FABRICATION AND DESIGN EVALUATION USING CAE TOOLS FOR A 3-AXIS VERTICAL MILLING MACHINE FOR SCULPTURED SURFACE MACHINING

A Thesis Report

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MASTER OF ENGINEERING
IN
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Submitted by

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CERTIFICATE

This is to certify that the thesis titled, "**FABRICATION AND DESIGN EVALUATION USING CAE TOOLS FOR A 3-AXIS VERTICAL MILLING MACHINE FOR SCULPTURED SURFACE MACHINING**" being submitted by "**Mr. AKHIL MAHAJAN, REGISTRATION NO. 801081001**", in partial fulfilment of the requirements for the award of degree of MASTER OF ENGINEERING in CAD/CAM & ROBOTICS ENGINEERING at THAPAR UNIVERSITY, PATIALA is a bonafide work carried out by him under our guidance and supervision and that no part of this thesis work has been submitted before for the award of any degree at any other place.

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i

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ABSTRACT

CNC technology has proved to be of great importance for manufacturing sector. The use of contemporary CNC systems along with higher computational and memory storage capacity of modern computer systems has redefined the shop floor automations strategies for modern manufacturing sectors. One of the biggest challenges today is how to make the CNC technology affordable, simpler and available to an extent that even a hobbyist/ occasional user can think of owing such technology. The aim of the present work is to evaluate the design of a new 3-axis CNC milling machine tool structure designed for sculptured surface machining using Finite Element Method (FEM) analysis in ANSYS.

The machine tool structure constitutes a significant part of total cost of a CNC system. Thus a simpler cost effective machine tool design can greatly reduce the overall cost of a CNC system. The problem is not only about the reduction in the machine tool fabrication, but other costs associated with usage, repair and up-gradation or enhancements should also be lesser. Thinking this as an overall objective a simplified machine tool unit for sculptured surface machining has been designed, and under the present work an overall shape to the conceptual design has been given. The main standard components required for developing the prototype of machine tool structure has been selected following the approach of kind of loading the machine tool will be subjected. The magnitude of cutting forces has been calculated after fixing the range of cutting parameters and types of cutting tools to be used. A systematic study has been done for the selection of the drive motors and router as spindle drive. The objective of maximizing the tool traverse in X, Y and Z direction so as to enable to cutting tool to approach the maximum machining volume for material removal subject to the constraint of sizes of standard available machine components used in CNC machines has also been studied.

Conceptual design for the machine tool which was achieved after a number of iterations, has been analysed using finite element method (FEM) using ANSYS for its stability under different loading conditions. The detailed FEM study has been carried out in two main phases, for analysis of overall machine structure and secondly for router assembly. As discussed in results and discussion, it has been found that the machine structure design is safe and the maximum level of deflections and stresses found during FEM analysis are in safe limit. The logical conclusions drawn from the present study have also been discussed along with the future scope of work in this area.

INDEX

CONTENTS		PAGE NO.
CERTIFICATE		i
ACKNOWLEDGEMENT		ii
ABSTRACT		iii
LIST OF FIGURES		vi
LIST OF TABLES		viii
CHAPTER 1: INTRODUCTION		**1-15**
1.1	HISTORY OF CARVING MACHINERY	1
1.2	AUTOMATION OF CUTTING OPERATIONS	2
1.3	AUTOMATION OF MACHINING OPERATION USING COMPUTER NUMERIC CONTROL (CNC)	3
1.4	MAIN COMPONENTS IN CNC MACHINING CENTRE	4
1.5	NC PART PROGRAM	5
1.6	MACHINE STRUCTURE	5
1.7	MACHINE CONTROL UNIT	12
1.8	TOOLING AND ACCESSORIES	13
1.9	PRESENT WORK	13
CHAPTER 2: LITERATURE REVIEW		**16-23**
CHAPTER 3: SELECTION OF DRIVE MOTORS		**24-30**
3.1	SELECTION OF SPINDLE MOTOR AND DRIVE MOTORS	26
3.2	SELECTION DRIVE MOTOR POWER	28
CHAPTER 4: SELECTION OF STANDARD COMPONENTS		**31-40**
4.1	DESIGN PROCEDURE	31
4.2	DESIGN OF MACHINE TOOL STRUCTURE PROTOTYPE	33
4.3	SELECTION OF STANDARD COMPONENTS	35

CHAPTER 5: COMPUTER AIDED ANALYSIS OF MACHINE STRUCTURE 41-68

5.1 GENERAL PROCEDURE FOLLOWED FOR FINITE ELEMENT METHOD (FEM) ANALYSIS IN ANSYS 41

5.2 FEM ANALYSIS OF MAIN MACHINE STRUCTURE INCLUDING ALL COMPONENTS 46

5.3 MECHANICAL STABILITY OF ROUTER MOUNTING SUB-ASSEMBLY 53

5.4 FABRICATION AND ASSEMBLY OF MACHINE COMPONENTS 66

CHAPTER 6: CONCLUSION AND FUTURE SCOPE OF WORK 69-70

6.1 CONCLUSIONS OF PRESENT WORK 69

6.2 FUTURE SCOPE OF WORK 69

REFERENCES 71-73

LIST OF FIGURE

FIGURE NUMBER		PAGE NO.
1.1	CNC MACHINE	3
1.2	SCHEMATIC DIAGRAM OF CNC	4
1.3	FIXED BASE ROUTER	8
1.4	PLUNGE ROUTER	9
1.5	ADVANTAGES OF RECIRCULATING BALL SCREW	11
1.6	ARRANGEMENT OF BALL SCREW	11
1.7	CIRCULAR ARC GROOVES	12
1.8	GOTHIC ARC GROOVES	12
3.1	BALL END MILL	24
3.2	FLAT END MILL	25
3.3	RADIUSED END MILL	25
3.4	STRAIGHT FLUTE TOOL	25
3.5	HELICAL FLUTE TOOL	26
3.6	WOOD CUTTING ROUTER	26
3.7	SERVO MOTOR FOR THE FEED DRIVES	30
4.1	DESIGN PROCEDURE	31
4.2	ITERATION 1.0	33
4.3	ITERATION 2.0	34
4.4	ITERATION 3.0	35
4.5	LOAD LAYOUT OF X-AXIS	38
4.6	LOAD LAYOUT OF Y-AXIS	39
4.7	LOAD LAYOUT OF Z-AXIS	39
5.1	FINITE ELEMENT METHODOLOGY	42
5.2	3-AXIS MACHINE TOOL MODEL FOR ANALYSIS	47
5.3	ACTIVE ELEMENTS IN MODEL FOR STRUCTURAL ANALYSIS	47
5.4	MESH OF MACHINE STRUCTURE	48
5.5	BOUNDARY CONDITIONS FOR THE MODEL	48
5.6	DETAILS OF THE EQUIVALENT STRESS	50
5.7	BACK OF Y-AXIS FOR VISUAL REALISM OF MAXIMUM EQUIVALENT STRESS	50
5.8	ZOOM IN OF BS CONNECTOR PLATE	51
5.9	DEFORMATION OF WHOLE STRUCTURE	51
5.10	DETAILED DEFORMATION ON THE VERTICAL COLUMN	52

5.11	DEFORMATION ON THE END PLATE OF Y-AXIS	52
5.12	DEFORMATION ON THE BACK PLATE OF Y-AXIS	53
5.13	ROUTER SUB-ASSEMBLY	55
5.14	MESH ELEMENTS IN ROUTER SUB-ASSEMBLY	55
5.15	BOLT PRETENSION AND FORCE APPLIED IN Z-DIRECTION	56
5.16	DURATION OF FORCE APPLIED ON TOOL TIP	56
5.17	EQUIVALENT STRESS ON MOUNTING BRACKET	58
5.18	GRAPH OF EQUIVALENT STRESS ON MOUNTING BRACKET	58
5.19	DEFORMATION IN MOUNTING BRACKET	59
5.20	GRAPH OF DEFORMATION IN MOUNTING BRACKET	59
5.21	SLIPPAGE BETWEEN ROUTER AND BRACKET MOUNTING WHEN FORCE IN Z-DIRECTION IS APPLIED	60
5.22	BOLT PRETENSION AND FORCE APPLIED IN X-DIRECTION	61
5.23	DEFORMATION IN MOUNTING BRACKET WHEN FORCE IS APPLIED IN X-DIRECTION	61
5.24	SLIPPAGE BETWEEN ROUTER AND BRACKET MOUNTING WHEN FORCE IN X-DIRECTION IS APPLIED	62
5.25	BOLT PRETENSION AND FORCE APPLIED IN Y-DIRECTION	63
5.26	DEFORMATION IN MOUNTING BRACKET WHEN FORCE IS APPLIED IN Y-DIRECTION	63
5.27	SLIPPAGE BETWEEN ROUTER AND BRACKET MOUNTING WHEN FORCE IN Y-DIRECTION IS APPLIED	64
5.28	BOLT PRETENSION AND FORCE APPLIED IN X AND Y-DIRECTION	65
5.29	DEFORMATION IN MOUNTING BRACKET WHEN FORCE IS APPLIED IN X AND Y-DIRECTION	65
5.30	SLIPPAGE BETWEEN ROUTER AND BRACKET MOUNTING WHEN FORCE IS APPLIED IN BOTH X AND Y-DIRECTION	66
5.31	MACHINE COMPONENTS	67
5.32	X-AXIS ASSEMBLY ON MACHINE STRUCTURE	68
5.33	THE MACHINE STRUCTURE	68

LIST OF TABLES

TABLE NUMBER		PAGE NO.
1.1	ADVANTAGES OF RECIRCULATING BALL SCREW	3
3.1	TOOL SELECTED FOR CARVING OPERATION	24
3.2	OPERATING PARAMETERS FIXED FOR MACHINE TOOL DESIGN	26
3.3	SPINDLE TORQUE AND FORCES ON MOTOR CALCULATIONS	27
4.1	STANDARD COMPONENTS PURCHASED FOR PRESENT PROJECT	35
5.1	PROPERTIES OF MILD STEEL	43
5.2	PROPERTIES OF HARDENED STEEL	44
5.3	PROPERTIES OF ALUMINIUM ALLOY	44
5.4	PROPERTIES OF HIGH SPEED STEEL	44

CHAPTER: 1

INTRODUCTION

Automation is associated with advancements in technology. In Mass production units, since the volume of production is very high with little or limited variety, special purpose machines, automated transfer lines have been developed and used. However for job or batch production general purpose machines are being used with the large variety of components and discontinuous demand of products. In the conventional general purpose machines, skilled operator is responsible for many inputs to the machine. The operator, in addition to the physical work of handling the machine, has to do a lot of information processing like reading the drawing and checking dimensions etc. Although the general purpose machines are very flexible but this flexibility is the cost of time and productivity. With the advent in NC technology the productivity on shop floor has been enhanced, but still the NC technology is a costly affair for small scale enterprises, especially the artisans and handicraft workers.

1.1 HISTORY OF CARVING MACHINERY

The carving has been great intent from the remotest ages as a foremost art for the decoration of wood. The tendency of human nature has always been to ornament every article in use. The North American Indian carves his wooden fish-hook or his pipe stem just as the Polynesian works patterns on his paddle. The native of Guyana decorates with a well-conceived scheme of incised scrolls, while the native of Loango Bay carve his spoon with a design of perhaps figures standing up in full relief carrying a hammock.

Figure-work seems to have been universal. To carve a figure in wood may be not only more difficult but also less satisfactory than sculpting with marble, owing to the tendency of wood to crack, to be damaged by insects, or to suffer from changes in the atmosphere. The texture of the material, too often proves challenging to the expression of features, especially in the classic type of youthful face. On the other hand, magnificent examples exist of the more rugged features of age. The beetling brows, the furrows and lines neutralizing the defects of the grain of the wood. In ancient work the surface may not have been of such consequence, for figures as a rule being painted for protection and especially colour.

The work is necessarily slow and requires substantial skill, making the works expensive. Other cheaper methods of decoration have driven carving from its former place. Machine work has much to answer for, and the endeavour to popularize the craft by means of the village class has not always achieved its own end". This statement has proven untrue, as the continued survival of the art and craft of wood carving can be demonstrated by the large number of wood carvers who have carried on or advanced the tradition in different parts of the world.

Wood in its various forms enters so largely into the industry of the nation that its economical and rapid conversion from forest trees into articles of general utility cannot be of paramount interest. Occasional efforts were made to perform the work mechanically, but not until the end of the eighteenth century were any definite and practical ideas described. Sir Samuel Bentham, an Englishman, patented in 1791 and 1793 principles which are in use today. His inventions included “planing machines with rotary cutters to cut on several sides of the wood at once veneer cutting machines, moulding and recessing machines, bevel sawing machines, saw sharpening machine, tenon cutting machine by means of circular saws, boring tools”.

1.2 AUTOMATION OF CUTTING OPERATIONS

The need for new technology to control machine movement was felt somewhat 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 successive settings of the tools were determined using the IBM purchased card reader, the US air force was the funding agency for the NC development. The Servomechanisms laboratory of MIT developed the first NC machine in 1952 [23].

The development of NC technology can be categorised into various generations as listed below:

1. **First Generation:** The control system of the first generation numerically controlled machines was built with vacuum tubes and associated devices. This was bulky, consumed a lot of power, while reliability and the storage capacity was poor.
2. **Second Generation:** 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. However, the number of printed circuit boards (PCB) was large. Since there were thousands of components and connections involved, the reliability was again was not so satisfactory.
3. **Third generation:** During mid-60s, the concept of Integrated Circuit (IC) revolutionised the electronics world further. Thyristor Controlled DC drive became popular during this

period. With the integration of NC machine tools helped in flexibility. From the totally hardwired design, the design of NC machine become soft wired. The other development was the concept of evolution of direct numerical control technique by which several NC machines could be controlled by the a single host computer.

4. **Fourth generation:** Towards the end of 1970, microprocessors came to use as a CPU of the computers. This change also impacted 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.

1.3 AUTOMATION OF MACHINING OPERATION USING COMPUTER NUMERIC CONTROL (CNC)

CNC is a microprocessor based control system that accepts a set of programmed instructions, process and sends output control information to a machine tool, accepts feedback information acquired from a transducer placed on the machine tool on both the instruction and the feedback, assures the proper motion, speed and operation occurrence.

Figure 1.1: CNC MACHINE [31]

There are two distinct types of control system need in CNC. The most significant among them is axis feed drive which controls the positions of the slides which in turn assures the accuracy and repeatability of the job produced. The machine tool control is supervised by a CNC controller which carries out the function like decoding, interpolation, diagnostics, area clearance program. The slides are moved by the feed drive servomotors through ball screw and the nut drive. The slide position is measured by either a resolver or an encoder mounted on a feed motor shaft or a linear feedback transducer.

Milling machines are probably the most versatile machine tools. Several well-known types are briefly described here, they are grouped in to horizontal and vertical milling machines based on the milling spindle. These machines are always equipped with continuous path control and have at least three and often five or more axis under control.

Milling machines rarely have automatic tool changers available and they can therefore be eliminated as option unless one wishes to indicate to the operator via digital readouts, the identification of the next tool with respect to length compensation and on milling path compensation in desirable.

For large milling machines different haft encoders or similar manual override possibilities are advantageous since they allow simplified alignment and trouble free restarting after breakage.

1.4 MAIN COMPONENTS IN CNC MACHINING CENTRE

The figure 1.2 shows the schematic diagram depicting the overall organization of CNC system

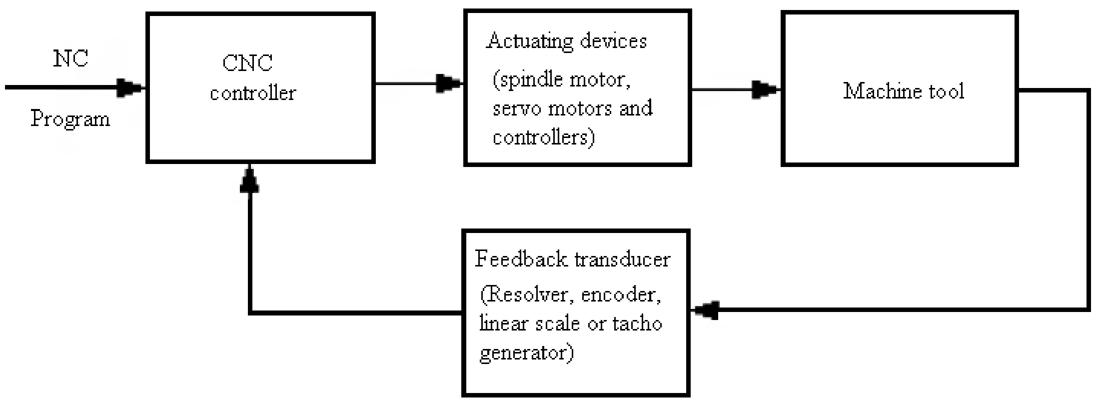

Figure 1.2: Schematic diagram of CNC [24]

The main components for a CNC system are listed below:

1. NC Part Program

2. Machine structure, which is composed of:
    a) Machine bed and columns
    b) Drive unit
    c) Spindle/Router
    d) Guideways and Slideways, which are further build on:
        i. Linear Motion ways
        ii. Ball Screw
3. Machine control unit (MCU)
4. Tooling

In the next sections of this chapter the above listed main CNC components are have been discussed in detail:

1.5 NC PART PROGRAM

In NC Machining centre controlling instructions are fed through a program to the machine. Such program is called Part Program. A part program is detailed set of sequential instructions defined to prescribe the relative movement between tool and work piece and hence complete the machining of the parts. Those set of instructions also control several other functions like spindle speed and feed selection, selection and controlling the tool and coolant etc. A part program is generated using geometric parameters of the part model. [24]

1.6 MACHINE STRUCTURE

Machines structure 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 [24]:

1. **Static load:** 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.

2. **Dynamic loads:** 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
   The effects of these vibrations can be reduced by following measures
   (a) Reducing the mass of the structure
   (b) Increase the stiffness of the structure
   (c) Improving the damping properties of structure

1.6.1 BEDS AND COLUMNS

The way machining forces are directed into the bed 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 [24]. This results in rigidity and accuracy without excessive dead weight. A major advantage is the reduced need for setting up work pieces in special jigs and fixtures thus reducing overall production time.

1.6.2 DRIVE UNIT

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

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:

1. Stepping frequencies and consequently possible feed rates are too low.

2. Maximum available torques are relatively low, that is acceleration characteristics are poor
3. Even short term overload can cause “dropping out of step”.
4. 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:

1. Motor with tachometer and brake
2. Control unit with power amplifier-for spindle drives with integrated field controller
3. Main transformer and smoothing choke
4. Mechanical clutch, with overload protection
5. Motor protection unit guarding against current overload or excess temperature
6. 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.

Developments over the past few years in drive technologies have made it possible to meet all the requirements of drive counts. Another important characteristic is uniformity and smoothness of motion jumps of oscillation being totally unacceptable. Even the slightest buzzing noise or any resonance of the motor will impair the surface finished of the machined work piece to a point where it may be rejected. To meet these requirements high speed / high rpm and low torque routers are preferred instead of high torque motors of spindle drive. General speed of spindle drive is from 25000 r.p.m to 33000 r.p.m.

1.6.3 SPINDLE/ ROUTER

For cutting, caving, milling and drilling on a work-piece in a machine tool various other operations spindles are used. For wooden carving machines spindle should have high rpm and low torque and special tools so that the wooden surface doesn't burn and give a smooth finish. Ordinary machine tool cannot achieve such high rpm which are in range of 25000 to 33000 rpm. So, special routers are used instead of ordinary spindles for wood machining:
The general requirement of spindles for CNC machine are:

1. High stiffness both static and dynamic
2. Running accuracy
3. Axial load carrying capacity
4. Thermal stability
5. Axial freedom for thermal expansion
6. High speeds of operations

a) Special requirement for Wood Carving Machines

i. Low reaction from components.
ii. Friction should not cause over heating of metallic cutting tool. That is why wood cutting is done at high speed and low feed.

b) Router for wood carving application

Figure 1.3: Fixed base router [32]

Figure 1.4: Plunge router [33]

The router is one of the most commonly used power tools used in wood working. As in technical sense for wood carving spindle is required which has high rpm and low 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 parts.

1.6.4 GUIDEWAYS AND SLIDEWAYS

Guide ways are used in machine tools to:

1. Control the direction or line of action of the carriage or the table on which a tool or a work piece is held.
2. 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 guideway. These errors further result in variety of tracking errors like pitch, yaw and the roll that are difficult to measure and correct. Further, over a period of usage, any kind of wear in the guideway reduces the accuracy of the guide motion, resulting in errors in movement and positioning.

The following points must be considered while designing guideways:

1. Rigidity

2. Damping capabilities
3. Geometric and kinematic accuracy
4. Velocity of slide
5. Friction characteristics
6. Wear resistance
7. Provision for adjustment of play
8. Position in relation to work area
9. Protection against fine chips and damage

These criteria vary depending upon a particular application and hence the selection of guideway 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 guideway.

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.

The guideways of CNC machines are composed of following parts:

(i) **Linear Motion ways:** Linear motion guideways are used in CNC machines which in turn use rolling elements in between moving and the stationary elements of the machines. They are used to overcome the relatively high coefficient of friction in metal to metal contacts and resulting in limiting the amount of wear, heat generation and improve the smoothness. Linear motion guideways have become very common in machine tools on account of their higher rapid transverse rates [24].

(ii) **Ball Screw:** Ball screws are primarily employed in feed mechanisms of CNC machine tools. When compared with conventional trapezoidal and ACME threads, ball screws provide many advantages, which are listed below:

- In a ball screw, the load between the threads of the screws is not transmitted by direct contact, but though immediate rolling members. The balls rotate between the helical grooves of the screws and nut in a manner similar to their function in ball bearings. An essential feature of almost all balls.

- Low coefficient of friction is of the order of 0.004 as compared to 0.1 to 0.5 which is typical of sliding friction power screws. Wear is therefore less and there is very little need of frequent adjustment.
- High transmission efficiency is there which is particularly marked at low values of helix angle of screw (2-5) degree that are typical of power screws This high efficiency allows larger thrust load to be carried with less torque.
- Frictional forces are virtually independent of the travel velocity and the friction at rest is very small; consequently the stick up phenomenon is absent ensuring uniformity of motion.

Advantages of Recirculating Ball Screw							
	Ball Screw	ACME Screw	Fluid Power	Belt, Chain	Rack and Pinion	Cam Follower	Pneumatic Cylinders
Inexpensive	●	●		●	●	●	
Low Power Use	●			●	●	●	
Low Maintenance	●						
High Accuracy	●				●		
High Repeatability	●						
High Efficiency	●						
High Load Capacity	●		●				
Compact Size	●	●			●		●

Figure 1.5: Advantages of recirculating Ball screw [34]

General arrangement of a ball screw

Figure 1.6: Arrangement of ball screw [35]

The basic of ball screw is to interpose a series of the balls between the screw and the nut. These balls roll in the grooves as the nut or screw moves and the rolling friction thus replaces the sliding friction of the conventional acme or trapezoidal screws.

The balls rolling in the grooves exit from the trailing end of the nut and are picked up by the return tube inserted from outside and are re circulated into the leading end of the nut. There are also systems in which the rolling balls circulate within the nuts. The ball screws can have circular or gothic arch grooves. Gothic arch grooves have a small axial clearance when they are used in single nut, while circular arch grooves allow little axial deformation and a greater load capacity.

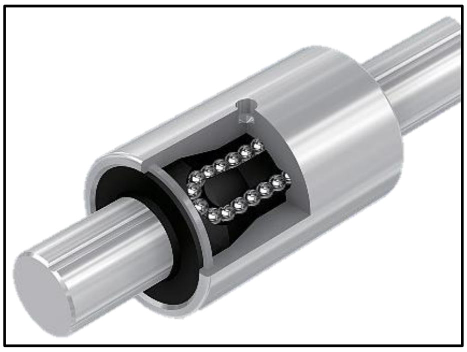

Figure 1.7: Circular arc grooves [36]

Figure 1.8: gothic arc grooves [37]

The greater the number of balls in a circulating system, the greater is the frictional resistance. Therefore use of an excessive number of balls is not recommended from designing point of view.

1.7 MACHINE CONTROL UNIT

The main function of the CNC system is to convert the part program instructions into motion of operative units of the machine tool through CPU and servo-control system. The system interprets this data and converts it into output signals. It is based on the use of numerical data for directly controlling the operative units of machine tool during a machining operation.

CPU decodes the part program instructions into position and velocity signals and feeds these signals to the servo-control unit which further generate suitable signals as command signals. The command values are converted into actual movement of operative units; by the servo-drive unit. The amount and rate of movement are controlled by CNC system depending upon the type of feedback system whether it is a closed loop or open loop. Position feedback signals are generated by the feedback devices like linear scales, rotary encoders and velocity signals are obtained by tacho-generators [24].

1.8 TOOLING AND ACCESSORIES

A touch trigger is the most valuable accessory which can be mounted in standard tool holders and loaded into the spindle via the automatic tool changer (ATC) to send information to the numerical controllers about the real condition of the work pieces. They require special software in the numerical controller to interpret these findings. Different tools are mounted on ATC according to their need in machining.

1.9 PRESENT WORK

Machining processes in the manufacture of wood products may be classified as sawing, peeling and slicing, planning, shaping, and routing, turning and boring, sanding and non-traditional machining processes such as cutting with laser beam and high-energy liquid jet. Wood carving from earlier times is done by means of hand cutting tools (knife) or a chisel and mallet leading to ornamentation of wooden object. The most common Indian woods used for carving are Sandalwood, Shisham (Talli), Babul, Sal, Teak. Decoration that is to be painted is usually carved in pine. The choice of wood depends upon requirement of carving being done. Traditionally, the carving process begins with general shaping process using gouges of various sizes. After that finishing is done which include polishing or applying lacquer for longer lifetime of work material.

The applications of the computer aided design (CAD) and computer aided manufacturing (CAM) technologies has proved to be of prime importance for carving operations whether in metal or wood. The CAD software can be used for 3-dimensional modelling the intricate part designs for carving applications in a very effective way. The geometric and topological information from these 3D part models can then be picked up by the integrated computer aided manufacturing modules/software, which can translate the 3D part geometry into the NC part program. But the use of such technologies for productivity enhancement is still a dream for a great portion of the population across the world, especially for those who are small scale entrepreneurs and the artisans/ craftsman, as the cost of owning such technologies is still very high. This is one of the main reasons, why most of the artisans/ craftsman do not use the automation for their customized manufacturing facility.

Another aspect is the reproducibility of the art work, as manufacturing without automated facilities/ systems these people cannot produce the identical finished artefacts or artistic goods in bulk. The higher investment in such systems was justified only for the mass or batch production systems where at least a few thousand components of each variety are produced. Also application of NC technology is better suited for manufacturing of components/ parts where the time required for actual machining constitute a significant part of the total production time per unit [25].

Even though there is drastic reduction in the cost of owning the NC technology, still it seems to be expensive for the small scale artisans/ craftsmen. This is because of the reason that the number of parts they produce are of very intricate shape which can only be produced in specially designed NC systems. Secondly, the number of units of each variety they produce are also very less.

The challenges such as shorter product life cycles, frequent design revisions, stringent demands on product quality and shorter times to market has imposed few more constraints before designing new automation strategies for small scale manufacturing systems. Moreover the manufacturing activities have become customer centric rather than manufacturer driven. Globalization has added further challenges for the artisans and craftsman across the world as they are now being forced to produce the better quality product in shorter time span and sell their products at reasonable prices. Thus there is an urgent need for cost effective automated system for the handicraft workers/ artisans, so that these people would be able to reproduce the identical finished products in smaller time duration and may be able to get righteously compensated for their skills.

In light of above facts, the present work has been planned to fabricate and carry out the CAE analysis of a low cost modular 3-axis NC vertical milling machine. The conceptual design of the prototype has been developed in Pro-Engineer. In the present dissertation work the detailed finite element method analysis in ANSYS 10.0 software has been done for the complete machine assembly, and router subassembly, as presented in the later chapters. The results showed that the machine structure behaves as desired under the estimated cutting loads, and the stresses and deflection are in the permissible range. After design validation, the fabrication for all specially designed low cost components was carried out. During design phase, basic fundamentals of “*Design for Assembly*” and “*Design for Manufacturing*” have been taken into consideration. It has been reflected in level of convenience while assembling the manufactured and purchased parts.

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 CNC carving machine for sculptured surfaces. Detailed study from basics has been carried out and an idea for building a modular structure was developed

Lin et al [1] has studied 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. The spindle head is always fed under linear guide mechanism through a ball screw driver. In this work finite element simulations and vibration tests were conducted and it was revealed that the preloading of linear guide greatly affects the vibration behaviour of spindle head. It was concluded that dynamic stiffness of spindle head could be enhanced by increasing the preload of linear guide.

Lee et al [2] observed that of the primary reasons for low productivity is large mass of the moving parts of machine tools, which cannot afford high acceleration and deceleration, encountered during operation. Moreover, the vibrations of the machine tool structure are among the other causes that restrict high speed operations. The slides of high speed CNC milling machines were designed with fiber reinforced composite materials to overcome this limitation. The vertical and horizontal slides of a large CNC machine were manufactured by joining high-modulus carbon-fiber epoxy composite sandwiches to welded steel structures using adhesives and bolts. These composite structures reduced the weight of the vertical and horizontal slides and increased damping by 1.5 to 5.7 times without sacrificing the stiffness.

Xiangsheng et al [3] analysed the influence of configuration parameters on dynamic characteristics on machine tools in working space, the configuration parameters have been suggested based on the orthogonal experiment method. Dynamic analysis of newly designed milling machine for producing turbine blades has been conducted by utilizing the model synthesis method. The finite element model is verified and updated by experimental modal analysis (EMA) of machine tool. The result obtained by modal synthesis method is compared with model finite element method (FEM) result as well. According to the orthogonal experiment method, four configuration parameters are considered as four factors for dynamic characteristics. The influence of configuration parameters on the first three natural frequencies is obtained by range analysis. It is pointed out that configuration parameter is the most important factor affecting the fundamental frequency of machine tools and it has less effect on lower-order modes of the system. The combination of configuration parameters which makes the fundamental frequency reach the maximum value is provided. It can be concluded that the influence of configuration parameters on the natural frequencies of machine tools can be analysed explicitly by the orthogonal experiment method, which offers a new method for estimating the dynamic characteristics of machine tools.

Hung et al [4] emphasised that prediction of machining stability is of great importance for the design of a machine tool capable of high-precision and high-speed machining. The machining performance is determined by the frequency characteristics of the machine tool structure and the dynamics of the cutting process expressed in terms of a stability lobe diagram. The aim of this study is to develop a finite element model to evaluate the dynamic characteristics and machining stability of a vertical milling system. Rolling interfaces with a contact stiffness defined by Hertz theory were used to couple the linear components and the machine structures in the finite element model. Using the model, the vibration mode that had a dominant influence on the dynamic stiffness and the machining stability was determined. The results of the finite element simulations reveal that linear guides with different preloads greatly affect the dynamic behaviour and milling stability of the vertical column spindle head system. These results were validated by performing vibration and machining tests. It can be concluded that the proposed model can accurately evaluate the dynamic performance of machine tool systems designed with various configurations and with different linear rolling components.

Abele et al [5] presented the state of the art in machine tool main spindle units with focus on motorized spindle units for high speed and high performance cutting. Detailed information is provided about main components of spindle units regarding the historical development, recent challenges and future trends. Advanced methods of modelling the thermal and dynamical behaviour of spindle units are shown in overview with specific results. Furthermore concepts of sensor and actuator integration were presented which all focused on increasing productivity and reliability.

Fenghe et al [6] observed that ram is a very important component of super heavy duty computer numerical control (CNC) floor type boring-milling machine and deformation of ram is a significant source causing errors in machining process. To compensate the deformation error of super-heavy-duty CNC floor type boring-milling machine, based on force analysis theory, the law and compensation measures of deformation of ram are investigated. Based on the principle of torque (force) balance of the ram components, the formulas of compensation forces and compensation torques were obtained. According to theoretical analysis results and the structural characteristics, rods compensation, hydrostatic pressure compensation and wire rope compensation measures have been conducted to compensate the deformation error of ram. The experiments and computer simulation results show that the straightness of the ram at its overhanging end meets the national machinery industry standards.

Namazi et al [7] considered majority of the chatter vibrations in high-speed milling which originates due to flexible connections at the tool holder–spindle, and tool–tool holder interfaces. This article presents modeling of contact stiffness and damping at the tool holder and spindle interface. The holder–spindle taper contact is modeled by uniformly distributed translational and rotational springs. The springs are identified by minimizing the error between the experimentally measured and estimated frequency response of the spindle assembly. The paper also presents identification of the spindle's dynamic response with a holder interface, and its receptance coupling with the holder–tool stick out which is modeled by Timoshenko beam elements. The proposed methods allow prediction of frequency-response functions at the tool tip by receptance coupling of tools and holders to the spindle, as well as analyzing the influence of relative wear at the contact by removing discrete contact springs between the holder and spindle. The techniques are experimentally illustrated and their practical use in high speed milling applications were elaborated.

Aknouche et al [8] described some interesting results about the tool wear. In a machining process of the North African Aleppo pine the correlation between the tool wear and cutting forces shows that, the running period is about 850m of cutting length. In this period, angle variation is unstable before it is around -1.1° in the stability zone. This confirms the existence of the first two separate areas that characterize the behaviour of the tool edge recession, which are running (abrupt wear) and the linear wear (known as stability period). As a consequence it can be said that an angle is a good criterion for tool wear estimation in these practical machining conditions. It is proposed to use the same approach with other tools. Another important aspect, particularly in the case of wood is to estimate the surface quality by measuring its roughness and wettability and to make the link with tool wear that is the transformation of wood by removal of matter generates new surfaces which have a precise functionality and will all receive protective films or glues. This highlights the need to characterise wood surface wettability. Depending on the various fields of application, a future aim of this study is to make practical recommendations to wood professionals, summarized in a good practice guide to enhance Aleppo pine solid woods value.

Cao et al [9] observed that chatter stability of machine tool is dependent on the dynamic behaviour of spindle systems. The alternative method was presented to predict the chatter stability lobes of high-speed milling with consideration of speed-varying spindle dynamics. Based on the dynamic model of the high-speed spindle, systematically investigations were carried out on the speed effects. It was found that the gyroscopic moment of the spindle shaft can increase the cross FRFs, but can hardly affect the direct FRFs (frequency response function) at the tool tip due to the damping of the spindle system. The centrifugal forces on both the shaft and bearings lower the overall spindle system stiffness evidently as the speed increases. It is shown that the stability lobes with consideration of speed effects shift to the low speed range significantly. The proposed method can provide a guideline to estimate the cutting performance of high-speed spindles during the design stage.

Jo¨nsson et al [10] highlighted that while designing CNC machine tools it is important to consider the dynamics of the control, the electrical components and the mechanical structure of the machine simultaneously. This paper describes the structure and implementation of a concept for real-time simulation of such machine tools using a water jet cutting machine as an application. The concept includes a real control system, simulation models of the dynamics of the machine and a virtual reality model for visualisation. However, already from the initial simulation results presented in the paper it could be concluded that the influence of structural flexibility on manufacturing accuracy is of importance at desired feeding rates and accelerations. The fully automated implementation developed in this work is a promising base for dealing with this trade-off between productivity and accuracy of the manufacturing process through multidisciplinary optimisation.

Zulaika et al [11] has presented an integrated approach for designing large milling machines, taking both mass reduction of mobile structural components and the maximum material removal rate into account. This approach considers milling operation and a productivity target as a starting point, and then deals with the design of the machine to achieve the targeted productivity with structural components of minimum mass. The procedure is based on modelling the interactions between process and machine by means of a stability model of the milling process in modal coordinates. The model allows the identification of the mechanical design parameters that limit the productivity as well as the threshold values that must be met to ensure the targeted productivity. Those values are reached in an iterative procedure that minimizes the mass of the critical structural components of the machine.

Son et al [12] described the development of a three-degrees-of-freedom (DOF) desktop reconfigurable machine tool. In this paper, the conceptual design of a desktop reconfigurable machine, which is capable of controlling the three DOF orientation of a spindle, was presented. Then, static and dynamic structural analyses were performed to characterize the effect of vibration on the manufacturing performance. The results demonstrated the feasibility of simultaneously controlling the position and orientation of the machine tool during the machining operation. Dynamic simulations and experimental results using a closed-loop control with position feedback were presented to illustrate the performance and features of the system. Unlike conventional full-scale manufacturing machines, the developed machine provided a number of advantages, including fast dynamic response, have simpler design, low cost, and a compact but relatively large workspace without motion singularities.

Dong et al [13] discussed the development and performance evaluation of a high-speed, 3-axis milling machine using a novel parallel kinematics x-y table. The x-y table is based on an inversion of the Oldham coupling. The advantages of this kinematic configuration include low inertia, uniform kinematic conditioning, and dynamically matched axis. The design of the x-y table makes this system particularly well suited for high-speed contouring in the x-y plane. The kinematics, dynamics, and mechanical design of the system have been evaluated. On the basis of linear and circular contouring experiments conducted to evaluate the system capabilities. The stiffness of both the mechanism as well as the direct drive actuation system were measured and reported. The experimental results demonstrate that the proposed mechanism offers an attractive combination of performance characteristics for high-speed contouring and high-speed machining.

Padayachee et al [14] developed reconfigurable Manufacturing System (RMS) paradigm address challenges in the design of manufacturing systems and equipment that will meet the demands of modern manufacturing. This research involved the development of Modular Reconfigurable Machines (MRMs) as an emerging technology in reconfigurable manufacturing. MRMs are mechanically modular machines. It focuses on aspects of the mechanical design and the development of a control system that supported the modularity and reconfigurability of the mechanical platform. This is complemented by a software architecture that has been developed with a focus on hardware abstraction for the management of an augmented mechanical and electronic architecture. The implications of MRMs for RMSs were discussed and key inhibitors to industrial implementation were identified.

Youssef et al [15] stated that modularity and reconfigurability of the building blocks of modern manufacturing systems have to be considered when evaluating their performance. The paper proposed a model for evaluating system availability and expected production rates for manufacturing systems that were composed of unreliable modular machines with multiple functionally parallel production units. The results show that machines with a larger number of modules, usually thought of as having lower availability, provided higher overall system availability in the case of machines with multiple spindles. Based on the new analysis and results, it is recommended that system designers favourably consider machines with multiple spindles rather than increasing the number of machines in parallel. These results provide an important support for the use of modular/ reconfigurable equipment compared with traditional equipment, in spite of the higher cost.

Gallardo et al [16] addressed the kinematics, including position, velocity and acceleration analyses, of a modular spatial hyper-redundant manipulator built with a variable number of serially connected identical mechanical modules with autonomous motions. First, the kinematics of the base module, a three-legged in-parallel manipulator, was formulated using the theory of screws. After that, the results thus obtained, were applied recursively for accomplishing the kinematic analyses of the hyper-redundant manipulator at hand.

Eyma et al [17] explained that the correlation between density and cutting forces in wood machining was not perfect and that some exceptions could not be explained. These difficulties have easily been explained by the anisotropy of wood material. It is appeared that mechanical properties could explain some exceptions in the relationship between density and cutting forces. The mechanical properties lead to further information on the wood species behaviour during machining and improve the accuracy of the relationship between cutting forces and wood species. Moreover, it appears clearly that parameters links to wood elastic behaviour were best factors to explain wood machining and strains involved (and not failure strain or fracture energy). In fact, wood machining seemed to be more a phenomena links to elastic stresses than to failure strains. These tests must be allowed to take into account the quickness and the impact during machining. So it is possible that this kind of tests leads to new information on wood behaviour during machining. Moreover, it appeared clearly that the measure of mechanical properties was done at low deformation quickness, in opposition to machining (high quickness). This difference could perhaps explain why correlations were not perfect and so give limits of the use of simple mechanical characteristics.

Bi et al [18] research presented an automated method to build kinematic and dynamic models for assembling modular components of modular robotic systems. By comparison with other approaches, the proposed method is applicable to any robotic configuration with serial, parallel, or hybrid structures. In addition, it is object oriented so that each modular component is an element with a sub-model and the overall model can be assembled from sub-models subject to the connection constraints.

Brisan et al [19] reconfigurable robotic system permits multiple configurations having different characteristics, using mostly modularized building blocks. The goal of paper is to investigate how the shape, dimensions and the distribution of singularities in the workspace for different configurations, with different degrees of freedom, of a reconfigurable robotic system, are changed. The computation of the workspace is based on the modularity property of the system. The presented results are also experimentally validated.

Neugebauer et al [20] reviewed current developments in mechatronic systems for metal cutting and forming machine tools. The integration of mechatronic modules to the machine tool and their interaction with manufacturing processes were presented. Sample mechatronic components for precision positioning and compensation of static, dynamic and thermal errors were presented as examples. The effect of modular integration of mechatronic system on the reconfigurability and reliability of the machine tools was discussed along with intervention strategies during machine tool operation. The performance and functionality aspects were discussed through active and passive intervention methods. A special emphasis was placed on active and passive damping of vibrations through piezo, magnetic and electro-hydraulic actuators. The modular integration of mechatronic components to the machine tool structure, electronic unit and CNC software system was presented. The paper concluded with the current research challenges required to expand the application of mechatronics in machine tools and manufacturing systems.

Wakasawa et al [21] after 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. The result is thought to be due to differences arising after the balls were packed. 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 daming the vibrations like giving vibration pads at the bottom of the modular structure this approach was eliminated.

Hung et al [22] predicted after observing machining stability is of great importance for the design of a machine tool capable of high-precision and high-speed machining. The machining performance is determined by the frequency characteristics of the machine tool structure and the dynamics of the cutting process, and could be expressed in terms of a stability lobe diagram. The aim of the study was to develop a finite element model to evaluate the dynamic characteristics and machining stability of a vertical milling system. Using the model, the vibration mode that had a dominant influence on the dynamic stiffness and the machining stability was determined. The results of the finite element simulations reveal that linear guides with different preloads greatly affect the dynamic behaviour and milling stability of the vertical column spindle head system. These results were validated by performing vibration and machining tests. It is concluded that the proposed model can be used to accurately evaluate the dynamic performance of machine tool systems designed with various configurations and with different linear rolling components.

CHAPTER: 3

SELECTION OF DRIVE MOTORS

The proposed work is to design a CNC machine tool structure for ornamental wood carving. The basic design criteria for design of CNC machine tool structure of wood carving is the estimation of cutting torque and moments that are encountered while cutting wood. Cutting forces/ torques required to mill or drill wooden parts can be obtained analytically or from an experimental study, which uses a force dynamometer coupled on machine base. Also it is required to estimate of how much torque/ power is required to move the power screw which in turn moves the columns. The forces acting on the columns are the weight of column itself and weight of components on them, which are bolted and the resistance offered by spindle when doing operation.

In this chapter the study for finding out the specified cutting forces and moment for sample of wood has been carried out for finding out the estimation of required spindle/ router power and power of drive motors. The carving operation is generally carried out by using different cutting tools of varying diameters, but in the present study, for designing the machine tool type with following specifications has been considered

Table 3.1: Tool selected for carving operations

Tool Type	Ball end mill, flat end mill, radiused end mill
Diameters	12.7mm, 6.35mm or lesser
Flute type	Straight or hellical
No. of teeth	2

Figure 3.1: Ball end mill [38]

Figure 3.2: Flat end mill [39]

Figure 3.3: Radiused end mill [40]

Figure 3.4: Straight flute tool [41]

Figure 3.5: Helical flute tool [42]

3.1 SELECTION OF SPINDLE MOTOR AND DRIVE MOTORS

For various operations on a work-piece in a machine spindles are used. For wooden carving machines spindle should have high rpm and low torque and special tools so that the wooden surface does not burn and give a smooth surface finish. This is the reason special routers are used for wood carving .Ordinary machine tool cannot achieve such high value of rpm which are in range of 25000 or above which can be obtained using special wood cutting router. The operating parameters selected for machine tool design has been listed in table 3.2

Figure 3.6: Wood cutting router [43]

Table 3.2: Operating parameters fixed for machine tool design

Tool types	Ball end mill, flat end mill, radiused end mill
Maximum Diameter	12.7mm
Minimum Diameter	6.35mm or lesser
Operational speed	33000 r.p.m for 6.35mm or lesser tool dia 25000 r.p.m for 12.7mm tool dia
Maximum feed rate	300 mm/min
Maximum depth of cut	For 6.35mm tool dia is 5mm maximum For 12.7mm tool dia is 8mm maximum

In table 3.3 the calculation for spindle torque requirements and tangential cutting forces has been done. Parameters taken below are for worst case condition possible while operating milling machine.

Table 3.3: Spindle torque and forces on motor calculations [26]

Parameter	**Case I**	**Case II**
Diameter of tool (D)	6.35mm	12.7mm
Revolutions min^{-1} (n)	33000 rpm	25000 rpm
Cutting speed (v) $\frac{\pi Dn}{1000X60}$	10.972 m/sec	16.624 m/sec
Depth of cut (t)	5 mm	8 mm
Max. Feed per minute (S_m)	300mm/min	300mm/min
Metal removal rate (Q) $\frac{btSm}{1000}$	9.525 cm^3/min	30.48 cm^3/min
Specific cutting force/ unit power [28] (U)	13 x 10^{-3} $kW/cm^3/min$	13 x 10^{-3} $kW/cm^3/min$
Power at the spindle (N) = U**.**Q	123.825 W	396.24 W
Power include an efficiency of 50%, Power (N_{new})	247.65 W	792.48 W
Tangential cutting force (P_z) = $\frac{N}{V}$	22.57 N	47.67 N
Torque at spindle (T_s) = 97.5 $\frac{N}{n}$	0.73 Nm	3.09 Nm

Tool of larger diameter used for roughing and lesser diameter used for finishing. From the table 3.3 it is clear that for the tool diameter 6.35mm spindle power requirements is 247.65 W, the standard router available is of 550W as presently the router mounting and brackets are designed for the router of 6.35 mm tool diameter. Therefore, the calculation done above justifies that router selected is sufficient to do carving operation. In present modular machine structure designing as discussed in table 3.3 force on the tip of the tool has been calculated as 22.57 N considering the safety factor as 2.0. Tangential force on the tool tip is taken as 45.14 N ≈ 46 N. To design the router mounting bracket this force is taken into consideration. Details of analysis for designing bracket mounting have been done in chapter 5 of this report.

But as the overall design criteria is determined by the largest tool size with its cutting parameters, thus keeping the provision for a bigger router of 1 KW power while designing the present machine. With some changes in router mounting for bigger router of 1 KW power, it can be mounted on the same machine. As drive motors are same for bigger router. So, the tangential force from the case II is considered for the calculation of torque in drive motors. Selection of motor powers has been discussed in the next section.

3.2 SELECTION OF DRIVE MOTOR POWER

As ball screw drives coupled with linear guide ways and block bearings has been used in machine structure. The drive motor with sufficient capacity required to drive the machine along X, Y and Z axis directions. The drive motor must be able to withstand the reaction force of 47.67 N ≈ 48 N which is generated while carving operation is being done, which provide hindrance for the motion. The maximum coefficient of friction between movable parts in machine structure like between ball screw and its flange is 0.02 and between block and linear guide is 0.03 [19]. Considering the maximum of two is used in table 3.3. Also, the force has been calculated for worst case scenario as discussed in case II of table 3.3. The worst case scenario includes the feed rate of 300 mm/min which is maximum for the drive motor, with largest tool diameter router that can be used here for carving which is 12.7 mm with the maximum depth of cut of 8 mm. It is important to consider the self-weight of the drive along with frictional reaction forces of block bearings on linear motion ways and ball screw. It is ball screw which will be cutting as final component/ joint transmitting the torque from the actuating drive motor to resist or overcome the axial driving forces. The maximum driving force/ torque is required for x-axis drive. Thus the calculation done will be carried out considering the self-load of machine structure on X-drive linear motion ways and ball screw. Under the above rated load conditions the maximum torque requirements from drive motors have been calculated as given below:

Maximum reaction force on the spindle, W_1 = 46 N (from table 3.3)

Coefficient of friction, μ =0.03

For X-axis:

Weight of Machine elements on x-drive ball screw, W_{sx} = 600 N (calculated from 3-D model in Pro-e)

Frictional force in direction parallel to axis of ball screw of x-axis drive, $W_{x1} = \mu.W_{sx}$ = 18N

Total force, $W_x = W_{x1} + W_1$

$= 64$ N

Similarly, for Y-axis:

Weight of Machine elements on y-drive ball screw, W_{sy} = 200 N (calculated from 3-D model in Pro-e)

Frictional force in direction parallel to axis of ball screw of y-axis drive, $W_{y1} = \mu.W_{sy} = 6$ N

Total force, $W_y = W_{y1} + W_1$

$= 6 + 46$

$= 52$ N

Similarly, for Z-axis:

Weight of Machine elements on y-drive ball screw, $W_{sz} = 50$ N (calculated from 3-D model in Pro-e)

Total force, $W_z = W_{sz} + W_1$

$= 50 - 46$ (-ve sign indicates the direction of cutting in negative direction

$= -2$ N

But when operation not being done the motor has to lift the weight of z-axis drive. so, the maximum weight the drive has to carry is 50 N.

From above calculation it is clear that maximum force generated from operation as well as weight of machine elements x-axis encounters the maximum force. Therefore, more power needed out of these three on x-axis motor. Considering, W= 66 N for further calculation:

Pitch of the ball screw, P = 5mm

Diameter of Ball screw, D = 16mm

Helix angle of the screw, $\alpha = \tan^{-1}\left(\frac{P}{\pi D}\right)$

$= \tan^{-1}\left(\frac{5}{16\pi}\right)$

$= \tan^{-1} 0.0995$

$= 5.68349°$

Friction angle, $ø = \tan^{-1}(0.03)$

$= 1.718°$

Torque required to drive ball screw, $T = \frac{D}{2} W \tan(\alpha + ø)$

$= \frac{16}{2} \times 66 \times \tan(5.68349 + 1.718)$

$= 68.5872$ N-mm

$= 6.85872$ N-cm

The actual drive motor selected for the prototype model of 3-axis vertical milling machine is 265 N-cm. The reason to select the drive motor is that if some times larger cutting force are encountered while machining of some other softer metals like aluminium or some hard spots in wood the drive should not stuck. Also there would be minimum heating for the larger motors as the carving operation for approximated cutting volume of machine which is 560 x

435 x 310 mm^3. It will take quite larger time for motors would be running continuously. Another reason is that the drive motors have been connected as direct drive using SOH couplings to enhance the precision of the machine. Although the drives will become more precise than geared or belt drives, but the suddenly increased load will be well taken up by these motors of high power rating.

Figure 3.7: Servo motor for the feed drives

Although requirements of motors in different directions are different as mentioned in above in the table. Considering the maximum torque out of these three which is in x-direction motor. For not complicating their movements and for the sake of simplicity for our controller selection of only one type of motor was done for all three axis. The motor having rated torque of 265-N cm was selected.

CHAPTER: 4

SELECTION OF STANDARD COMPONENTS

The aim of the proposed work is to design a CNC machine tool structure for ornamental wood carving. Under this objective the work is to evaluate the design and fabricate a suitable prototype for CNC machine tool structure using the suitably selected standard and precision components available in market and custom design of base structure of NC machine which will suit the wood carving requirements as well as meeting the criteria of maximization of effective cutting tool movement within boundary of machine tool for optimization of material cutting space. Generalised design procedure followed for the present work is as given below:

4.1 DESIGN PROCEDURE:

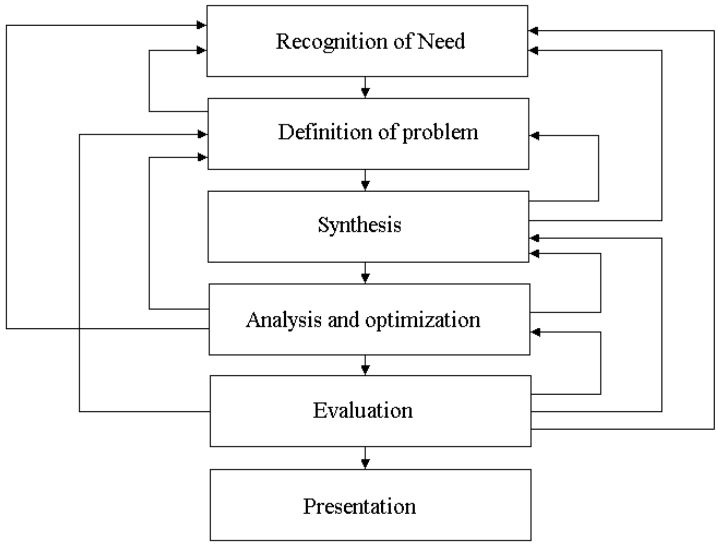

Figure 4.1: Design Procedure[29]

All these steps have been followed as per the flow chart given above. A brief description about the steps followed for the present work has been detailed below:

1. Recognition of need

Productivity enhancement for a great portion of the population across the world, especially for those who are small scale entrepreneurs and the artisans/ craftsman, as the cost of owning such technologies is still very high. As a result many artisans/ craftsman find it difficult to meet the requirement of production levels to break even. Therefore it was figured that a low budget CNC machine for wooden sculptured machining is needed.

2. Synthesis (Mechanism)

Desired motion of the mechanism or group of mechanism figured out in which arrangement of the parts which will give the motion required. A 3-axis machine with Cartesian co-ordinates using the work volume for maximization of workspace of the machine, to be able to carve out on a piece of wood.

3. Analysis and optimization

Both analysis and optimization require that we construct or devise abstract models of the system which will admit some form of mathematical analysis. It simulates the real physical system very well. Analysis of forces on the machine for deciding the power of motors and router selection, which have been done in the previous chapter. Analysis on the basis of considering the standard earth gravity and forces on the spindle which is detailed in the next chapter.

4. Evaluation

Selected the member best suited for each member of the machine. Size of each member of the machine by considering the force acting on the member and the permissible stresses for the material used. It should be kept in mind that each member should not deflect or deform than the permissible limit. The size of member modified to agree with past experience and judgement to facilitate manufacture. The modification may also be necessary by consideration of manufacturing to reduce overall cost. After doing explore in the market and suitable components selection was done and analysed for each and every component in the machine. After taking into budget constraint some of the components were selected and so do the material for each component.

5. Presentation

Draw the detailed drawing of each component and the assembly of the machine with complete specification for manufacturing processes suggested. The component as per drawing is manufactured in the workshop. Manufacturability and assembly according to fitment as taking under consideration the distortion after welding, misalignment of holes while tapping giving way for elliptical holes corresponding to those adjacent to tapped holes for better fitment. Grinding bolts to make them better for visual realism. Detailed drawing of every component before the production, to assist for manufacturing on the shop floor. Detailed drawing done on the basis of operation being done like distributing the work firstly to make it to proper size and then the second operation to do holes.

4.2 Design of machine tool structure prototype

Proposed work was to evaluate design of a machine tool structure using CAE tools with an overall idea that what a machine would look like. With help of software solid works structural design and axis drives of the machine were optimized for work volume of the machine.

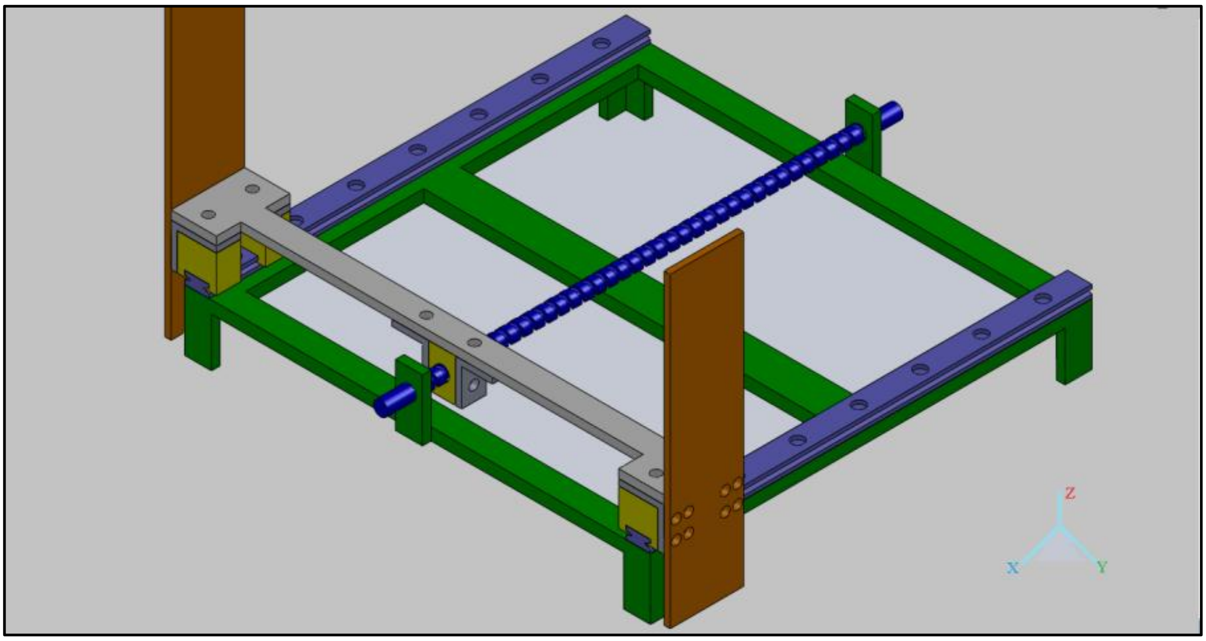

Figure 4.2: Machine tool base design in first Iteration

Iteration 1.0

First iteration was to develop a basic design of structure starting from the base with its components in x-axis direction. It was decided to fabricate machine designed from simple material available in the market, making it convenient at user end to manufacture its own spares using conventional machine tool and thus eliminating any idea to get spares from any store. Then mild steel plates were selected as low cost raw material. Guide rails were fitted, length of x-axis was constrained, vertical columns were placed during the modelling stage it was observed that it required use of too many fasteners to connect the different structural members involved in the structure. It may adversely affect the accuracy and may also result in a noisy working of the proposed machine. The modular structure was too much exposed which will be prone to outside dust which in turn can reduce the service life of the moving parts and the components associated with them in the machine. Also proper utilization of the bounding box which envelops the whole machine wasn't there. As shown in the figure there was no use of area below the stand which in figure 4.2 is shaded green. A structure has more chances of its vibrations when its legs are not locked with each other from the base.

Iteration 2.0

With new ideas a new model started to take a shape of modular structure which was better version than the last one. After a good brain storming sessions, a new plan for modular structure was made, details of which has been shown in figure 4.3. Improved version than the previous as it negate all the possible factors that lead to inefficiency of the machine.

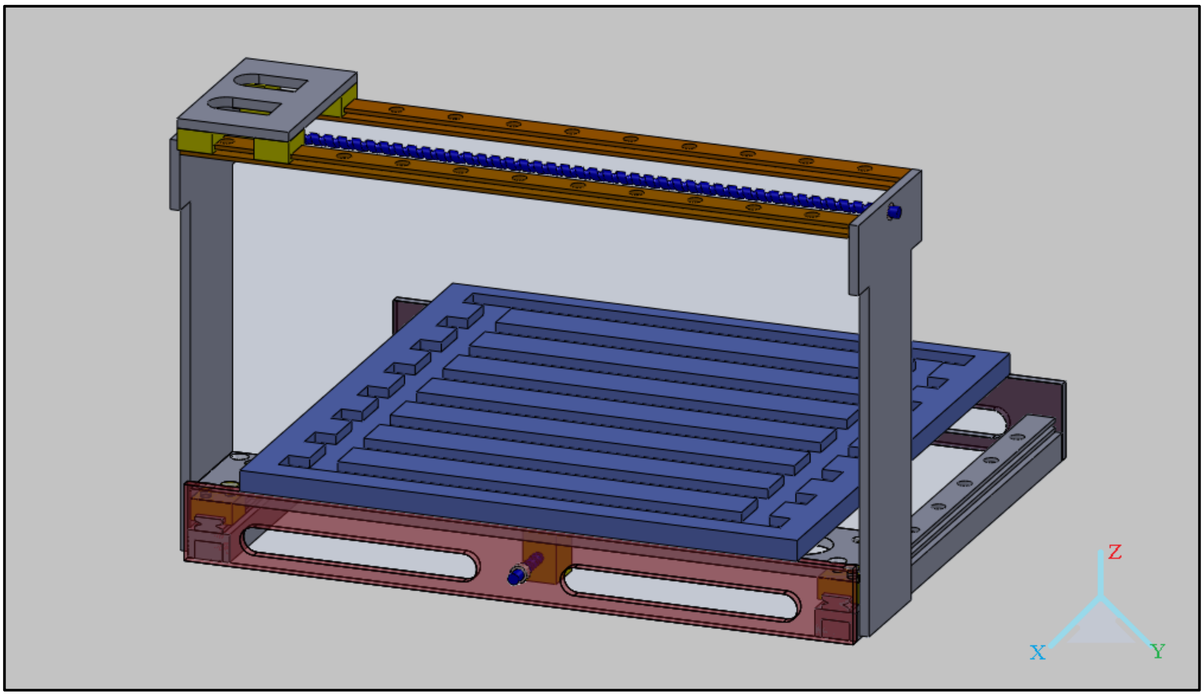

Figure 4.3: Machine tool base design in second Iteration

The base area is utilized fully. Designs of vertical columns were to be revised, as it came into view that it may lack counter weight support and no room for fasteners in components adjoining the vertical columns at the bottom. Also on the top of the columns the guide rails for X-Axis movement restrict the area movement of Z-Axis and its placement was major problem in this model.

Iteration 3.0

There were many problems when starting with x-axis as base. Criteria set for work started from y axis. When the dimensions of y-axis was constrained and adjusted according to the movement for z-axis. It was observed that modelling should be done starting from making a yz assembly. It was decided to use parametric software rather than non-parametric solid works as many things were interconnected like if any chances done on the machine structure also change in its assembly. Software called Pro-engineer was used for designing. After number of iterations within the system component by component, a structure shown in figure 4.4 was developed, many small parts installed on the machine for the visual realism.

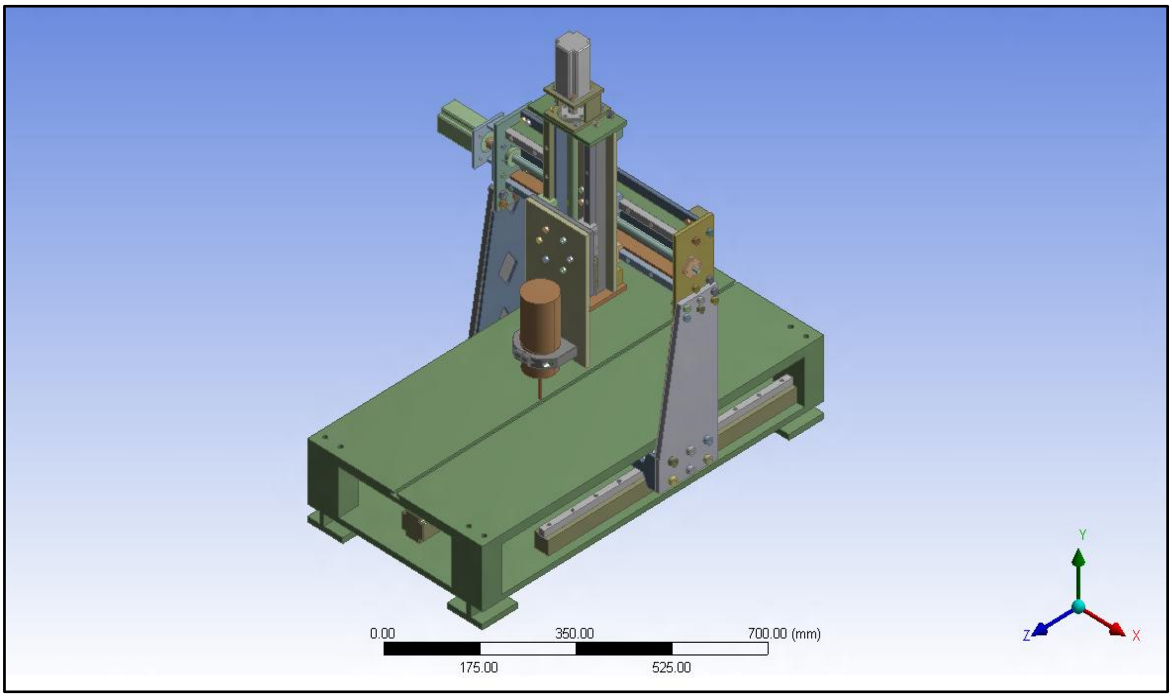

Figure 4.4: Iteration 3.0

Subsequent modifications were required during the CAE analysis and evaluation phase of the propsed strucrure.The model built in Pro-e was convereted into IGES format for analysis using CAE tools. Details of which has been given in next chapter.

4.3 SELECTION OF STANDARD COMPONENTS

Considering the factors like cost, availability and customization, the following major standard components have been selected as given in table 4.1

Table 4.1: Standard component purchased for present project.

S. No.	**Part name**	**Major Specification/ requirements**	**Quantity**
1.	Linear Motion Ways	Cross-section: 15x15 mm^2	6
2.	Block Bearings	Cross-section: 15x15 mm^2 Preloaded	12
3.	Ball Screw and Ball Screw Flange	16mm, Preloaded Pitch 5mm/revolution	3
(i)	Fixed End Horizontal Holding block (BK), for X-drive Ball Screw	To Hold 16mm Ball Screw towards drive motor side	1

(ii)	Float End Horizontal Holding block (BK), for X- drive Ball Screw	To Hold 16mm Ball Screw on free side	1
(iii)	Fixed End Vertical Holding block (FK), for Y and Z-drive Ball Screw	To Hold 16mm Ball Screw towards drive motor side	2
(iv)	Fixed End Vertical Holding block (FF), for Y and Z-drive Ball Screw	To Hold 16mm Ball Screw on free side	2
4.	Drive Motors Type: Stepper Motor	Pulses per revolution: 200 Torque: 265N-cm	3
5.	Router	Max. Rpm: 33,000 rpm, Power: 550 Watt	1
6.	Limit Switches	24 volt, micro limit switches	6
7.	Encoders	Type: Incremental Encoder Pulses per revolution: 1000	3

Detailed criteria for selection of above mentioned major standard components have been explained in subsequent sections.

4.3.1 Linear Guide ways

A linear guideway allows a type of linear motion that utilizes rolling elements such as balls or rollers. By using re-circulating rolling elements between the rail and the block, a linear guideway can achieve high precision linear motion. Compared to traditional slide, the coefficient of friction for the linear guideway is only 1/50 [27]. Because of restraint effect between the rails and the blocks, linear guideway can take up loads in up/ down and the left/right directions. With these features, linear guideways greatly enhance moving accuracy, especially when accompanied with precise ball screws.

The selection of LM ways is dependent upon the type and value of loading the machine drive will be subjected. The selection criteria for block bearing are dependent upon the LM ways selected.

Advantages and Features of Linear Guideways

1. High positional accuracy
2. Long life with high motion accuracy:

When a load is driven by linear motion guideway, the frictional contact between the load and the bed desk is rolling contact. The coefficient of friction is only 1/50 of the traditional contact, and the difference between the dynamic and the static friction is small. Therefore, there would be no slippage while the load is moving.

The model of LM ways selected has following design characteristics [27]:

Basic Dynamic load rating C = 11.38 kN

Basic static load rating C_o = 16.97 kN

Static Rated moment:

M_R = 0.12 kN-m

M_P = 0.10 kN-m

M_γ = 0.10 kN-m

The definition of basic static load rating is static load of constant magnitude and direct resulting in total deformation (permanent) of 0.0001 times the diameter of the rolling element and the raceway at the contact point subjected to the largest stress. The maximum static load applied to linear guideway must not exceed the basic static load rating.

A large safety factor is especially important for linear guideway which are subjected to impacts. Static safety factor considered 5.0 which as suggested by the manufacturers of linear guideway [27]. These factors of safety include the load conditions when blocks and rails are subjected to impacts and vibrations.

To Calculate the Working load capacity the following mathematical expression is used [27]:

$f_{sl} = \frac{Co}{P}$

Where,

f_{sl} = Static safety factor for simple load

Co = Static load rating (kN)

P = Calculated working load

Considering C_o =16.97 kN and f_{s1} = 5.0, using equation 4.1, the value of working load is given below:

$P = \frac{Co}{fsl}$

$P = \frac{16.97}{5}$

P = 3 kN

Even factor of safety has been considered as 5.0, and the rails can withstand up to 3000 N of load. Actual self-weight of model is around 600 N approx. This 600 N of whole structure weight will come above the rails on X-axis which will take more load than other two axis rails. Two blocks on single rail is used to minimize the moment caused due to motion of blocks. It also distributes the load evenly and helps the ball screw to move freely.

Moment on Linear Motion rails [27]

$f_{sm} = \frac{Mo}{M}$

f_{sm} = Static safety factor for moment

Mo = Static permissible moment

M = Calc. applying moment

$f_{sm} = \frac{Mo}{M}$

$5 = \frac{0.12}{M}$

M = 0.024 KN-mm

CALCULATION OF LOAD ON ONE BLOCK OF X-AXIS:

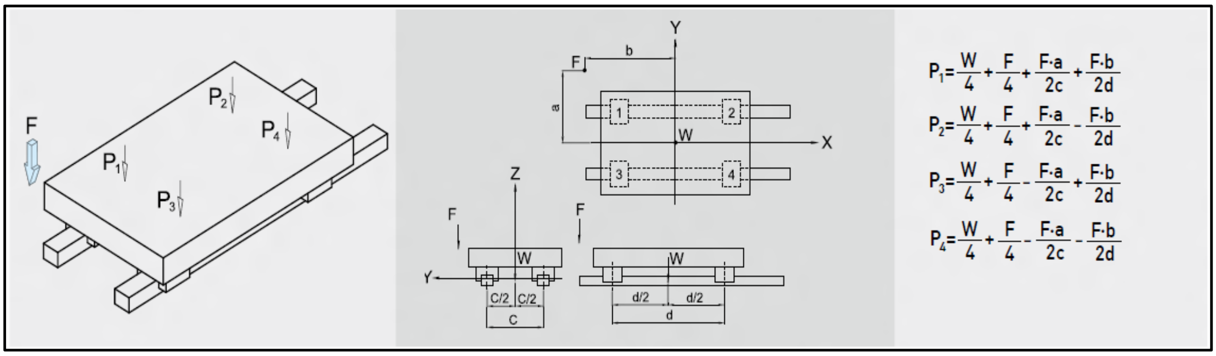

Figure 4.5: LOAD LAYOUTS OF X-AXIS [27]

Weight of the plate over the four blocks (W) = 100 N

Force is the weight hanging on each vertical column (F) = 250 N

a, b, c, d are the distances as shown above

a = 235mm, b = 4.5mm, c = 375mm, d = 70mm.

P_1 = 166 N, P_2 = 134 N, P_3 = 166 N, P_4 = 134 N

CALCULATION OF LOAD ON ONE BLOCK OF Y-AXIS:

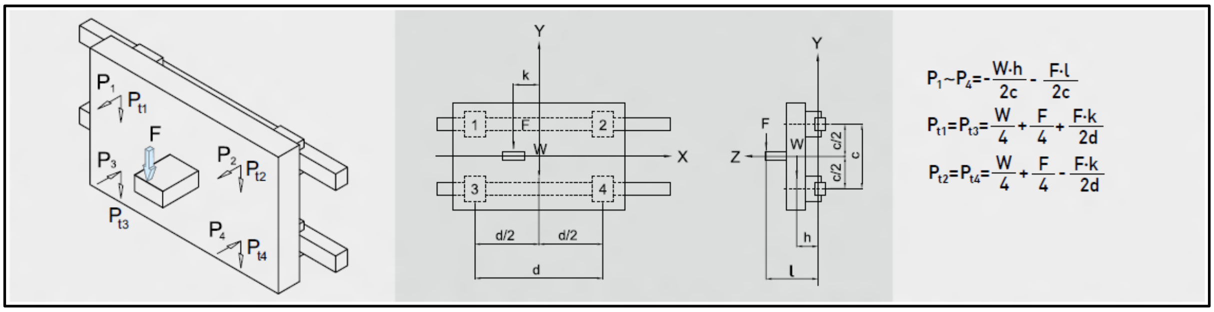

Figure 4.6: LOAD LAYOUT OF Y-AXIS [27]

Weight of the plate over the four blocks (W) = 15 N

Force is the weight hanging (F) = 215 N

l, h, c, d are the distances as shown above

l= 33mm, h = 30mm, c = 104mm, d = 70mm.

P1 = P2 = P3 = P4 = -31.947 N (negative is for the direction)

$P_{t1} = P_{t3}$= 57.5 N

$P_{t2} = P_{t4}$ =57.5 N

CALCULATION OF LOAD ON ONE BLOCK OF Z-AXIS:

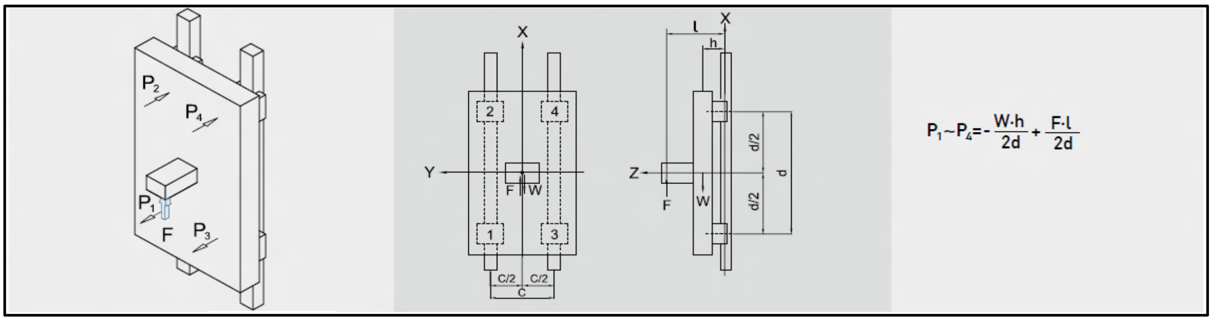

Figure 4.7: LOAD LAYOUT OF Z-AXIS [27]

Weight of the plate over the four blocks (W) = 15 N

Force is the weight hanging (F) = 50 N

l, h, c, d are the distances as shown above

l= 33mm, h = 30mm, c = 104mm, d = 70mm.

P1 = P2 = P3 = P4 = 8.5714 N

Calculating the Nominal life [27]

Basic dynamic load rating is used for calculation of service life of linear guideway. Flaking occurs eventually when rolling elements of linear guideway are continuously subjected to repeated stresses, the raceway surfaces show fatigue.

$L = \left(\frac{C}{P}\right)^3.50$ Km

L = Nominal Life

C = Basic dynamic load rating

P = Actual load on one block

Considering the maximum load (P) = 166 N

$L = \left(\frac{C}{P}\right)^3.50$ km

$= \left(\frac{11380}{166}\right)^3.50$ km

= 1,61,09,146.32 km

Calculating the Service life [27]

$L_h = \frac{L.\ 10^3}{v_e.60} = \frac{\left(\frac{C}{P}\right)^3.\ 50 \text{ x } 10^3}{v_e.60}$

L_h = Service life (hr)

L = Nominal life (Km)

v_e = Speed (m/min)

$\frac{C}{P}$ = Load factor

Movement of machine in Rapid (v_e) = 5000mm/min

$L_h = \frac{1{,}61{,}09{,}146.325467 \text{ x } 10^3}{5 \text{ x } 60}$

= 53697154.418 hrs.

CHAPTER: 5

COMPUTER AIDED ANALYSIS OF MACHINE STRUCTURE

In this chapter design evaluation and fabrication of prototype of a 3-axis vertical milling machine has been discussed. “ANSYS 10.0” has been used as CAE tool for design analysis. Basic aim of the present work is to develop a low cost CNC 3-axis vertical wood carving machine. The machine grade steel materials are very costly and further the machining cost is higher for these machine grade steels. Also a number of post machining operations like heat treatments and surface grinding are required to get the components made from such materials to their final usable shape. Considering the factor of cost for CNC machining of machine grade steel and the associated cost of post machining treatments it was decided to use mild steel as material for making components for the present prototype. It would be helpful to the future users of the machine (artisans in remote locations) as for replacement of machine parts/ spare parts can be made available using conventional machining with fair accuracy as well as cost effectively.

For the effective utilization of this product end users can easily find component replacement as the majority of the components are developed from flat mild steel plates of various thicknesses. Thus this machine finds its user market among the small scale entrepreneurs and the artisans/ craftsman, for who cost of owning such technologies is still very high. Cost is one of the main reasons, why most of the artisans/ craftsman do not use the automation for their customized manufacturing facility. As a result many artisans/ craftsman find it difficult to meet their production requirement.

The next challenge was the working life and stability of the machine prototype using mild steel as material. The criteria for the better machine safety and service life can be achieved with increasing factor of safety for the designed components which would compensate for using alternative costly machine grade steel.

Finite element analysis using ANSYS has been done to verify the structural deflection in the machine drives and the deflections in the overall machine prototype model which has been developed by our research team. The other major component which is heart of machine is spindle/ router. The mounting to be developed for spindle must be sufficiently rigid, mechanically stable under cutting load conditions and should be able to withstand the high frequency vibrations that router/ spindle generate during carving operations.

For the purpose of CAE analysis of machine prototype the main machine model has been divided into two main parts as written below:

(i) Main machine structure including all machine elements.

(ii) Study of mechanical stability of router mounting sub-assembly.

The subsequent sections of this chapter present the detailed procedure followed for CAE analysis of above mentioned machine parts.

5.1 GENERAL PROCEDURE IN FINITE ELEMENT METHOD (FEM) ANALYSIS IN ANSYS

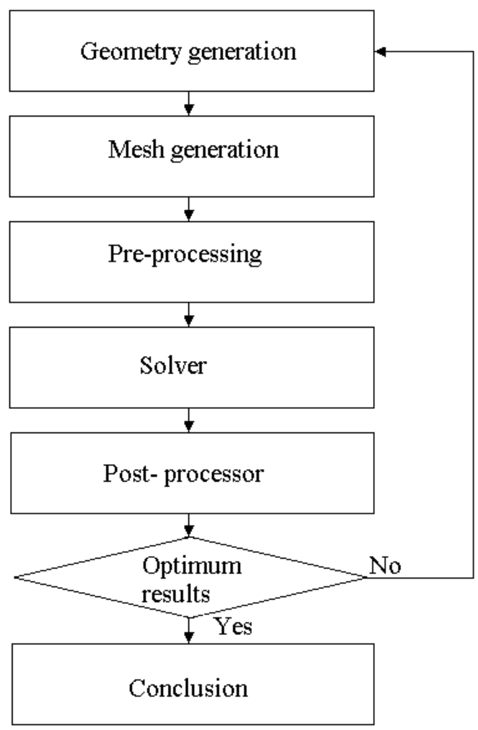

Figure 5.1: Finite element methodology

The general FEM methodology as followed in ANSYS has been shown in figure 5.1. The stepwise procedure adopted in the ANSYS software for analysis of the machine tool structure has been explained below:

5.1.1 Geometry Generation

Geometry refers to 3-dimensional part model which is used as an initial input for FEM analysis. The 3D geometry of the machine tool has been developed in Pro-E, which is further saved as IGES format before it is translated to ANSYS. After getting the 3D model of the complete machine tool assembly in ANSYS, the next important step is to define the connections between various components in assembly to form complete machine and its systems. When geometry is borrowed in ANSYS environment the model connections between the components are also generated automatically, which are program controlled, but the connections may or may not be exactly according to type that exists physically between two components. In case of the present model of machine structure the contacts were given manually. ANSYS support many types of contacts/ connections and some of them used in designing the machine structure were bonded, frictional, translational, and revolute.

The machine tool structure has to be tested for static load conditions. Hence, the static structural module was considered which would be best for this kind of analysis.

Materials properties used for type of material used for each component in machine structure were added. Two properties essential for analysis in ANSYS are Young's Modulus and Poisson's Ratio. Four different materials were used in the present model. The essential properties of these materials have been listed in table 5.1, table 5.2, table 5.3, and table 5.4.

Table 5.1 Properties for Mild Steel

S. No.	**Description/ Material Property**	**Value/ Magnitude**
1	Density	7.85×10^{-6} kg/mm^3
2	Young's Modulus	2×10^5 MPa
3	Poisson's Ratio	0.3
4	Compressive Yield Strength	250 MPa
5	Tensile Yield Strength	250 MPa
6	Tensile Ultimate Strength	460 MPa
7	Reference Temperature	22°C
8	Bulk Modulus	1.6667×10^5 MPa
9	Shear Modulus	76923 MPa

Table 5.2 Properties for Hardened Steel

S. No.	Description/ Material Property	Value/ Magnitude
1	Density	$7.85x10^{-6}$ kg/mm^3
2	Young's Modulus	$2.08ex10^5$ MPa
3	Poisson's Ratio	0.33
4	Compressive Yield Strength	250 MPa
5	Tensile Yield Strength	250 MPa
6	Tensile Ultimate Strength	460 MPa
7	Reference Temperature	22°C
8	Bulk Modulus	$2.0392x10^5$ MPa
9	Shear Modulus	78195 MPa

Table 5.3 Properties for Aluminium Alloy

S. No.	Description/ Material Property	Value/ Magnitude
1	Density	$2.77x10^{-6}$ kg/mm^3
2	Young's Modulus	71000 MPa
3	Poisson's Ratio	0.33
4	Compressive Yield Strength	280 MPa
5	Tensile Yield Strength	280 MPa
6	Tensile Ultimate Strength	310 MPa
7	Reference Temperature	22°C
8	Bulk Modulus	69608 MPa
9	Shear Modulus	26692 MPa

Table 5.4 Properties for High Speed Steel

S. No.	Description/ Material Property	Value/ Magnitude
1	Density	$7.85x10^{-6}$ kg/mm^3
2	Young's Modulus	$2.04x10^5$ MPa
3	Poisson's Ratio	0.31
4	Compressive Yield Strength	250 MPa
5	Tensile Yield Strength	250 MPa
6	Tensile Ultimate Strength	260 MPa
7	Reference Temperature	22°C
8	Bulk Modulus	$1.7895x10^5$ MPa
9	Shear Modulus	77863 MPa

5.1.2 Mesh Geneneration

Grid Generation (mesh generation) is sub division of domain into a number of smaller sub domains (grid/ cells/ control volumes). Many elements and nodes are genrated according to the size of the geometry and parts inside the part model or assembly.

Mesh is automatically generated in ANSYS taking default relevance center as coarse. If the solution for the coarse mesh converges, then one can proceed further with the analysis, otherwise if the solution doesn't converge the mesh can be made finer according to need for the converged solution. The relevance option allows controlling the fineness of the mesh for the entire model, like it is possible to indicate a preference towards high speed (-100) or high accuracy (+100) solutions. Finer the mesh more accurate is the result. The coarse mesh is less accurate. As one moves in scale from coarse to fine, the number of mesh elements and nodes are increased. Finer mesh uses more elements, more time and ultimately more system resources.

5.1.3 Pre Processing

(a) Boundry Conditions/ Constraints

In the next step appropriate boundary conditions are required to be specified like kind physical constraints. Boundary conditions are set according to the problem definition.

(b) Application of Load

The type of load condition has to be precisely applied at the appropriate location and direction in the model. The type of loading in present analysis includes; the cutting force on the cutter and the consideration of the self-weight of the machine components in the assembly.

5.1.4 Solver

(a) Type of FEM evaluation criteria

Mainly three types of evaluation criterias are used in FEM analysis, which are Von-Mises, Trescas, and minimum strain energy. The Von-Mises criteria is generally used in materials which are ductile in nature. Thus in the present work the Von-Mises criteria has been used as the material used for different components of machine tool are ductile and the componenets has been accepted to be safe untill the deformations are in elastic limit.

(b) Convergence criteria:

In ANSYS the load is increased gradually so that maximum value of load it achieved from zero to highest level in a predecided time interval. During the present study time interval has been decided by carrying out a number of trial studies. Initally time period value has been chosen as 5 seconds, but the results do not converge for this value of time period. Then higher values of load buildup time have been choosen and finally a suitable time period value is selected. Analysis setting for the application of load in Ansys is applied in two ways which are:

a) Sub step method and

b) Auto time stepping method.

It was preferred to use auto time stepping. In auto time stepping, as the load is applied from zero to its maximum. It divides this increasing nature of load in number of seconds. It is done by hit and trial so we can iterate for minimum, maximum and step end time for the step size. In auto time stepping next step depends on the previous one that is, if the previous step has been converged it will try to increase the step size towards the maximum step size given by the user. It tries to make a non-linear graph for time of the step size. If the solution doesn't converge for particular range in analysis bisection can be occurred decreasing time in the step size and try converging for lesser time.

5.1.5 Post Processing

The converged results from the FEM solver can be finally produced in suitable presentable format using post-processor. Using ANSYS post-processor the converged results has been prepared in the form of contour plots for stress levels, strain rate, and deformation. These three types of contour plots have been visually studied for deriving the logical results. If the level of stresses or deformation are within the acceptable range the design is accepted, otherwise the design has to be modified so as to bring the level of stresses or deformation within an accepteable range.

5.2 FEM ANALYSIS OF MAIN MACHINE STRUCTURE CONSIDERING ALL COMPONENTS

Using the FEM Methodology the two main systems of the machine prototype have been analysed in "ANSYS 10.0" for the type of loadings they will be subjected. The section below presents the results for the FEM analysis of these two systems of the machine prototype.

5.2.1 Geometry Generation

For faster solution from FEM solver it is desired that we should try to minimise the number of mesh elements and nodes in the model. To achieve this, the machine table and the base table on which bottom rails are fastened using machine screws has been suppressed. These components are rigid structure which doesn't make much difference for the drive motion and deflection. Thus the base of the bottom rails have been taken as fixed supports.

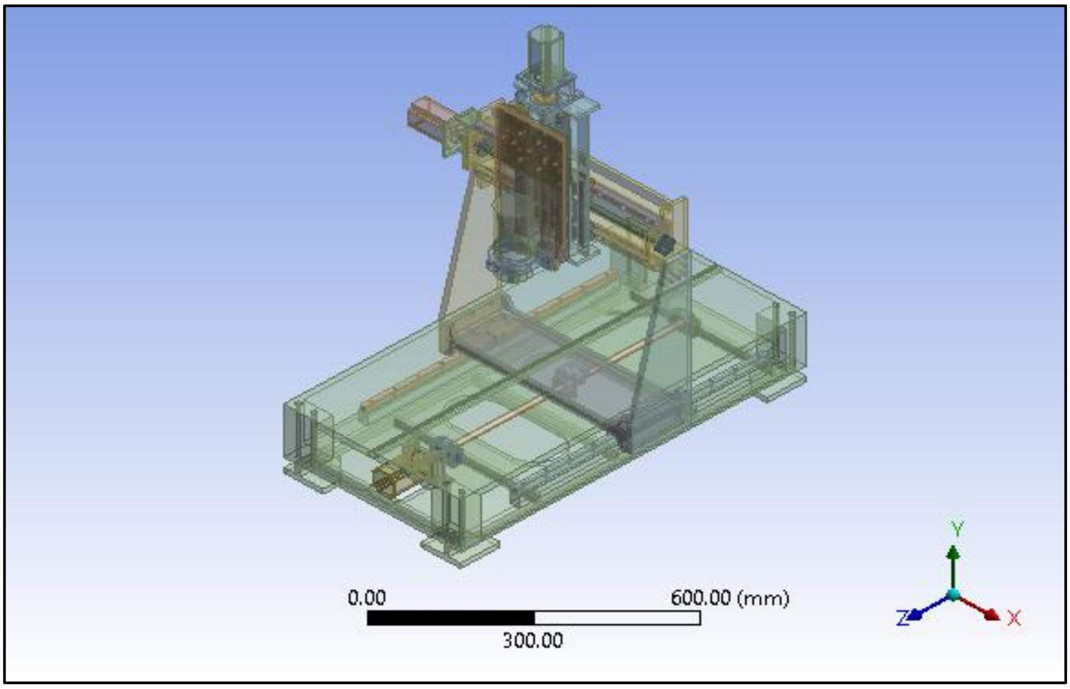

Figure 5.2: 3-Axis Machine Tool model for Analysis

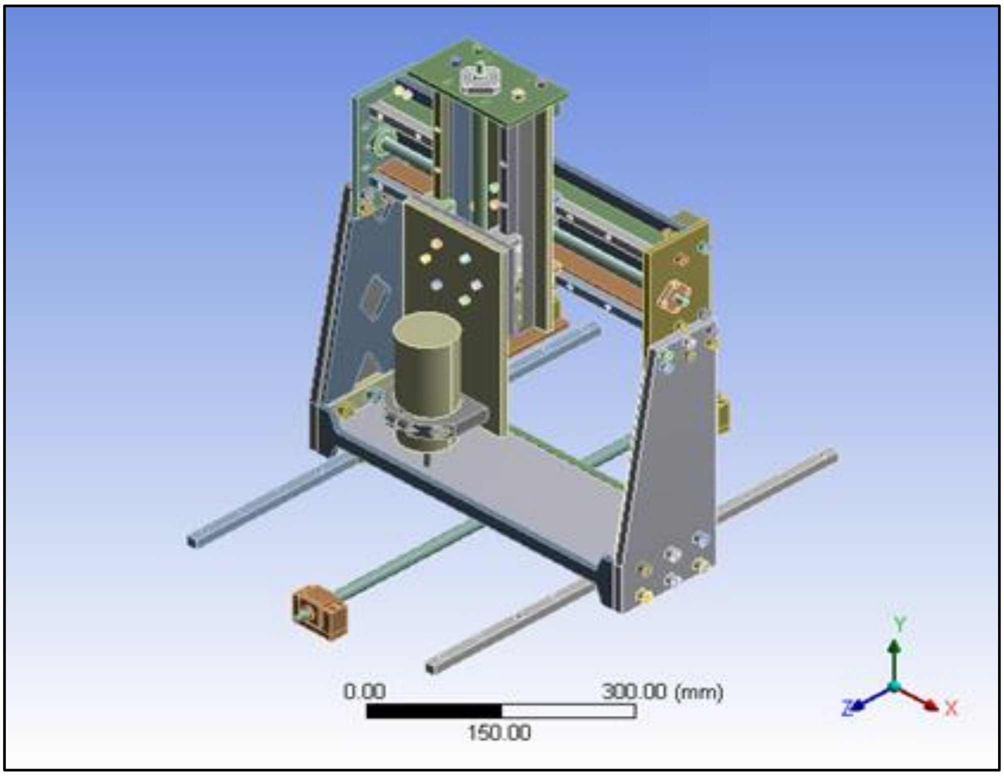

Figure 5.3: Active elements in model for Structural Analysis

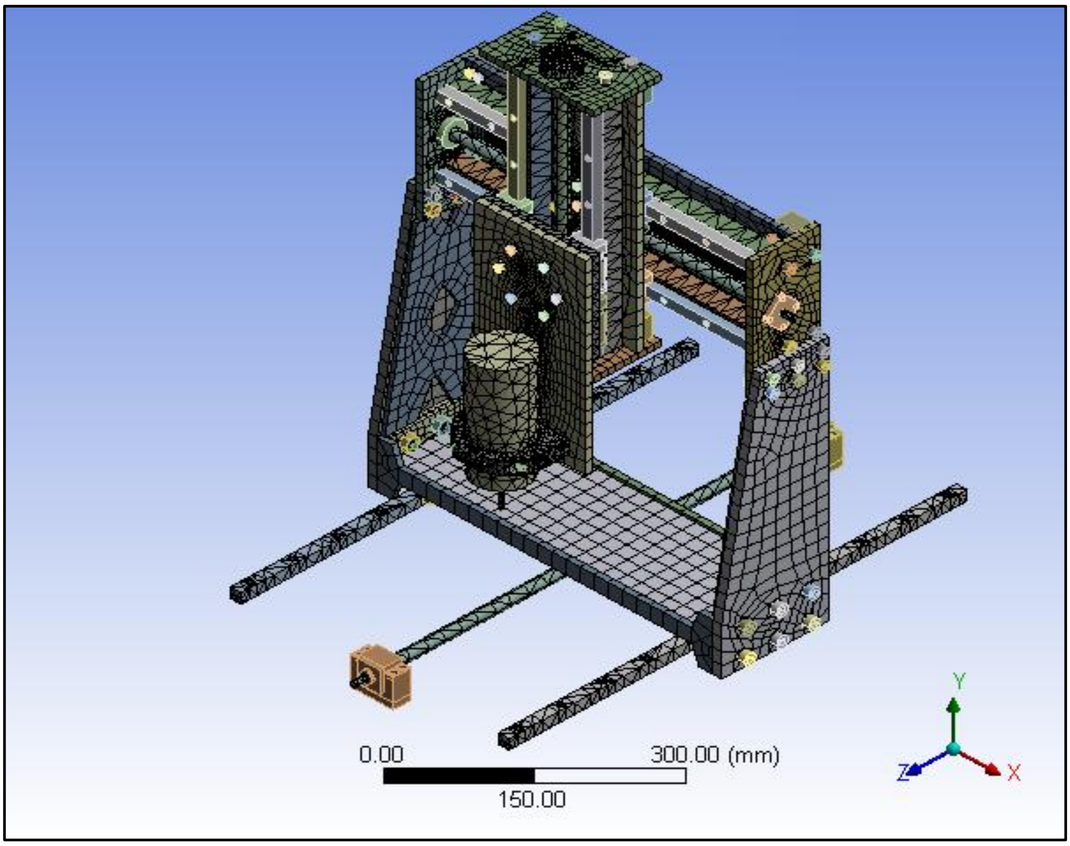

Figure 5.4: Mesh of the machine structure

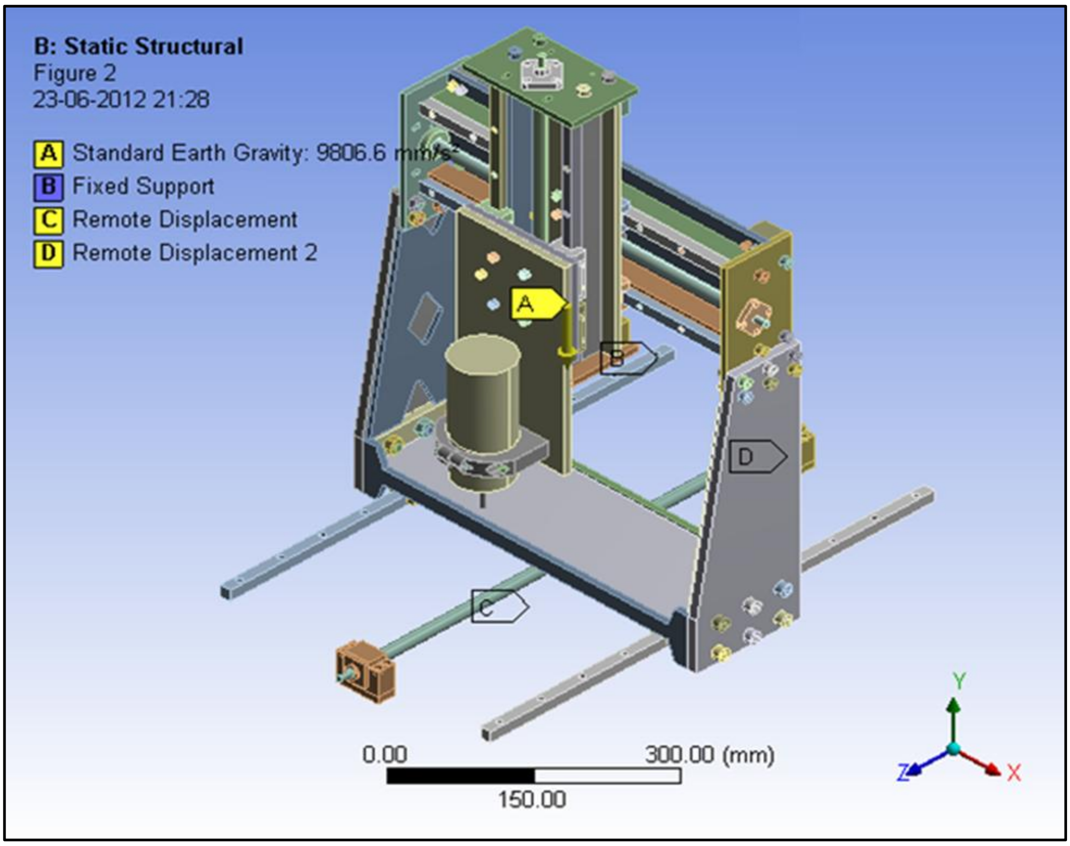

Figure 5.5: Boundary conditions for the Model

The dimensions of bounding box of total machine assembly chosen for FEM analysis is length (along X-axis) is 647mm, width (along Y-axis) is 884mm and height (along Z-axis) is 880mm. The geometric model selected for FEM analysis as shown in figure 5.2. This model has a total number of bodies equal to 344 and active bodies are 278.

5.2.2 Mesh Generation

Relevance center for automatic mesh generation for this analysis is taken as medium. The automatic meshing generates a total of 305949 nodes and number of elements generated are 99156, as shown in figure 5.4.

5.2.3 Pre-Processor

(a) Boundry conditions / constraints

(b) Application of load

In Pre- processor boundary conditions are given by constraining parts that are rigid with respect to ground. In this model as shown in the figure 5.5, the rails in x-axis direction have been constrained, as denoted by abbreviation “B-fixed support”. The flanges of x-axis ball screw are also fixed, which are denoted as C and D. After defining these boundary conditions, load is applied in the form of standard gravitational effect on the each component of whole structure in negative of Y-direction as shown in figure 5.5.

5.2.4 Solver

(a) Type of FEM analysis

Von-Mises criteria has been used in this study, which gives us equivqlent stress corresponding to the applied loads and boundry conditions. Static structural scheme is used, alongwith Von-Mises criteria for FEM study considering the satndard earth gravity for the study of effcet of self weight of machine structure on the deformation/ deflection/stress levels on different machine components.

(b) Convergence criteria

Auto time stepping is used for this analysis as the standard gravity is given from zero to its maximum magnitude level of 9806.6 mm/s^2. The FEM solver divides this increasing nature of gravity in user given time period. After a number of iterations the time period value selected was 5 seconds.

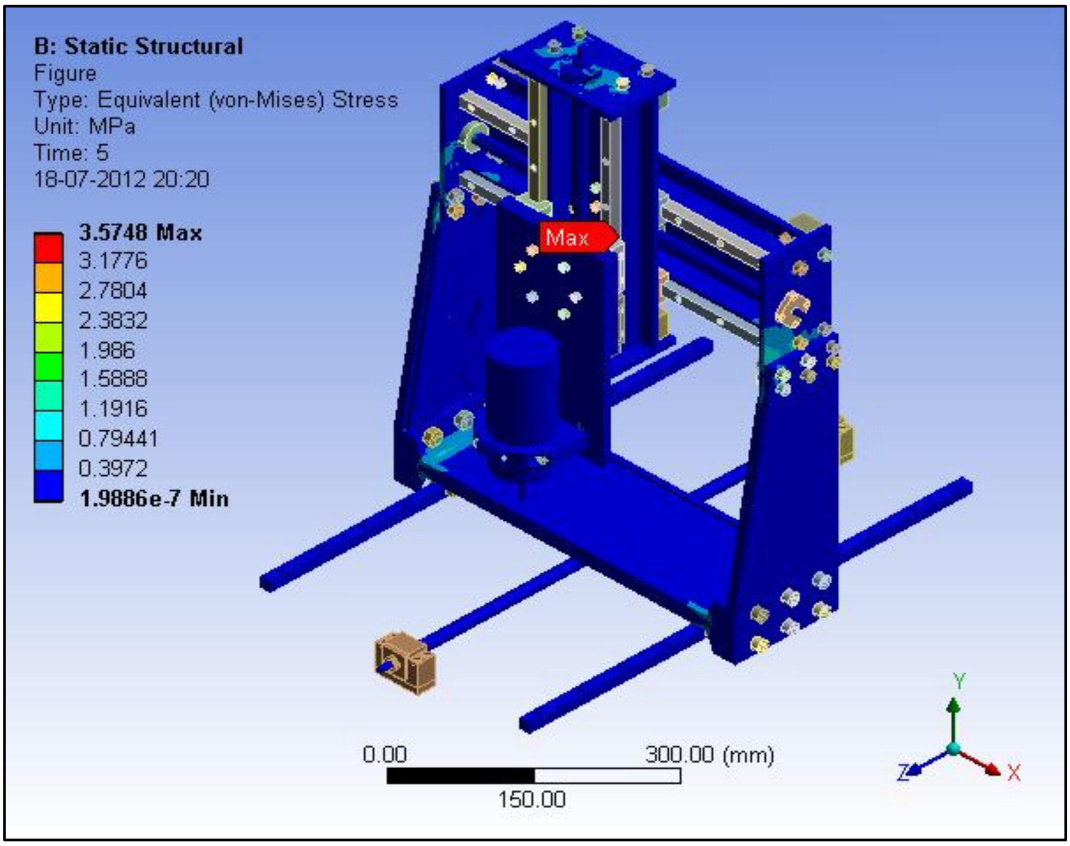

Figure 5.6: Details of the Equivalent Stress

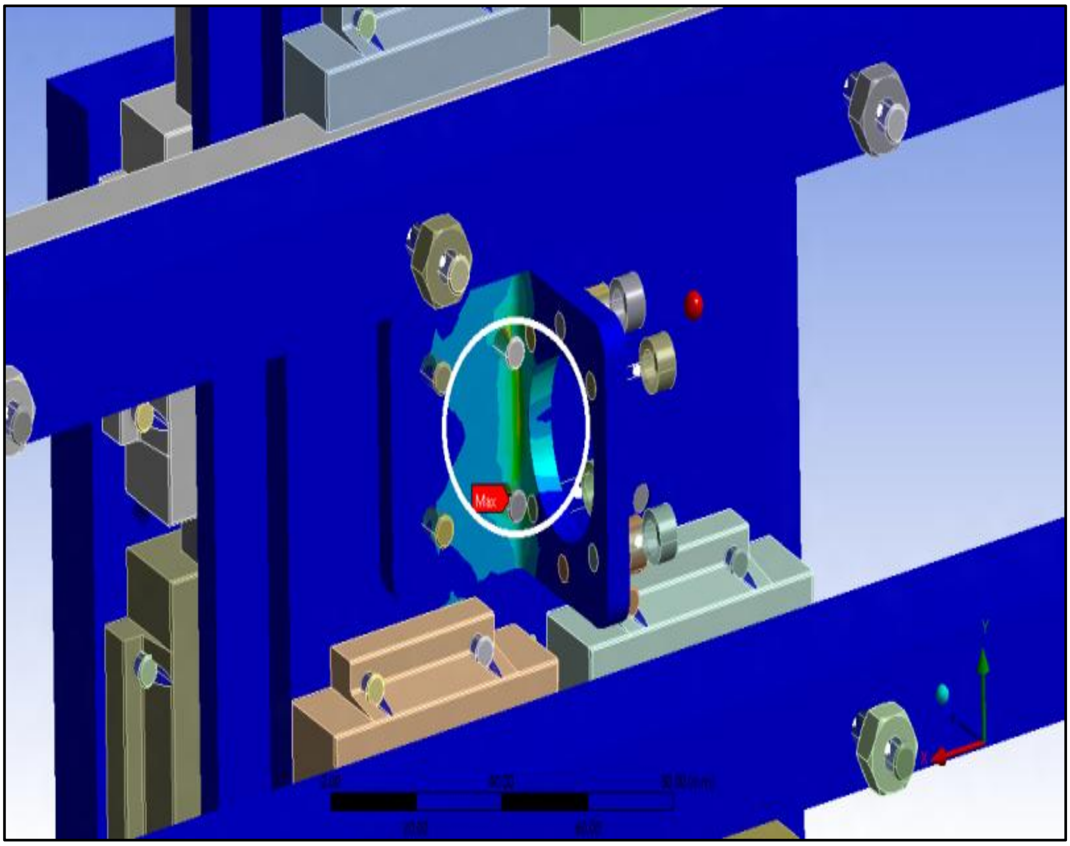

Figure 5.7: Back of y-axis for visual realism of maximum equivalent stress

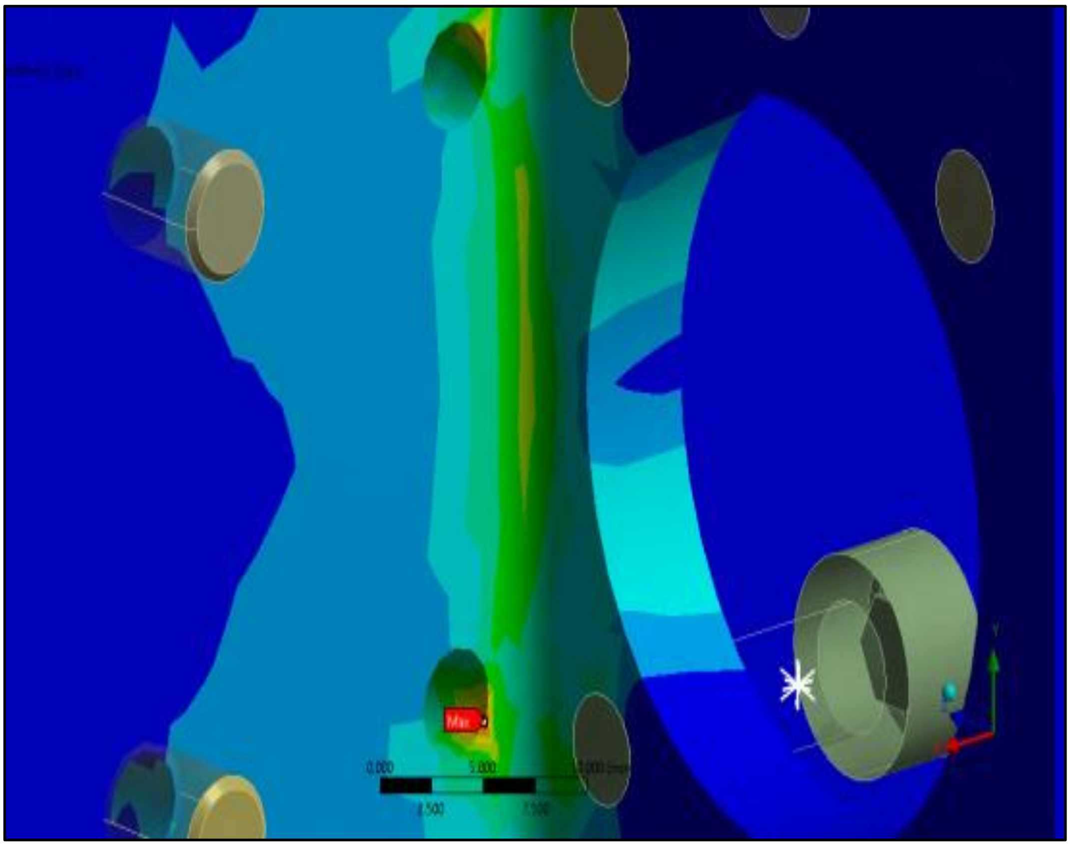

Figure 5.8: Zoom in of BS connector plate

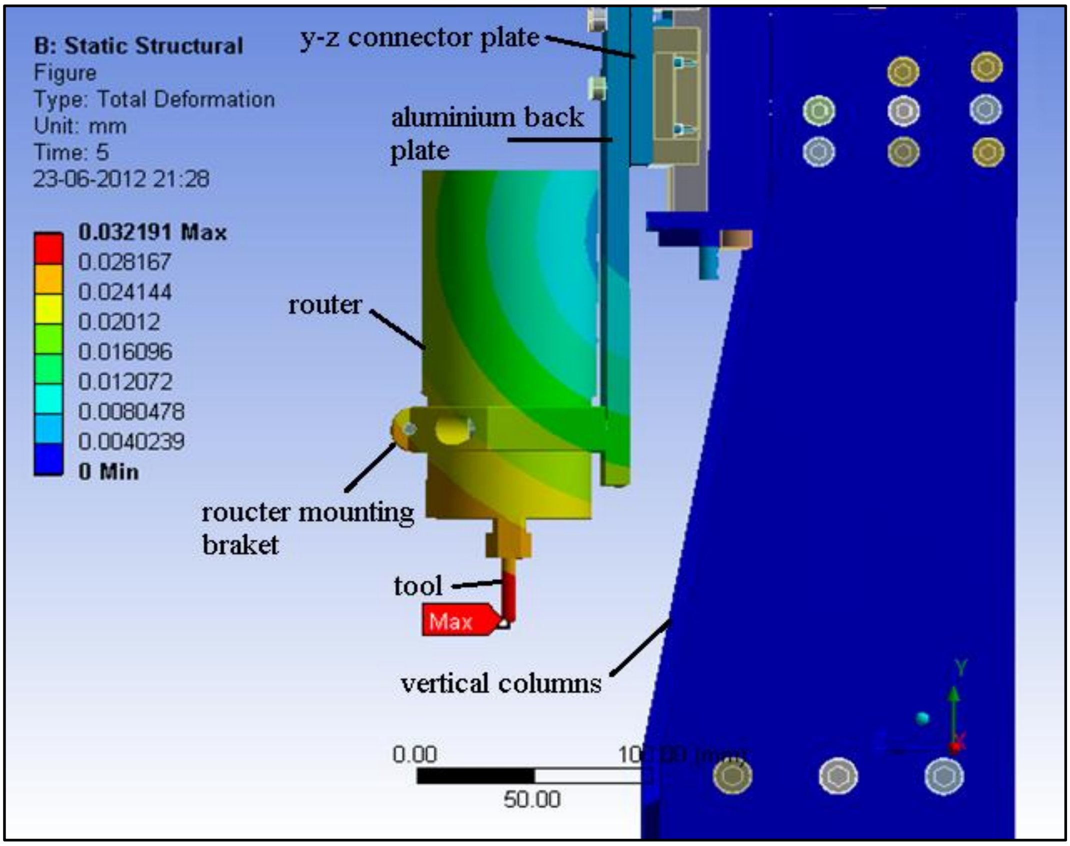

Figure 5.9: Deformation of whole structure

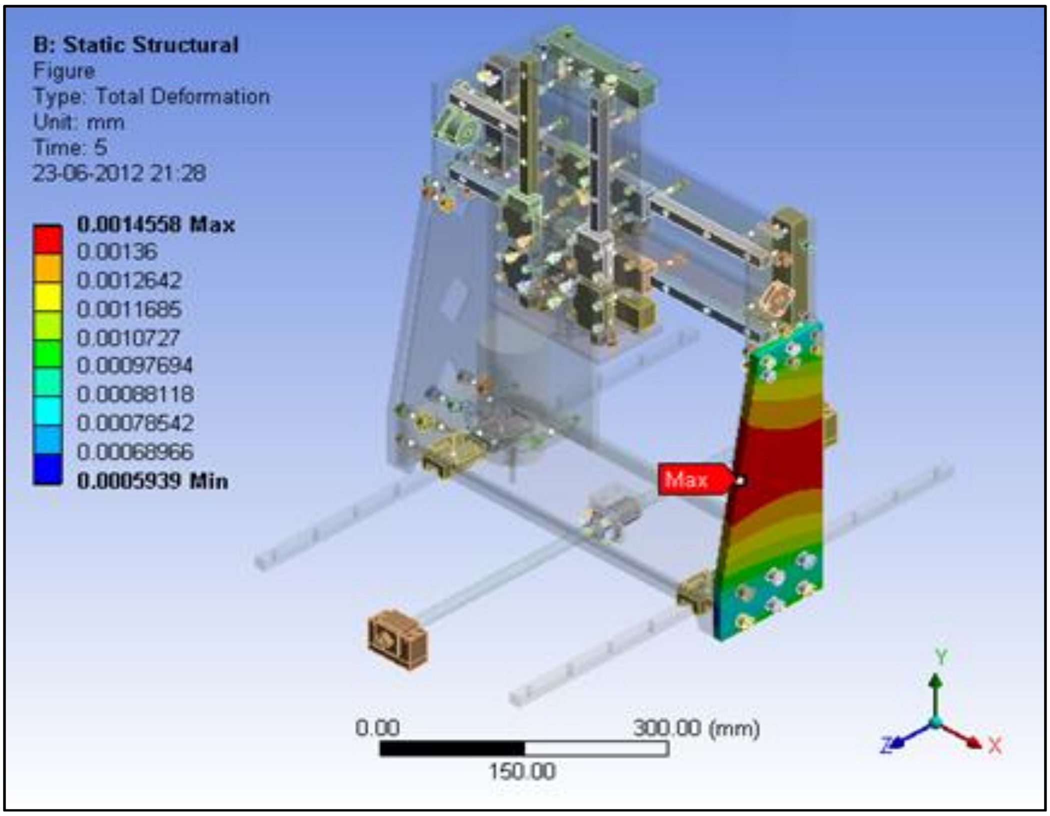

Figure 5.10: Detailed deformation on the vertical column

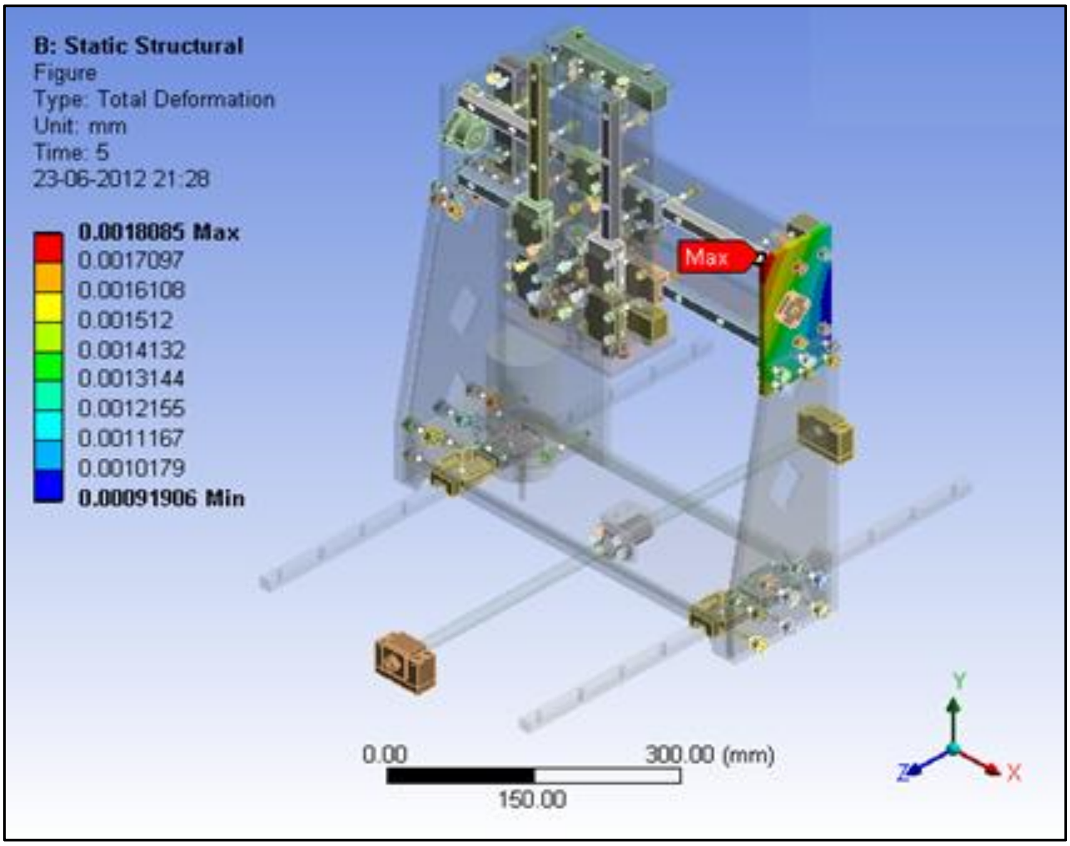

Figure 5.11: Deformation on the end plate of y-axis

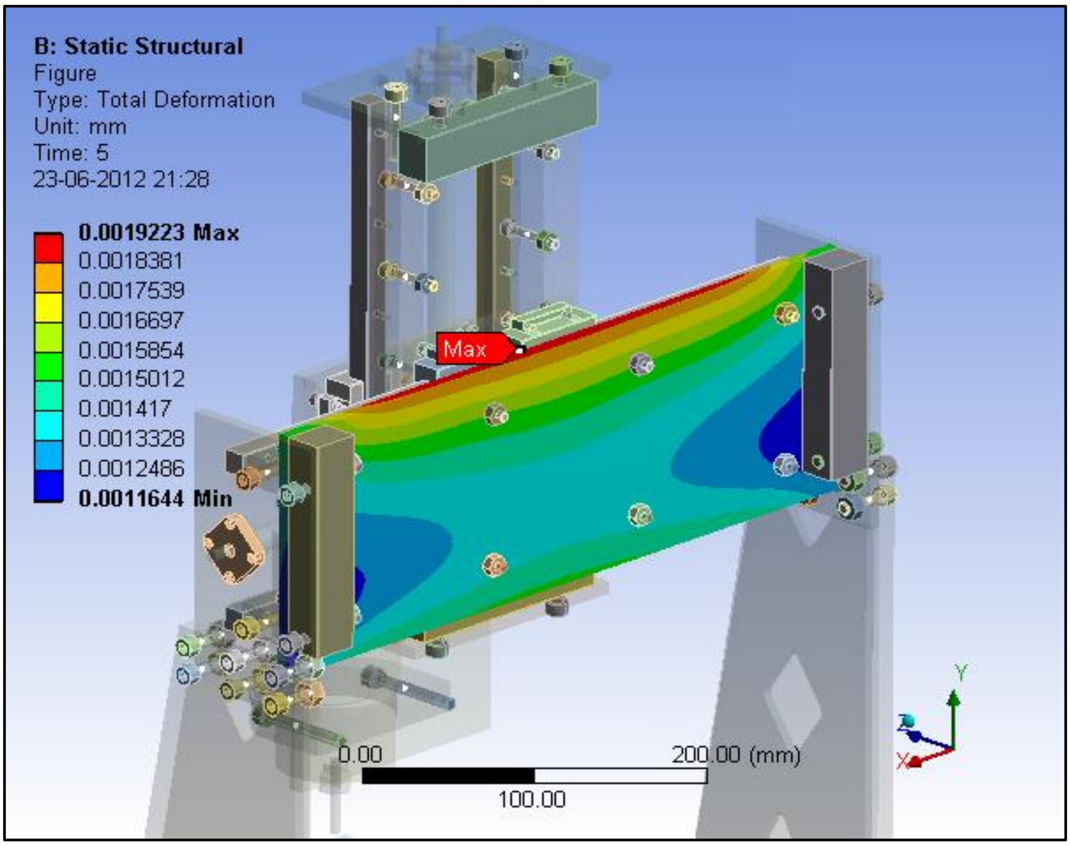

Figure 5.12: Deformation on the back plate of y-axis

5.2.5 Results for static structural analysis of machine prototype

a) Equivalent stress (Von-Mises)

As apparent from the figure 5.6, the maximum equivalent (Von-Mises) stress under the load of standard gravity is 3.5748 MPa. The part at which maximum equivalent stress is BS connector and its material is mild steel, the yield tensile strength of mild steel is 250Mpa. So, the structure under gravitational load is quite safe. The figure 5.7 and figure 5.8 shows the enlarged view of the component and the position of the maximum stress. This region is at the location where the BS connector is fastened with machine screws to the ball-screw flange of Y-axis drive unit.

(b) Maximum Deformation

The maximum deformation for the whole structure with standard gravity has been shown in figure 5.9. The magnitude of the maximum deformation is 0.03 mm which is on the tip of the cutting tool. This deformation is due to overhang which is justified for any cutting tool, otherwise from the coloured chart it is clear that till y-z connector plate of z-axis the deformation is of range 0.004, which is negligible. The overhang can be avoided as it is clear that deformation starts getting worse from the aluminium back-plate on which router mounting bracket have been attached. To avoid this deformation, aluminium back plate can be modified by adding two T-section stiffeners in front side vertically on it. This will reduce the deformation of this plate considerably. Another remedy could be to select a hard aluminium alloy material for this plate like aluminium 6061. Deformations on other major parts are shown in the figure 5.10, 5.11 and 5.12. This structure is quite safe as the values of the deformations are around 0.1micron. The maximum stress levels are also less than 4.0MPa.

5.3 MECHANICAL STABILITY OF ROUTER MOUNTING SUB-ASSEMBLY

5.3.1 Geometry Generation

After modelling has been done, part model is converted to IGES format for the design evaluation in Ansys software as discussed earlier. This model was modified for second type of analysis which is study of mechanical stability of router mounting sub-assembly. So, it was decided for problem solving that only that much part model should be activated which has router mounting bracket and some attachments adjacent to it. As discussed in the previous section that, till y-z connector assembly of z-axis machine structure is almost rigid under the load of gravity which is negligible as the deformation value is $1/10^{th}$ of micron. The geometric model selected for FEM analysis as shown in figure 5.13, which has a total number of bodies equal to 344 and active bodies are 14.

5.3.2 Mesh Generation

Mesh is automatically generated in ANSYS taking relevance center as medium. After meshing is done total number of nodes formed was 39148 and number of elements generated was 17186.

5.3.3 Pre-Processor

(a) Boundry conditions / constraints

(b) Application of load

In this part of analysis, the connector plate of z-axis drive assembly is fixed which is denoted as “B”, as shown in the figure 5.15. The gripping bolt used in front of the router mounting bracket has been given pretension of 3400N, which allows the mounting bracket to make a firm grip around the neck of the router. The router is supposed to slip inside the router mounting bracket when excessive cutting force is applied along Z-axis in upwards direction through the cutting tool. The magnitude of permissible cutting reaction force acting upwards along z-axis for tool of diameter 6.37mm has been taken as 46N. The aim of this study is to assure the proper gripping and no slippage of router in up or downwards direction upto a 46N cutting force, but a slippage in upwards direction in case the cutting tool mounted on router experiences greater cutting forces.

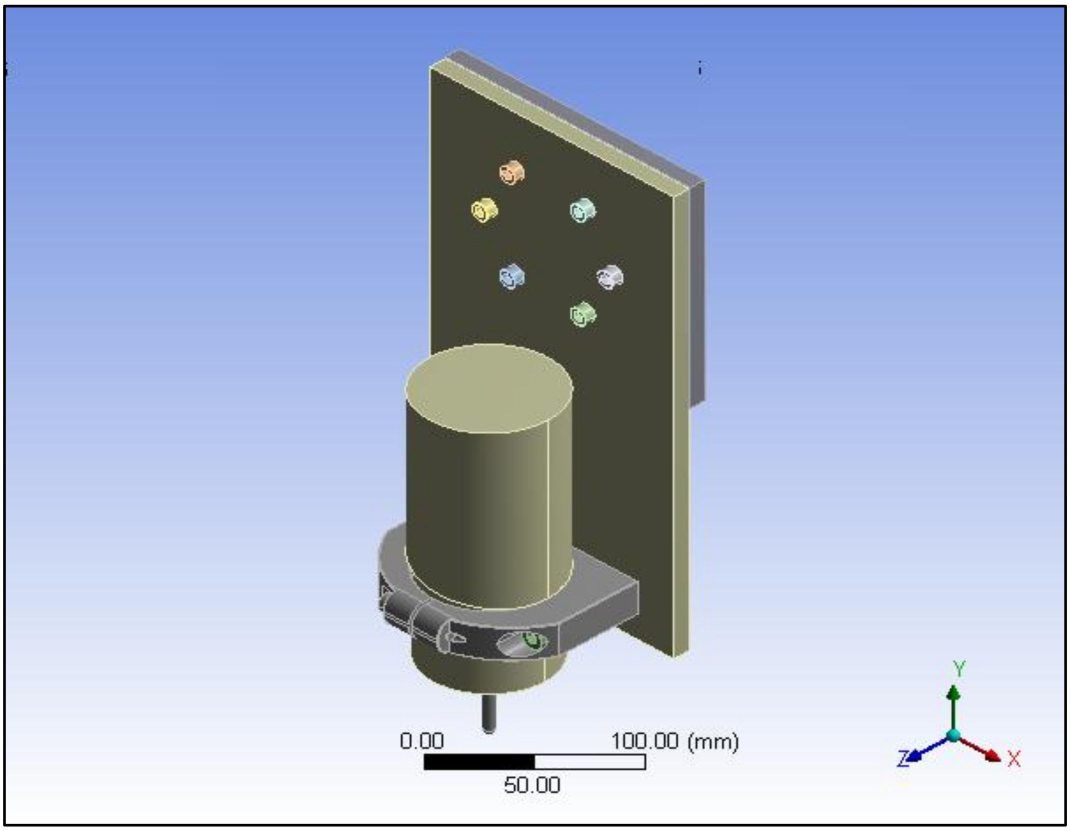

Figure 5.13: Router sub-assembly

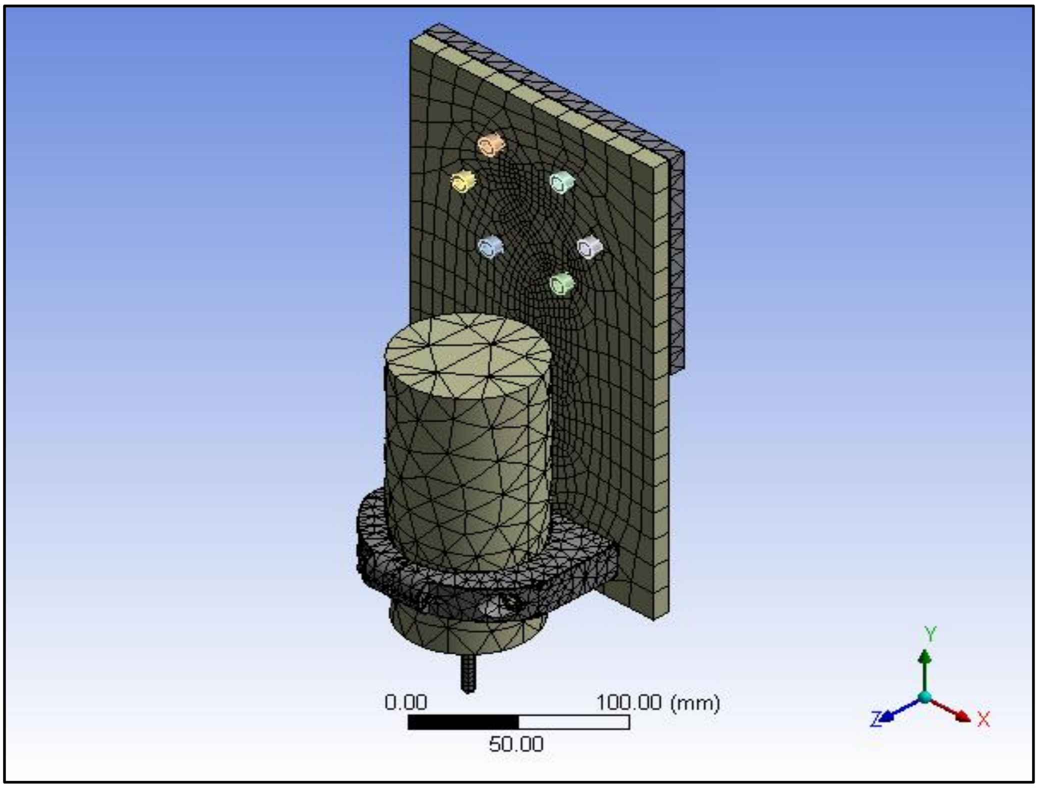

Figure 5.14: Mesh elements in router sub-assembly

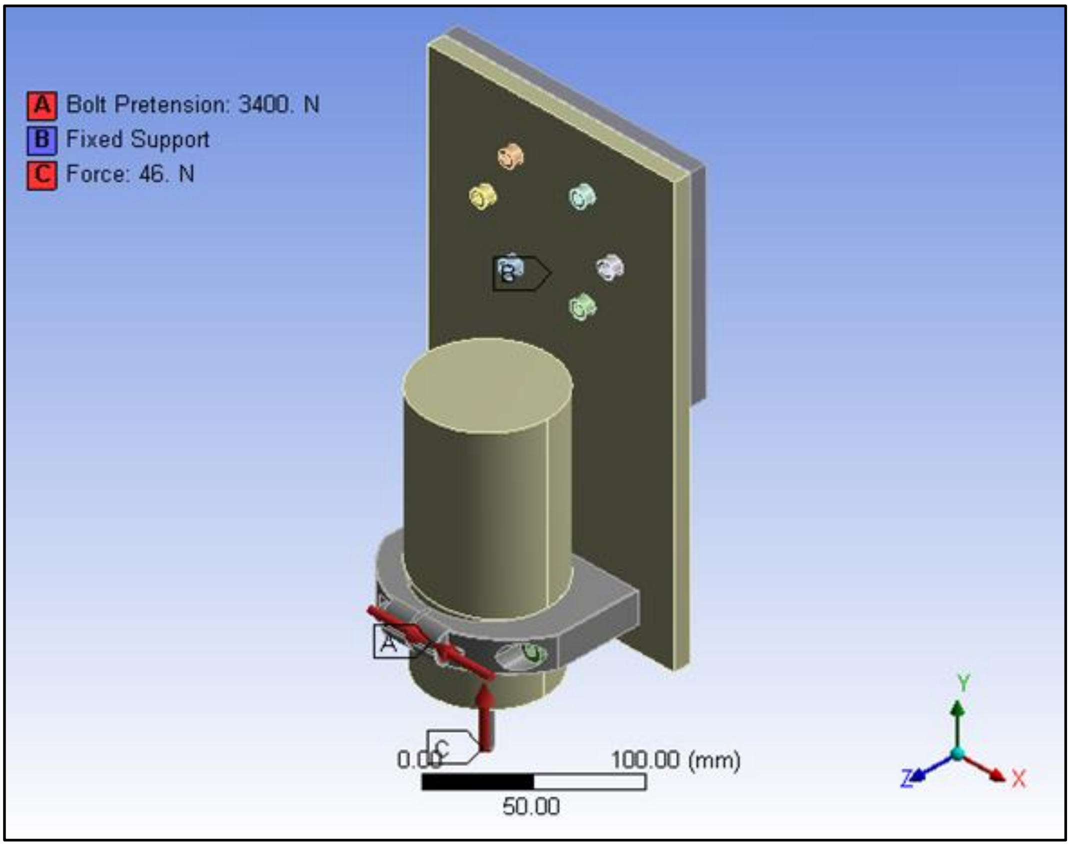

Figure 5.15: Bolt pretension and force applied in Z-direction

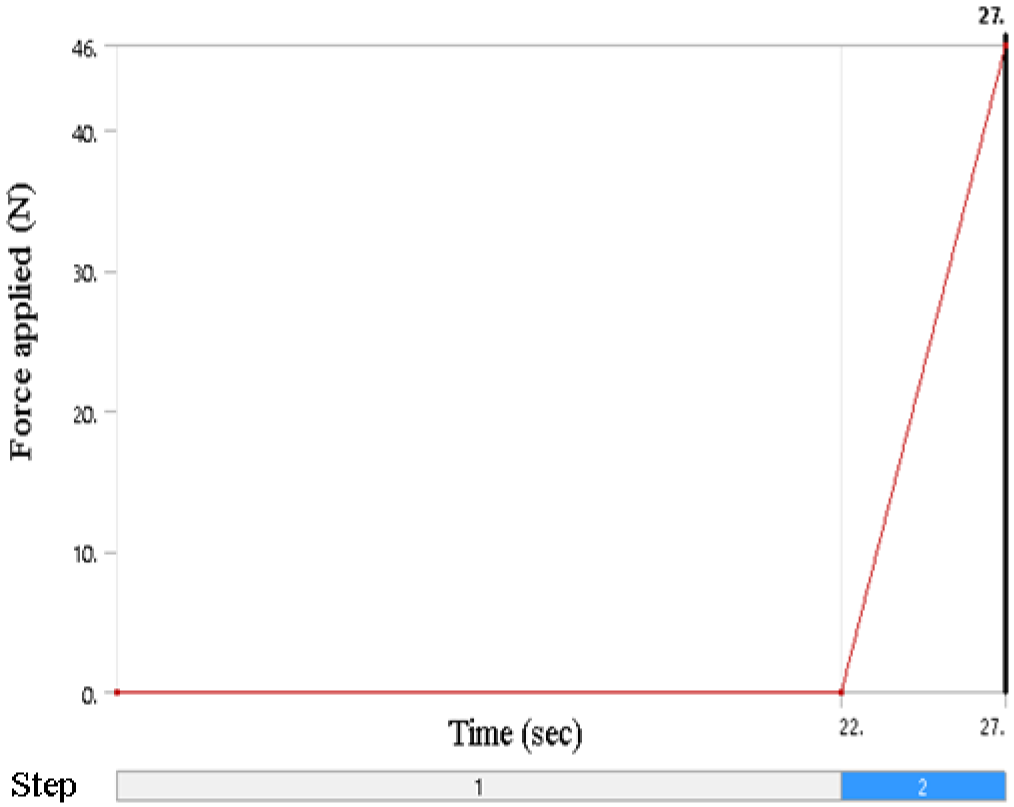

Figure 5.16: Duration of force applied on the tool tip

5.3.4 Solver

Initially when bolt pretension was kept 1000 N and force of 46N was applied on the tool tip, the router slipped right through. The grip was soft in this case. Now the gripping force was increased gradually, until the deformation and slippage were minimized. The bolt pretension finally selected after several iterations was 3400N. The solver was set in two steps, the first step was to tighten the grip around the router through the mounting bracket when bolt in front was given pretention value of 3400N. As this analysis was also done using auto time stepping the bolt pretension was gradually increased from zero to 3400N in 22 seconds. The force applied on the tool was gradually built from zero to 46 N in 5 second time that is from 22nd second to 27^{th} second. Time period from 0^{th} to 22^{nd} seconds is only to check the types of deformities on router mounting when pretention bolt is tightened in the router mounting bracket. Deformation occurred during this phase is of no importance. The time period from 22^{nd} to 27^{th} second is the time when the reaction force is applied on tool tip to check the router slippage in mounting bracket. It is important to check if deformations occurred in time period 22^{nd} to 27^{th} second are in safe limits or not.

5.3.5 Post Processor

The results for router mounting subassembly have been derived for application of forces in four different directions separately. The result obtained from post-processor are in the following two formats: (a) Equivalent (Von-Mises) stress, and (b) Deformation

5.3.5.1 Results of FEM analysis when reaction of cutting force is acting upwards along Z-axis direction

(a) Equivalent (Von-Mises) stress

As shown in figure 5.17 and figure 5.18, the maximum deformation has occurred only in router mounting bracket when a bolt pretention of 3400N is applied along with the cutting reaction force in upwards direction along Z-axis of 46N. The magnitude of the maximum stress is 202.41 MPa, whereas the safe limit for stress in aluminium is 280. Thus this design of router mounting bracket is safe under this loading condition.

(b) Deformation

Under the influence of the 3400 N gripping force the maximum deformation of 0.74342 mm has been seen in the aluminium router mounting bracket. But when the cutting reaction force of 46N is fully applied in upwards direction along Z-axis the maximum deformation observed is 0.72588 mm, as shown in figure 5.19 and figure 5.20. If considering the time interval from 22 to 27 seconds when the reaction of cutting force of 46 N is applied on the mounting bracket, it gets an overall additional deformation/deflection of -0.01754 mm which is also under safety limit.

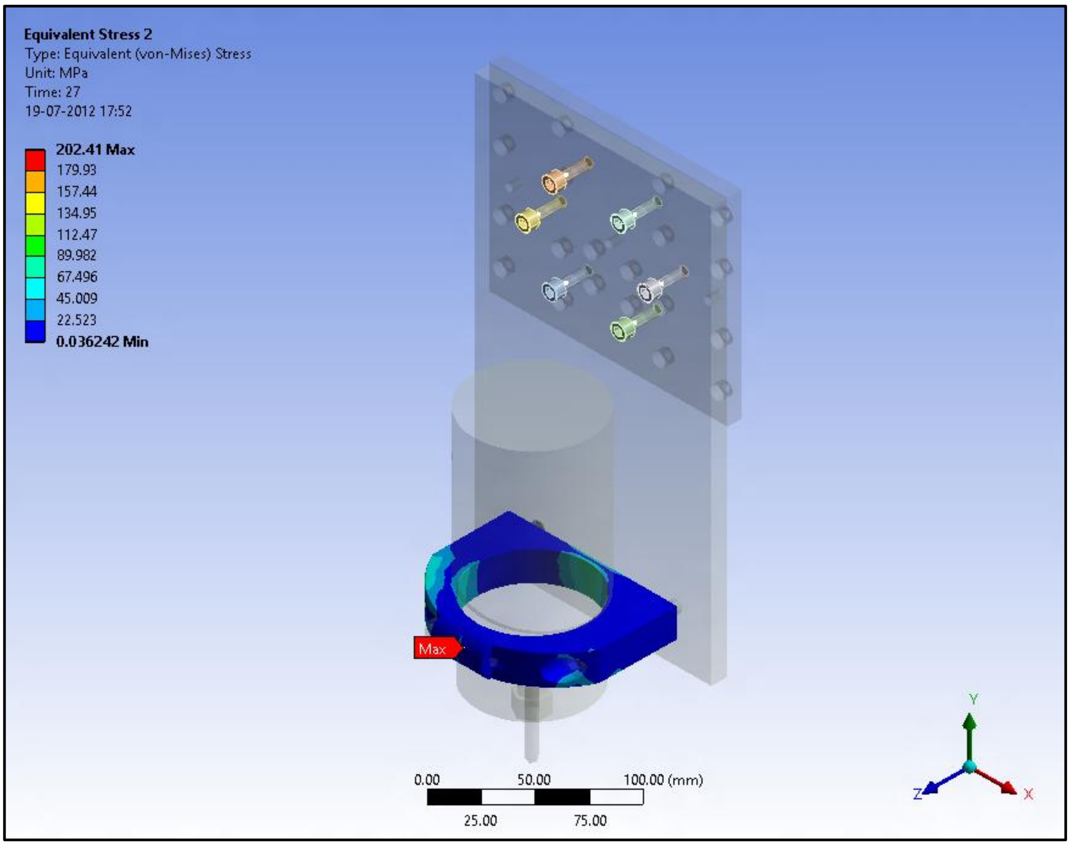

Figure 5.17: Equivalent stress on mounting bracket

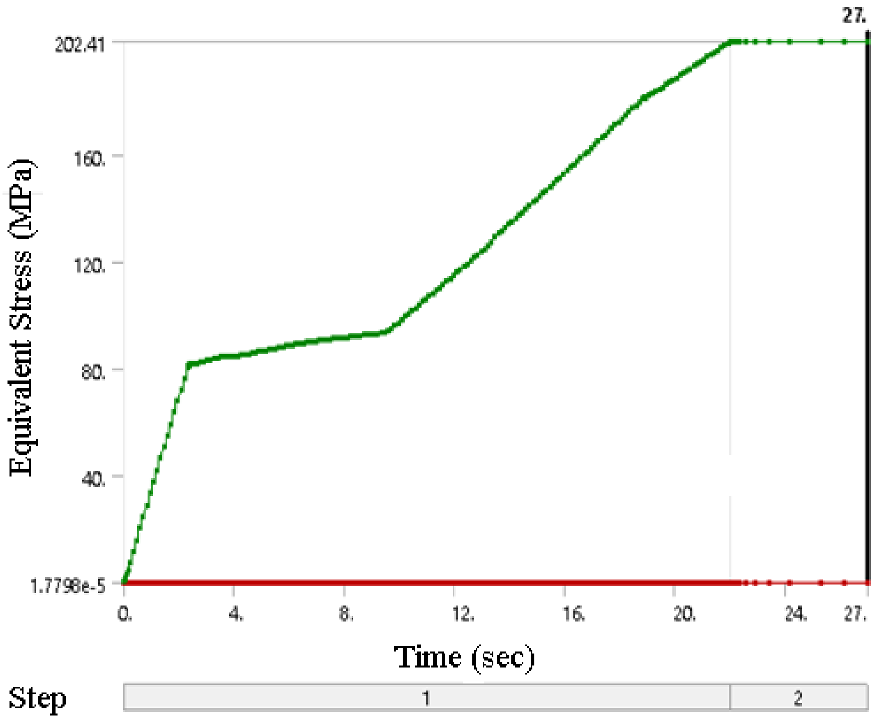

Figure 5.18: Graph of Equivalent stress on mounting bracket

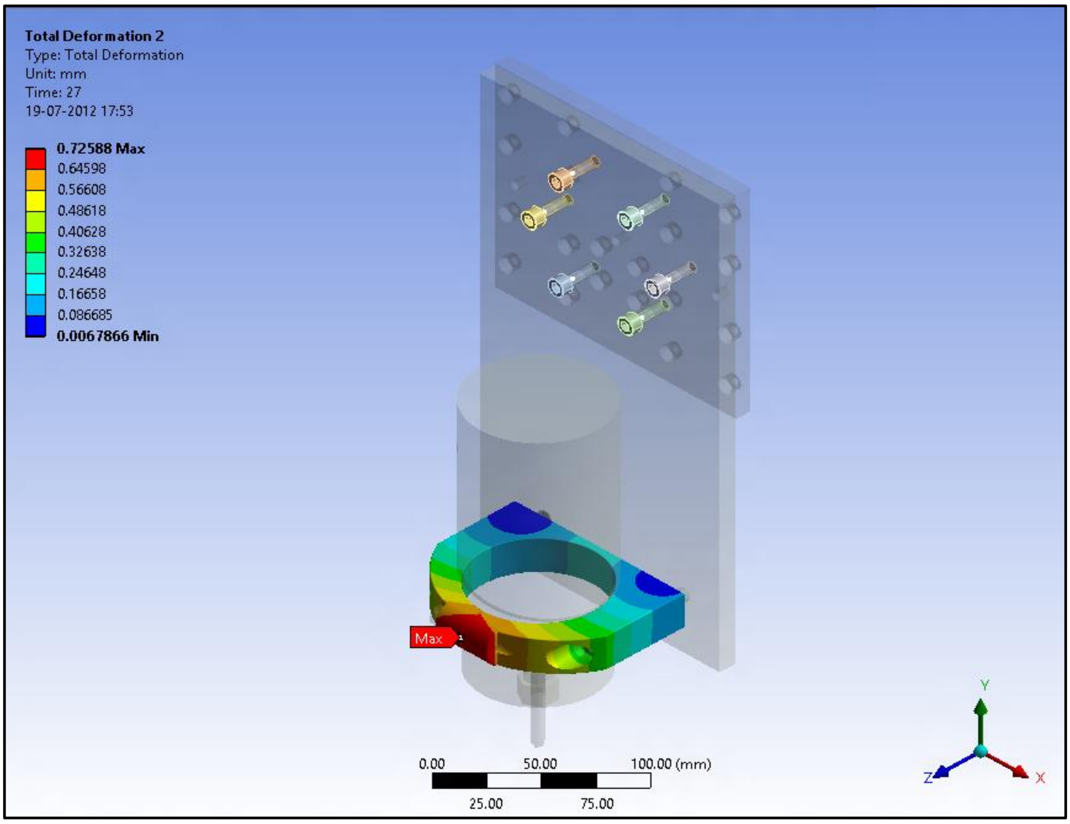

Figure 5.19: Deformation in mounting bracket

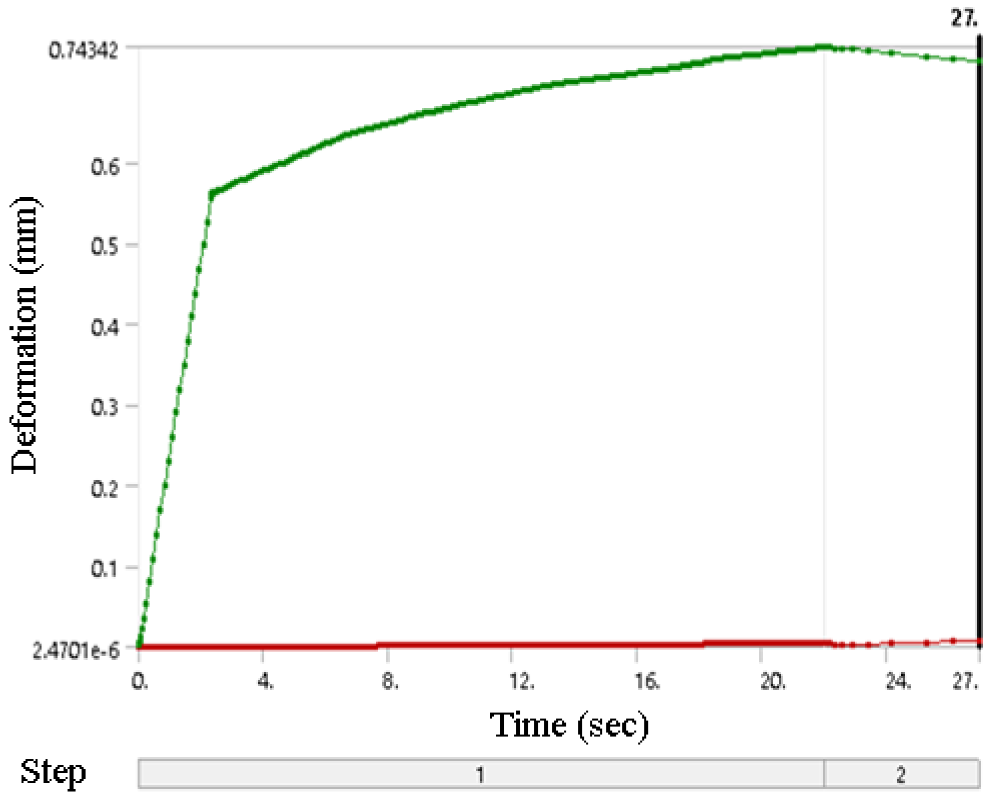

Figure 5.20: Graph for deformation in mounting bracket

(c) Slippage between router and mounting bracket in vertical direction

The maximum slip between router mouting bracket and router after 22 second when the bolt have been tightened with a pretention of 3400N is 0.083587mm and maximum slip after aplication of reaction of cutting force of 46N in upwards direction along z-direction is 0.082946 mm as shown in figure 5.21. The value of cofficient of sliding friction between two mating surfaces of aluminium has been taken as 1.35 [30]. If considering the time interval from 22 to 27 seconds when the reaction of cutting force is applied, the net slippage is 0.000641 mm in upwards direction which is also under safety limit, as compared to the deflection of 0.03mm at tool tip under the selfload condition as explained in section 5.2.

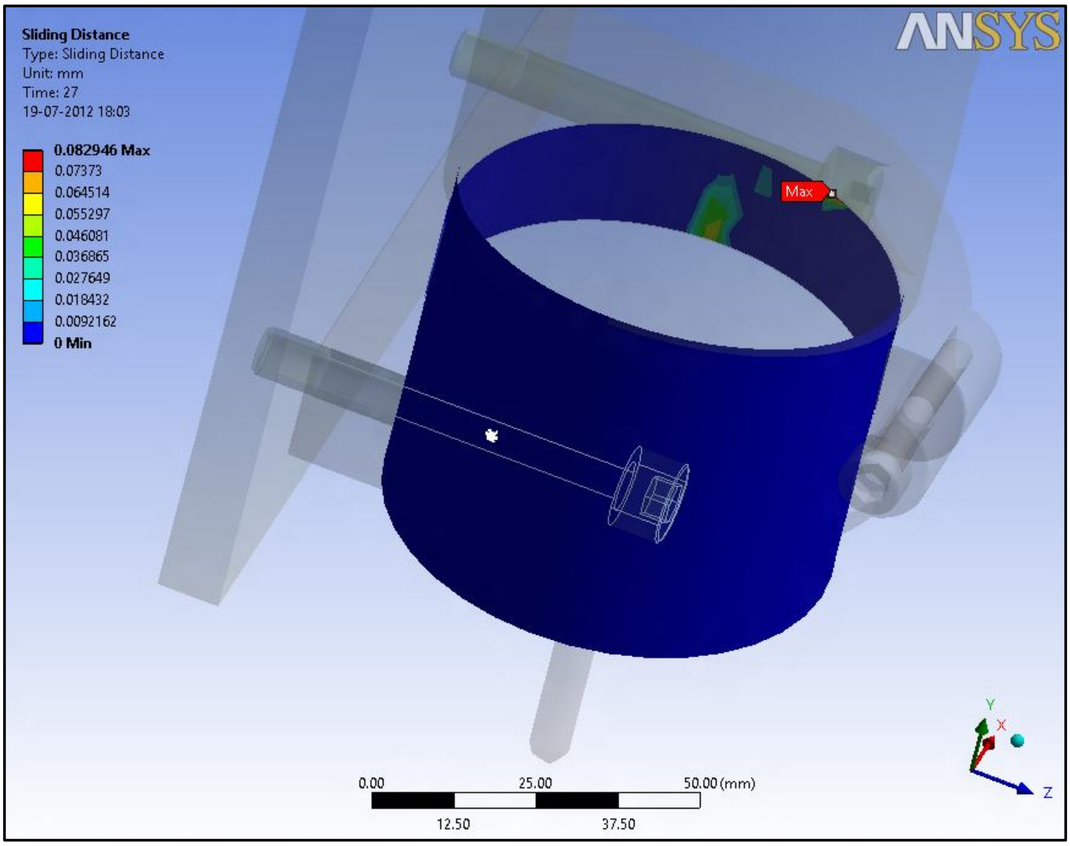

Figure 5.21: Slippage between router and bracket mounting when force in z–direction is applied

5.3.5.2 Results of FEM analysis when reaction of cutting force is applied in X-direction

(a) Deformation of router mounting assembly

The maximum deformation after applying bolt pretention of 3400N, after 22 second has been observed as 0.73343 mm. After the application of reaction of cutting force of magnitude 46N in x-direction from the duration of 22nd to 27th second the maximum deformation is observed as 0.68838 mm. If considering the time interval from 22 to 27 seconds when the cutting force is applied, the mounting bracket gives the deflection of 0.045mm, as shown in figure 5.22 and figure 5.23. With the change of material or change of thickness of aluminum plate on which router mouting is fastened, this deflection of 0.045mm can be reduced further.

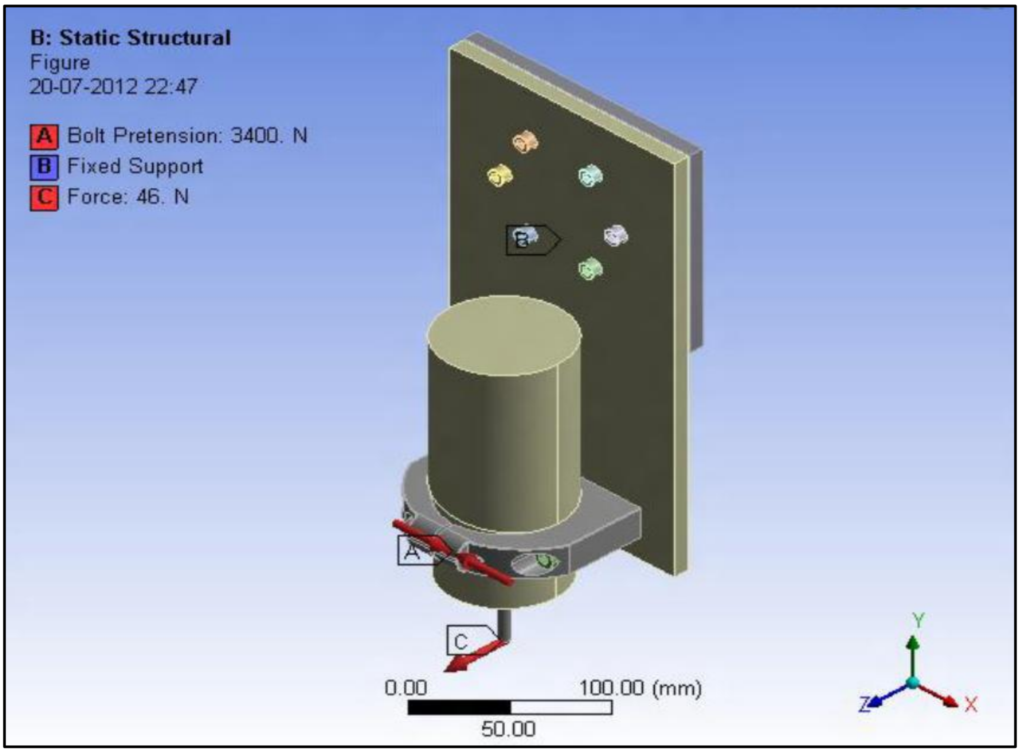

Figure 5.22: Bolt pretension and force applied in x-direction

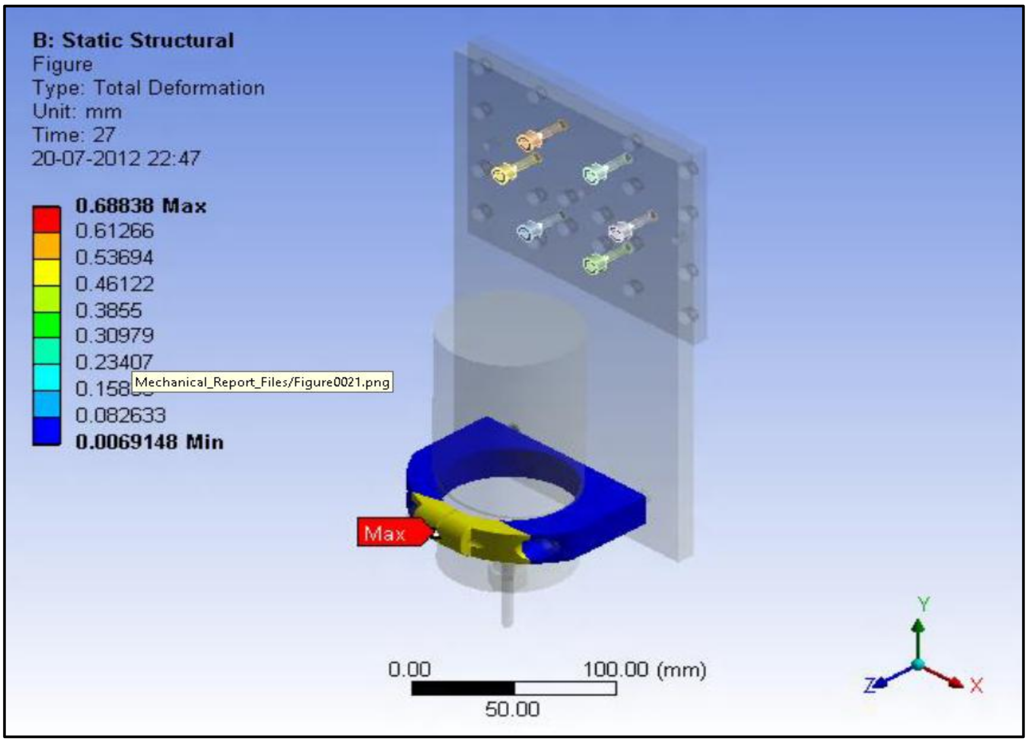

Figure 5.23: Deformation of mounting bracket when force applied in x- direction

(b) Slippage between router and mounting bracket

The maximum slip between router mouting bracket and router after 22 second when the bolt have been tightened with a pretention of 3400N is 0.078mm and maximum slip after aplication of reaction of cutting force of 46N in X-direction is 0.0897mm as shown in figure 5.24. If considering the time interval from 22 to 27 seconds when the reaction of cutting force is applied, the net slippage is 0.0117 mm in upwards direction which is also under safety limit.

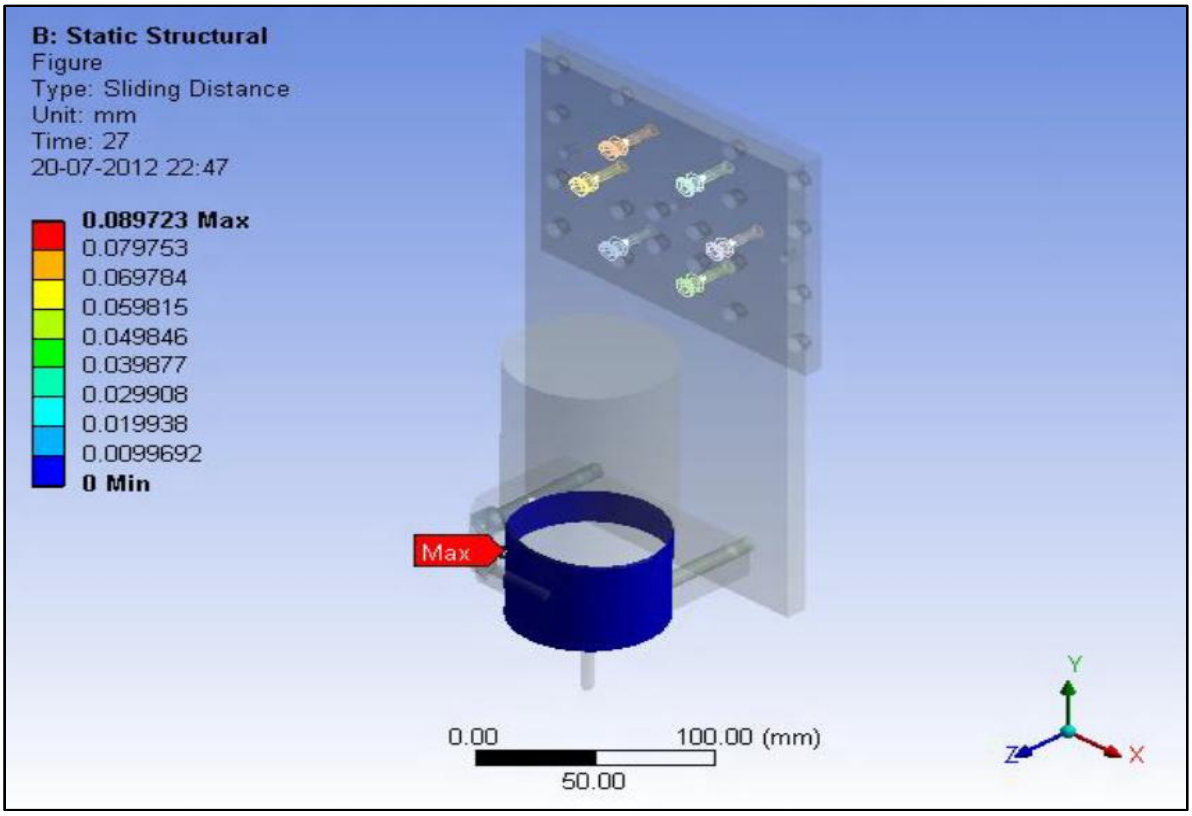

Figure 5.24: Slippage between router and bracket mounting when force in x–direction is applied

5.3.5.3 Results of FEM analysis when reaction of cutting force is applied in Y-direction

(a) Deformation of router mounting assembly

The maximum deformation after applying bolt pretention of 3400N, after 22 second has been observed as 0.74269mm. After the application of reaction of cutting force of magnitude 46N in Y-direction from the duration of 22nd to 27 second the maximum deformation is observed as 0.76532mm. If considering the time interval from 22 to 27 seconds when the cutting force is applied, the mounting bracket gives the deflection of 0.023mm, as shown in figure 5.25 and figure 5.26.

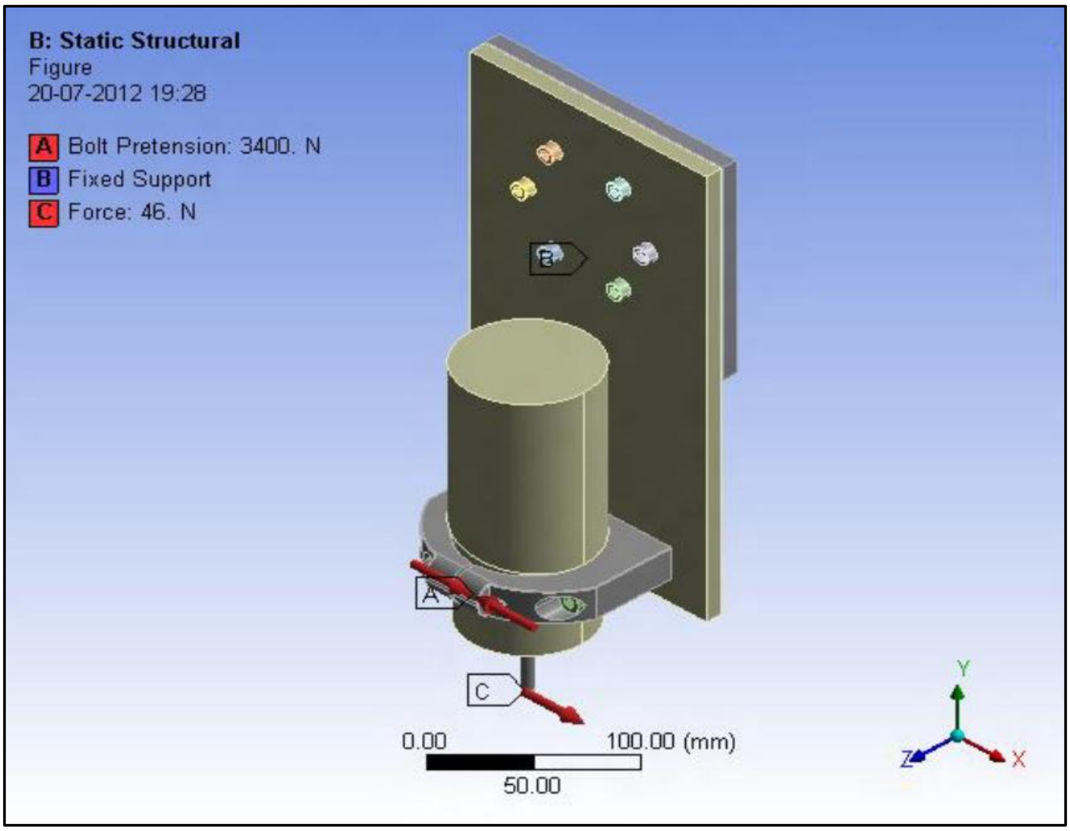

Figure 5.25: Bolt pretension and force applied in y-direction

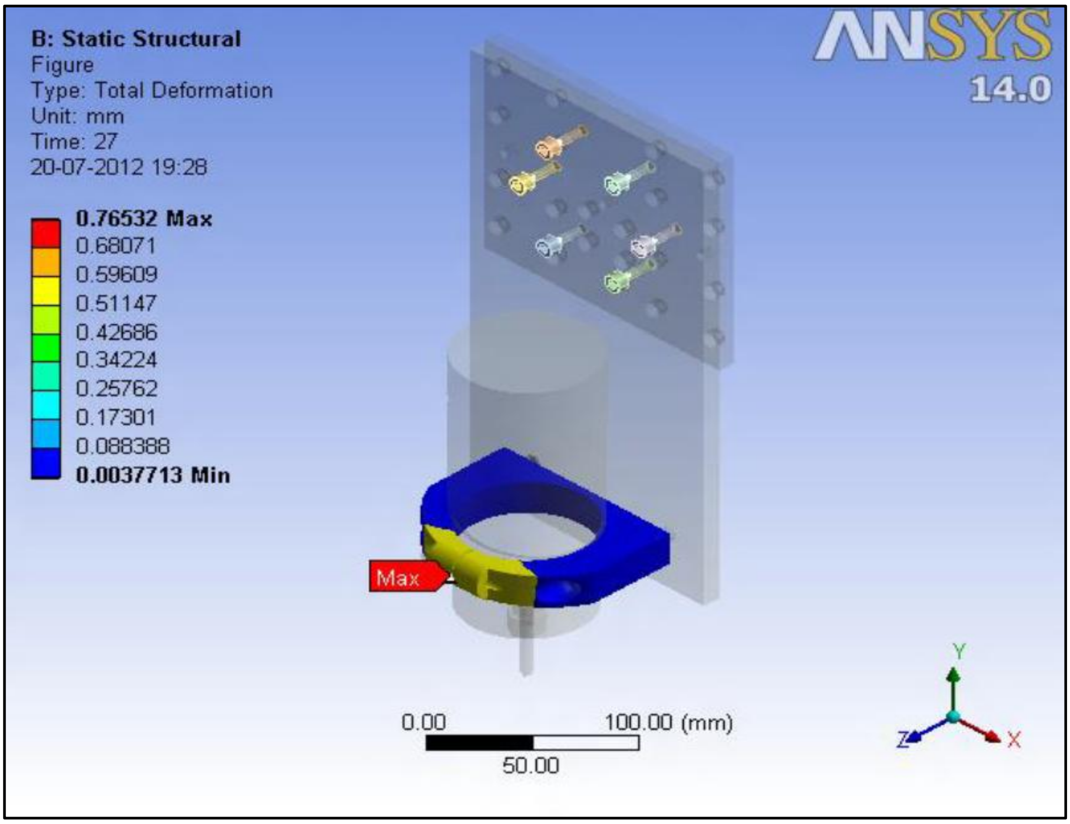

Figure 5.26: Deformation of mounting bracket when force applied in y-direction

(b) Slippage between router and mounting bracket

The maximum slip between router mouting bracket and router after 22 second when the bolt have been tightened with a pretention of 3400N is 0.0743mm and maximum slip after aplication of reaction of cutting force of 46N in Y-direction is 0.070084mm as shown in figure 5.27. If considering the time interval from 22 to 27 seconds when the reaction of cutting force is applied, the net slippage is 0.004mm in upwards direction which is under accepted limit.

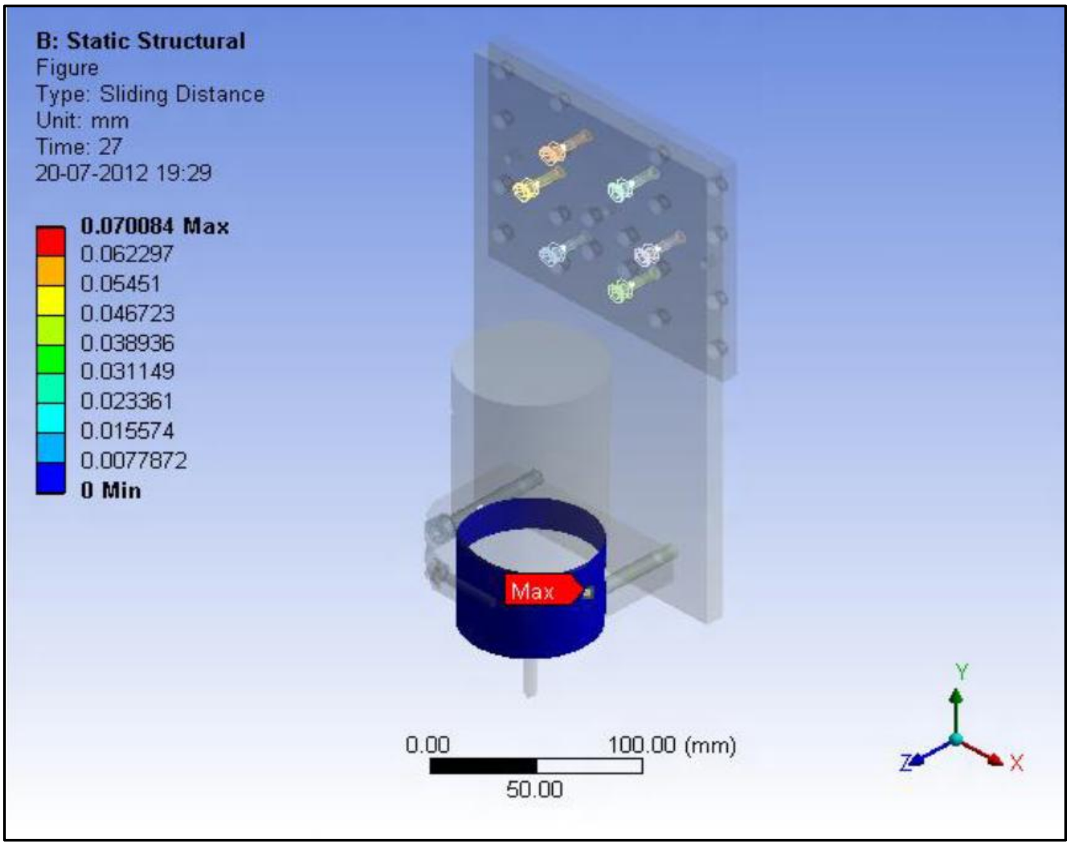

Figure 5.27: Slippage between router and bracket mounting when force is applied in both y–direction

5.3.5.4 Results of FEM analysis when reaction of cutting force is applied in X and Y-direction simultaneously

(a) Deformation of router mounting assembly

The maximum deformation after applying bolt pretention of 3400N, after 22 second has been observed as 0.74225mm. After the application of reaction of cutting force of magnitude $46/\sqrt{2}$ N in X-axis as well as Y-axis direction from the duration of 22^{nd} to 27 second the maximum deformation is observed as 0.71206mm. If considering the time interval from 22 to

27 seconds when the cutting force is applied, the mounting bracket gives the deflection of 0.03mm, as shown in figure 5.25 and figure 5.26, which is also under safety limit.

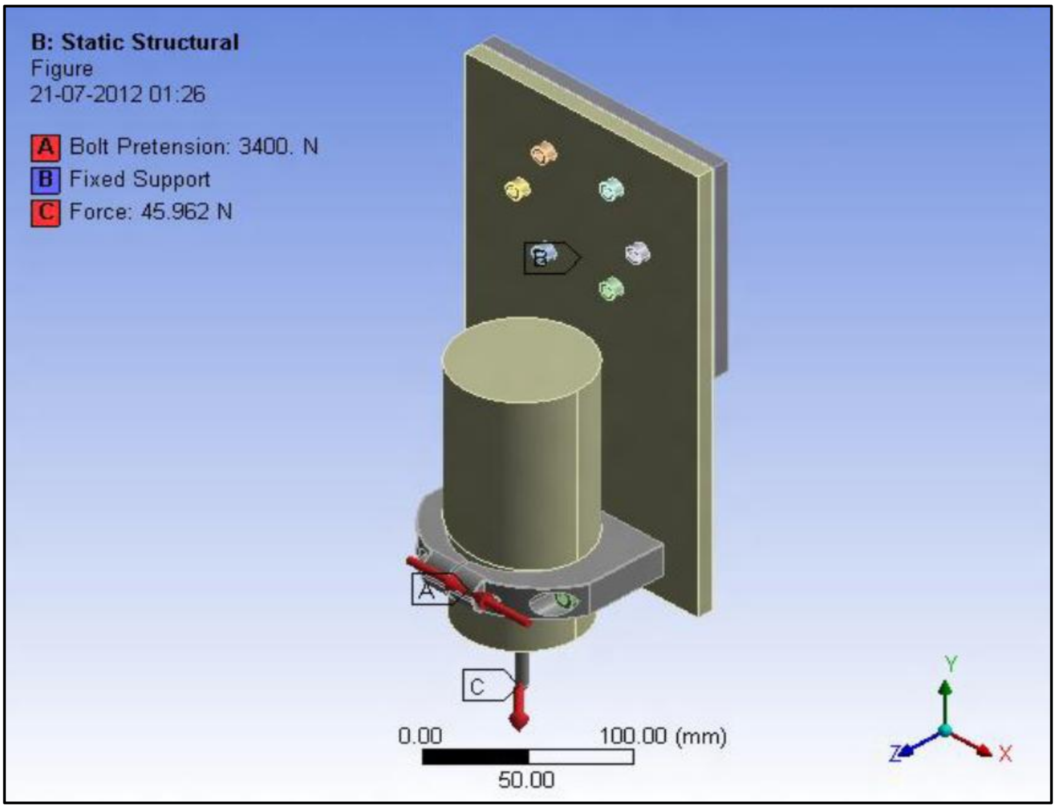

Figure 5.28: Bolt pretension and force is applied from both x and y-direction

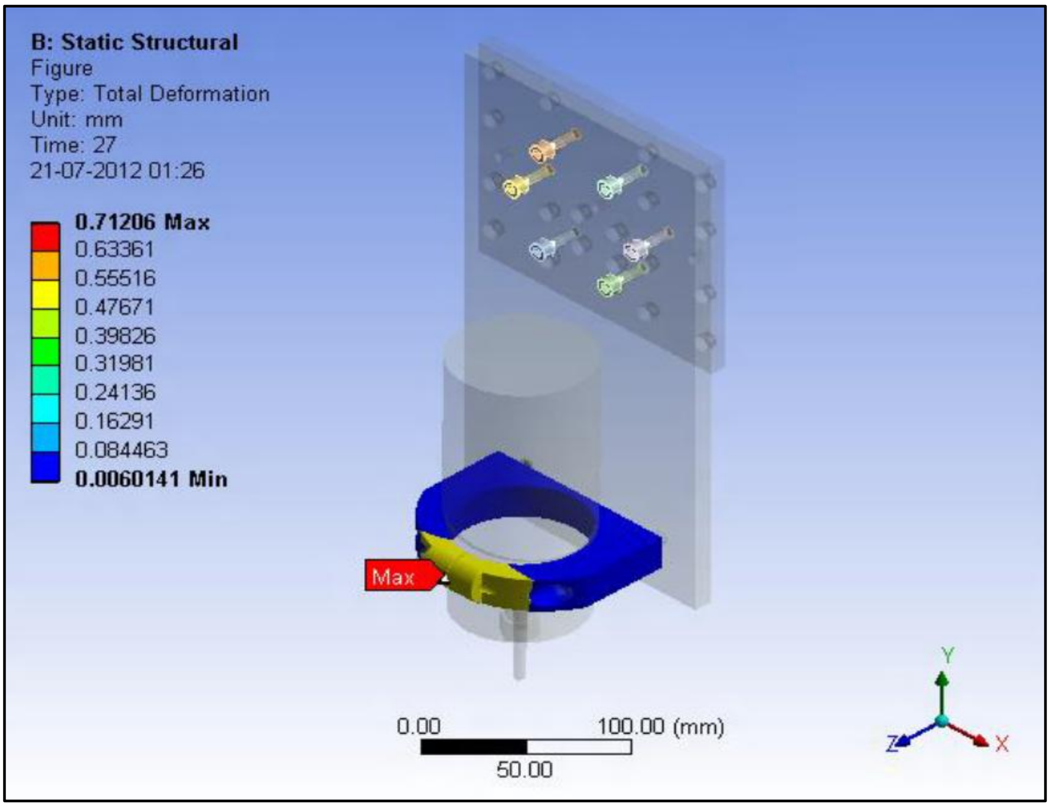

Figure 5.29: Deformation of mounting bracket when force applied in x and y- direction

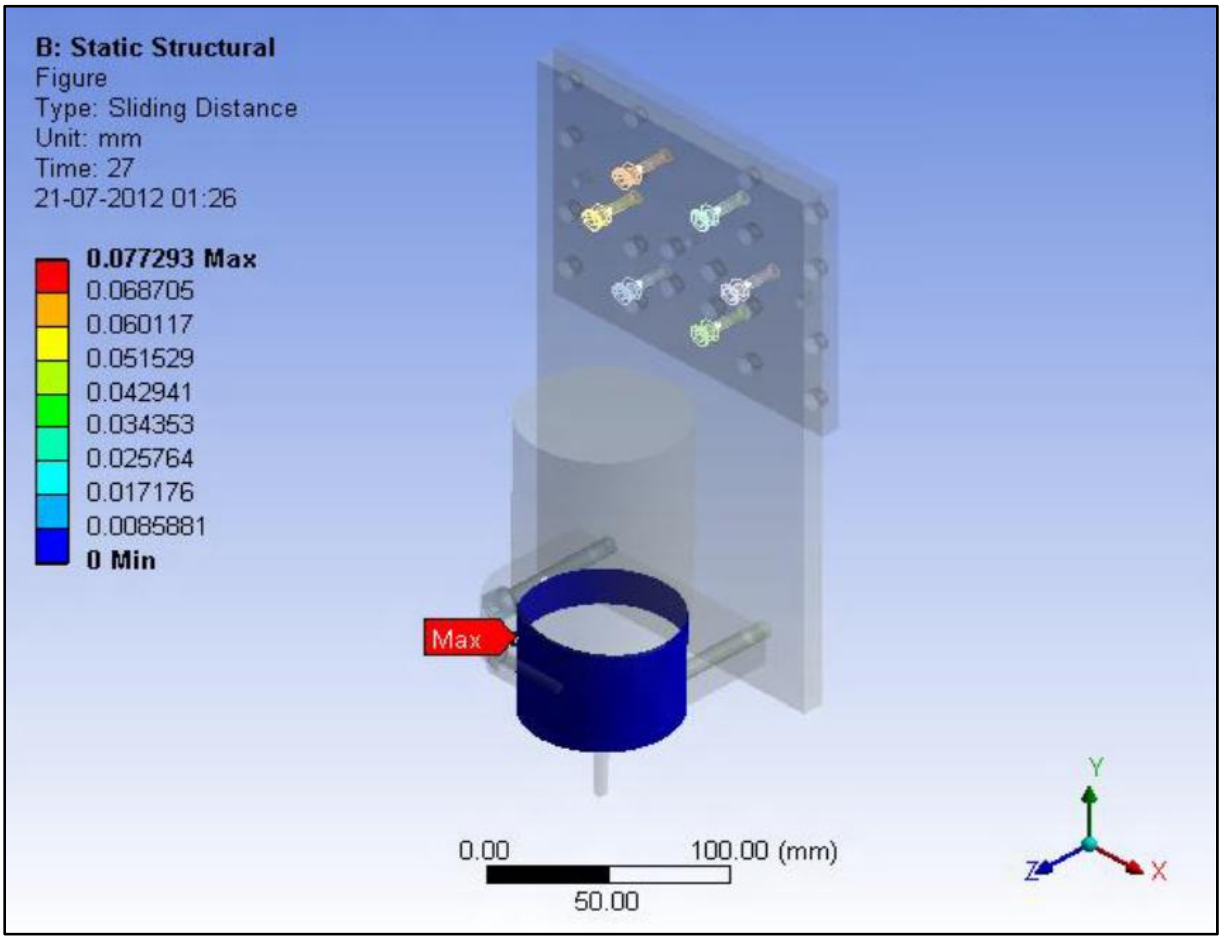

Figure 5.30: Slippage between router and bracket mounting when force is applied in both x and y–direction

(b) Slippage between router and mounting bracket

The maximum slip between router mouting bracket and router after 22 second when the bolt have been tightened with a pretention of 3400N is 0.77293mm and maximum slip after aplication of reaction of cutting force of 46N in Y-direction is 0.05967mm as shown in figure 5.30. If considering the time interval from 22 to 27 seconds when the reaction of cutting force is applied, the net slippage is 0.017627mm in upwards direction which is under accepted limit.

5.4 FABRICATION AND ASSEMBLY OF MACHINE COMPONENTS

The verification of the machine components for their mechanical behaviour as presented in the above sections gave a clear picture about the safety and suitability. Although the analysis and fabrication part was being carried out simultaneously, but it build the confidence level about the behaviour of machine components while assembly. The design was created considering the concepts of "*Design for Manufacturing*" and "*Design for assembly*". The concept of "*Design for Manufacturing*" is helpful for ease of manufacturing while "*Design for assembly*" concepts have been proved to be of great use for ease of assembly and dis-assembly. Some of the components for machine tool prototype have been machined in-house at Central workshop of Thapar University, Patiala. While manufacturing of critical pats had been sub-contacted. In addition as listed in 3rd chapter some ready-made standard components have also been purchased. These parts were assembled in the central workshop of Thapar University. The figure 5.31 shows some of unassembled parts/ subassemblies, while figure 5.32 shows the base part of the machine prototype. The figure 5.33 shows the final complete assembly of the machine prototype along with the names of some of the major subassemblies.

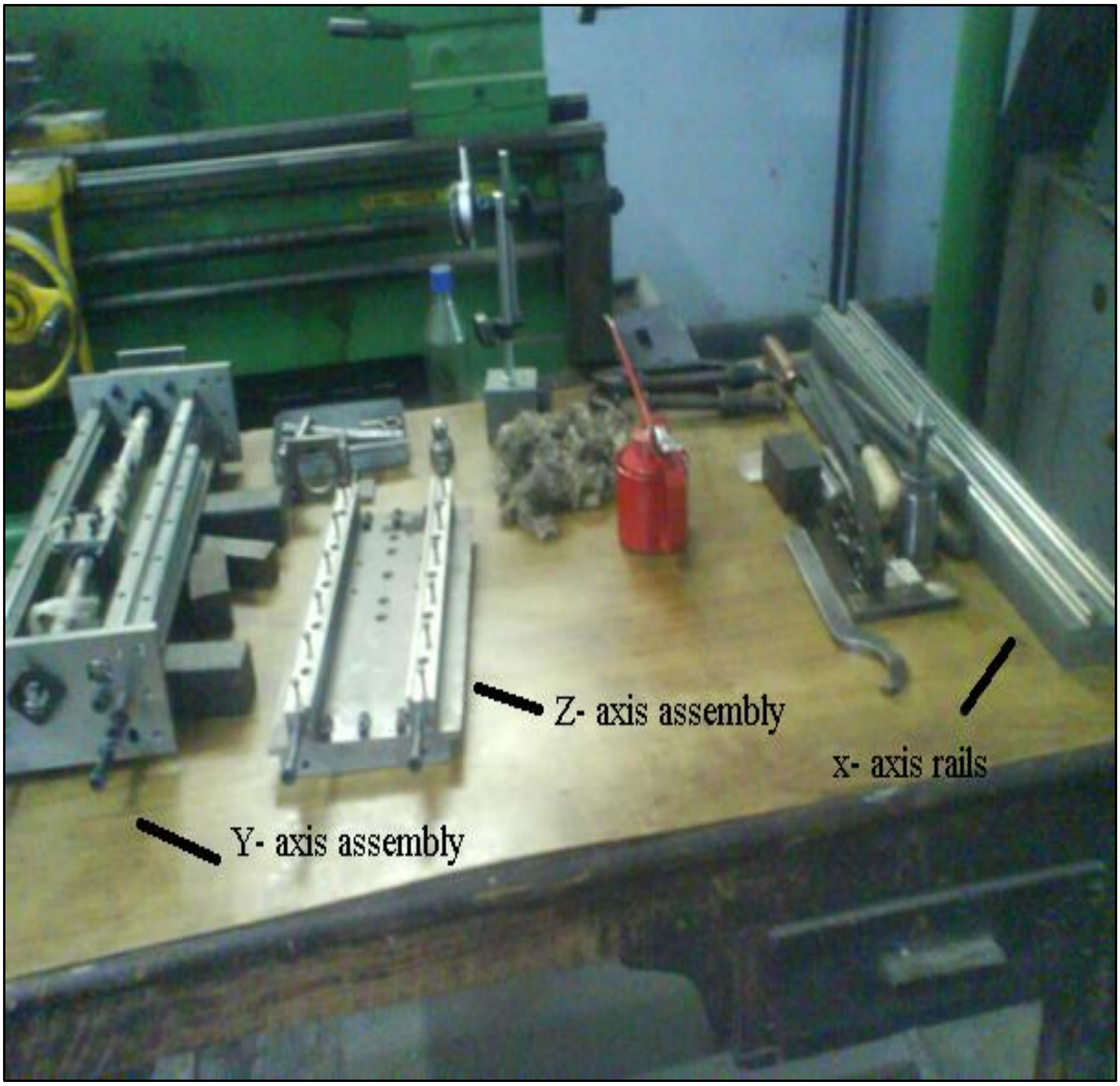

Figure 5.31: Machine components

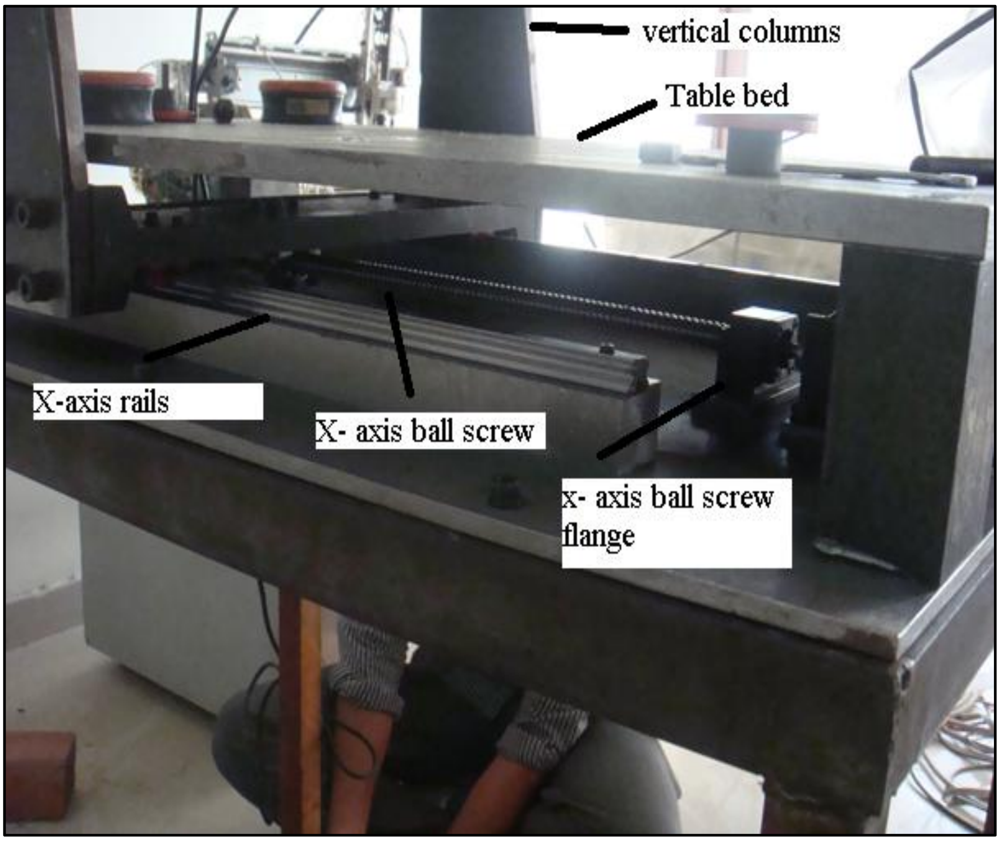

Figure 5.32: X-axis assembly on machine structure

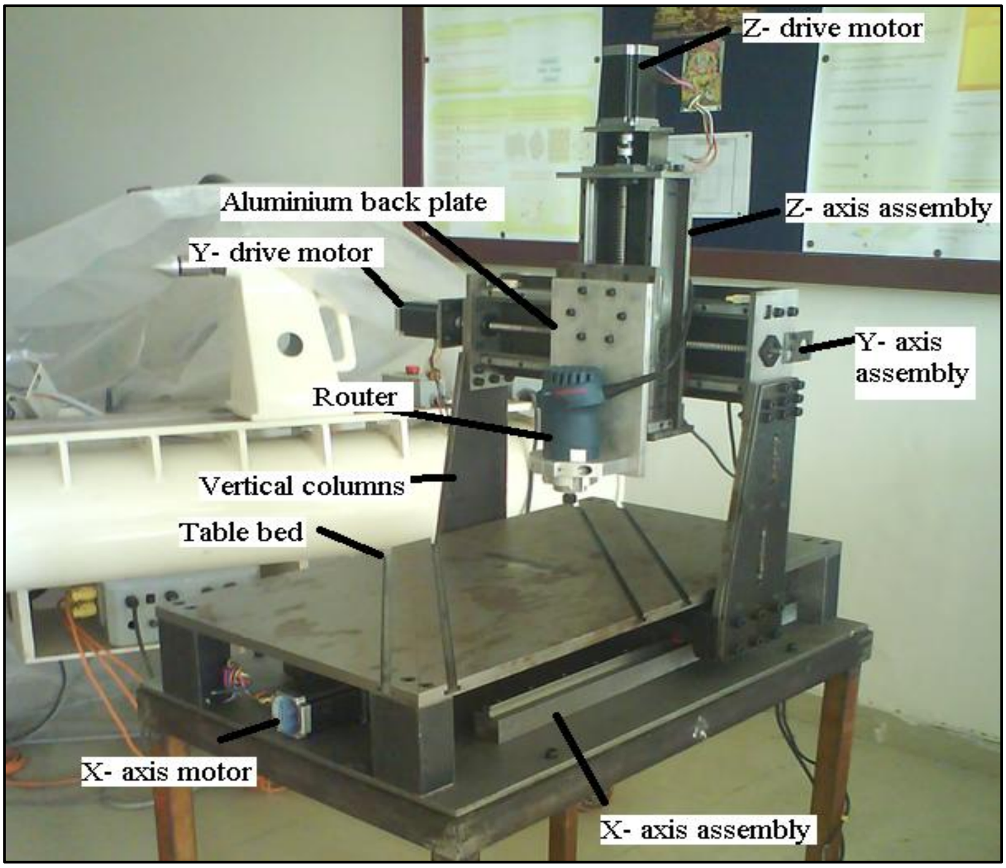

Figure 5.33: The machine structure

CHAPTER: 6

CONCLUSIONS AND FUTURE SCOPE OF WORK

6.1 Conclusion of Present Work

In the present work a detailed study for the design development and evaluation has been done. It is apparent from the results shown in the chapter 5 that the design of the machine structure is safe and suitable. One of the approaches followed for design of machine tool was modular design which has many advantages like simplification of constituent components in terms of their design, manufacturing and easy replacement after their service life. All these aspects have been elaborately discussed in the previous chapters. The conclusions drawn from the present study are as follows:

(i) The design of machine structure is safe for the standard gravity force.

(ii) The design of machine structure is safe for maximum feed rate value of 300mm/min and maximum depth of cut of 8mm with tool of 12.7mm diameter, in addition considering a factor of safety is 2.0.

(iii) The router mounting which is designed for usage with a tool of 6.35mm diameter to cut maximum depth of cut of 5mm at a feed rate of 300mm/min has also proved to be safe under required usage conditions. The design for this subassembly can be further strengthened by changing the thickness or the material of aluminium plate used for clamping of router mounting.

(iv) The maximum slippage observed in the router after application of reaction cutting force is also under safe limits.

(v) Assembly of components of the prototype was convenient as expected because there were not any significant deflections because of self-weight of machine components/ subassemblies.

6.2 Future Scope of Work

The present work has been more focused towards FEM analysis for the structural strength of the machine components. A lot of work can be carried out for the further refinements of the work carried out in this thesis work as listed below:

(i) Study of the machine structural behaviour considering the machine component made up of machine grade steels or high strength alloys, along with the trend of change in cost of material and cost of manufacturing.

(ii) Study of vibration characteristics of the machine tool.

(iii) Study of dynamic characteristics of router assembly and machine tool as a whole.

(iv) Refinement and up-gradation of conceptual machine design using sensitivity analysis can be carried out for optimisation of weight/strength ratio of the machine tool.

(v) Possibilities of extending the present design to a low cost Mill turn center can be further explored.

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