Methodology for Design and Selection of Components of 3-Axis CNC Granite Carving Machine Tool

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In partial fulfilment of the requirements for
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by

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CERTIFICATE

I hereby declare that the thesis entitled **"Methodology for Design and Selection of Components for 3 axis Granite Carving Machine Tool"** is an authentic record of my work carried out as requirements for the award of the degree of **Master of Engineering** in **CAD/CAM Engineering** at **Thapar University, Patiala** under the supervision of **Mr. Ravinder K. Duvedi & Mr. Sandeep Sharma**, Assistant Professor, Mechanical Engineering Department, Thapar University, Patiala during the period July, 2012 to July, Year . The matter embodied in this report has not been submitted in partial or full to any other university or institute for the award of any degree.

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It is certified that the above statement made by the student is correct to the best of my/our knowledge and belief.

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Nomenclature

L_T : Length of tool

D: Diameter of tool

d: Depth of cut

f: Feed rate

Z: No. of teeth

f_Z: Feed per rotation

w: Width of cut

V_c: Cutting speed

N: Spindle rpm

L_X: Length of X, axis traverse respectively

α: Maximum acceleration for X, Y and Z-axis drives

X : Nominal X-axis position

Y: Nominal Y-axis position

Z: Nominal Z-axis position

K_C: Maximum Specific cutting energy considered for hardest granite

Q: Material removal rate

P_f: Spindle power required

F_T: Tangential cutting force

K': Geometry factor of the tip of the tool

T_P: Spindle torque required for plunging

P_P: Spindle power required for plunge milling

F_P: Thrust Force in plunge milling

F_m: Maximum cutting force out of face and plunge milling

P_S: Maximum spindle Power required for face or plunge milling

L_f: Loss factor

P_{max}: Maximum Spindle power required

$P_{standard}$ Nearest Standard spindle power available

m_s: Mass of standard spindle

σ_b: Bearing stress of selected Bracket material

B_t: Thickness of bracket plate

M_{bkt}: Mass of designed Bracket

W_{bkt}: Weight of the bracket

d_b: Diameter of clamping holes on bracket calculated from different cutting conditions

σ_{bcal}: Analytically calculated bearing stress from Bracket

m_{asm1}:Total mass of Spindle and Bracket together (sub assembly-1)

W_{asm1} :Total weight of Spindle and Bracket together (sub assembly-1)

W_i: Inertia load of spindle only in x-a xis direction

W: Total load acting on each bolt of sub assembly-1

N_b: Number of bolt used to clamp spindle + bracket on Z-axis drive plate

W_{dt}: Direct Tensile load on each bolt

L_v: Vertical distance of bolts from twisting edge

L: Distance of the force vector from the twisting edge

w_b: Load on the bolt per unit distance

W_t: Tensile load on bolt

W_{mtb}: Maximum tensile load on group of bracket clamping bolts

σ_{bolt}: Tensile stress for bolt material

S_b: Factor of safety for clamp bolts

d_{bx}: Diameter of the bracket clamping bolt for cutting in x-axis

e_y: Eccentricity of the load when calculating bolt diameter for cutting in Y-axis

W_s: Direct shear load on each bolt

M_{ie}: Turning moment produced by the load W due to eccentricity e_y

L_c: Distance of bolt from centroid of sub assembly 1

S_{fb}: Secondary shear load on bolt

R_s: Resultant shear load on bolt

R_{smax}: Maximum Resultant Shear load on bolt

τ_{bolt}: Allowable shear stress for bolt material

d_{by}: Diameter of the bracket clamping bolt for cutting in Y-axis

W_P: Load on clamping bolts in plunge cutting

e_Z: Eccentricity of load in plunge cutting

W_{te}: Equivalent tensile Load

W_{se}: Equivalent shear Load

d_{btz}: Diameter of the bracket clamping bolt in tension in plunge cutting

d_{bsz}: Diameter of the bracket clamping bolt in shearing in plunge cutting

M_X, M_Y, M_Z Maximum bending moment in guide way in X, Y and Z axis direction respectively

l_{tx}: Distance for force in X-axis cutting and point of moment in guide way

l_{ty}: Distance for force in Y-axis cutting and point of moment in guide way

l_{pz}: Distance for force in Z-axis cutting and point of moment in guide way

M_{Xa}, M_{Ya}, M_{Za}: Allowable maximum bending moment in guide way in X, Y and Z axis direction respectively

L_{fs}: Distance out from ball screw that force is being applied

l_{bb}: Center to center spacing of bearings

f_{bb}: Resultant force on bearings by ball screw

$F_{\mu b}$: Friction force on each bearing

μ_b: Coefficient of friction of round shaft

M_{Zback}:Mass of the back plate of Z-axis of round shaft

M_{asm2} :Mass of subassembly2 comprising of mass of subassembly 1 and back plate of Z

N_{mx}, N_{my}, N_{mz}: Motor's rated rotational speed in X, Y and Z direction respectively

S_x, S_y, S_z : Maximum Linear Speed of slide in X, Y and Z direction respectively

l_x, l_y, l_z : Lead of screw in X, Y and Z slide respectively

t_{acc}: Acceleration time

$\alpha_x, \alpha_y, \alpha_z$: Acceleration of slide in X, Y and Z direction respectively

$F_{fRx}, F_{fRy}, F_{fRz}$: Guide surface resistance in X, Y and Z slide respectively

F_{ax}, F_{ay}, F_{az} : Axial load on ball screw in X, Y and Z direction respectively

η_{sc}: Sliding screw efficiency

T_{tx}, T_{ty}, T_{tz}: Driving torque to obtain n X, Y and Z direction respectively

d_{bsc}: Diameter of ball screw according to maximum allowable stress

σ_{max} : Maximum permissible von mises stress

d_{bav}: Standard ball screw diameter available

L_{fx}, L_{fy}, L_{fz}: Length of ball screw in X, Y and Z direction respectively

J_1: Mass moment of Inertia of pinion

J_2: Mass moment of Inertia of gear

J_3 : Mass moment of Inertia of ball screw

J_4 : Mass moment of Inertia of work and table

J_t : Total Mass moment of Inertia

P_{rev} : Pitch of ball screw in rev/mm

ρ : Density of ball screw material

N_b : Number of bolt used to clamp spindle + bracket on Z-axis drive plate

$r_{bx}, r_{by} r_{bz}$,: Radius of ball screw rod in X, Y and Z direction respectively

μ : Co-effcient of friction of ball screw

e : Transmission efficiency of Ball screw

T_α: Acceleration torque

T_g: Gravity torque

T_f Friction torque

T_{total}: Total torque

J_a: Mass moment of Inertia of driving side

J_l: Mass moment of Inertia of load side

J_K: Mass moment of Inertia of coupling

J_{motor} :Mass moment of Inertia of motor

J_{bs}: Mass moment of Inertia of ball screw

J_{schl}: Mass moment of Inertia of slide and work piece

Abstract

This thesis report is an endeavour towards developing a methodology for the design and selection of standard and non-standard components for CNC machine tool structure for ornamental carving operations on granite stone. The ornamental carving was previously a craftsmen job but with the passage of time CNC machines replaced them. These machines are capable of producing much high products within quite less span of time. The objective of this study was to develop a methodology for design and selection of CNC machine tool components. This whole work is basically categorised as a 3 step process. Firstly the empirical relationships in which a methodology was developed for the design and selection of standard components for machine tool. Thus the standard components such as ball screw, round guideways and linear motors were selected from some standard manufacturers. The details of this chapter are explained in the 3rd chapter. Secondly the work is done for the design of supporting non-standard elements. The term support element refers to those non-standard components which were designed exclusively for this particular machine tool structure, with the purpose of supporting and providing stiffness to the standard components such as the bracket was designed for supporting the spindle motor. Other such components are the double c-sectioned channel for supporting the Y axis sub-assembly. Then in the third step, all the components designed and selected were analyzed for stability and stiffness under the application of static and dynamic forces with the help of ANSYS 10 (FEM tool).The results of this step are explained in the 4rd chapter of the report. Finally a conclusion and the future scope of the work have been discussed.

Chapter 1

INTRODUCTION

Ornamental carving from the ancient times has been done on materials like wood, marble. Granite and other stones etc. with the help of hand tools like hammers and chisels etc. Later on some basic machines like lathe and milling centres also came into use, but manual interference was required to a large extent. With the increasing void between the production and demand , the need for a fast and accurate system was realised that could do the needful and then came into light the CNC machines. Now a days the CNC machine tool ranges from 2 to 5 or 3+2 axis but here the study is specifically focused on the structural refinement of 3-axis machine tool structure for granite sculpturing. Presently there are number of 3-axis CNC machine tool structures available for granite sculpturing but the aim of this study is to optimally design a machine tool structure, which is neither over designed nor under designed in order to withstand the loads acting on it. Moreover the study aims at designing a reconfigurable for example it could be made from standard components readily available in the market. Last but not the least, for someone to pursue the study of design for these structures, the design process is not readily available or revealed. One needs to go through a large amount literature. So a comprehensive method of designing a machine tool structure is presented here. The objective of this study is to design a 3-axis machine tool structure for sculpturing/carving operations on granite sheet. The machine tool structure must be so designed, so that it should withstand the maximum force generated due to cutting operation & due to the inertia of the moving parts. Secondly, the machine tool structure should be re-configurable in nature in the sense that, it should be easy to assemble and should be made of standard components generally available in the market. Lastly, the design procedure should be so expressive, compact and simple in nature, so that it may be quite easy to understand & implement it, for someone willing to design or modify a 3-axis machine tool structure.

1.1 ORNAMENTAL CARVING [1]

Stone sculpture is the process of forming 3-dimensional, visually interesting shapes from stone. It is an ancient activity where pieces of rough natural stones were shaped by the controlled removal of stone. Countries like India, Egypt, an most of the Europe have quality stones in abundance. Monumental sculpture are used in large works including the architectural sculpture, for buildings etc.

1.2 GRANITE (STRUCTURE & PROPERTIES) [2]

Granite is an extremely hard, metamorphic stone composed of calcite $CaCO_3$. It is a very hard, crystalline, and primarily composed of feldspar, quartz accompanied by one or more dark minerals.. Granite is the hardest building stone, and granite slabs and granite tiles occupy a prominent place among dimensional stones. It is formed by recrystallization of limestone under intense heat and pressure of geological processes. This process results in a stone with very tight crystalline structure but with a definite porosity and because of this structure, marble can be polishes and can be used for sculptures. Due to highly dense grain, it is impervious to stain. Granite is also used for wall cladding, roofing, flooring, and a variety of other interior and exterior applications.

Table 1.1: Properties of granite stone [2]

S No	Parameters	Values
1	Hardness	3 to 4 on moh's scale
2	Density	2.5 to 2.8 $\times 10^3$ Kg/m^3
3	Compressive Strength	1800 to 2100 Kg/cm^2
4	Water Absorption	Less than 1%
5	Porosity	Quite low

1.3 AUTOMATION OF CUTTING PROCESSES [1] [3]

The intense need for new technology to control machine was realized around 1940 to meet the challenges in the production of aerospace components. The major contribution to this development was made by persons who developed a technique to machine accurate templates to develop helicopter blades. This involved calculation of few hundred points on a curve and drilling them in precision mill. The Servomechanisms laboratory of MIT developed the first NC machine in 1952. The development of NC technology can be categorised into various generations as listed below:

1.3.1 Generation first

The control system of the first generation numerically controlled machines used vacuum tubes and other devices. This was bulky, also consumed a lot of power, while reliability and the storage capacity was poor.

1.3.2 Generation second

The advent of PCB in to the field of electronics impacted to reduce the size of the controller for the NC machine. These machines were built with transistors. But the number of printed circuit boards (PCB) was large. The reliability was also not good as so many components we used at a time.

1.3.3 Generation third

During mid-60s, the concept of Integrated Circuit (IC) revolutionised the electronics world further. Thyristor controlled DC drive became popular during this time. With the integration of NC machine tools helped in flexibility. From the totally hardwired design, the design of NC machine become soft wired.

1.3.4 Generation fourth

Towards the end of 1970, microprocessors came to use as a CPU of the computers. This change also influenced in the design of NC machine tools. Initially 8 or 16 bit microprocessors were used. Later control systems with several processors were introduced. The reliability of the system was improved. Today many CNC systems are based on 32 bit as well as 64 bit microprocessors.

1.4 MACHINE TOOL [4] [5]

Machine tools are the class of machines which are used to manufacture different types of components for example lathe, milling machine and CNC machines. Here the study is specifically focused on 3-axis machine tool structure for carving on granite sheets. The structure of a 3-axis machine tool constitutes the load carrying and supporting member of the CNC machine. All the motors driver mechanisms and the other functional assemblies of machine tools are aligned to each other and rigidly fixed to the machine structure. The machine structure is subjected to static and dynamic forces and it is therefore essential that the structure does not vibrate beyond the limits under the action of these forces. The machine structure component is also influenced by the consideration of manufacture, assembly and operation. The basic design involved in the design of machine structure are discussed below.

1.4.1 Static loads involved in machine tool design

The static load of a machine tool results from the weights of the slide and the work piece and the forces due to cutting. To keep the deformation of the structure due to static loading within permissible limits the structure should have adequate stiffness and proper structural configuration.

1.4.2 Dynamic loads involved in machine tool design

Dynamic load is a term used for the constantly changing factors acting on the structure while movement is taking place. These forces cause the whole machine to vibrate. The origin of such vibration is

a) Unbalanced rotating part

b) Improper meshing of gears

c) Bearing irregularities

d) Interrupted cuts while machining

1.5 KEY COMPONENTS OF A MACHINE TOOL [5]

A 3-axis machine tool structure comprises of several key components which govern it's shape , size, weight and quickness in operation. All these components are described below.

a) Bed and Columns
b) Slideways / Guide ways
c) Ball screws
d) Linear motors / Drive unit
e) Spindle motor

a) Beds and columns

The way machining forces are directed into the bed and columns of the machine can have considerable influence on accuracy. Cutting forces are transmitted in a loop from the spindle when the operation on the work-piece is being done and then to the bed and back towards the spindle. It is necessary to minimize the length of force loop. This results in rigidity and accuracy without excessive dead weight. In a machine tool, the bed is the most critical components which possess the complicated mechanical structure coupled with the sophisticated load bearing conditions. With the ever-increasing demand in higher machine precision, the requirement for bed and column stiffness is also increasing.

b) Slideways / Guideways

The basic function of guideways used in machine tools is to:

i. Control the direction or line of action of the carriage or the table on which a tool or a work piece is held.
ii. To absorb all the static and dynamic forces.

The shape and size of work produced on a CNC machine tool depends on the accuracy of the movement and on the geometric and kinematic accuracy of the guideways. The geometric relationship of the slide and the guide way to the machine base determines the geometric accuracy of the machine. Kinematic accuracy depends on the straightness, flatness of the guideways. These errors further result in variety of tracking errors like pitch, yaw and the roll that are difficult to measure and correct. The following points must be considered while designing guideways:

a) Damping capabilities.
b) Rigidity.
c) Geometric and kinematic accuracy
d) Velocity of slide
e) Friction characteristics

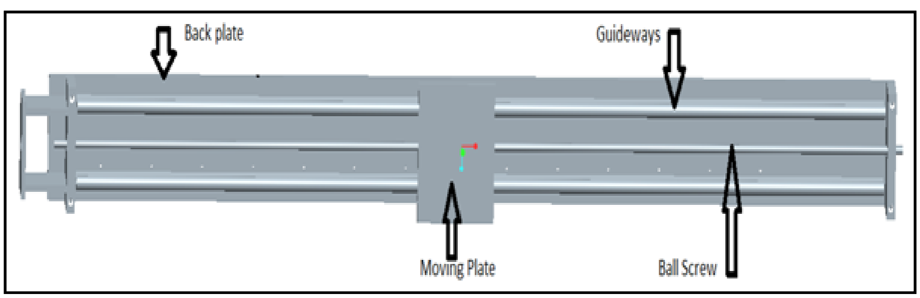


Figure 11:1 : Guideways for machine tool

f) Wear resistance
g) Provision for adjustment of play
h) Position in relation to work area

i) Protection against fine chips and damage

These criteria vary depending upon a particular application and hence the selection of guideways and their geometry can be quite critical in some cases. This will ensure uniform wear on guideways. Guideways are primarily of two types:

a) Friction guideways.

b) Antifriction linear motion guideways.

Frictional guideways are not used in CNC machines as they are most widely applied in conventional machine tools due to their low manufacturing cost and good damping properties. These guideways operate under condition of friction. It acts under stick-slip phenomenon to reduce the possibility of stick slip there should be a minimum but constant friction between the surfaces in contact. Antifriction linear motion guideways are used in CNC machines where friction plays a vital role in the movement of the parts. Generally the coefficients of friction in the components of CNC machine are very less.

c) Ball screw

Ball-screws consist of a screw spindle and nut integrated with recirculating balls, bearings and a ball return mechanism. The ball screw drive is an assembly that converts rotary motion to linear motion. The balls in this assembly form the connection between the nut and the screw.. The forces transmitted are distributed over the ball bearings, giving a comparatively low relative load per ball. With rolling elements the ball screw drive has a very low friction coefficient. Ball screw drives typically provide mechanical efficiency of greater than 90% so.. The major characteristics of ball-screws are:

a) High efficiency and reversibility.

b) Backlash elimination and high stiffness.

c) High lead accuracy.

d) Predictable life expectancy.

e) Low starting torque and smooth running.

f) Quietness.

d) Linear motors

Mainly three types of drive technologies are used in modern CNC system, which are:

a) Stepper drives

b) D.C. servo drives

c) A.C. drives

Figure 1:2 : Linear motor [6]

Stepping motor would appear to be particularly well suited to small NC machine tools, having low feed rates of the range of 300 mm/ min because they are able to directly convert digital path data into actual mechanical displacement of axes. They need no analog intermediate equipment no feedback from techno generators, and no path measuring systems. Additionally they require practically no maintenance as these motors are fully enclosed, also these drives are relatively economical. Despite having the above listed advantages the stepper drives are less preferred due to following reasons:

a) Stepping frequencies and consequently possible feed rates are too low.

b) Maximum available torques are relatively low, that is acceleration characteristics are poor

c) Even short term overload can cause “dropping out of step”.

d) Resolution is insufficient so that even at a resolution of 0.004 inches per step the rapid traverse speeds achievable are too low.

Today, modern NC machine tools have servomotor drivers fitted as standard equipment, and they provide infinitely variable feed and speed rates both the machine axes and the main spindle. Usually such a servo drive comprises of the following components:

a) Motor with tachometer and brake

b) Control unit with power amplifier-for spindle drives with integrated field controller

c) Main transformer and smoothing choke

d) Mechanical clutch, with overload protection

e) Motor protection unit guarding against current overload or excess temperature

f) A measuring element flanged directly on to the drive shaft, a path measuring system for feed drives.

For spindle drivers a gearbox with fixed or variable gearing to achieve fine adjustment of the motor speed in order to move should match speed and torque requirements at the spindle. It is the task of the feed drive to accurately position each individual axis of a machine tool and to precisely control all movements so that work piece with specified tolerance can be machined. The required accuracy require demands a strict adherence to the NC programmed position. With the simultaneous motion control of several axis, the relative motion between the cutting tool and work piece create three dimensional motion. So, dynamic behaviour is the main criterion for the selection of a drive. All programmed movements must be executed with the minimum possible declaration or overshoot, independent of other factors such as cutting power fluctuations or varying frictional losses. So, feed motion of axis of an NC machine tool have to be very precise, with as little deceleration and high repeatability as possible, in order to fulfil the accuracy requirements demanded. All motions must be independent of any counteraction forces much as those resulting for cutting or inertia. Positioning speed should be as fast as possible to avoid any lags in any programmed parameters.

e) Spindle motor

Figure 1:3: Spindle motor [7]

The router / spindle motor is one of the most commonly used power tools used granite working. As in technical sense for granite carving spindle is required which has comparatively low rpm and high torque. Generally, there are two types of routers viz. fixed base router and plunge router. Plunge router is used more often as they are more versatile in use. Plunge routers are used where there is high precision related work. Fixed base models have a certain advantage in centricity of the guide bush and base due to less movable part For carving on granite sheets the power requirement of the spindle motor is generally as high as 15 to 30 KW. Moreover the weight of the spindle is quite high as 50 to 100 Kg. So the bracket mounting and the other components are required to be of high stiffness rigidity. The basic requirements of the spindle motor are:

a) High stiffness both static and dynamic
b) Running accuracy
c) Axial load carrying capacity
d) Thermal stability
e) Axial freedom for thermal expansion
f) High speeds of operations

SCHEMATIC OF PURPOSED STRUCTURE

Below are the schematics for the purposed machine tool structure. The coordinate system shown in the figure is the reference one and the whole design process is developed using this system.

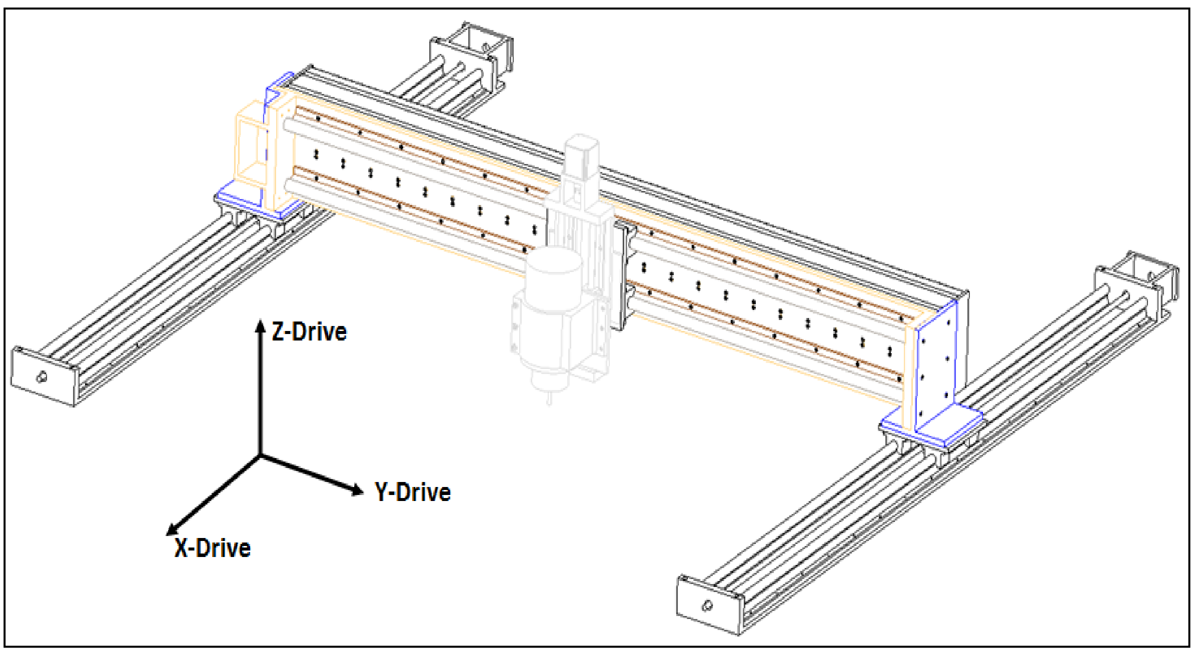


Figure 1:4: Proposed model of machine tool.

PRESENT WORK

Present work is on developing a methodology for design and selection of non standard components of the CNC machine tool structure for ornamental carving on granite. The main objective of the work was to streamline the design process for selection of standard off the shelf components for the CNC machine tool structure as well as to design of the support elements. The relevant literature was studied which has been explained in the next chapter of literature review. Subsequently in the 3^{rd} chapter, detailed procedure of CNC machine tool design has been discussed. Further the analysis of selected standard components has been verified using ANSYS under static and dynamic loading conditions. Results of the analysis have been used to suitably modify and redesign some of non standard components in order to withstand the loading conditions. All the formulations used in the methodology are programmed in an excel sheet in the same order.

Chapter 2
LITERATURE REVIEW

In this chapter the literature relevant to the present work has been presented. As the objective of the present work is to design a low cost, reconfigurable machine tool structure for ornamental carving on granite, detailed study from basics has been carried out and an idea for building a modular structure was developed.

2.1 MODULAR MACHINES

The concept of modularity in machine tools refers to a single machine that can perform various operations, thus reducing the production cost and time of manufacturing. This concept has emerged from F.M.S (Flexible Manufacturing Systems) and applied in to the design of machine tools. Lots of study has been done till date on modularity of machine tools, some of which is discussed below.

Alejandro *et al.* [8] documented a paper on the design, refinement and implementation on the reconfigurable machine tools that provided a flexible platform for turning and milling. They said that the concept of reconfigurability emerged after the FMS (Flexible Manufacturing Systems) became prominent. His work basically focused on the reconfigurability of mill-turns. Further he made a prototype of such a machine and presented the features and causes of failure of such machines.

Moriwaki [9] did a comprehensive survey and presented his findings on multifunction machine tools and their kinematic configurations. He concluded that various kinds of machine tools have been designed in recent past but still there is a need of developing machine tool which have high accuracy and can perform of high speeds.

Padayachee & Bright [10] discussed the design and barriers for industrial implementation of modular machines. They told that the modular machines have been built to meet the demands of the growing industry. The modularity permits the kinematic architecture and processing functions of the machine to be reconfigured to meet changing production requirements. The modular machines could be easily assembled in different orientations to perform different operations. Five problems were identified in modular machines that posed a hindrance in their industrial implementation which included the complex geometry of the machine tool and the un even mass distribution of machine parts.

Jani and Janez [13] did a study on the diamond tool for machining on granite and their wear with an aim to present the and describe the structure of diamond tools for CNC machine tools and to characterize the principle of diamond tool wear and forces which cause it's wear. It was concluded that the right combination of diamond grains types and matrix hardness enables productivity, quality machining and long tool life.

2.2 CUTTING FORCES DURING CARVING OPERATION

Calculation of cutting forces is the base of designing a machine tool In our present study the design of machine tool starts with the calculation of cutting forces and then further proceeding to the next elements. Below is some work presented on developing models for calculation of cutting forces on taking in to consideration various types of granites.

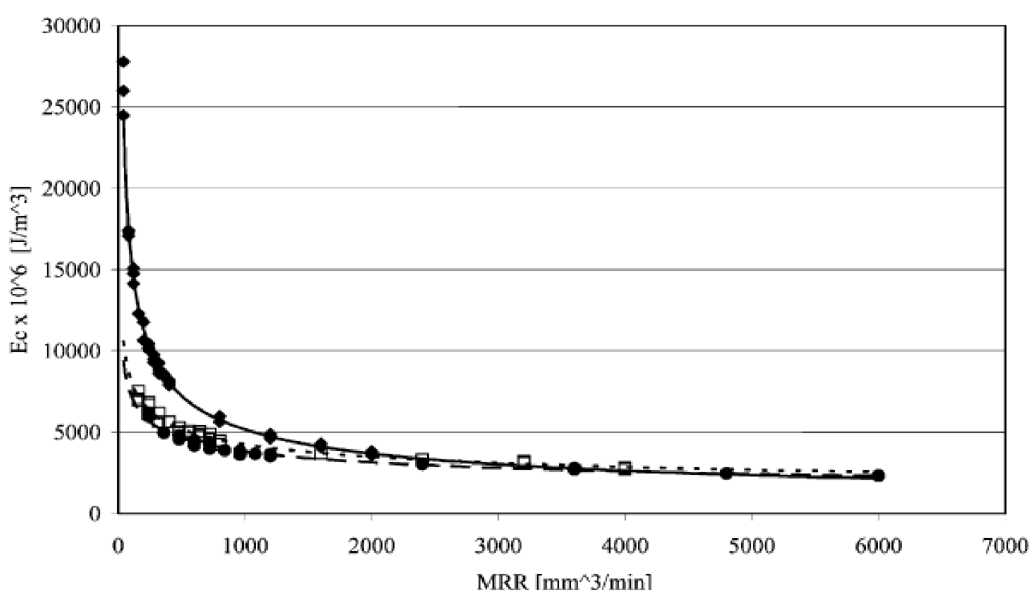


Figure 2:1 : Specific cutting energy vs. MRR [14]

Polini and Turchetta [14] did a study on force and specific energy in stone cutting by diamond mill. They presented an empirical model to calculate the cutting force (radial and tangential) during the cutting operations on granite stone by diamond milling tool. The study revealed that how the chip thickness is related to energy and cutting force of the work piece. ANOVA demonstrated that the cutting force strongly depends by both the depth of cut and the feed speed. By means of the equivalent chip thickness or the material removal rate (MRR) the influence of parameters were considered. This study was helpful to find the value of specific cutting energy of granite whose value was $5000 \times 10^6 J/m^3$ and is a function of cutting force.

2.3 WEIGHT REDUCTION AND BALANCING OF MACHINE TOOL STRUCTURE

As the weight of a machine tool is one of the prime hindrance in it's fast and rapid movement, so efforts have been made to reduce it and make the structure more stiff. Below are presented some studies which focused on the reduction of the weight in the machine tool structure by different approaches such as using sandwich structures, using material with high stiffness etc.

Cho *et al.* [15] in their study concluded that the mass reduction of the components of the machines is good in order to obtain high performance as small machine tool structures have low structural stiffness. In his experimental work several machine tool components were fabricated and assembled using mechanical joining and adhesive bonding. The results revealed that the re-designed structure was 36.8% lighter, and the structural stiffness was increased by 16% with higher loss factors (2.82–3.64%).

Lee *et al.* [16] in their study revealed that a machine tool should have high transfer speed and high cutting speed also as the air movement of the tool too i.e. air cutting time amounts to 70% of the total machining time One of the reason provoking this study was that the machine tools ahve large and heavy components which cannot bear high acceleration. Thus he designed a machine tool by using sandwich or composite structures materials These composite structures reduced the weight of the vertical and horizontal slides by 34% and 26%, respectively, and increased damping by 1.5–5.7 times without sacrificing the stiffness.

2.4 DRIVE ASSEMBLY

Rails and drive units of a machine tool are the components which decide it's strength and smoothness in operation. A constant effort has been made to improve the working conditions of these components by different methods. Below are mentioned some of those studies.

Lin *et al.* [19] has studied the effect of preloading of linear guides on dynamic characteristics of vertical column spindle system which is of importance for enhancing the structural performance of a vertical milling machine. In this work the FEM simulations were conducted and it was known that the preloading of the guides affected the vibration characteristics of the machine tool. It was concluded that dynamic stiffness of spindle could be increased by increasing the preload of linear guide.

Wakasawa *et al*. [21] in their research elaborated that main factors generating the damping capacity in machine structure by packed balls are the collision and friction among the packed balls and between the packed balls and the inner surface of the square section of the rail. These square sections can be installed anywhere on the machine particularly where do damp the vibrations, where it is needed that the vibrations should not be travelled further. The damping ratio could be successfully regulated by the values of the repulsion coefficient for glass balls of different diameters closely packed in structures. This kind of thing can be installed in our system of increase in damping capacity of machine and also damping of chatter and various other vibrations. However, due to other design constraint and other method of damping the vibrations like giving vibration pads at the bottom of the modular structure this approach was eliminated.

Verl & Frey [25] developed a correlation between feed velocity and preloading in the ball screw drives.. Increasing the rotational velocity led to a gain of the inner forces. Also the correlation between rotational speed and friction produced by a ball screw can be used for the effective pretension within a ball screw nut. Thus a new approach when calculating the life expectancy for a feed drive.

2.5 CONCLUSION

From the literature survey discussed above it can be concluded that there is a large amount of research being done on different parts / key components of the machine tools individually, but for the complete design no consolidated document is there where one can find the design methodology. Therefore this study aims at developing a complete and detailed design methodology for the design of machine tool structure so as to enable one to easily pursue with the design or modifications in the machine tool structure.

Chapter 3
DESIGN METHODOLOGY

The design process of a 3-axis machine tool structure comprises of a step by step hierarchical model for the selection of the key components and secondly for the structural design and optimization of other non standard components. This step by step model is based on some empirical relationships which govern the shape, size and strength of the component required. The design started with the predefinition of the material i.e. granite and the size of the granite sheet to be sculptured upon. It was followed by the determination of the cutting forces produced during the cutting operations (end milling and plunging). These forces were the function of specific cutting energy of the material and were calculated with the help of mathematical formulation which is described below in this chapter. After the calculation of the cutting forces, the spindle power required for end and plunge milling was calculated, followed by the selection of a standard spindle motor for granite cutting available in the market. When the spindle motor was selected, then a bracket is designed in order to hold the spindle motor in correct position without deflection. The bracket design was done with the help of ANSYS 10, which is a standard FEM tool for structural and dynamic analysis. Standard cutting tools available in the market were also selected, which actually determined the traverse of the vertical / Z-axis of the machine tool. The tool and spindle along with the bracket was termed as sub-assembly_1 which was then mounted on the Z drive with the help of screws. The design of these screws was done with the help of machine design procedures. The Z drive was actually a set of rails along with a ball screw upon which a traversing plate is there. This plate was driven by the ball screw arrangement where a nut being fixed on the moving plate. The selection of this Z drive (round shaft) was done from the catalogue of a standard manufacturing company [26]. Similarly the Y and X drives were chosen according to the load being experienced and the length required by them. Thus all the 3 axis of the machine were selected and finally assembled with the help of a modelling tool called Pro-E. The structural analysis of each component were done simultaneously The finally a machine tool structure was obtained which had the deflection under required limits. Below is presented a flow chart of the complete design process and the empirical relationships for each step are then discussed below. NO

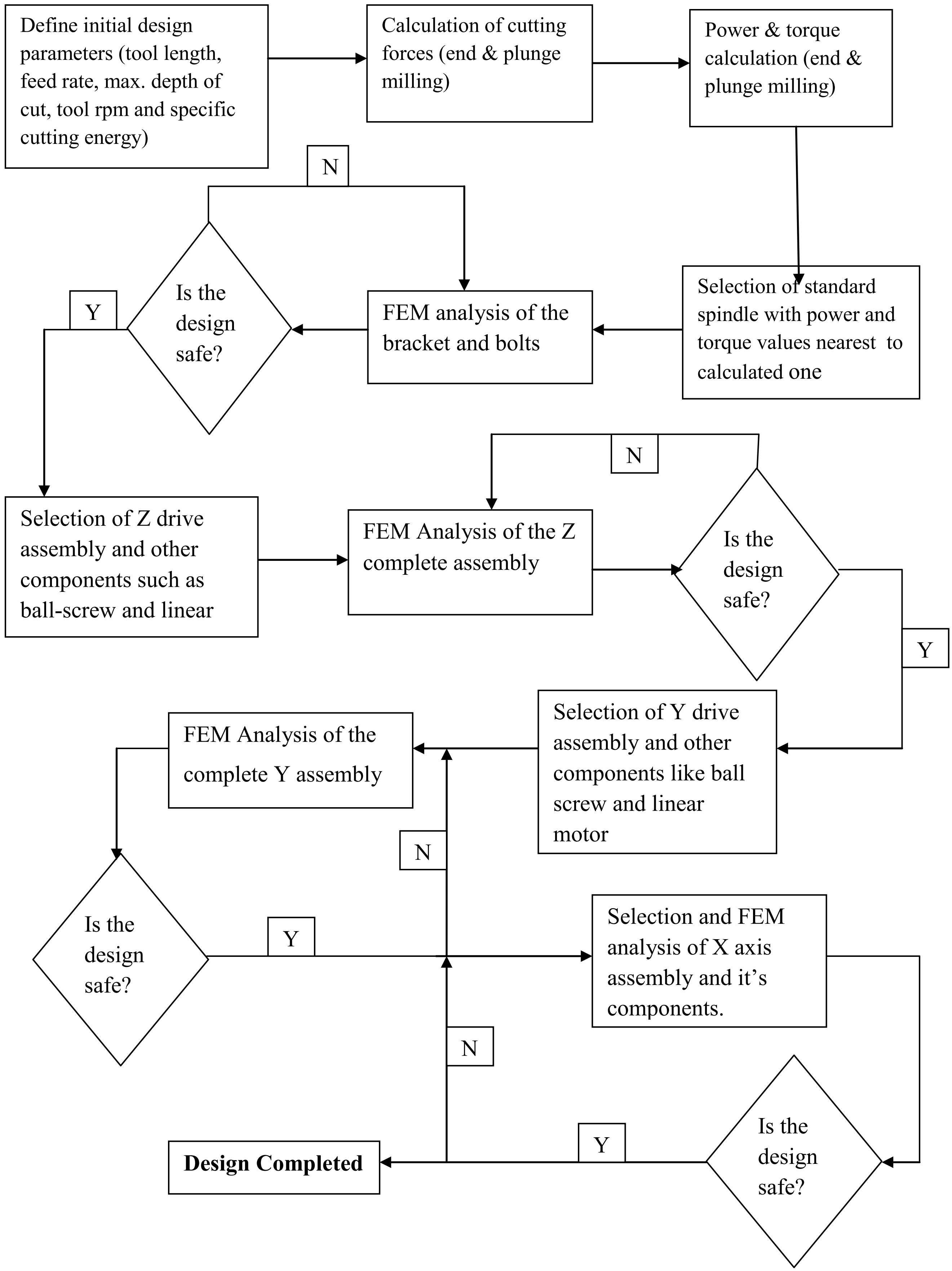


Figure 3:1: FLOW CHART OF DESIGN METHODOLOGY

3.1 DESIGN CONSTRAINTS

In the study for the design of this machine tool for granite sculpturing it was learnt that for such a design, some initial parameters need to be fixed. These parameters basically define the purpose for designing. In our case, these parameters were selected on the basis of surveys done in the local market and on the internet. The design of this machine tool structure started with these initial constraints which actually governed the whole design procedure. These initial parameters range from maximum and minimum tool length to the material property such maximum specific cutting energy etc. These initial constraints are not universal, but for our design methodology we have chosen and fixed them according to market and web survey conducted by author. All these initial parameters are listed below.

Table 3.1: Initial design parameters

S. No.	INITIAL CONSTRAINTS	SYMBOLS
1	Maximum tool length (mm)	L_t
2	Diameter of the largest tool (mm)	D
3	Maximum depth of the cut	d
4	Maximum feed rate	f
5	Cutting speed of the tool	V_c.
6	Maximum acceleration of the drives	α
7	Specific cutting energy of the granite material	K_e
8	Tool speed (RPM)	N
9	X, Y and Z traverse of the machine tool	L_x, L_y, L_z

After the definition of these initial constraints, the design parameters are calculated. Based upon these parameters, the selection and design of the whole machine components is done. These parameters are calculated below.

3.2 CUTTING FORCES ACTING ON TOOL DURING MILLING OPERATION [14]

The cutting force actually refers to the tangential cutting force that a rotating tool encounters while performing milling operations. The cutting force is basically the function of specific cutting energy. This specific cutting energy is a material property (granite in this case) and it means that, this much amount of energy is required to remove a unit volume of the work material. With the help of an empirical relations shown below, it's possible to find out the cutting forces (F_t) in the different cutting directions. There are two types of cutting operations considered in this case i.e. *end milling* and *plunge milling*. In case of end milling the cutting force is tangential, while in the plunge milling case it’s acting upwards in the direction along the tool axis. The cutting force for both these operations is calculated below.

3.2.1 End milling operation

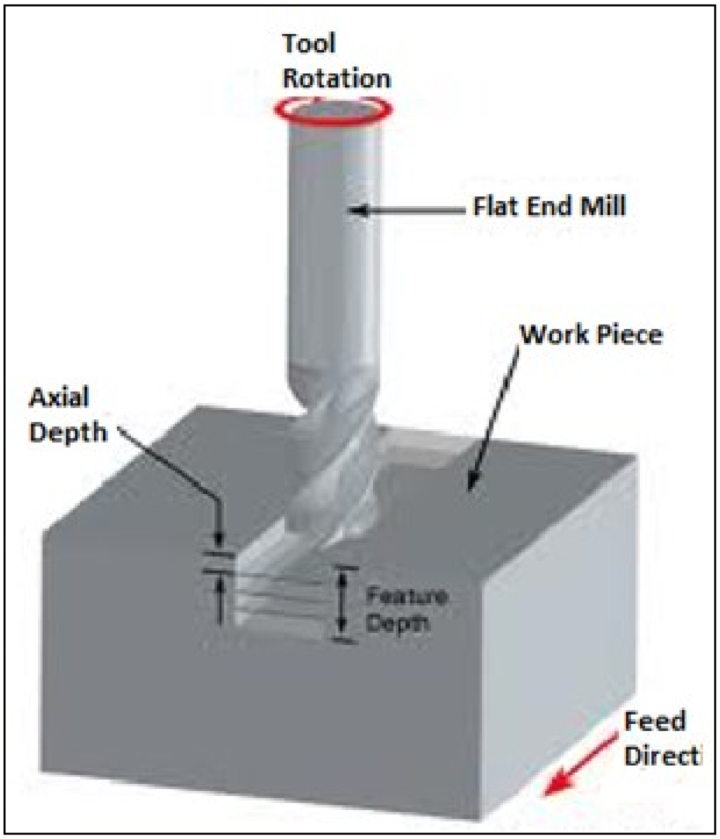


Figure 3:2: End milling [27]

In end milling the milling tool once inserted in to the workpiece moves ahead and removes the material for a fixed width and depth of the cut. The empirical relationship for the cutting force calculation in the case of end milling is given below.

$$K_e = \frac{F_t \times V_c}{f \times b \times t} \qquad \text{Eq. 3.1}$$

Here the numerator term is basically the power required for the cutting operation and the denominator term is the MRR. So by fixing the parameters like cutting velocity and feed rate, it’s possible to find out the cutting force. Otherwise, it can also be evaluated experimentally by using a dynamometer.

3.2.2 Plunge milling

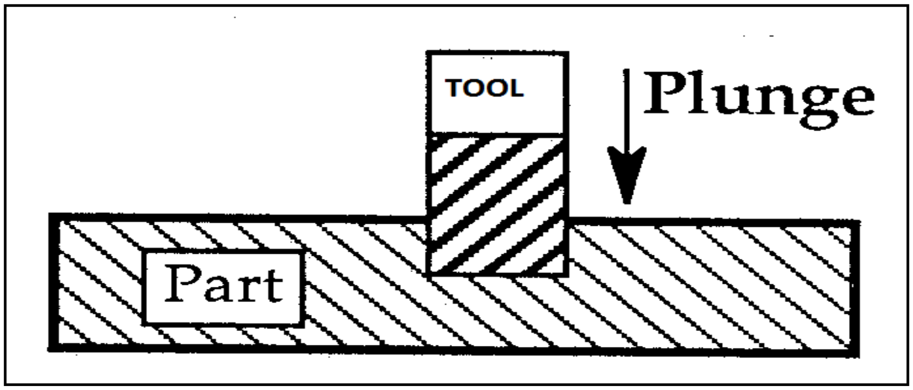


Figure 3:3 : Plunge milling [27]

In the case of plunge milling, the rotating tool inserts vertically in to the material of the work piece. The tool along with the spindle experiences a vertical thrust force upwards which is given by the formulation below.

Thrust Force $F_a = \frac{(\mathrm{K} \times \mathrm{K_c} \times \mathrm{f_z} \times \mathrm{D})}{2}$ Eq. 3.2

K = Coefficient of Geometry of the tip of the tool

K_c = Specific cutting force in N/m^2

f_z = Feed per tooth in m/tooth

3.3 SPINDLE MOTOR SELECTION [14]

The spindle is the component that carries and rotates the cutting tool against the forces encountered during milling operation. Spindle must have sufficient torque and power to perform these operations. Again in accordance to the variation in power calculation, there are basically two types of cutting operations performed viz. milling and plunging. When the tool moves in the plane of the work piece it's called milling and when it moves vertically up and down, then its plunging. The formulation for the power requirement during end milling and plunge milling are given below.

3.3.1 Power calculation in end milling

In the case of end milling, the tool experiences tangential cutting force which restricts the rotation of the tool. To encounter this force the spindle motor must be of required power and torque specifications. While selecting the spindle motor for the machine tool, following points should be considered

- Maximum available rpm of the spindle.
- Weight of the spindle unit.
- Maximum Torque available.
- As a structure it must be capable of withstanding the cutting forces.

Power for end milling operation in KW can be calculated as: $P_{end} = \frac{K \times Q}{1000}$ Eq. 3.3

Torque required for the same in KW can be calculated as: $T_{end} = \frac{F_t \times D}{2}$ Eq.3.4

K = coefficient - specific cutting power in $W/m^3/min$

Q = Material removal rate in m^3/min

N = RPM of spindle

The material removal rate M.R.R in m^3/min is given as: $Q = \frac{a_e \times a_p \times f}{1000000}$ Eq. 3.5

a_p = Axial engagement of the tool in mm

a_e = Radial engagement of the tool in mm

f = Feed rate in m/min

3.3.2 Power calculation in plunge milling

Plunging action occurs when the rotating tool plunges or is inserted in to the work material. The spindle motor must be capable to deliver the required power and withstand the axial force generated during this operation.

Power in KW $P_p = \frac{(Kc \times D \times fz \times Vc)}{240000}$ Eq.3.6

Torque in N-m $T_p = \frac{K_c \times D^2 \times f_z}{8}$ Eq.3.7

K = Coefficient of Geometry of the tip of the tool

K_c = Specific cutting force in N/m^2

f_z = Feed per tooth in m/tooth

So the selection of spindle motor comprises of the calculations of tangential and axial cutting force on work material required to perform the machining operations and these forces depend on the specific cutting energy of the material and on the method of machining too. In vertical milling process the two types of cutting forces - tangential and axial, occurs due to simple milling and plunging respectively The value of the power coming out to be maximum from either of the milling or plunging operations should be selected as spindle power. Moreover the spindle power should be more than the calculated power by a factor of 30% .Therefore, $P_{actual} = 0.3 \times (P_p)$ Eq.3.8

3.4 SELECTION OF CUTTING TOOL [13] [28]

The stone engraving tools are basically steel tools of different shapes and profiles with diamond granules sintered on its cutting surface. Thus the basic phenomenon of stone cutting /engraving is abrasion process. The diamond granules act as abrasives, which remove the stone material when the rotating tool comes in contact with it. Diamond tools are key elements in the machining of natural stone, especially hard stones such as granite. Diamond tools are mostly used in CNC. They are used in different machining technologies like cutting, abrasive milling, abrasive drilling, grinding and polishing. All diamond tools

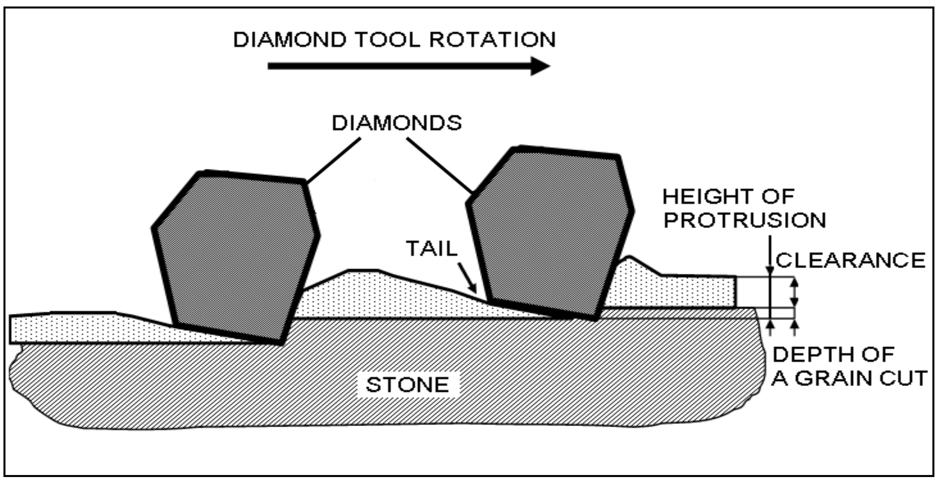


Figure 3:4: Abrasion Process in Stone Cutting

The support gives the right shape to the tool, transmits the kinetic energy from the machine axes to the diamond grains and absorbs the stresses generated during machining. Diamond grains constitute the cutting edges. They are characterized with grain size, shape and concentration. The matrix is a metallic alloy fixing the abrasive grains on the tool support in order to make machining possible. It assures both, the cutting ability and long tool life. For effective and quality machining with diamond tools it is necessary to get the right combination of matrix and diamond grains. Hard stones like the granite have to be machined with tools that have a harder matrix and suitable diamond grains The importance of the stone CNC machining technologies is increasing. Nowadays, the shapes of stone products are becoming increasingly complex. To machine complex shapes, CNC technologies and different shapes of diamond tools are needed. All diamond tools have the same machining principle - grinding. Tools for CNC machining technologies mainly differ from each other in shape and grain size. Typical CNC machining technologies and basic tools for each technology are cutting, drilling, milling grinding and polishing

Figure 3:6 : Different granite cutting tool [13]

3.5 DESIGN OF BRACKET FOR HOLDING THE SPINDLE [29]

The role of the bracket is to hold the weight of the spindle and support all cutting forces encountered by the tool without deflection. Therefore followings points are required to be considered while selecting and designing the bracket of the machine tool.

a) Material of the bracket plate.
b) Number of bolts required to clamp the spindle with bracket.
c) Pitch of the bolt and side margin against the bearing(tearing)strength of bracket(thickness of the bracket plate)
d) Diameter of bolt.
e) Material of the bolt and nut.

The bracket and the spindle collectively are termed as Sub-assembly_1. There are 3 different cases in which the bracket and its bolts must hold good for proper milling operations. Following are the cases which need to be considered while designing the bracket and bolt size for clamping purpose, but before that there is a need of defining the axis if the machine tool structure.

3.5.1 When cutting tool moving along the X axis

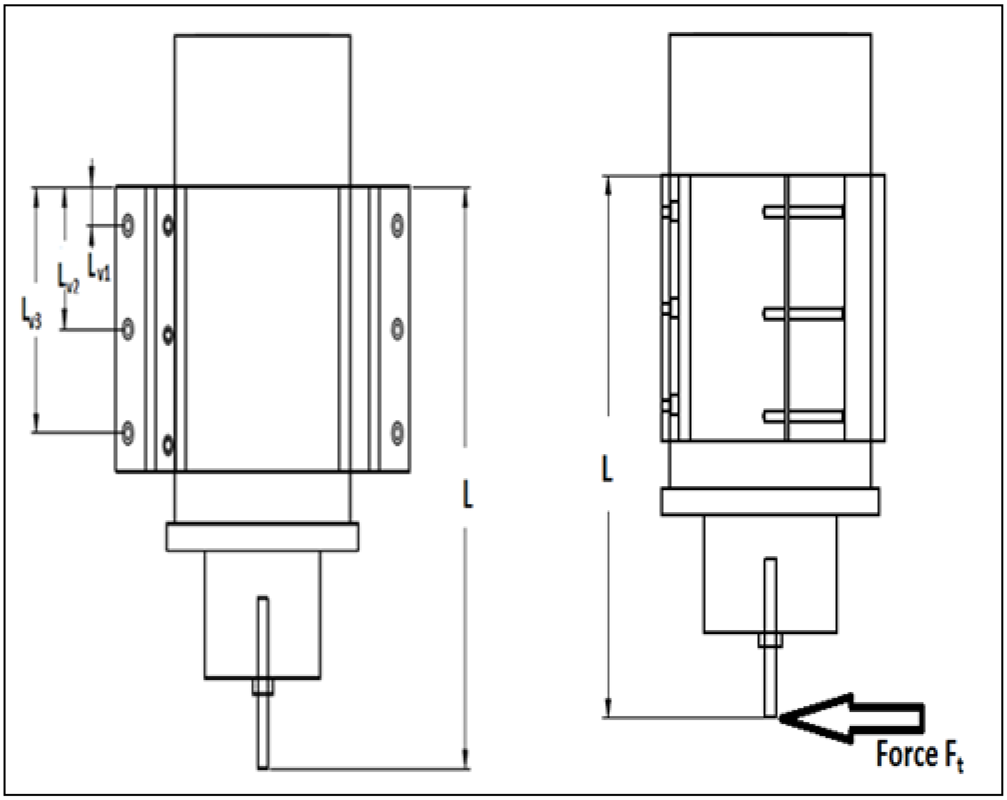


Figure 3:7 : Tool moving in X direction

Inertia of Sub-assembly_1 in X direction acting on the bolts: $W_{inet} = m \times \alpha$ Eq. 3.9

Total load acting on the bolt: $W = W_{inet} + F_t$ Eq. 3.10

If the total number of bolts be "n", then, direct tensile load on each bolt

$W_{dt} = \frac{W_{total}}{n}$ Eq. 3.11

Let the vertical distances of the bolts in the 1st, 2nd, and 3rd row from the tilting edge be L_{v1}, L_{v2} & L_{v3} respectively.

Load (w) on the bolt per unit distance $w = \frac{W \times L}{2({L_1}^2+{L_2}^2+{L_3}^2)}$ Eq. 3.12

Tensile load (W_{t1}) on each bolt at distance L_{v1} $W_{t1} = \mathrm{w} \times L_{v1}$ Eq.3.13

Tensile load (W_{t2}) on each bolt at distance L_{v2} $W_{t2} = \mathrm{w} \times L_{v2}$ Eq.3.14

Tensile load (W_{t3}) on each bolt at distance L_{v3} $W_{t3} = \mathrm{w} \times L_{v3}$ Eq.3.15

Total tensile load (W_1) on bolt at distance L_1 $W_1 = W_{dt} + W_{t1}$ Eq.3.16

Total tensile load (W_2) on bolt at distance L_2 $W_2 = W_{dt} + W_{t2}$ Eq.3.17

Total tensile load (W_3) on bolt at distance L_3 $W_3 = W_{dt} + W_{t3}$ Eq.3.18

Maximum Tensile load (W_t) will be chosen from the above three equations for W_1, W_2 & W_3

Diameter of the bolt material(dc) can be calculated from the equation below ,where σ_{bolt} is the allowable tensile stress for bolt material

$$W_t = \frac{\pi}{4}(d_c)^2\sigma_{bolt} \quad \text{Eq.3.19}$$

3.5.2 When cutting tool moving along Y direction

In this case while the too is moving left to right performing the end milling operations, the load acting on the joints being same as in eq.3.10 i.e. W, the eccentricity of the load being "e" and the number of bolt be "n" then the direct shear load on the each bolt is given as:

Direct shear load on each bolt $W_s = \frac{W}{n}$ Eq.3.20

Turning moment produced by the load due to eccentricity $M_{ie} = P \times e$ Eq. 3.21

Now the distance of the 1st ,2nd ,3rd ,4th ,5th & 6th bolt from the centroid (say 6 number of bolt) be $L_{c1}, L_{c2}, L_{c3}, L_{c4}, L_{c5}, L_{c6}$ respectively as shown in the figure.

Secondary shear load on bolt 1: $(F_1) = \frac{(\mathrm{M_{ie}}) \times \mathrm{L_{c1}}}{\mathrm{L_{c1}}^2 + \mathrm{L_{c2}}^2 + \mathrm{L_{c3}}^2 + \mathrm{L_{c4}}^2 + \mathrm{L_{c5}}^2 + \mathrm{L_{c6}}^2}$ Eq. 3.22

Secondary shear load on bolt 2: $(F_2) = F_1 \times \frac{l_2}{l_1}$ Eq.3.23

Secondary shear load on bolt 3: $(F_3) = F_1 \times \frac{l_3}{l_1}$ Eq. 3.24

Secondary shear load on bolt 4: $(F_4) = F_1 \times \frac{l_4}{l_1}$ Eq.3.25

Secondary shear load on bolt 5: $(F_5) = F_1 \times \frac{l_5}{l_1}$ Eq.3.26

Secondary shear load on bolt 6: $(F_6) = \mathrm{F}_1 \times \frac{\mathrm{l}_6}{\mathrm{l}_1}$ Eq.3.27

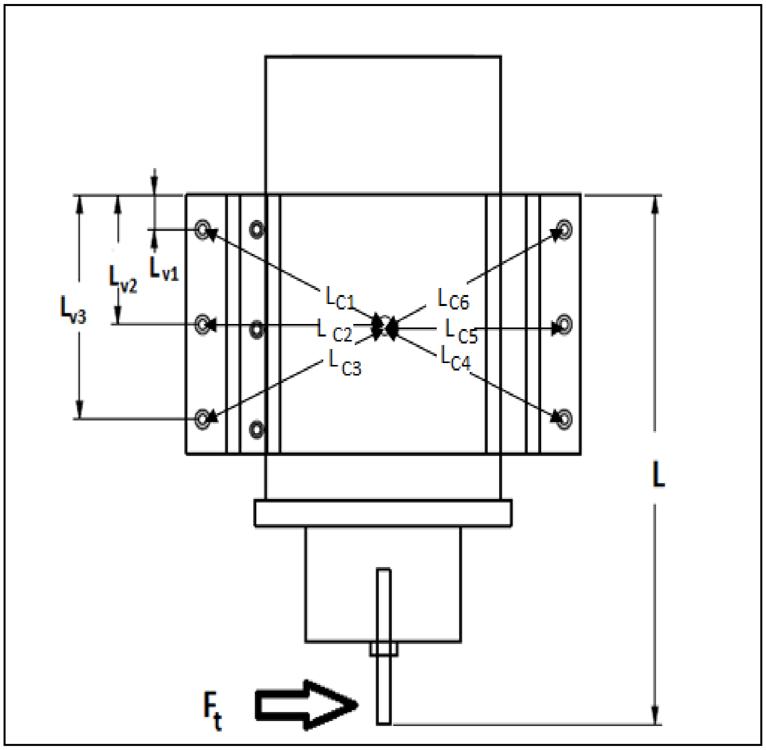


Figure 3:8 : Tool moving in Y direction

Angle between direct and secondary shear load for bolt 1 = cosθ1

Angle between direct and secondary shear load for bolt 2 = cosθ2

Angle between direct and secondary shear load for bolt 3 = cosθ3

Angle between direct and secondary shear load for bolt 4 = cosθ4

Angle between direct and secondary shear load for bolt 5 = cosθ5

Angle between direct and secondary shear load for bolt 6 = cosθ6

Resultant shear load on bolt 1: $(R_1) = \sqrt{P_s^{\,2} + F_1^{\,2} + 2P_sF_1cos\theta1}$ Eq.3.28

Resultant shear load on bolt 2: $(R_2) = \sqrt{P_s^{\,2} + F_2^{\,2} + 2P_sF_2cos\theta2}$ Eq.3.29

Resultant shear load on bolt 3: $(R_3) = \sqrt{P_s^{\,2} + F_3^{\,2} + 2P_sF_3cos\theta3}$ Eq.3.30

Resultant shear load on bolt 4: $(R_4) = \sqrt{P_s^{\,2} + F_4^{\,2} + 2P_sF_4cos\theta4}$ Eq.3.31

Resultant shear load on bolt 5: $(R_5) = \sqrt{P_s^{\,2} + F_5^{\,2} + 2P_sF_5cos\theta5}$ Eq.3.32

Resultant shear load on bolt 6: $(R_6) = \sqrt{{P_s}^2 + {F_6}^2 + 2P_sF_6cos\theta 6}$ Eq.3.33

Maximum resultant shear load (R) will be the maximum from these above six equations.

Thus the diameter of bolt (dc) can be found from the following equation, where τ_{bolt} is the maximum permissible shear stress of the bolt material.

$R = \frac{\pi}{4}{d_c}^2\tau_{bolt}$ Eq. 3.34

3.5.3 When cutting tool moving in Z direction

When cutting in the vertically downwards direction the bolts clamping the bracket with the back plate experience both shear force and tension. To calculate the diameter of the bolts required to serve the purpose, the following methodology is followed.

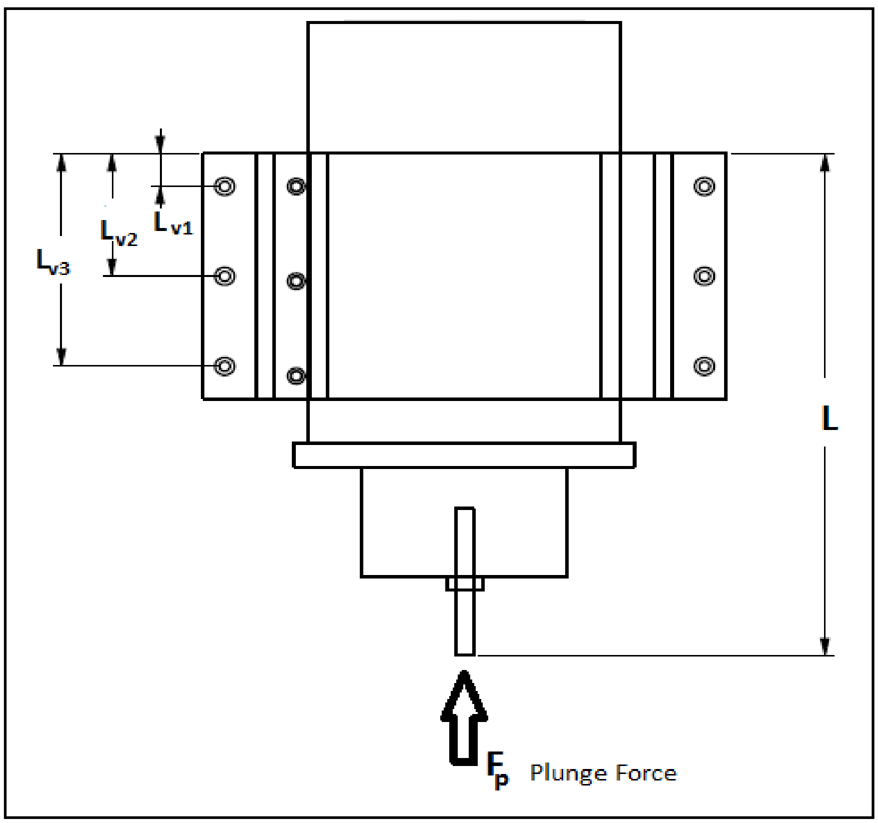


Figure 3:9 : Tool moving in downward direction

Load W_p on clamping bolts in plunge cutting $W_p = W_i + F_p$ Eq.3.35

Direct shear load on each bolt $W_{ds} = \frac{W_p}{N_b}$ Eq.3.36

Vertical distances of the bolts in the 1st, 2nd, and 3rd row from the tilting edge be L_{v1}, L_{v2} & L_{v3} respectively thus the bolt at maximum vertical distance from twisting edge be calculated as $L_t = max.(L_{v1}, L_{v2}, L_{v3})$

Maximum tensile load on bolt $W_t = \frac{W_{ds} \times e_z \times L_t}{2 \times (L_{v1}^2 + L_{v2}^2 + L_{v3}^2)^2}$ Eq.3.37

Equivalent tensile Load $W_{te} = \frac{1}{2}(W_t + \sqrt{(W_t^2 + 4 \times w_s^2})$ Eq. 3.38

Equivalent shear Load $W_{se} = \frac{1}{2}(\sqrt{(W_t^2 + 4 \times w_s^2})$ Eq. 3.39

Diameter of the bracket clamping bolt in tension $d_{btz} = \sqrt{\frac{4 \times s_b \times W_t}{\pi \times \sigma_{bolt}}}$ Eq.3.40

Diameter of the bracket bolt according to shearing $d_{bsz} = \sqrt{\frac{4 \times W_{se}}{\pi \times \sigma_{bolt}}}$ Eq. 3.41

3.6 DESIGN AND SELECTIONS OF DRIVE ASSEMBLY COMPONENTS [26]

The drives of a 3-axis CNC router comprises of basic components like ball-screw, guide-ways, linear motors, coupling etc. In the drive of a CNC router there are guide-ways upon which a plate on which load is mounted moves back and forth. The movement of this plate is governed by a ball-screw and nut arrangement. The power required for rotating this ball-screw is given by linear motors. The couplings are responsible for transmitting he power from the linear motors to the ball-screw. The guide ways of a machine tool have a varying range from linear guide-ways, roller guide-ways to round-shaft guide-ways, each having their own benefits. The ball-screw too has a lot of options to be fit into different operating conditions. They vary in the type of mounting methods to the type of nuts used etc. The linear motors also come in different models varying from NEMA 23 to 123 etc. according to the power and torque requirement of the machine tool structure. Below is given the design methodology for all these prime components.

3.6.1 Design of ball-screw [30]

A ball screw is the machine tool member which transmits rotary motion from the drive motor to the linear motion to the slide way. With the Ball Screw, balls roll between the screw shaft and the nut to achieve high efficiency, high accuracy and high rigidity without backlash. Its required driving torque is only one third of the conventional sliding screw. As a result, it is capable of not only converting rotational motion to straight motion, but also converting straight motion to rotational motion. As the Ball Screw is ground with the highest-level facilities and equipment at a strictly temperature controlled factory, thus it's accuracy is assured under a thorough quality control system that covers assembly to inspection. Moreover the all screw requires a minimal starting torque due to its rolling motion, and does not cause a slip, which is inevitable with a sliding motion. Therefore, it is capable of an accurate micro feeding. The ball screw is subjected to following loads:

a) The thrust or axial force due to the load of work piece,
b) Cutting force, friction of the slide way and inertia force.
c) Torsion force to overcome this force.

Below are mentioned the step by step procedure for selecting ball screw as a standard component.

a) Selecting the initial conditions for ball screw selection

There are some conditions that need to be initialized for the selection of a ball screw. This is the first step towards selecting a ball screw.

Table 3.2: Initial conditions for ball screw selection

S No	Conditions	Units (if applicable)
1	Transfer orientation	Horizontal (X or Y), Vertical (Z)
2	Transferred mass , m	kg
3	Frictional coefficient of the guide surface, μ	N
4	Guide surface resistance , f	N
5	External load in the axial direction , F	N
6	Desired service life time , L_h	hours
7	Stroke length , L_s	mm
8	Operating speed , V_{mzx}	m/s
9	Acceleration and deceleration time t_1 and t_3	s
10	Even speed time, t_2	s
11	Acceleration, $\alpha = \frac{V_{max}}{t_1}$	m/s^2

12	Driving motor type (DC servomotor, stepping motor, etc.)	
13	The rated rotational speed of the motor, N_{mo}	min^{-1}

b) Axial clearance

The axial clearance of the precision and rolled Ball Screw is the clearance between the ball and the bearing surface area. The permitted value of this clearance is specified in various literatures in accordance to DIN standards. To avoid this clearance and the displacement under the axial load the concept of preload is used, where initially some axial force is applied to avoid the clearance (as shown in the fig. below) by means of spacers.

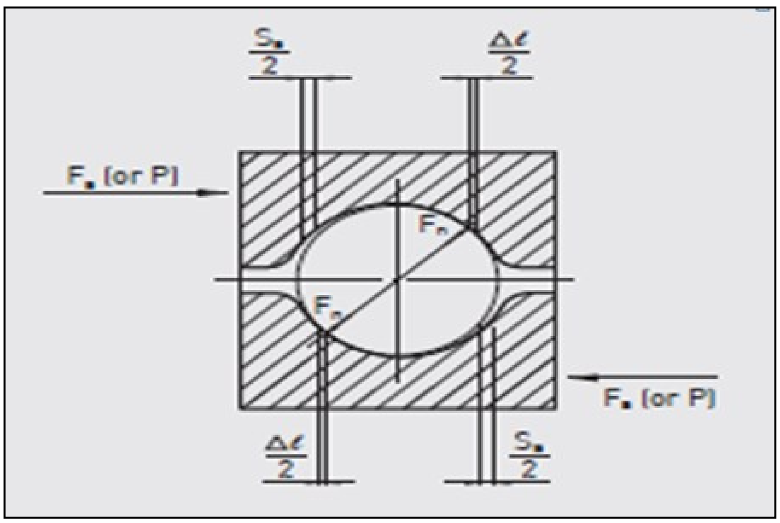


Figure 3.9: Axial Clearance of the ball screw [27]

c) Preload

A preload is provided in order to eliminate the axial clearance and minimize the displacement under an axial load. When performing a high accuracy positioning a preload is generally applied. When a preload is provided to the ball screw, rigidity of the nut increases. The elastic displacement curves of a ball-screw with and without preload are shown in the figure below.

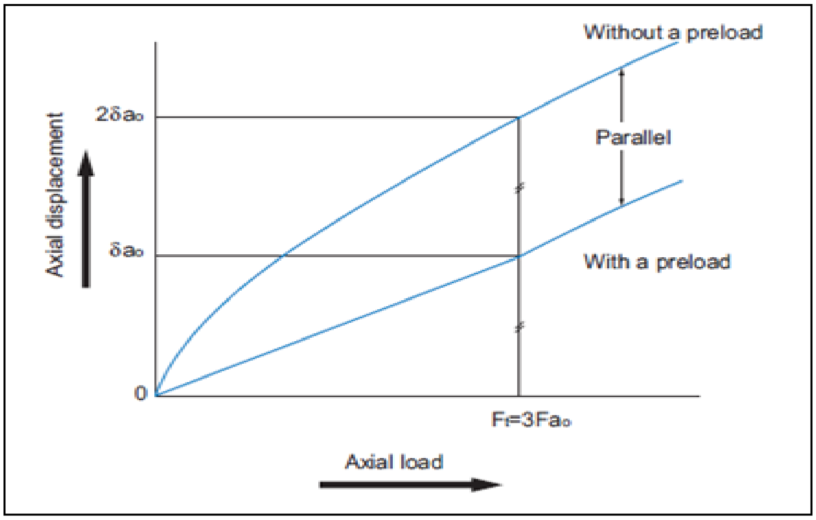


Figure 3.10 : Effect of preload [30]

d) Selecting the length, lead and diameter

The length, lead and diameter of the ball screw are selected in accordance to diameter of the shaft and DIN standards. The standard values of these factors are listed in the tables provided in the references.

e) Mounting methods

The permissible axial load and the permissible rotational speed vary with mounting methods for the screw shaft. Therefore, it is necessary to select an appropriate mounting method according to the conditions. There are three different mounting methods for mounting ball-screws on to a machine tool structure as shown below..

a) Fixed-supported.
b) Fixed-Free.
c) Fixed-Fixed

The fixed-supported mount is most commonly used. The second one i.e. fixed-free is used where ball-screws is of short length or with low speeds and are in expensive. The fixed-fixed mounting is used in the case of over constrained mountings and where high stiffness, accuracy and high speed is required. These ball-screws need to be pre-stretched to avoid buckling in case o thermal expansion.

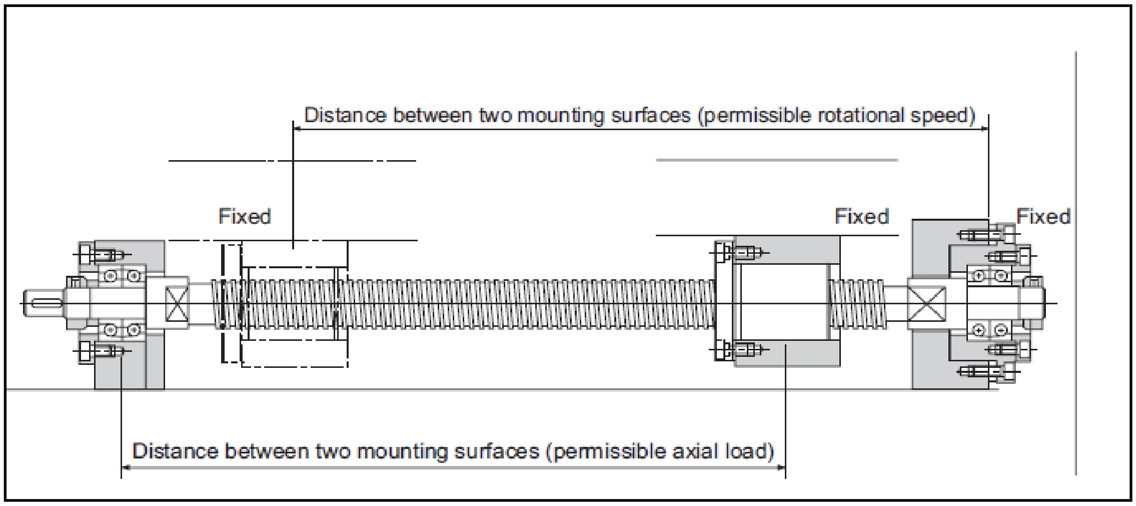


Figure 3:10 : Fixed-supported mounting [30]

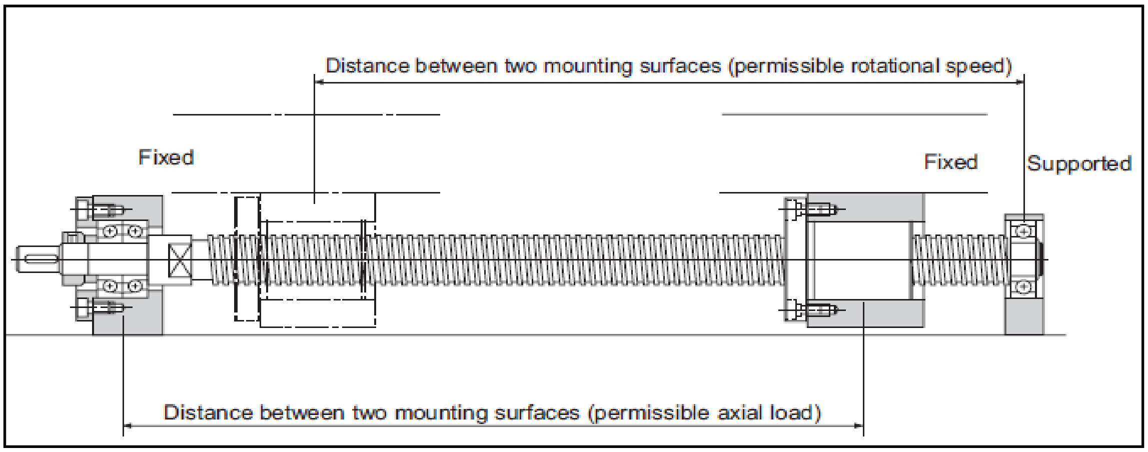


Figure 3:11 : Fixed-Free mounting [30]

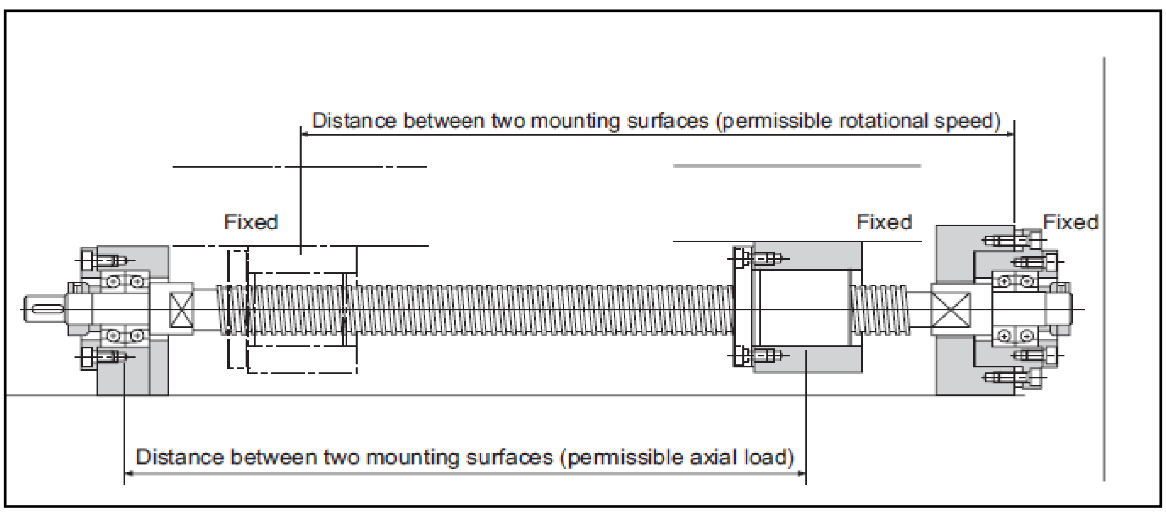


Figure 3:12 : Fixed-Fixed mounting [30]

f) Selecting the nut configuration for ball screw [29]

The nuts of the Ball Screws are categorized by the ball circulation method into the return-pipe type, the deflector type and end the cap type. These three nut types are described as follows. In addition to the circulation methods, the Ball Screws are categorized also by the preloading method.

- **Return Type**

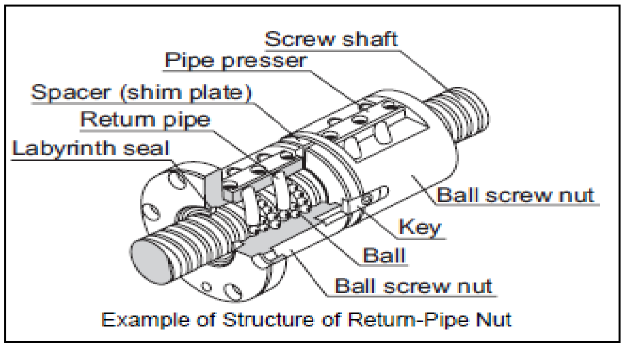


Figure 3:13 : Return Type Nut [30]

These are most common types of nuts that use a return pipe for ball circulation. The return pipe allows balls to be picked up, pass through the pipe, and return to their original positions to complete infinite motion.

- **Deflector Type**

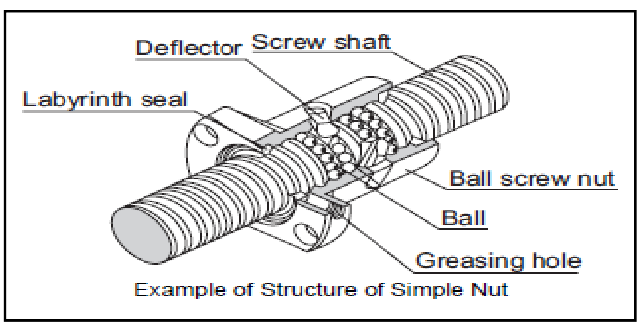


Figure 3:14 : Deflector Type Nut [30]

These are the most compact type of nuts. The balls change their travelling direction with the help of a deflector, pass over the circumference of the screw shaft and return to their original positions to complete their infinite motion.

- **End Cap Type**

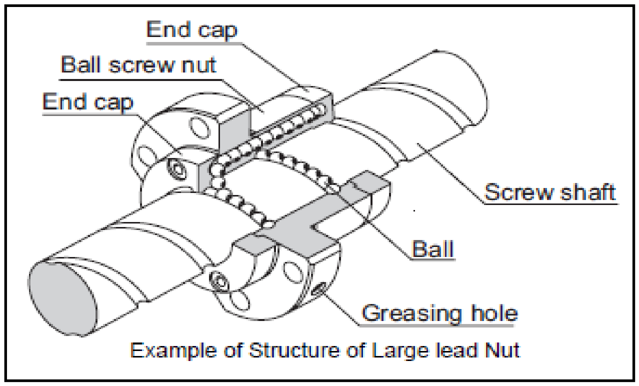


Figure 3:15 : End Cap Type Nut [30]

These nuts are most suitable for the fast feed. The balls are picked up with an end cap, pass through the through hole of the nut, and return to their original positions to complete an infinite motion.

3.6.2 Calculation of the axial load for a load carrying ball screw

a) In horizontal mounting

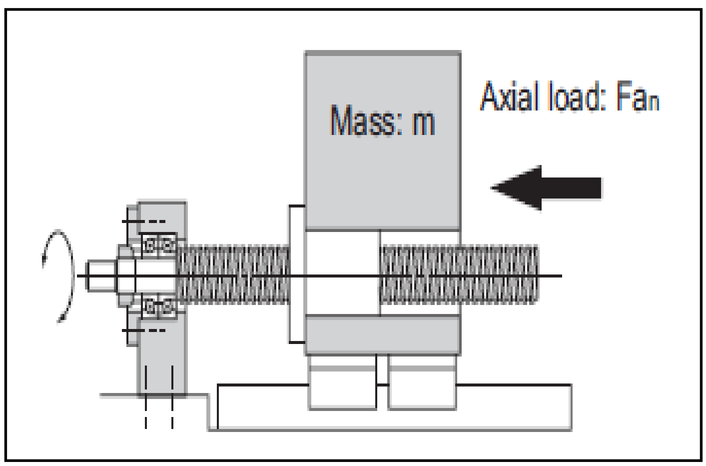


Figure 3:16 : Horizontal mounting of load [30]

When the load is acting on a horizontally mounted ball-screw, then the axial force acting to reciprocate the work piece back and forth is given by following equations.

$$\boldsymbol{F_{a1} = \mu \times mg + f + m\alpha} \quad \text{Eq.3.42}$$

$$\boldsymbol{F_{a2} = \mu \times mg + f} \quad \text{Eq.3.43}$$

$$\boldsymbol{F_{a3} = \mu \times mg + f - m\alpha} \quad \text{Eq.3.44}$$

$$\boldsymbol{F_{a4} = -\mu \times mg - f - m\alpha} \quad \text{Eq.3.45}$$

$$\boldsymbol{F_{a5} = -\mu \times mg - f} \quad \text{Eq.3.46}$$

$$\boldsymbol{F_{a6} = -\mu \times mg - f - m\alpha} \quad \text{Eq.3.47}$$

F_{a1} : Axial load during forward acceleration

F_{a2} : Axial load during forward uniform motion

F_{a3} : Axial load during forward deceleration

F_{a4} : Axial load during backward acceleration

F_{a5} : Axial load during uniform backward motion

F_{a6} : Axial load uring backward deceleration

$\alpha\ (acceleration) = \frac{V_{max}}{t_1}$ m = mass transferred

V_{max} = maximum speed & t_1 = acceleration time

μ = coeffctient of friction & f = guide surface resistance without load

b) In vertical mounting

When the load is acting on a vertically mounted ball-screw, then the axial force acting to reciprocate the work piece back and for the is given by following equations. The diagram for the vertical mounting is presented below in which a load of mass M is acting vertically downwards and the ball screw is resisting the moment produced by the weight of the load.

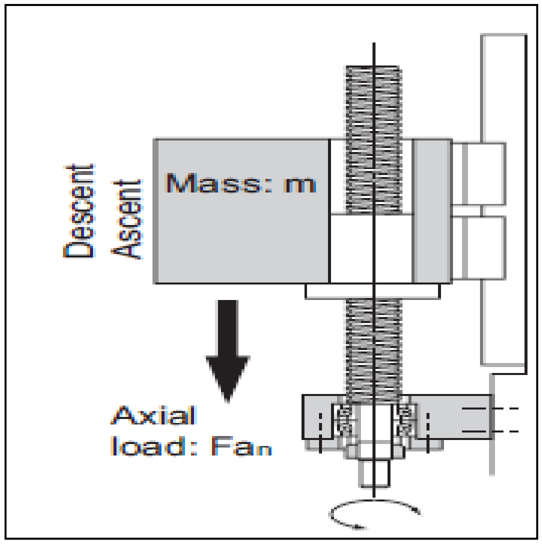


Figure 3:17 : Vertical mounting [30]

$$\boldsymbol{F_{a1} = mg + f + m\alpha} \quad \text{Eq.3.48}$$

$$\boldsymbol{F_{a2} = mg + f} \quad \text{Eq.3.49}$$

$$\boldsymbol{F_{a3} = mg + f - m\alpha} \quad \text{Eq.3.50}$$

$$\boldsymbol{F_{a4} = mg - f - m\alpha} \quad \text{Eq.3.51}$$

$$\boldsymbol{F_{a5} = mg - f} \quad \text{Eq.3.52}$$

$$\boldsymbol{F_{a6} = mg - f + m\alpha} \quad \text{Eq.3.53}$$

F_{a1} : Axial load during upward acceleration

F_{a2} : Axial load during upward uniform motion

F_{a3} : Axial load during upward deceleration

F_{a4} : Axial load during downward acceleration

F_{a5} : Axial load during uniform downward motion

F_{a6} : Axial load uring downward deceleration

α (acceleration) $= \frac{V_{max}}{t_1}$ m = mass transferred

V_{max} = maximum speed   & t_1 = acceleration time

μ = coeffctient of friction   & $f = guide\ surface\ resistance\ wit\square out\ load$

3.7 DESIGN AND SELECTION OF GUIDEWAYS

The guideways / slideways are available in various shape and size options having their own strengths and drawbacks. These variations in their shape and size ,facilitates their use in different conditions. The round shaft guideways are used in this case.

3.8 ROUND SHAFT [26]

The advantages of selecting the round shaft over linear guideways are :

a) Dust proof, in case of machining of wood the chips are in the form of dust which can cause blockage of the transmission system.
b) They require low maintenance in contaminated environments.
c) Self lubricating and do not require additional grease.

Figure 3:18 : Aluminum rail support [27]

3.8.1 Round shaft selection

The following factors should be considered while selecting the slides for a machine tool structure.

a) Length of stroke and size
b) Mounting orientation.
c) Load to be carried out.
d) Maximum velocity.
e) Acceleration and deceleration rate.
f) Service life.
g) Environment of work.

3.8.2 Calculation for the Bending Moment Produced

The basic formula for calculating the bending moment of the machine drive unit in any direction is given below. The application of this formula is given in the following equations

$\text{Bending moment} = (\text{Cutting Force}) \times (\text{Dis. of cutting force from COG of arrangement}$

Applying this formula in the actual machine structure we the following equations are obtained for calculating the bending moment in X and Y direction due to tangential and plunging force.

Maximum bending moment in Y due to tangential force : $M_y = F_t \times l_{tz}$ Eq.3.54

Maximum bending moment in Y due to plunge force : $M_y = (F_p - m_{asm1}) \times l_{tx}$ Eq. 3.55

Maximum bending moment in X due to tangential force : $M_x = F_t \times l_{tx}$ Eq. 3.56

3.9 SELECTION OF DRIVE MOTORS [6]

The function of the driving / linear motor is to rotate the ball screw and thus move the drive of the machine tool structure. For the selection of proper motor and its size the torque required should be calculated, for which these point should be considered.

a) Rate of acceleration, deceleration and required velocity for the desired motion.
b) Inertia, frictional and other load torques to be encountered.
c) Condition and Environment of operation.

3.9.1 Motor types

There are different types of motors used for ball screw rotation and all of them are listed below. The benefits and shortcomings of all these moors are listed in the following table. Each motor operates on a different concept, has a different applicability and has a totally different construction. The features of each type of motor is listed in the table given below.

a) Stepper motor
b) DC motor
c) Brushless DC motor
d) Induction Motor(AC motor)

Table 3.3 : Types of motors [6]

Characteristics	Stepper	DC	Brushless	AC	Comment

	motor	**motor**	**DC motor**	**motor**	
Low cost	Yes	Yes	Yes	No	Lowest cost is of stepper motor with increase in cost from DC to Brushless DC
Smooth Operation	No	Yes	Yes	No	High performance commutation techniques, such as sinusoidal commutation, will contribute to make brushless DC motor operation smoother
High Speed	No	Yes	Yes	No	Stepper motor usually do not go above 3000 rpm
High Power	No	No	Yes	Yes	Stepper and DC motor do not come over the range of 1 kW
High torque to size ratio	Yes	No	Yes	Yes	Brushless DC motor provides a better spectrum of torque over speed, while stepper motor performance drops significantly at higher speeds
Ease of use	Yes	No	No	No	No feedback and no servo tuning required
Simplest control circuitry	No	Yes	No	No	Other than DC motor all require more than one amplifier circuit.

3.9.2 Calculation of peak torque for the drive motor

The basic criteria for the selection of the drive motors is their torque rating. There are different types of torques but peak torque is the basis of selection. Now the total or the peak torque (T_p) of the drive motor is given by the equation below. This is the factor on which a linear motor can be selected. The peak torque is the maximum amount of torque that a motor can deliver.

$$T_p = T_a + T_f + T_v + T_g \qquad \text{Eq.3.57}$$

T_α = Acceleration torque

T_f = Friction torque

T_ω = Viscous torque

T_g = Gravity torque

The formulation for all these torques are given below.

- Acceleration torque (T_α)

Also named as pull up torque it is called as acceleration torque because at this moment speed changes from zero to maximum velocity.

$$T_\alpha = \frac{j_t \times \alpha}{e} \qquad \text{Eq.3.58}$$

j_t = Total inertia

α = Acceleration

e = Transmission efficiency

Where the total inertia is given as $J_t = J_1 + \frac{N1}{N2^2}^2 (J_2 + J_3) + J_4$ Eq.3.59

J_1 = inertia of pinion

J_2 = Inertia of gear

J_3 = Inertia of feed screw

J_4 = Inertia of table and work piece.

The inertia of the feed screw, table and work piece are given as follows.

$J_3 = \frac{\pi L \rho r^4}{2}$ Eq.3.60

$J_4 = W \times \left[\frac{\delta}{\pi \times \frac{\theta}{180}}\right]^2$ Eq.3.61

L = Length of feed screw

ρ = Density of feed screw material

r = Radius of feed screw rod

W = Weight of work and table

δ = Table movement per pulse

θ = Step angle per pulse

- Friction torque (T_f)

It is the torque caused by the frictional force between the two objects in contact move.

$T_f = \frac{W*\mu}{2\pi*P*e}$ Eq.3.62

μ = Co-efficient of friction

P = Pitch of feed screw

e = Transmission efficiency

- Gravity torque (T_g)

$Tg = \frac{0.0016 \times M_{asm}}{P \times e}$ Eq.3.63

M_{asm} = Mass of the assembly being moved by the motor.

P = Pitch of feed screw

e = Transmission efficiency

3.10 SUPPORTING COMPONENTS

The structure of the machine tool has some non standard supporting components which were designed and then analysed, such as bracket for spindle mounting and the double C-section channel behind the Y sub assembly to provide rigidity to the whole structure and stop it’s deflection. A closed C-section channel structure has been used in this machine tool structure with the purpose of providing rigidity to the machine tool structure. Without this the machine tool assembly would deflect a lot. Firstly an I-section was used, but it could not solve the purpose. Then a double C-section was used with stiffeners fitted in to it. This method reduced the deformation within the permissible range. Simultaneously an excel sheet was programmed for the formulations in the design methodology and was written in the same order to that of the methodology.

Chapter 4
RESULTS AND ANALYSIS

The complete results & analysis reports for the design process of the 3-axis machine tool structure for ornamental carving on granite stone is discussed in this chapter. An MS-Excel sheet for the complete design process was programmed, in which the step by step procedure was listed along with the formulations used. Firstly, the initial parameters were input into this sheet and according to the output values the key components were designed and selected. The steps to be described in this chapter are similar to that discussed in the previous chapter-3 of design methodology. Secondly, a model of the machine tool was prepared in Pro-Engineer (modelling software). All the designed and selected standard components were assembled together to make the final assembly of the machine tool. As the modelling of each part and assembly was done, the FEM analysis of each component and assembly were performed parallely on ANSYS 10. If the deflection of a component was within limits then that iteration of model and analysis was accepted, or otherwise was remodelled and again analysed in the FEM tool. Finally after so many iterations (some of which are shown in this chapter), the final model came into being, which could bear the loads of the machine tool structure.

4.1 INITIAL DESIGN CONSTRAINTS

In reference to the table no () in chapter 3 of design methodology, where all the initial design constraints were defined, here is presented the table for actual values for those parameters. These initial constraints are not universal, but for our design methodology we have chosen them according to our market and web survey. They may vary from the rest.

Table 4.1 : Design constraints

S No	Initial design constraints	Values	Units
1	Maximum tool length, L_t	80	mm
2	Diameter of the largest tool, D	20	mm
3	Maximum depth of the cut, d	8	mm
4	Maximum feed rate, f	1	m/min
5	Cutting speed of the tool, V_c	942	mm/min
6	Maximum acceleration of the drives, α	0.066	m/s^2
7	Specific cutting energy of the granite material, K_e	5000×10^6	J/m^3
8	Tool speed, N	15000	rpm
9	Traverse of the machine tool	$3 \times 2 \times 0.04$	m

4.2 CUTTING FORCES PRODUCED IN MILLING OPERATIONS

In reference to the equation no 3.1 & 3.2 in the previous chapter for calculating cutting force for both end milling and plunge milling, the results are declared below. The value of cutting force for both these cases was calculated on the basis of the initial parameters above mentioned.

Table 4.2 : Values of forces

S No	Type of forces	Value	Units
1	Tangential cutting force in end milling	849.25	N
2	Thrust force in plunge milling	500	N

4.3 SELECTION OF SPINDLE MOTOR

The function of the spindle motor is to rotate the cutting tool with required speed and torque. For this purpose the spindle must be of desirable power and torque capacity. In reference to the eq. No 3.3 &3.6 of the previous chapter, the required power for end and plunging milling operations came out to be 13.33 and 7,85 KW with a torque requirement of 8.49 and 5 N-m respectively. Choosing the maximum value and multiplying with a loss factor of 1.25, the required power comes out to be 16.667 KW. The spindle available in the market nearest to this required value was of 18 KW having a self weight of 47 Kg.

Table 4.3 : Power and torque requirements

S No	Parameters	Power	Units
1	Power requirement in end milling	13.33	kW
2	Power requirement in plunge milling	7.85	KW
3	Torque required in end milling	8.49	N-m
4	Torque required in plunge milling	5	N-m

4.4 MATERIAL PROPERTY

During the analysis of different components in the ANSYS the material property od each component has to be defined. In this case the components are made of materials like stainless steel, aluminum alloy, structure steel and so on. The specific material property chart of each component is defined below.

4.4: Material properties

Material	Component	Density(ρ) (kg/m^3)	Poisson Ratio (μ)	Young's Modolus (E) (GPa)	Ultimate Tensile & Yeild Strength (MPa)
Aluminum alloy 6061	Bracket & Rail Support	2700	0.33	69.9	124 & 55.5
Stainless steel	Bolts , Rails	7750	0.31	193	586 &207
HSS	Tool	8230	0.25	207	960 & 690
Structured steel	All structured components	7850	0.29	205	550 & 430

4.5 DESIGN OF BRACKETS AND BOLTS

The bracket was designed with the purpose of withstanding with the cutting forces and self weight of the assembly without being deflected while holding the spindle motor along with tool in its proper position. It was so designed that clamping force applied on the spindle could be varied and more over 6 no. of bolts were placed symmetrically in order to clamp it properly with the back plate.

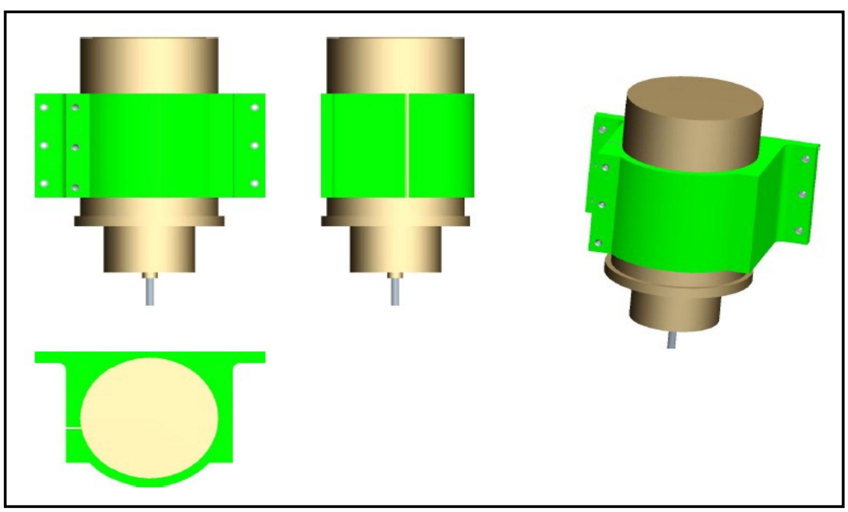
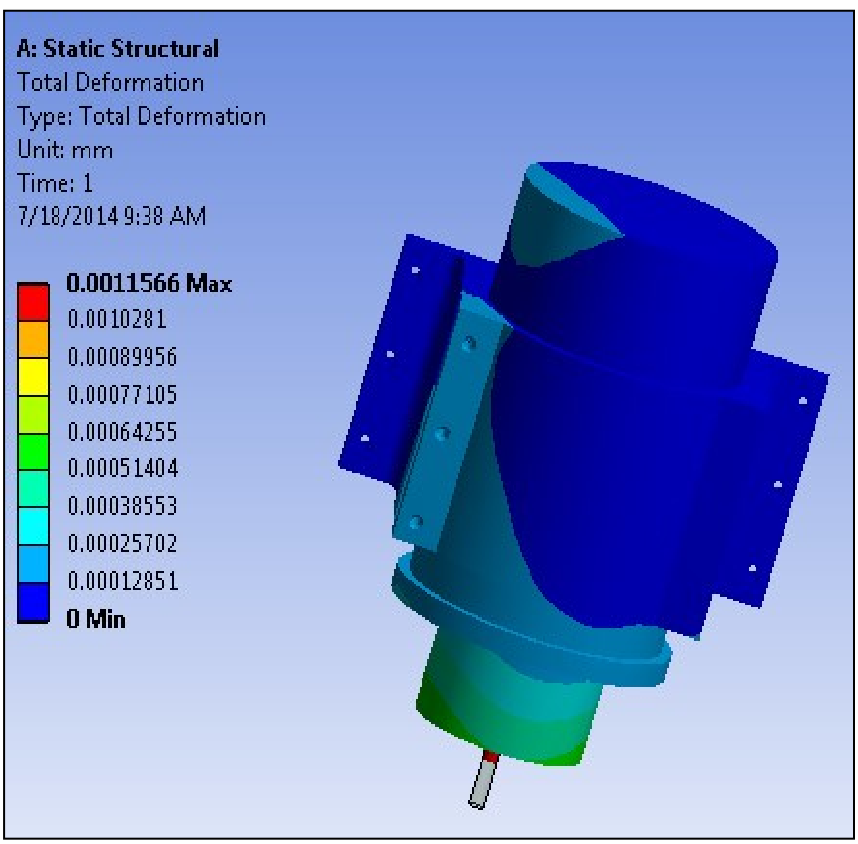


Figure 4:1: Different views of sub-assembly_1

The material of bracket being aluminium alloy and bracket is of structured steel. The bolt material is stainless steel. The material and standard size of the bolts was chosen from the catalogue of unbrako manufacturers. For the design of the bolts, 3 different cases need to be discussed, but before that a new term needs to be introduced i.e. Sub-assembly_1, which is the combination of spindle along with the bracket. The mass of the spindle when analysed in Pro-Engineer came out to be 5.96 kg and the spindle mass as discussed above was 47 Kg. So the total mass of the Sub-Assembly_1 came out to be 52.96 kg.

Case 1 : When cutting force acting along X-axis

In reference to the equation no.3.19 for calculating the diameter of the bolts in the previous chapter, the tensile load on each bolt due to the inertia of sub-assembly_1 and the cutting force, the direct tensile load on each of the 6 bolts came out to be 142.12 N. Now due to the moment produced the tensile load at 1st,2nd and 3rd horizontal row of bolts came out to be 119.03 N ,446.36N and 773.70 N respectively



4:2 : FEM Analysis of bracket for force applied in Y direction

Now the total tensile load i.e. the sum of direct tensile load and that acting at different distances on each bolt of the 1st,2nd and 3rd horizontal row came out to be 261, 588 and 915 N respectively. The orientation of applying force and the FEM analysis for the structural rigidity of the bracket is shown below. The results of the diameter of the bolt chosen and the result of ANSYS analysis is also shown.

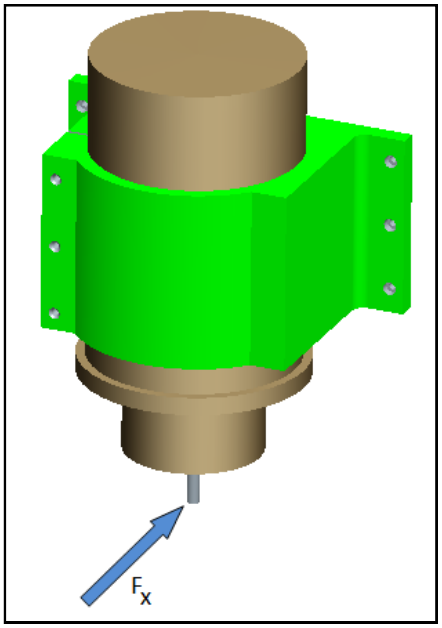


Figure 4:3 : Force acting along X direction

Thus in reference to the equation 3.19 for calculating the bolt diameter in the previous chapter, the diameter of the bolt having tensile stress 650000000 N/m^2 and considering a FOS of 3 comes out to be 2.3 mm. The results of the case discussed above for the design of bracket and bolts have been summarized in the table below.

Table 4.5: Diameter of bolts for X direction movement of tool

Case 1 (Tool cutting in X direction)			
S No	**Parameters**	**Values**	**Units**
1	Mass of sum-assembly_1 (Spindle +Bracket)	52	Kg
2	Total load acting on the bolts, W	852.75	N
3	Total load on each bolt, W/N_b ($N_b$being no. of bolts)	142.125	N
4	Max. vertical distance of a bolt row from twisting edge, L_{v3}	0.130	mm
5	Maximum tensile load on any of the bolt rows, W_{mtb}	915.8	N
6	Diameter of the clamping bolt, d_{bx}	2.3	mm

Case 2 (When cutting horizontally in Y-axis)

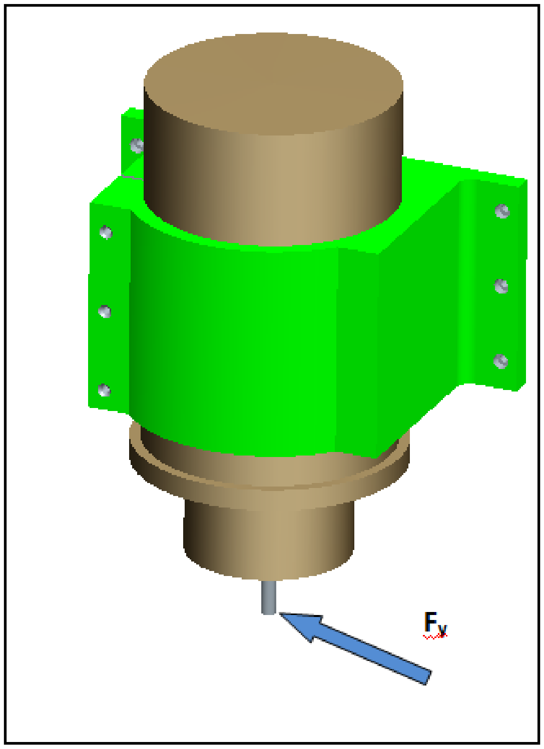


Figure 4:4 : Force acting along Y direction

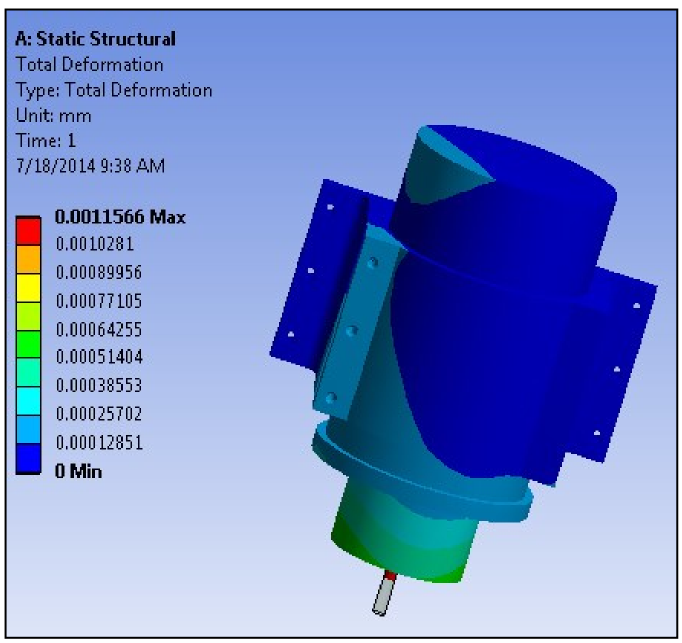


Figure 4:5: FEM analysis of bracket for force applied in Y direction

Now after calculating the angle between the direct and secondary shear load of all the 6 bolts starting from 1st to 6th in order as shown in the diagram below, the resultant of the direct and the secondary shear load according to the equation no 3.28 to 3.33 comes out to be 338, 337, 338, 338, 337, 338 N respectively. Choosing the maximum out of all the resultants the value the diameter of the bolt according to equation no 3.34 and with allowable shear stress to be 325000000 N comes out to be 2.1 mm. The results of the case discussed above for the design of bracket and bolts have been summarized in the table below.

Table 4.6 : Diameter of bolts for Y direction movement of tool

Case 2: (Tool cutting in Y direction)			
S No	**Parameters**	**Values**	**Units**
1	Load acting on the bolted joints, W	852.75	N
2	Eccentricity of the load, e_y	0.0245	m
3	No. of bolts, N_b	6	
4	Direct shear load on each bolt, $W_s = W/N_b$	142.1254	N
5	Max. secondary shear load, S_{fb6}	253.1223	N
6	Max. resultant of direct & shear load, R_{s6}	388.875	N
7	Diameter of the bolt, d_{by}	2.1	mm

Case 3 (When cutting vertically in Z direction)

This is the case of plunge cutting i.e. when the tool plunges in to the granite sheet vertically in the Z direction.

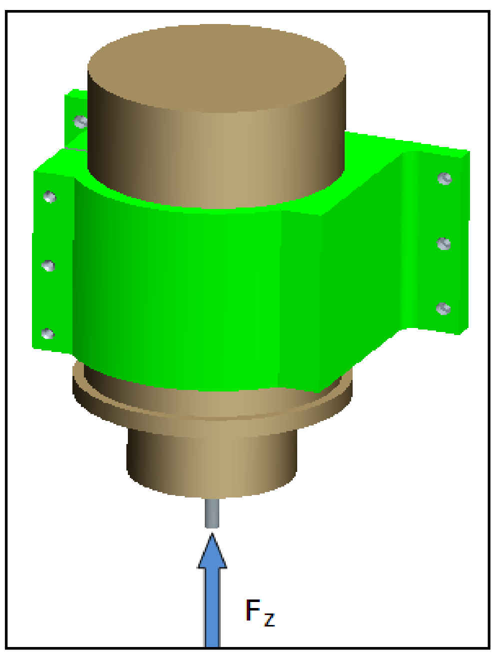


Figure 4:6 : Force acting along vertically upwards (-Z) direction

The load on the clamping bolts, which is the sum of the inertial and the plunging force comes out to be 503 N and thus the direct shear load on each bolt to be 83.9 N. After calculating equivalent tensile and shear loads from the equations 3.37 and 3.38 which is 95.9 and 84.6 N respectively, the value of the diameter of the clamping bolts comes out to be 1 mm (reference equation no 3.41). The results of the case discussed above for the design of bracket and bolts have been summarized in the table below.

Table 4.7 : Diameter of bolt for Z direction movement of tool

S No	Case 3: (Tool plunging in vertically downwards "-Z" direction)		
1	**Parameters**	**Values**	**Units**
2	Load on the bolts (Inertial force + Thrust force), W_p	503.49	N
3	Direct shear load on each bolt, $W_{ds} = W_p/N_b$	83.91	N
4	Eccentricity of load, e_z	0.0951	m
5	Equivalent tensile load, W_{te}	95.08	N
6	Equivalent shear load, W_{se}	84.67	N
7	Diameter of bolt in tension, d_{btz}	0.8	mm
8	Diameter of bolt on shear load, d_{bsz}	1.00	mm

4.6 DESIGN AND SELECTION OF Z AXIS DRIVE ASSEMBLY

The drives assembly for the machine tool structure consists of many key components such as ball-screws, round guideways, linear motors, couplings etc. As in our study we have predefined our objective of designing the machine tool with the help of standard components, so in our survey we came across a manufacturer making standard round shaft driven drive assemblies as a complete structure, comprising of ball screws, round guideways etc. The complete design procedure of all these components have been discussed in the previous chapter and now the results of that analysis have been discussed below.

4.6.1 Calculation of Z traverse

The calculation for the traverse of the Z drive depends on the factors such as length of mounting plate and clearance required on both sides etc. Here the sub-assembly_2 also comes in to play. The sub-assembly_2, in addition to sub-assembly_1 comprises of the back plate of the z axis drive. The complete details of this is given below.

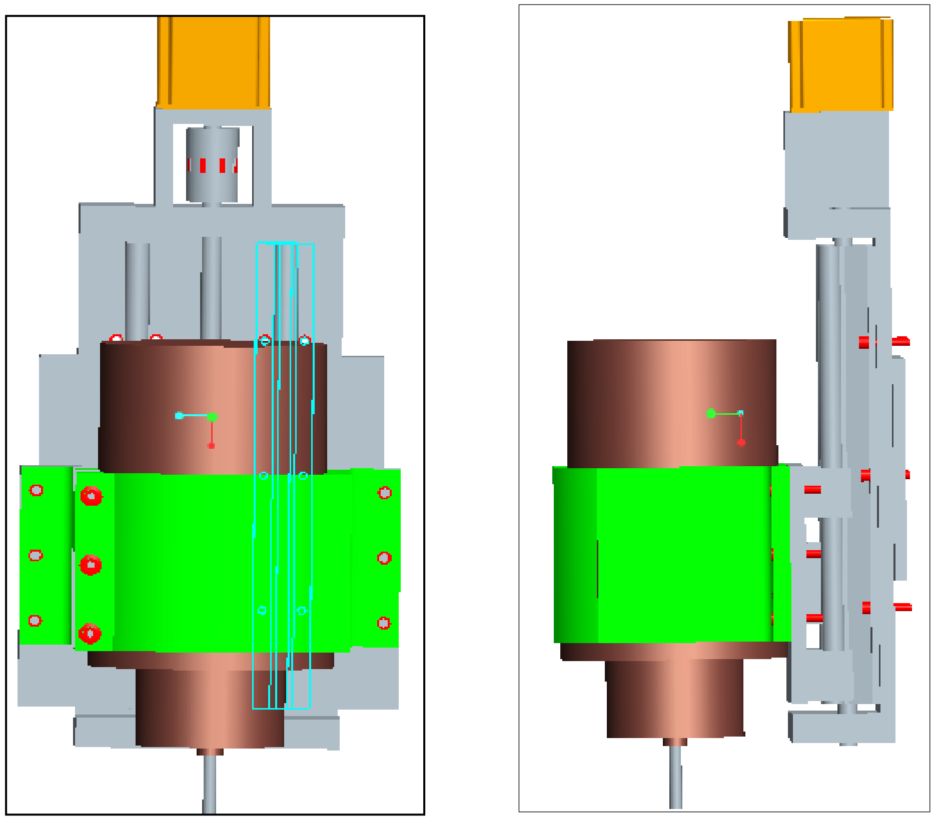


Figure 4:7 : Z axis drive assembly

Table 4.8 : Z axis traverse

S No	Parameters	Values	Units
1	Mass of sub-assembly_2	62	kg
2	Z- axis Traverse required, L_z	130	mm
3	Length of the Bracket mounting plate, L_{pz}	177.8	mm
4	Clearance required on each end of Z-axis slide, L_{cz}	25	mm
5	Total length of the Z drive, L_{dz}	357.8	mm
6	Standard Z-axis slide length available, S_{lz}	406.4	mm

4.6.2 Selection of round shaft guideways

As discussed earlier, that the Z-drive as a whole unit comprising of all the standard components have been selected from the **PCB** linear website (Round shaft technology catalogue). The selection procedure for round shaft guideways has been discussed earlier in the previous chapter and now the results have are being discussed. The diameter and length of the round shaft, when selected by the standard procedure (mentioned in the catalogue) came out to be 15.875 mm and 406.4 mm respectively. But when the drive was analysed it was realised that this model would not serve the purpose, as the deflection were out of permissible ranges. Therefore the next standard model of same length was selected with a rail diameter of 19.5 mm and it very well resisted all the forces. The complete detail of the selection procedure and the iterations involved in this design is given below. Material properties for the structure and their applications are presented in the following table.

a) Iteration 1 (FEM analysis of sub-assembly_2)

In first iteration of the sub-assembly_2 it was analyzed against moments produced in the X, Y and Z directions because of the tangential and the plunging force of 850 and 500 N respectively. During the analysis it was revealed that the total deformation was 23 µm (max-X Deformation-10 µm; max-y Deformation-5 µm; max-Z Deformation-10 µm).

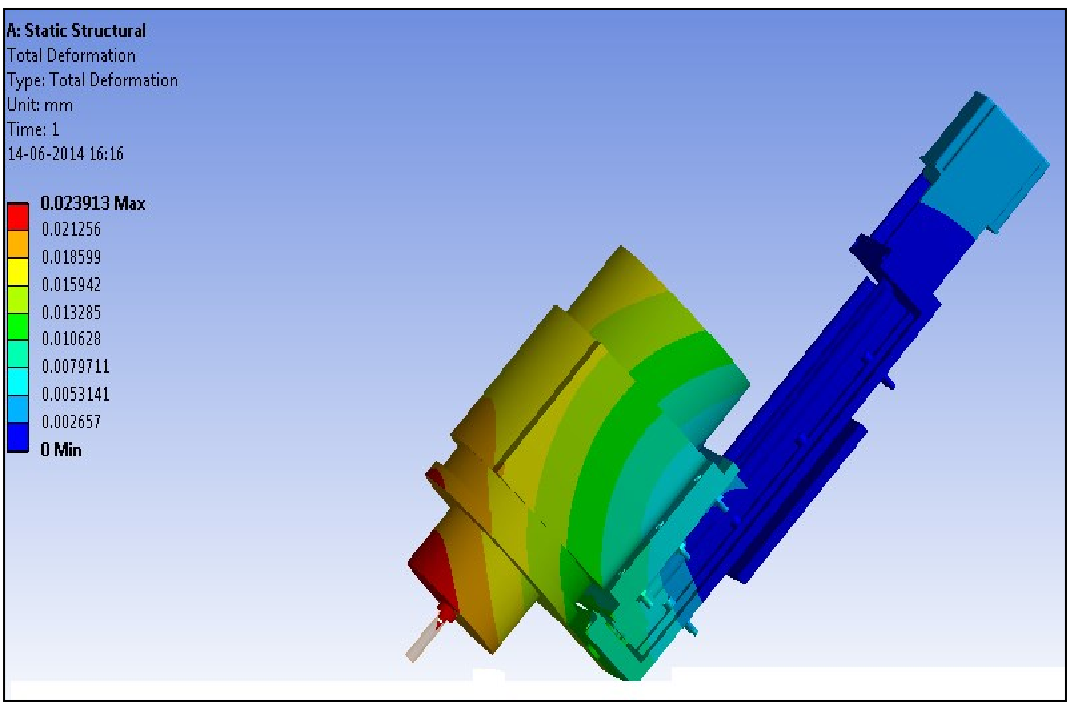


Figure 4:8 : Iteration 1 , Total deformation in X direction

b) Iteration 2 (FEM analysis of sub-assembly_2)

In the 2nd iteration the rail diameter was increased to 19.5 mm and the assembly was analysed .Below are the results for the total deflection in the case loads applied in X, Y and Z direction respectively. The maximum total deflection in this case came out to be 7 µm (max-X Deformation-2 µm; max-y Deformation-1 µm; max-Z Deformation-6 µm).

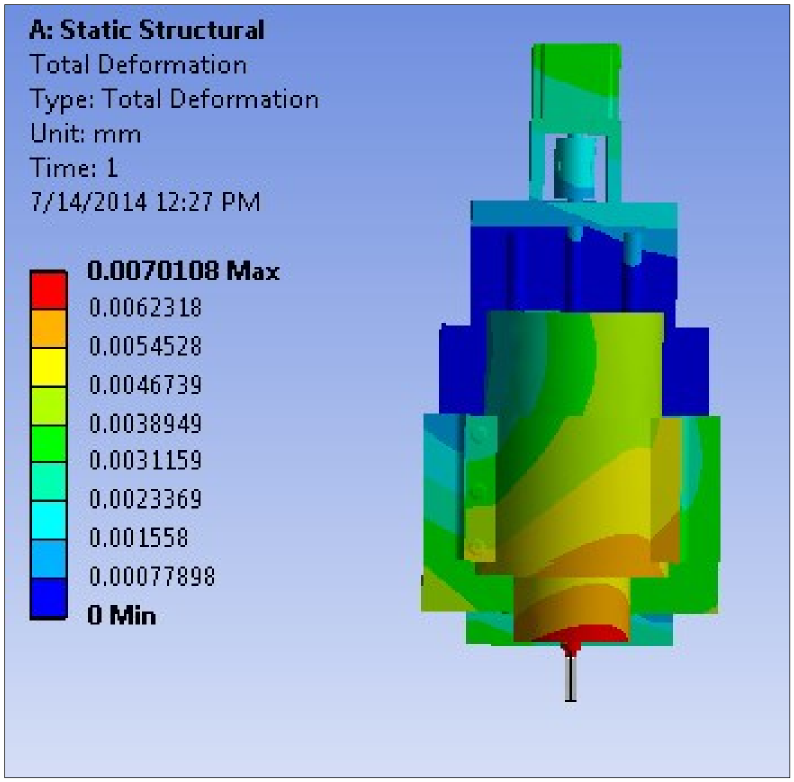


Figure 4:9 : Iteration 2, Total deformations in X direction

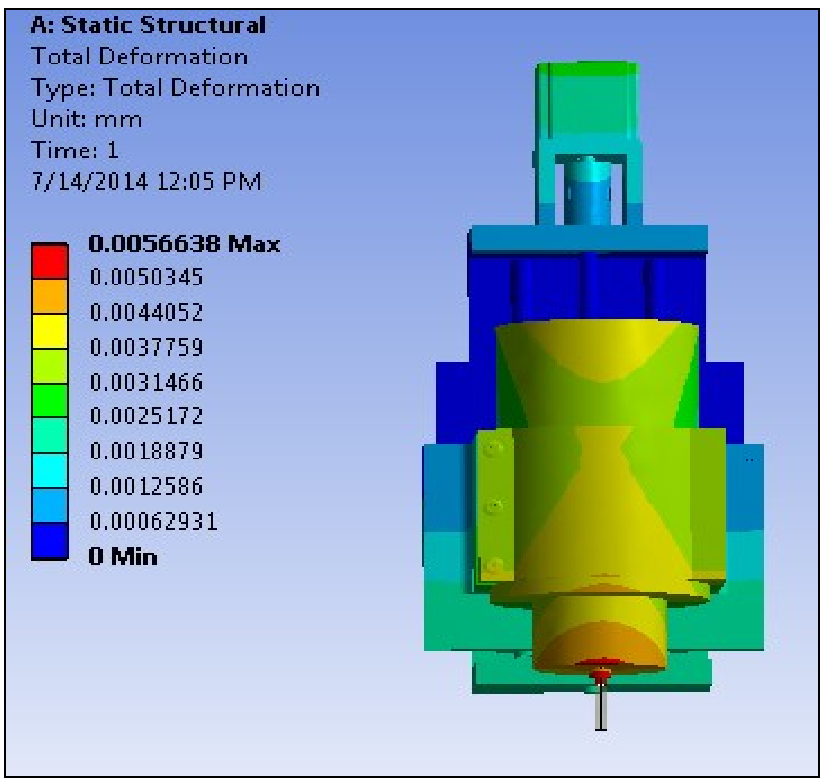


Figure 4:10 : Iteration 2, Total deformation in Y direction

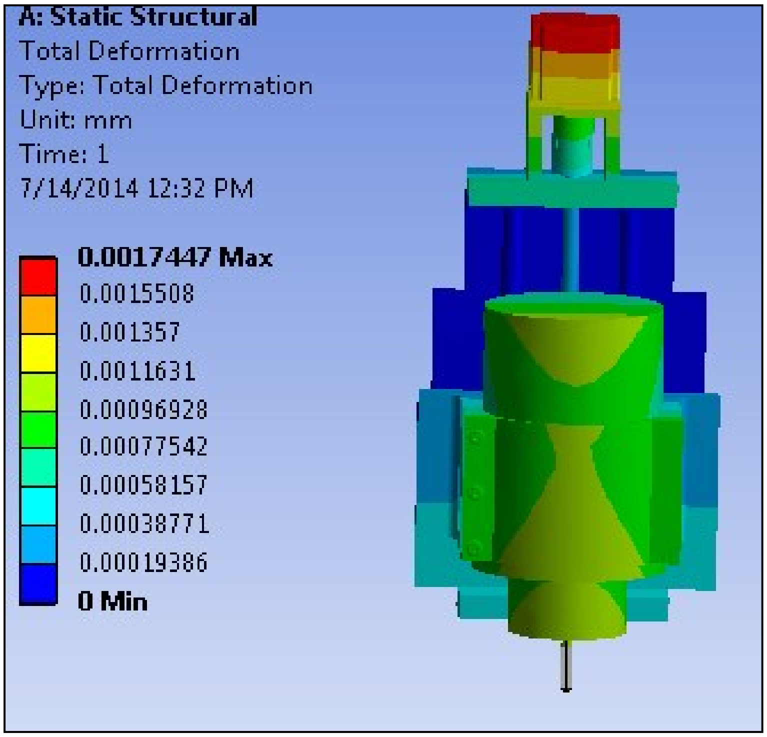


Figure 4:11 : Iteration 2, Total deformation in Z direction

Table 4.9 : Selection of diameter for Z round shaft guideways

Selection of round shaft guideways for Z axis drive assembly		
Parameters	**Values**	**Units**
Maximum Bending Moment in Y axis due to tangential force, M_Y	277.7070	N-m
Maximum Bending Moment in Y axis due to plunge force, M_y	2.8036	N-m
Maximum Bending Moment in X axis due to tangential force, M_x	121.8684	N-m
Allowable bending moment in X direction, M_{ya}	311	N-m
Allowable bending moment in Y direction, M_{xa}	311	N-m
Total deformation in iteration 1 (rail diameter to be 15.875mm)	23×10^{-6}	m
Total deformation in iteration 2 (rail diameter to be 19.5 mm)	7×10^{-6}	m

4.6.3 Selection of ball screw

The ball screw being a standard component of the Z drive unit, was selected by the standard procedure as described below. The motor rated speed being 60rpm and the minimum linear speed being 0.005m/s, the lead of the screw came out to be 0.005m. As the value of the acceleration calculated came out to be 0.033 m/s^2 and the guide surface resistance being

531.2 N, the maximum value among of all the axial loads during acceleration and deceleration came out to be 651.59 N. The sliding screw efficiency being 0.85, the driving torque required to obtain the thrust force of the ball screw came out to be 0.6103 N-m and the diameter according to the maximum allowable stress came to be 10mm. But the standard available screw diameter was 16mm. After analysis on ANSYS 10 the ball screw came to be 19.5 mm. The brief description of this procedure is given below.

Table 4.10 : Selection of ball screw diameter

Ball screw design / selection for Z axis drive assembly		
Parameters	**Values**	**Units**
Motor rotational speed, N_m	60	rpm
Maximum linear speed, s	0.005	m/s
Lead of the ball screw, l	0.005	m
Linear acceleration, α	0.033	m/s^2
Guide surface resistance, F_{fr}	531.24	N
Maximum of all the axial loads, $F_{a\ max}$	651.59	N
Screw efficiency (sliding), η_{sc}	0.85	
Torque required to overcome thrust, T_t	0.6103	N-m
Diameter of the screw, d_{scr}	10	mm
Diameter of standard ball screw, $d_{standard}$	15.875	mm
Diameter of standard ball screw selected after analysis, $d_{standard\_1}$	19.5	mm

4.6.4 Selection of linear motors for Z axis drive assembly

The function of the linear motor is to rotate the ball screw by providing it with the required torque. The ball screw in turn moves the plate of the drive unit. The type of motor used in this design is brushless DC motor having low cost, smooth operation and with high torque and power as it's specifications. The selection procedure for the motor selection was described in the previous chapter and the results came out that the peak torque requirement for Z axis motor was 3.44 N-m. The standard motor which could satisfy this need of Z axis was NEMA 23. The overall results are described below.

Table 4-11 : Selection of Z motor torque

SELECTION OF LINEAR MOTOR FOR Z AXIS DRIVE UNIT		
PARAMETERS	**VALUE**	**UNITS**
Inertia of feed screw, J_3	0.0003704	$Kg - m^2$
Inertia of work and table, J_4	0.0015	$Kg - m^2$
Acceleration torque, T_a	0.0158	N-m
Gravity torque, T_g	0.6183	N-m
Friction torque, T_f	3.0182	N-m
Transmission efficiency, e	0.8	N-m
Thrust Torque (Driving torque to obtain thrust force), T_t	0.6103	N-m
Peak torque, T_p	4.2626	N-m

4.7 DESIGN OF Y AXIS DRIVE ASSEMBLY UNIT

The design of the Y axis drive unit is same as that of the Z axis. Same no. of components like round shaft guideways, ball screw and drive motors being there. For this design firstly the sub-assembly_3 was defined which was the combination of sub-assembly_2 and Z axis base drive unit. So the complete Z axis assembly is termed as sub-asssembly_3. The mass of this assembly being 82,5 Kg. Further the traverse of the Y axis drive was calculated and it came out to be 2304 mm, but the standard length available was 2438 mm which was then chosen. Then came the selection of round guideways, in which the maximum bending moments in the X and Y directions came out to be 301.4 and 360.08 N-mm respectively. Whereas the permissible bending moments in these directions were 373 and 501 N-m respectively. Thus the model with rail diameter 25mm was selected for the Y axis drive unit but the ANSYS analysis revealed that the deflection was high so the next model with rail diameter 31.25 was selected which in fact served the purpose. Similarly the selected diameter of the ball screw came out to be 18 mm whereas the standard available diameter of the ball screw was 2.32 mm. Proceeding further in the design and selection procedure, the peak torque of the linear motor came out to be 3.44 N-m. Thus the whole drive assembly components were selected. After the selection of components a back structure was also fixed to the back of Y assembly, which provided the strength against the bending of the complete structure. Several iterations were performed on these back structures to red [3] [4]uce the deflection of the structure. The complete results and the iterations for the design of this drive unit are presented below.

For reducing the deflection in the Y-sub assembly supporting the Z-sub assembly, there is needed a support element at the back of the Y assembly. So many different structures were tried for this purpose like I-section or C- section and finally an assembled double c-section was made which reduced the total deflection in the structure to $13\mu m$.

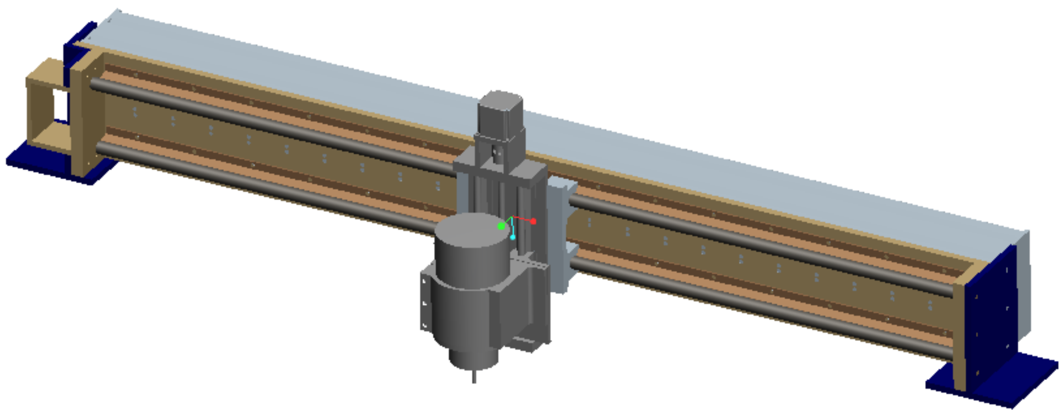


Figure 4:12: Y assembly view

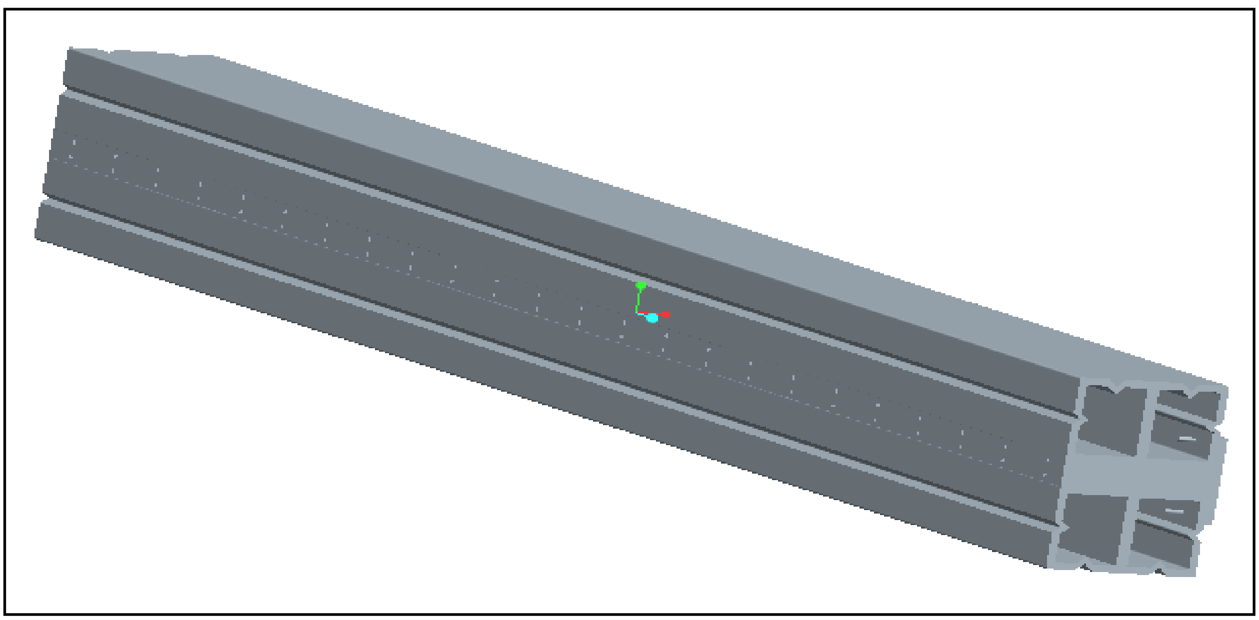


Figure 4:13 : Supporting double C-section assembly

4.7.1 Calculation of Y traverse

The calculation for the traverse of the Y drive depends on the factors such as length of mounting plate and clearance required on both sides etc. Here the sub-assembly_3 also comes in to play. The sub-assembly_3, in addition to sub-assembly_3 comprises of the base unit assembly of the z axis drive. The complete details of this is given below:

Table 4.12: Y axis traverse

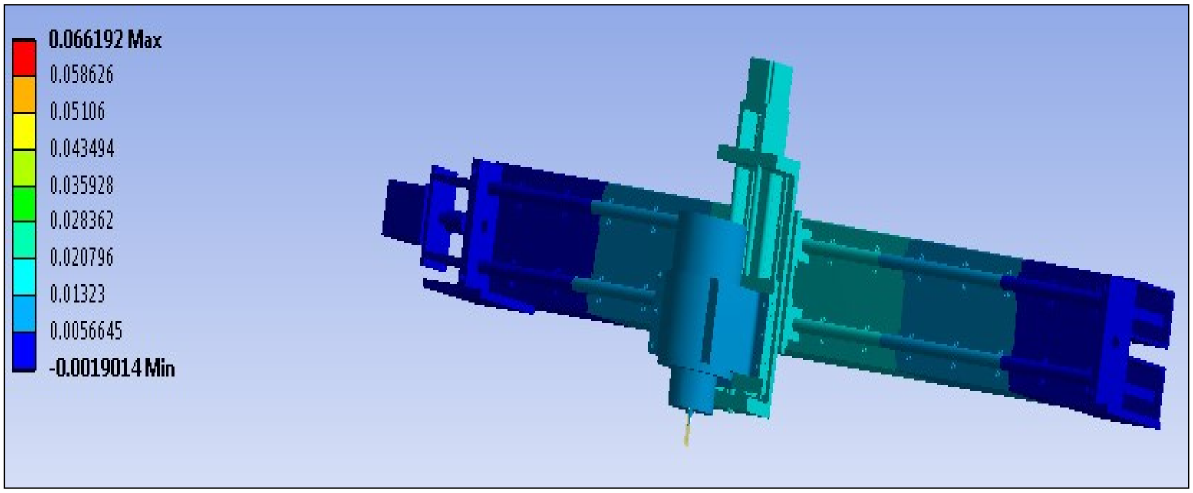
S No	Parameters	Values	Units
1	Mass of sub-assembly_3	82.5	kg
2	Y- axis Traverse required, L_y	2000	mm
3	Length of the Bracket mounting plate, L_{py}	254	mm
4	Clearance required on each end of Y-axis slide, L_{cy}	25	mm
5	Total length of the Y drive, L_{dy}	2304	mm
6	Standard Y-axis slide length available, S_{ly}	2438	mm

4.7.2 Selection of round shaft guideways

As discussed above, again the Y-drive as a whole unit comprising of all the standard components have been selected from the PCB linear website (Round shaft technology catalogue). The selection procedure for round shaft guideways has been discussed earlier in the previous chapter and now the results have are being discussed. The diameter and length of the round shaft, when selected by the standard procedure (mentioned in the catalogue) came out to be 25.4 mm and 2438 mm respectively. But when the drive was analysed it was realised that this model would not serve the purpose, as the deflection were out of permissible ranges. Therefore the next standard model of same length was selected with a rail diameter of 31.25 mm and it very well resisted all the forces. The complete detail of the selection procedure and the iterations involved in this design is given below. But here there is a need to define sub-assembly_3 which is the combination of sub-assembly_2 and Z axis base drive assembly.

a) Iteration 1 (FEM analysis of sub-assembly_3)

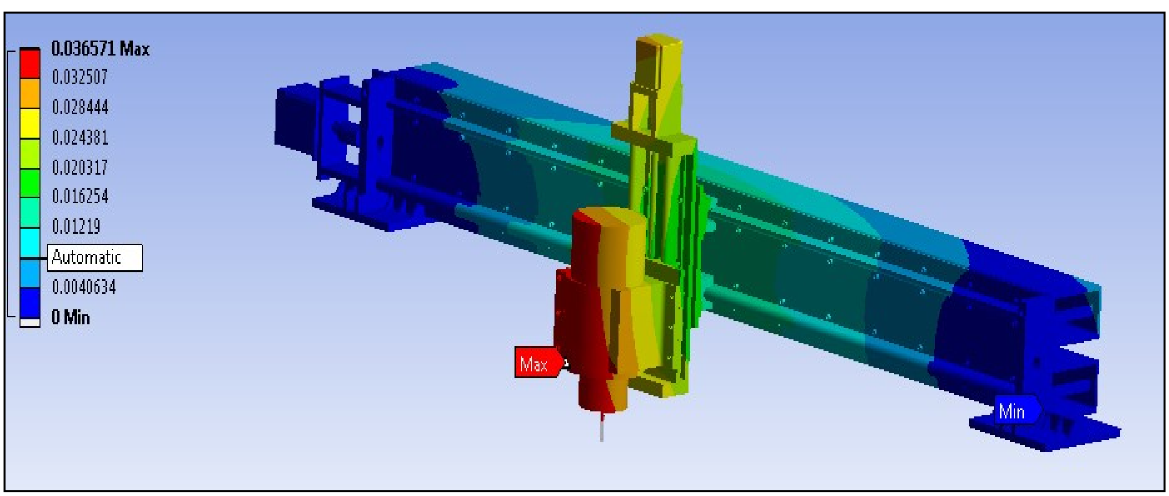
In first iteration of the sub-assembly_3 it was analyzed against moments produced in the X, Y and Z directions because of the tangential and the plunging force of 850 and 500 N respectively. In this iteration the rail diameter for Y axis was 25.4 mm and the thickness of the stiffeners and plate for the double C section at the back was also 8 mm.During the analysis it was revealed that the total deformation was 66 µm (max-X Deformation-10 µm; max-y Deformation-5 µm; max-Z Deformation-10 µm).



4:14: Iteration 1 of Sub-assembly_3

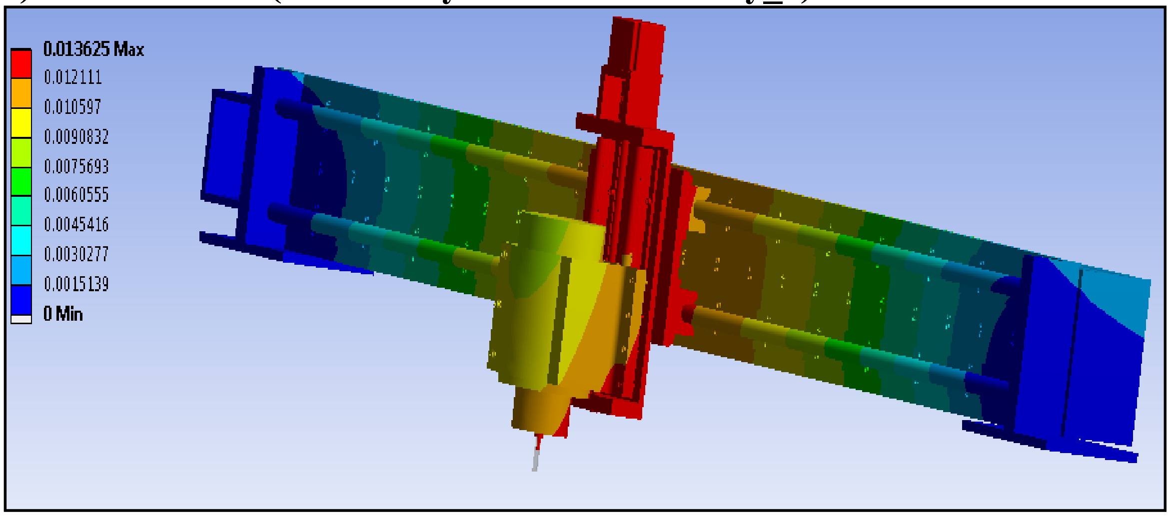
b) Iteration 2 (FEM analysis of sub-assembly_3)

In this iteration the rail diameter for Y axis was same i.e. 25.4 mm but the thickness of the stiffeners and plate for the double C section at the back was 10 mm.The maximum total deflection in this case came out to be 7 µm (max-X Deformation-2 µm; max-y Deformation-1 µm; max-Z Deformation-6 µm).



4:15 : Iteration 2 of Sub-assembly_2

c) Iteration 3 (FEM analysis of sub-assembly_3)



4:16 : Iteration 3 of sub-assembly_3

In this iteration the diameter of the rails was chosen to be 31.25 mm and the thickness of the back structure was 10mm. But the back structure was geometrically changed. There were end plates foxed to the back structure and a support was provided in between the 2 C-sectioned channels. Thus the deflection reduced to 13 μm. In this case which was quite acceptable

Table 4-13: Selection of round shaft guideways for Y assembly

Selection of round shaft guideways for Y axis drive assembly		
Parameters	**Values**	**Units**
Maximum Bending Moment in Y axis due to tangential force, M_Y	360	N-m
Maximum Bending Moment in Y axis due to plunge force, M_y	112	N-m
Maximum Bending Moment in X axis due to tangential force, M_x	302	N-m
Allowable bending moment in X direction, M_{ya}	501	N-m
Allowable bending moment in Y direction, M_{xa}	373	N-m
Total deformation in iteration 1 (rail diameter to be 25.4 mm)	66×10^{-6}	m
Total deformation in iteration 2 (rail diameter to be 25.4 mm)	36×10^{-6}	m
Total deformation in iteration 3 (rail diameter to be 31.25 mm)	13×10^{-6}	m

4.7.3 Selection of ball screw

The ball screw being a standard component of the Y drive unit, was selected by the standard procedure as described below. The motor rated speed being 60rpm and the minimum linear speed being 0.0064m/s, the lead of the screw came out to be 0.0064m. As the value of the acceleration calculated came out to be 0.0423 m/s^2 and the guide surface resistance being 1076 N, the maximum value among of all the axial loads during acceleration and deceleration came out to be 2848 N. The sliding screw efficiency being 0.85, the driving torque required to obtain the thrust force of the ball screw came out to be 3.38 N-m and the diameter according to the maximum allowable stress came to be 18mm. But the standard available screw diameter was 16mm. The brief description of this procedure is given below.

Table 4.14 : Selection of ball screw diameter

Ball screw design / selection for Z axis drive assembly		
Parameters	**Values**	**Units**
Motor rotational speed, N_m	60	rpm
Maximum linear speed, s	0.0064	m/s
Lead of the ball screw, l	0.0064	m
Linear acceleration, α	0.0423	m/s^2
Guide surface resistance, F_{fr}	1076	N
Maximum of all the axial loads, $F_{a\,max}$	2848	N
Screw efficiency (sliding), η_{sc}	0.85	
Torque required to overcome thrust, T_t	3.38	N-m
Diameter of the screw, d_{scr}	18	mm
Diameter of standard ball screw, $d_{standard}$	25.4	mm
Diameter of standard ball screw selected after analysis, $d_{standard\_1}$	25.4	mm

4.7.4 Selection of linear motors for Y axis drive assembly

The function of the linear motor is to rotate the ball screw by providing it with the required torque. The ball screw in turn moves the plate of the drive unit. The type of motor used in this design is brushless DC motor having low cost, smooth operation and with high torque and power as it's specifications. The selection procedure for the motor selection was described in the previous chapter and the results came out that the peak torque requirement for Z axis motor was 3.44 N-m. The standard motor which could satisfy this need of Y axis was NEMA 43. The overall results are described below.

Table 4.15 : Selection of Y axis linear motor

SELECTION OF LINEAR MOTOR FOR Y AXIS DRIVE UNIT		
PARAMETERS	**VALUE**	**UNITS**
Inertia of feed screw, J_3	0.00074222	$Kg - m^2$
Inertia of work and table, J_4	0.0038	$Kg - m^2$
Acceleration torque, T_a	0.0373	N-m
Gravity torque, T_g	1.9102	N-m
Friction torque, T_f	2.148	N-m
Transmission efficiency, e	0.8	N-m
Thrust Torque (Driving torque to obtain thrust force), T_t	7.3887	N-m
Peak torque, T_p	11.4408	N-m

Chapter 5
CONCLUSION

The present study was an effort towards the development of a methodology for the design and selection of standard and non standard components for a CNC granite carving machine tool. By using the empirical relationships firstly, the standard components such as ball screw, round shaft guideways and spindle motor were selected. Further the non standard or the supporting elements such as bracket for spindle holding and double c-section for back support were designed using the concepts of machine design and analysis on ANSYS (an FEM tool). The complete design and selection was a hierarchical process which started from the tool selection, to spindle selection, to bracket design and then followed by selection of ball screws and round shaft rails. Each component was simultaneously analyzed and then a final structure was modelled and analyzed. After performing all the analysis work, which was a rigorous process of applying load in different directions and orientations, some results were found out, from which it was concluded that the deformation in each component analyzed was within the limits. This meant that the designed machine tool was safe and efficient enough in order to withstand the predefined loading conditions.

Future scope

The presented wok on the development of methodology for design and selection of components for 3 axis machine tool can be extended in future in some ways which have been discussed below.

1. The 3rd drive axis assembly which is the X axis can be designed and analyzed same as done in the case of Z axis and Y axis.
2. As of now the Structural analysis have been performed but the modal and the dynamic analysis can also be performed to analyze the machine tool structure against vibrations and inertia respectively.
3. The present work can also be extended in the direction of making this machine tool more rigid and light weight [16], by using composite materials or sandwich type structural elements.
4. Moreover in place of round shaft guideways some other arrangements, which are more stiff and suitable for granite cutting can be used with high material properties.

Chapter 6
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