used when low carbon steel will be welded by electrodes.**

Table 53: Standards of Materials and Explanation of Used Standards

11. CONCLUSION

Machining via CNC Mini Lathe Machine with using cylindrical Aluminium workpiece that has maximum 150MPa yield strength and calculations by basing some design constraints such as maximum depth of cut 2mm and 0.2mm/rev feed rate gives 202.5N tangential force will be applied from cutting tool to workpiece. Motor calculations shows that 2.3kW power will be enough to transmit 3000rpm exits from motor shaft and transmitted as 2000rpm to spindle shaft due to ratio of pulleys in motor-spindle shaft configuration which equals 1.5. Calculations by basing material of workpiece is Aluminium due to design constraints prove that tabletop lathe machine designed may machine Aluminium's that have maximum 150MPa with 2.5 Factor of Safety. These variables are enough to work without giving troubles in configuration that includes motor, v-belt and pulley system instead of gear belt drive, 2 shafts and bearings. Hand-made calculations and configurations was also tested in ANSYS Workbench and ANSYS showed that most critical parts in that's shafts for both motor and spindle may turn and transmit torque in regular basis. Cost analysis within materials showed that no exceeding 7500USD budget and by using these materials, mini-lathe machine works safely, and no failure will occur due to calculated results.

12. RECOMMENDATIONS

- Place of automation panel may be changed. Motor should be placed inside to be more protected by design.
- A cover should be placed to avoid spilling tool around or preventing any incidents because of tool spilling.
- Different materials may be selected to a power screw because selecting only one material may not be best solution every time.
- Tool post and tailstock may be manufactured by ourselves because our cnc lathe machine is integrated with todays technology and materials that have made with iron may be not be best fit because materials made by iron are mostly used in traditional lathe machines.
- Slides may be produced as to be inclined form. Like, degree of slides may increase or decrease by programmable controller. With this concept, more different machining operations may be done.
- Tailstocks may design to be movable by programming. Which means that no handwheel and no bolt-nut connection must not be done. This consideration supplies a whole-body motion rather than towards-backwards motion of live centre such as having more programmable parts instead of traditional and manually remoted matches with the integrated concept of the CNC Mini-Lathe Machine.

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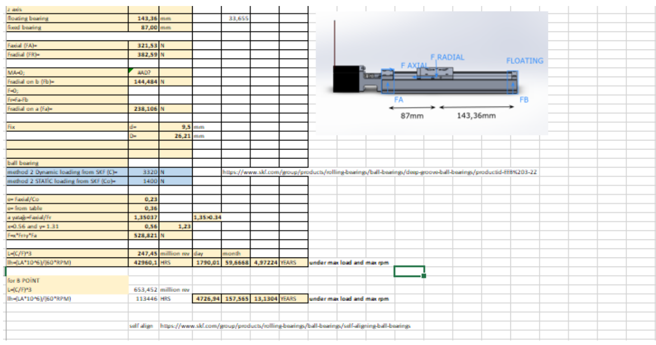
13.APPENDIX

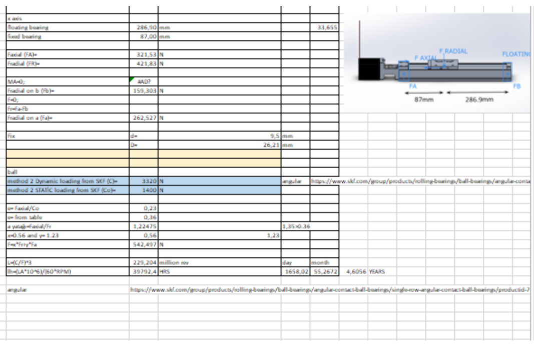
Outline of Schematic A2: Engineering Data

	A	B	C	D	E
1	Contents of Engineering Data			Source	Description
2	Material				
3	AISI 1045 - Cold Rolled			C:	
*	Click here to add a new material				

Properties of Outline Row 3: AISI 1045 - Cold Rolled

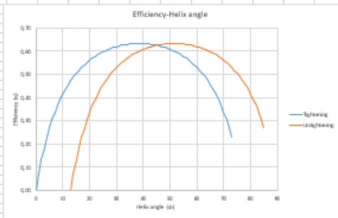
	A	B	C	D	E
1	Property	Value	Unit		
2	Material Field Variables	Table			
3	Density	7870	kg m^-3		
4	Isotropic Elasticity				
5	Derive from	Young'...			
6	Young's Modulus	72	GPa		
7	Poisson's Ratio	0,29			
8	Bulk Modulus	5,7143E+10	Pa		
9	Shear Modulus	2,7907E+10	Pa		
10	Tensile Yield Strength	580	MPa		
11	Tensile Ultimate Strength	680	MPa		





ρ' (degree)	ϕ	tan(ϕ)	tan(ϕ+ρ')	tan(ϕ-ρ')	tightening efficiency tan(ϕ)/tan(ϕ+ρ')	untightening efficiency tan(ϕ-ρ')/tan(ϕ)	efficieny mean	pitch(mm)
13,0	0	0,00	0,23		0,00		0,00	0,00
	1	0,02	0,25		0,07		0,04	0,46
	2	0,03	0,27		0,13		0,07	0,86
	3	0,05	0,29		0,18		0,09	1,20
	4	0,07	0,31		0,23		0,11	1,51
	5	0,09	0,32		0,27		0,13	1,77
	6	0,11	0,34		0,31		0,15	2,01
	7	0,12	0,36		0,34		0,17	2,22
	8	0,14	0,38		0,37		0,18	2,41
	9	0,16	0,40		0,39		0,20	2,58
	10	0,18	0,42		0,42		0,21	2,73
	11	0,19	0,45		0,44		0,22	2,87
	12	0,21	0,47		0,46		0,23	3,00
	13	0,23	0,49	0,00	0,47	0	0,24	3,12
	14	0,25	0,51	0,02	0,49	0,07	0,28	3,68
	15	0,27	0,53	0,03	0,50	0,13	0,32	4,17
	16	0,29	0,55	0,05	0,52	0,18	0,35	4,61
	17	0,31	0,58	0,07	0,53	0,23	0,38	4,99
	18	0,32	0,60	0,09	0,54	0,27	0,41	5,33
	19	0,34	0,62	0,11	0,55	0,31	0,43	5,64
	20	0,36	0,65	0,12	0,56	0,34	0,45	5,91
	21	0,38	0,67	0,14	0,57	0,37	0,47	6,15
	22	0,40	0,70	0,16	0,58	0,39	0,48	6,38
	23	0,42	0,73	0,18	0,58	0,42	0,50	6,58
	24	0,45	0,75	0,19	0,59	0,44	0,51	6,76
	25	0,47	0,78	0,21	0,60	0,46	0,53	6,93
	26	0,49	0,81	0,23	0,60	0,47	0,54	7,08
	27	0,51	0,84	0,25	0,61	0,49	0,55	7,22
	28	0,53	0,87	0,27	0,61	0,50	0,56	7,34
	29	0,55	0,90	0,29	0,62	0,52	0,57	7,46
	30	0,58	0,93	0,31	0,62	0,53	0,57	7,56
	31	0,60	0,97	0,32	0,62	0,54	0,58	7,65
	32	0,62	1,00	0,34	0,62	0,55	0,59	7,74
	33	0,65	1,04	0,36	0,63	0,56	0,59	7,82
	34	0,67	1,07	0,38	0,63	0,57	0,60	7,88
	35	0,70	1,11	0,40	0,63	0,58	0,60	7,95
	36	0,73	1,15	0,42	0,63	0,58	0,61	8,00
	37	0,75	1,19	0,45	0,63	0,59	0,61	8,05
	38	0,78	1,23	0,47	0,63	0,60	0,61	8,09
	39	0,81	1,28	0,49	0,63	0,60	0,62	8,13
	40	0,84	1,33	0,51	0,63	0,61	0,62	8,16
	41	0,87	1,38	0,53	0,63	0,61	0,62	8,18
	42	0,90	1,43	0,55	0,63	0,62	0,62	8,20
	43	0,93	1,48	0,58	0,63	0,62	0,62	8,21
	44	0,97	1,54	0,60	0,63	0,62	0,62	8,22
	45	1,00	1,60	0,62	0,62	0,62	0,62	8,22
	46	1,04	1,66	0,65	0,62	0,63	0,62	8,22
	47	1,07	1,73	0,67	0,62	0,63	0,62	8,21
	48	1,11	1,80	0,70	0,62	0,63	0,62	8,20
	49	1,15	1,88	0,73	0,61	0,63	0,62	8,18
	50	1,19	1,96	0,75	0,61	0,63	0,62	8,16
	51	1,23	2,05	0,78	0,60	0,63	0,62	8,13
	52	1,28	2,14	0,81	0,60	0,63	0,61	8,09
	53	1,33	2,25	0,84	0,59	0,63	0,61	8,05
	54	1,38	2,36	0,87	0,58	0,63	0,61	8,00
	55	1,43	2,48	0,90	0,58	0,63	0,60	7,95
	56	1,48	2,61	0,93	0,57	0,63	0,60	7,88
	57	1,54	2,75	0,97	0,56	[illegible]	[illegible]	[illegible]

$$\eta = \frac{F_{pre} \cdot P}{2 \cdot \pi \cdot M_T}$$



ρ' (degree)	ϕ	tan(ϕ)	tan(ϕ+ρ')	tan(ϕ-ρ')	tightening efficiency tan(ϕ)/tan(ϕ+ρ')	untightening efficiency tan(ϕ-ρ')/tan(ϕ)	efficieny mean	pitch(mm)
13,0	0	0,00	0,23		0,00		0,00	0,00
	3	0,05	0,29		0,18		0,09	1,20
	6	0,11	0,34		0,31		0,15	2,01
	12	0,21	0,47		0,46		0,23	3,00
	15	0,27	0,53	0,03	0,50	0,13	0,32	4,17
	18	0,32	0,60	0,09	0,54	0,27	0,41	5,33
	22	0,40	0,70	0,16	0,58	0,39	0,48	6,38
	26	0,49	0,81	0,23	0,60	0,47	0,54	7,08
	36	0,73	1,15	0,42	0,63	0,58	0,61	8,00

x axis

x axis		
P (threads per inch)= 1/pitch diameter	5	
lead screw length=	417 mm=10.43 inch	
pitch =	5 mm (0.2 inch)	
Weight	94 lb (43 kg)	
efficiency	0.43	
J total		
Jw=	94 /(386*0,43))*(1/(2*pi*P))^2	
Jw=	0,000573813	lb-in-sec'2
j screw	0,00001963	lb-in-sec'2
density= 0.28 lb/in^3 (steel)		
Jtotal=	0,000593442	lb-in-sec'2
Taccel=	0,023863757	lb*in
Ftotal=	18,8	lb
Trun=	0,60	lb*in
Tmotor=Taccel+Trun	0,62	lb*in
Tmotor=Taccel+Trun	10,56	oz*in
Tmotor=Taccel+Trun	0.074	n*m
J motor	0,37	oz in^2
	0,00006	lb-in-sec'2
Jtotal/j motor<10	9,890692547	

z axis		
P=threads per inch	5	
lead screw length=	265mm=10.43 inch	
pitch =	5 mm (0.2 inch)	
weight=	85 lb (39 kg)	
J total		
Jw=	85 /(386*0,43))*(1/(2*pi*P))^2	
Jw=	0,0005189	lb-in-sec'2
j screw	0,0000125	lb-in-sec'2
density= 0.28 lb/in^3 (steel)		
Jtotal=	0,000531	lb-in-sec'2
Taccel=	0,021367	lb*in
Ftotal=	17,0	lb
Trun=	0,54	lb*in
Tmotor=Taccel+Trun	0,56	lb*in
Tmotor=Taccel+Trun	8,96	oz*in
Tmotor=Taccel+Trun	0.063	n*m
J motor	0,37	oz in^2
	0,00006	lb-in-sec'2
Jtotal/j motor<10	8,855818461	

max speed=	V/(2*L*IMAX*SPR)	
	0,011111111	rev/sec
p max=	Imax*V	
	96	watt

https://www.allaboutcircuits.com/tools/stepper-motor-calculator/

1,332

IMAX=	2	A
L=	2,7	Mh
voltage=	48	v
steps/rev=	400	steps
steps/rev=		

my card = 400 pulses/rev	**400**	**pulses/rev**	basic step angle=1.8 degree	360 degree/1.8 degree=200 step per revolution		stroke=
distance=	**10**	**mm**				
feed speed=	**6,66**	**mm/sec**	1sn 6.66mm 1mm		45	
alumunium 10 mm = 1,5 seconds	**1,50**	**seconds**				
my screw pitch= (5 mm)	**5,00**	**mm**	0.5 pitch have accuracy(netlik)			
10mm*(1 rev/5 mm)*(400 pulses /rev)	800	pulses	10 mm için toplam gereken pulses			
800 pulses /1,5 seconds	533	pulses/sec	10 mm ilerleme için gereken hız	F tangential	208.5N	safety =1,54
(5 mm/1 rev)*(1 rev/400 pulses)=accuracy	0,0125	(accuracy)	ne kadar yüksek olursa düşük resolution (ne ka	F feed force	88.5N	
10 mm*(1rev/5 mm)	2	rev	10mm yol alması için	Resultant force	226.5N	
(2 revs /1.5 sec)	1,33	rev/sec		Resultant force with safety	350 N	
1.3 revs/sec convert to rpm	80	rpm		Weight	35,67	kg
1.25*h=800 pulses	1,25			Extra Unexpected Weight	2,13	kg
h=	640	pulses/sec		Neva 17048 Motor	0,7	kg
640(pulses/sec)*1/400(rev/pulses)	1,60	rev/sec		Up Power Screw System	1	kg
1 sec =	1,6	rev/sec		Component	3	kg
1.6 rev/sec =	96	rpm		Cutting Tools	0,5	kg
Torque calculations				Total Weight	43	kg
my weight is 100 lb (43kg)	**100**	**lb**				
1 g = 386.4 in/s^2	386,4	in/s^2				
(100 lbs /1 g)=payload mass	0,259	(lb-s^2)/in				
(0.259*0.001)= jcarriage	0,000259	lb-s^2-in	to	0,00410	oz-in-s^2	
j screw=jcoupler+jlead=from table	0,020	lb-in^2	to	0,00083	oz-in-s^2	
my motor rotor inertia(Nema17048)	**0,37**	**0z-in^2**	to	0,00095	oz-in-s^2	
convert to oz-in-s^2						
jtt=jmotor+jgearbox+(jcoupler +jlead screw+jcarriage)/i'2	0,00588	0z-in-s^2		2,034482759		
Taccel = jttl*(Δ speed/Δtime)	0,037632	oz-in-rev	1.4 rev/sec'a 0.25 sec ulaşır			
convert to oz-in (oz-in-rev*2*pi/rev)	0,236449382	0z-in	times 2 pi (360 degree)			
Torque=Taccel+Trun	0,236449382	0z-in				
Tmeasured	1,655145677	0z-in				
margin max torque	58	0z-in	to	**0,4**	**n*m**	
holdining max torque (Nema 17048)=	**0,59 n*m**					
karşılaştırma(jscrew+jcarriage)/jmotor	9.,8		Per a rule of thumb, the load to motor inertia ratio should be kept below 10.			

```
clc; clear;
Ba = input('Enter the friction angle value (you can only enter 15 btw 30) : ');
ar = input('Enter the rake angle value ( you can only enter 5 btw 7) : ');
Qc = (pi/4)-((Ba*0.0174532925)-(ar*0.0174532925));
Qcc = Qc*57.2957795;
disp('The shear angle value is ');
disp(Qcc);
b = input('Then, enter the depth of cut value (you can only enter 0.5 mm btw 2 mm) [b] : ');
h = input('Enter the uncut chip thickness value (you can only enter 0.05mm btw 0.2) [h] : ');
shearF = input('Enter the maximum Shear Strength value : ');
Ftc = b*h*shearF*(cos((Ba-ar)*pi/180)/(sin(Qc)*cos((Ba+Qcc-ar)*pi/180)));
Ffc = b*h*shearF*(sin((Ba-ar)*pi/180)/(sin(Qc)*cos((Ba+Qcc-ar)*pi/180)));
disp('Tangential Force is');
disp(Ftc);
disp('Feed Force is');
disp(Ffc);
FS = 2;
r = input('Enter the workpiece radius : ')
Torque = Ftc * r * FS;
    disp('Applied Torque is')
    disp(Torque);
    Power_W= input('Enter the spindle speed (RPM) : ');
    Power = (Torque*Power_W*2*pi)/60;
    disp('Required power is ');
    disp(Power);
```

MECH/MECT4902 DESIGN PROJECT MEMO

Year: 2021 Semester: Fall

Team No: 6

Advisor: Assist. Prof Dr. Umut Karagüzel

Project Title: Design of A Tabletop Lathe

Meeting No: 1 Date: 17.10.2021.

Attendants: Ege Kubilay, Ahmet Furkan Tüfekçi, Caner Sözen

Discussed Topics:

- Making research wanted about lathe machines were requested by our advisor and results were showed and talked about it.

Work to be done until next meeting and responsible team member(s):

- Benchmarking about lathe machines will be done to next meeting.
- Preparation of Gannt Charted requested.

Prepared By: Ege Kubilay

MECH/MECT4902 DESIGN PROJECT MEMO

Year: 2021 Semester: Fall

Team No: 6

Advisor: Assist. Prof Dr. Umut Karagüzel

Project Title: Design of A Tabletop Lathe

Meeting No: 2 Date: 21.10.2021

Attendants: Ege Kubilay, Ahmet Furkan Tüfekçi, Caner Sözen

<u>Discussed Topics:</u>

- Gannt Chart was presented and required changes were talked.
- Results of benchmarking analysis were showed. Afterwards, budget constraint determined and also, mechanical constraints which is all project will be done due to workpiece that is made of aluminum.

Hand drawing about our thought lathe machine were requested by our advisor

<u>Work to be done until next meeting and responsible team member(s):</u>

- Hand drawing about our thought lathe machine were requested by our advisor
- Determining critical points in lathing operation and finding the critical points requested to be researched and found

Prepared By: Ege Kubilay

MECH/MECT4902 DESIGN PROJECT MEMO

Year: 2021 Semester: Fall

Team No: 6

Advisor: Assist. Prof Dr. Umut Karagüzel

Project Title: Design of A Tabletop Lathe

Meeting No: 3 Date: 28.10.2021

Attendants: Ege Kubilay, Ahmet Furkan Tüfekçi, Caner Sözen

<u>Discussed Topics:</u>

- Shear Force and Analysis of other forces
- Articles that will be useful through the timeline of project
- Formulas about cutting force in the book of Automotive Mechatronics
- Width of tool post length of post per revolution

<u>Work to be done until next meeting and responsible team member(s):</u>

- Equivalent force calculation
- Friction angle calculation

Prepared By: Ege Kubilay

MECH/MECT4902 DESIGN PROJECT MEMO

Year: 2021 Semester: Fall

Team No: 6

Advisor: Assist. Prof Dr. Umut Karagüzel

Project Title: Design of A Tabletop Lathe

Meeting No: 4 Date: 01.11.2021

Attendants: Ege Kubilay, Ahmet Furkan Tüfekçi, Caner Sözen

<u>Discussed Topics:</u>

- Results of equivalent cutting force and friction angle

<u>Work to be done until next meeting and responsible team member(s):</u>

- Development of design by basing hand drawing

Prepared By: Ege Kubilay

MECH/MECT4902 DESIGN PROJECT MEMO

Year: 2021   Semester: Fall

Team No: 6

Advisor: Assist. Prof. Dr Umut Karagüzel

Project Title: Design of A Tabletop Lathe

Meeting No: 5   Date: 08.11.2021

Attendants: Ege Kubilay, Ahmet Furkan Tüfekçi, Caner Sözen

Discussed Topics:

- Safety Factor of the Lathe Machine
- Power Screw System and selection of screw and motor connection

Work to be done until next meeting and responsible team member(s):

- Spindle Motor will be selected
- Free Body Diagram will be draw
- SolidWorks drawing on beginner level

Prepared By: Ege Kubilay

MECH/MECT4902 DESIGN PROJECT MEMO

Year: 2021 Semester: Fall

Team No: 6

Advisor: Assist. Prof. Dr Umut Karagüzel

Project Title: Design of A Tabletop Lathe

Meeting No: 6 Date: 11.11.2021

Attendants: Ege Kubilay, Ahmet Furkan Tüfekçi, Caner Sözen

<u>Discussed Topics:</u>

- Detailed discussion about Power Screw system
- Material Selection about main parts of lathe machine

<u>Work to be done until next meeting and responsible team member(s):</u>

- Development and detailed version of SolidWorks 3D drawing

Prepared By: Ege Kubilay

MECH/MECT4902 DESIGN PROJECT MEMO

Year: 2021 Semester: Fall

Team No: 6

Advisor: Assist. Prof. Dr Umut Karagüzel

Project Title: Design of A Tabletop Lathe

Meeting No: 6 Date: 18.11.2021

Attendants: Ege Kubilay, Ahmet Furkan Tüfekçi, Caner Sözen

Discussed Topics:

- Material Selection for detailed reasons
- Pugh Matrix for Materials
- SolidWorks 3D drawing

Work to be done until next meeting and responsible team member(s):

- Revision of Power Screw calculations

Prepared By: Ege Kubilay

MECH/MECT4902 DESIGN PROJECT MEMO

Year: 2021 Semester: Fall

Team No: 6

Advisor: Assist. Prof. Dr Umut Karagüzel

Project Title: Design of A Tabletop Lathe

Meeting No: 7 Date: 02.12.2021

Attendants: Ege Kubilay, Ahmet Furkan Tüfekçi, Caner Sözen

<u>Discussed Topics:</u>

- Critical parts were shown in motor part
- V- belt and pulley system and reasons for selecting
- Calculations of Power Screw System and feedbacks are given.
- Usage of bearing in the system

<u>Work to be done until next meeting and responsible team member(s):</u>

- Calculations will be done from start

Prepared By: Ege Kubilay

MECH/MECT4902 DESIGN PROJECT MEMO

Year: 2021 Semester: Fall

Team No: 6

Advisor: Assist. Prof. Dr Umut Karagüzel

Project Title: Design of A Tabletop Lathe

Meeting No: 9 Date: 09.12.2021 (Interim Presentation Date)

Attendants: Ege Kubilay, Ahmet Furkan Tüfekçi, Caner Sözen

<u>Discussed Topics:</u>

- Who will be the presenter for tonight?
- Topics to be discussed in interim presentation
- Feedbacks about presentation file.

<u>Work to be done until next meeting and responsible team member(s):</u>

Prepared By: Ege Kubilay

MECH/MECT4902 DESIGN PROJECT MEMO

Year: 2021 Semester: Fall

Team No: 6

Advisor: Assist. Prof. Dr Umut Karagüzel

Project Title: Design of A Tabletop Lathe

Meeting No: 10 Date: 16.12.2021

Attendants: Ege Kubilay, Ahmet Furkan Tüfekçi, Caner Sözen

<u>Discussed Topics:</u>

- Remained parts and calculations of all team members

<u>Work to be done until next meeting and responsible team member(s):</u>

- ANSYS WorkBench Analysis

Prepared By: Ege Kubilay

MECH/MECT4902 DESIGN PROJECT MEMO

Year: 2021 **Semester: Fall**

Team No: 6

Advisor: Assist. Prof. Dr Umut Karagüzel

Project Title: Design of A Tabletop Lathe

Meeting No: 11 **Date: 23.12.2021**

Attendants: Ege Kubilay, Ahmet Furkan Tüfekçi, Caner Sözen

<u>Discussed Topics:</u>

- Calculations of Motor Shaft
- ANSYS WorkBench analysis results
- Non-clear points in calculations of bed and slides

<u>Work to be done until next meeting and responsible team member(s):</u>

- Keyway Calculations
- ANSYS WorkBench Analysis of Spindle Shaft and correction of static analysis spindle shaft

Prepared By: Ege Kubilay

MECH/MECT4902 DESIGN PROJECT MEMO

Year: 2021 Semester: Fall

Team No: 6

Advisor: Assist. Prof. Dr Umut Karagüzel

Project Title: Design of A Tabletop Lathe

Meeting No: 12 Date: 31.12.2021

Attendants: Ege Kubilay, Ahmet Furkan Tüfekçi, Caner Sözen

<u>Discussed Topics:</u>

- Keyway calculations
- Final version of ANSYS WorkBench Analysis
- Remained parts and calculations

<u>Work to be done until next meeting and responsible team member(s):</u>

- Strength of materials calculations of bed

Prepared By: Ege Kubilay

MECH/MECT4902 DESIGN PROJECT MEMO

Year: 2022 Semester: Fall

Team No: 6

Advisor: Assist. Prof. Dr Umut Karagüzel

Project Title: Design of A Tabletop Lathe

Meeting No: 13 Date: 06.01.2022

Attendants: Ege Kubilay, Ahmet Furkan Tüfekçi, Caner Sözen

<u>Discussed Topics:</u>

- Strength of Materials calculations of bed
- Keyway Calculations

<u>Work to be done until next meeting and responsible team member(s):</u>

- Revised version of Keyway Calculations
- Weldment Calculations

Prepared By: Ege Kubilay

MECH/MECT4902 DESIGN PROJECT MEMO

Year: 2022 Semester: Fall

Team No: 6

Advisor: Assist. Prof. Dr Umut Karagüzel

Project Title: Design of A Tabletop Lathe

Meeting No: 14 Date: 13.01.2022

Attendants: Ege Kubilay, Ahmet Furkan Tüfekçi, Caner Sözen

Discussed Topics:

- Weldment Calculations
- Preparation of report

Work to be done until next meeting and responsible team member(s):

Prepared By: Ege Kubilay