Lean Manufacturing of Lathe Machine

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Major Projet

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(Computer Integrated Manufacturing)

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This is to certify that

I. The thesis comprises my original work towards the degree of Master of Technology in Mechanical Engineering(CIM) at Nirma University and has not been submitted elsewhere for Degree.

II. Due Acknowledgment has been made in the text to all other material used.

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Abstract

This dissertation work aims to understand the role of lean manufacturing in design and optimum utilization of product. Today market demands for more efficient product at low cost. To fulfill the market demands it is necessary to improve design of product constantly and to decrease its manufacturing cost. Lean thinking is a process which emphasize on elimination of non value added services and it augurs maximum efficiency of product Research & Development. Here lean principle is implemented to design optimum utilization of conventional lathe machine components from a manufacturer's point of view.

Existing lathe bed structure has been studied for stress analysis and modal analysis. Optimization of existing lathe bed has been carried out using value stream mapping (VSM) principle and finite element tools.

The optimized design proposed by FE analysis have been studied for functional and manufacturing feasibilities. A modified design of lathe bed has been proposed with total weight reduction of 10%. An alternate material (Al-Cu alloy) has been proposed for equivalent performance. A weight reduction of 63% has been noticed with compare to the existing lathe bed made of GCI.

Thus with lean manufacturing concept it can be used to optimize the structure without impeding its functional requirements.

Keywords:Lean Manufacturing, Finite Element Analysis, Optimization

Contents

List of Figures

List of Tables

Nomenclature

P_Z	Main cutting force, N
P_X	Feed force, N
P_Y	Thrust force, N
β	Damping coefficient
P	Power, Kw
K	Material factor
z	No of cutting edges in contact
v	Cutting speed, m/min.
a	Depth of cut, mm
f	Feed, mm/rev.
P_V	Vertical forces, N
P_H	Horizontal forces, N
E	Young's moduls of Elasticity, N/mm^2
L	Length, mm
δ	Deflection, mm
D_w	Maximum diameter of workpiece, mm
L_w	Maximum length of workpiece, mm

Abbreviation

VSM	Value Stream Mapping
FEM	Finite Element Method
BEM	Boundary Element Method
FEA	Finite Element Analysis
DMU	Decision Making Unit
PDM	Product Data Management
CSM	Conventional Speed Machining
HSM	High Speed Machining
CAD	Computer Aided Design
CAM	Computer Aided Manufacturing
CAE	Computer Aided Engineering
CIM	Computer Integrated Manufacturing
TPS	Total Production System
R&D	Research and Development
GCI	Gray Cast Iron
Al-Cu	Alluminium & Copper

Chapter 1

Introduction

1.1 Introduction

How many people in the manufacturing industry can truly say that they have not heard of LEAN? Not many. Yet how many of these believe in lean, have implemented lean, are the passionate change agents who have convinced senior stake holders that lean is the way forward for their company? Less. Much Less. Lean is a revolution-it isn't just about using tools, or changing a few steps in our manufacturing processes-it's about the complete change of our businesses-how the supply chain operates, how the directors direct, how the managers manage, how employees people go about their daily work. Everything. So what is this revolution, and how is it impacting the machine tools industries? The background of lean thinking is based in the history of Japanese manufacturing techniques which have now been applied world-wide within many types of industry. The Fig. 1.1 represents Lean Manufacturing process.[1]
The bodies of contemporary machine tools are required to be rigid and efficiently damp vibration. Gray cast iron is the basic material mostly used in machine tool body construction because of its high stiffness (E = 120130GPa) and damping coefficient (β= 0.0085). Sand casting is the method for making bodies of cast iron. That technology is energy-consuming, but is well mastered and easy to automate. In

machines requiring less accuracy man uses welded bodies. The manufacturing cost is lower for small lot production (lack of costly casting models), so it is necessary to apply stress-relief annealing. Welded bodies have higher stiffness (E = 190210GPa), but lower damping co-efficient (β = 0,002) in comparison to gray cast iron ones. Assuming the same rigidity, a welded body is twice lighter than a body of cast iron. From some time now machine tools manufacturers' use mineral casting and composite material in body building. Mineral casting consists of a binding agent (methacrylate, epoxy or polyester resin) and a filler (sandstone, basalt, granite or quartz grit), and is of high damping coefficient (β= 0,020,03) and sufficient mechanical properties (E = 3050GPa). Assuming the same rigidity, the weight of a mineral casting body is similar to that of cast iron. Mineral casting is widely used for main bodies of precision machine tools (primarily lathes and grinders). Composite materials are used for moveable bodies, where particular stiffness and weight minimizing is a priority. The subject of our consideration is the stress and displacement analysis under static and dynamic load of the modern design lathe body made of gray cast iron. Bearing in mind the complexity of the geometrical shape of the lathe body, the finite element method is used as a basic tool in the analysis. The main aim is to evaluate the possibility of reducing the body weight by means of decreasing internal wall thickness also provide alternate light weight material.[2]

Structural optimization is very well known technique to improve design and product development process. It is classified in three types. They are topology Optimization, Shape optimization, & Size Optimization. In industry Topology Optimization is very well known technique to optimize weight and cost. But exposure to shape and size optimization is very less. This thesis work focuses on topology and shape optimization, various techniques for the optimization, role of software in optimization process and result comparison. Ansys is used as a versatile tool for doing optimization as well as stress and displacement analysis. The aim for this optimization is to reduce 20% weight of the component which will directly reduce material cost by leaps and bounds. The detail study of lathe machine reveals that 75% of its material cost is

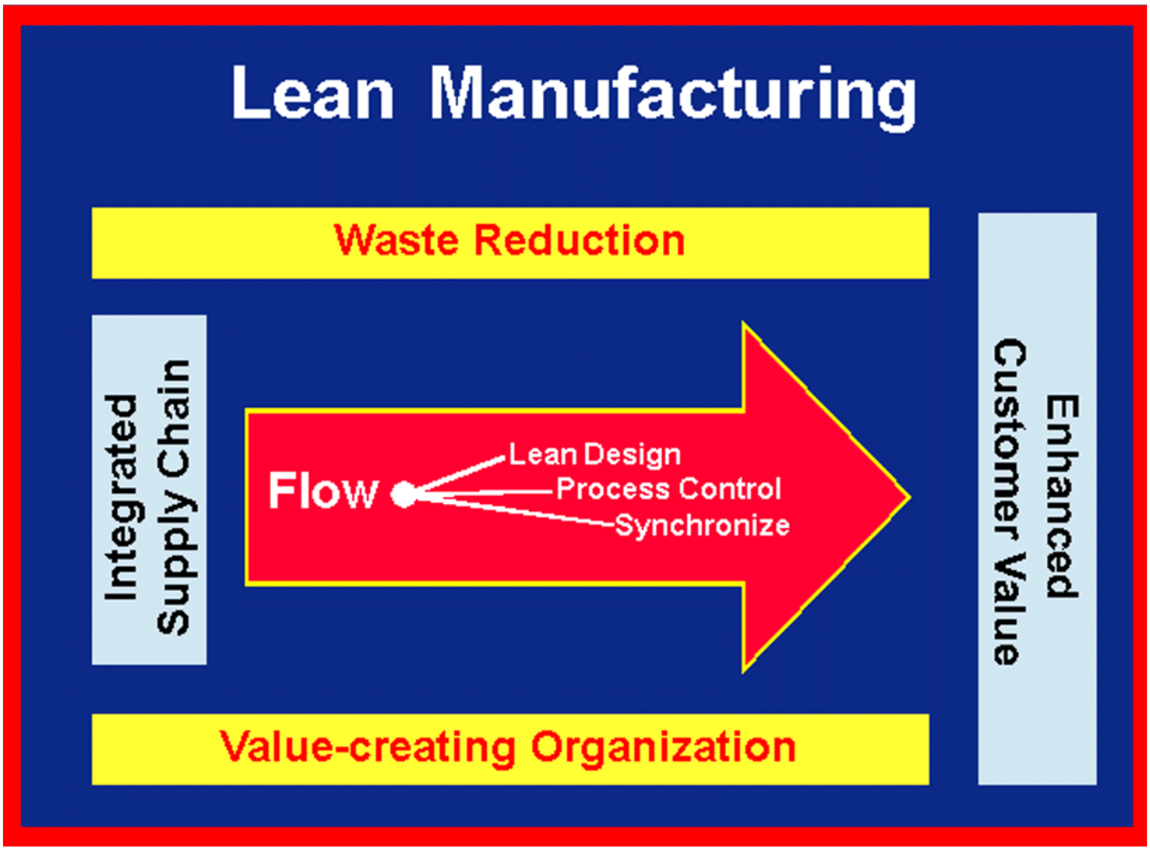

Figure 1.1: Lean Manufacturing

consumed by lathe bed and rest all other components are standard components. Lean manufacturing principle suggest to improve only those components which adds value to customer and eliminates waste from it. The standard components do not required to be lean as they required to be replaced quite often during performance of lathe machine by end users. This study leads to conclusion that lathe bed is vital and essential component of lathe machine and it requires to be lean designed.

1.2 Objectives Of Present Work

- Study lathe machine components from lean manufacturing prospectives.

- Analytical calculation of cutting forces acting on lathe machine.
- Static & Transient Dynamic Stress and Displacement study of lathe bed using Finite Element Analysis.
- Comparison and study between Grey cast iron and Al-Cu alloy as casting material for lathe bed.
- Topology optimization of lathe bed.

1.3 Methodology

1.3.1 Study lathe machine components from lean manufacturing prospectives

a. Identify number of components in a lathe machine.

b. Segregation of lathe components into standard and non standard components.

c. Elimination of standard components from lean principle.

d. Use non standard components into consideration for lean principle.

1.3.2 Analytical calculation of cutting forces acting on Lathe machine

a. Identify the sources and pattern of the forces that develops in lathe machine during machining.

b. State the effects of the forces in lathe machine and its operations.

c. List various conditions of turning process parameters and evaluate the maximum force.

d. Visualize and represent the forces originated and distributed in lathe machine.

1.3.3 Value Stream Mapping (VSM)

- Carry out a Value stream mapping for a center lathe bed.
  - Current state.
    - Detail study of turning operation in a light weight center lathe bed. Elimination of components such as headstock, tailstock and carriage as they are assembled from standard components. Study the effect of maximum forces acting on lathe bed and obtain static structural and transient dynamic results under loading condition.
  - Future state.
    - Proposed new modified & optimized lathe bed model for the same forces developed in existing conditions. Compare grey cast iron & Ai-Cu composite as a casting material for lathe bed. Justify your selection.

1.3.4 Modeling & analysis of lathe bed using Ansys & Solid Works

- Create a light weight center lathe BED solid model using CAD software Solid Works. Consider lathe manual for assistance and measure physical dimensions where ever necessary to create parts.
- Define material properties to lathe bed component.
- Carry out static structural and transient dynamic FEM analysis using FEA software Ansys.

Chapter 2

Literature Review

2.1 Introuction

Traditional manufacturing systems are built on the principle of economies of scale. Here, the large fixed costs of production are depreciation-intensive because of huge capital investments made in high-volume operations. These fixed costs are spread over large production batch sizes in an effort to minimize the total unit costs of owning and operating the manufacturing system. Large work-in-process inventories are also characteristic of traditional manufacturing. The resultant "batch and queue" operation produces large numbers of a particular product and then shifts sequentially to other mass-produced products. As an alternative to batch-and-queue, high-volume, and inflexible operations, the principles of the Toyota Production System (TPS) have been widely adopted in recent years throughout the US.[3]
Application of TPS principles have led to lean manufacturing (also called lean production, or lean thinking) in which production and assembly cells consisting of product focused resources (workers, machines, floor space, etc.) are closely linked in terms of their throughput times and inventory control. These cells are typically U-shaped or rectangular and lend themselves to,

a. Smooth (balanced) work flow across a wide variety of products.

b. Elimination of waste.

c. High quality output.

d. Flexible operation.

e. Low total unit production costs.

Economic benefits attributable to lean manufacturing include reduced lead-time and higher throughput, smaller floor space requirements, and lower work in process. In factories using lean manufacturing, large machines characteristic of batch-and-queue processes (typically referred to as "monuments") are often no longer aligned with lean work cells and are not needed or desired. Instead, smaller more flexible machines are typically organized into work cells devoted to the production of a family of products.

Workers then operate the machines in the cell to minimize the cycle time for a family of products, minimize inventory, and maximize quality. In existing factories, eliminating monuments and investing in new, smaller machines can be troublesome to managers who were responsible for originally approving a high-volume batch-and-queue manufacturing process. Scrapping a massive piece of equipment, which still has a sizeable book value, can be viewed as admitting that a mistake was made years ago by investing in manufacturing technology that quickly became obsolete. Therefore, the decision to abandon (or replace) high-volume monolithic machines in favor of cellular manufacturing systems that employ TPS and lean manufacturing principles can be extremely difficult for managers to make, fraught with subjective factors beyond economics.

The benefits seen within non-process industries (see Fig. 2.1), such as the automotive industry, are well documented:

- Decreased lead times for customers.
- Reduced inventories for manufacturers.

Figure 2.1: The Benefits of LEAN[1]

- Improved knowledge management.
- More robust processes (as measured by less errors and therefore less rework).

This makes lean a very real and physical concept especially for manufacturing. Lean production has now expanded and lean thinking has been applied to all aspects of the supply chain. There are many well documented examples of the application of 'lean thinking' to business processes such as project management (Melton, 2003); construction, design, and so on. Lean can be applied to all aspects of the supply chain and should be if the maximum benefits within the organization are to be sustainingly realized. The two biggest problems with the application of lean to business processes are the perceived lack of tangible benefits and the view that many business processes are already efficient. Both assumptions can be challenged (Melton, 2004).
There are many tangible benefits associated with lean business processes. A lean business process will be faster, e.g. the speed of response to a request for the business process will be faster, and as most business processes are linked to organizational supply chains, then this can deliver significant financial benefits to a company. The perception that a business process is already efficient is all too often an illusion. Func-

tionally, many business processes may appear very efficient, however the application of Lean Thinking forces us to review the whole supply chain in which the business process sits, and this frequently reveals bottlenecks and pockets of inefficiency. But for now let us return to the world of manufacturing within the process industries.[1] The external loads acting on the lathe body are the basic forces needed in stress state analysis and strength evaluation of the considered machine tool set. The main external static loads are cutting forces developed in extreme conditions of cutting processes and the gravity forces of the lathe body and the other lathe sub-sets as headstock, slide and tailstock. Dynamic forces resulting from inertia forces developed during machine slide start up and stop in the direction parallel to slide ways were also considered. Since cutting forces and inertia forces of the slide are transmitted to the bed indirectly by the headstock, slide and tailstock, appropriate sets that simulate the machine sub-sets with a quite high rigidity were elaborated, in order to determine the inter-reactions between the sub-sets and the lathe body. The working forces acting on the lathe sub-sets, see Fig. 2.2, were determined for the case of rough turning.[2]

The forces acting on the lathe body were determined according to the diagrams shown in Fig.2.2 and 2.3. In order to determine the reactions between the slide and lathe bed the slide set was modeled with bar elements having a high rigidity and loaded by cutting forces and their masses. The support reactions acting on the lathe bed are determined within the regions where the load is transmitted from slide to bed at four support points, i.e. at blocks and at the axis of the ball screw. It was assumed that the blocks will transmit the forces along axes X and Y (according to the presented coordinate system). However, the ball-screw will transmit the force along the Z axis.[2]

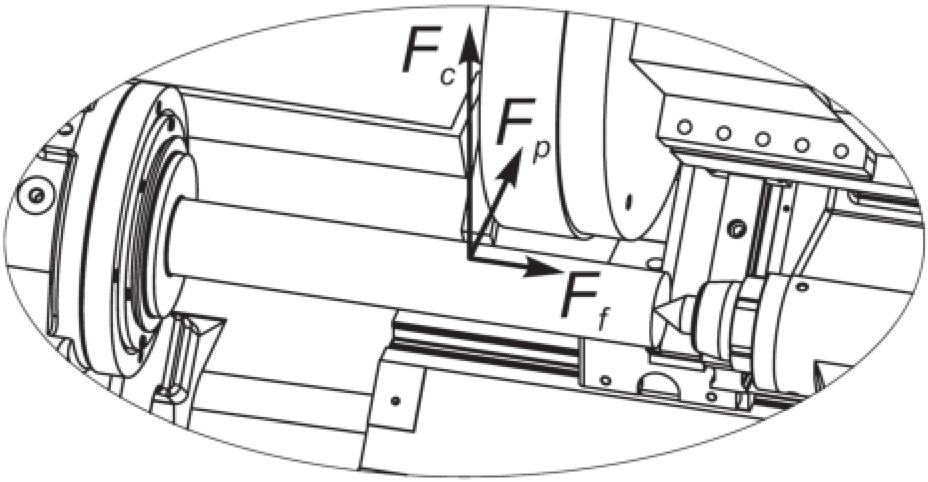

Figure 2.2: Cutting Forces[2]

A similar approach was applied to determine the load coming from headstock and tailstock. Two bar systems were modeled. The first model was loaded by the cutting force and the masses of specific elements see Fig. 2.4b. The second model, shown in Fig. 2.4c, was loaded by the force coming from the contact between the tailstock and the machined element. The support reactions were determined at the fixing region between the headstock and the body at 4 points of the tailstock support, and along the cylinder driving tailstock. The load acting on the lathe body was the sum of the determined reactions.[2]

2.2 Value Stream Mapping (VSM)

A value stream is a collection of all actions (value-added as well as non-value-added) that are required to bring a product (or a group of products that use the same resources) through the main flows, starting with raw material and ending with the

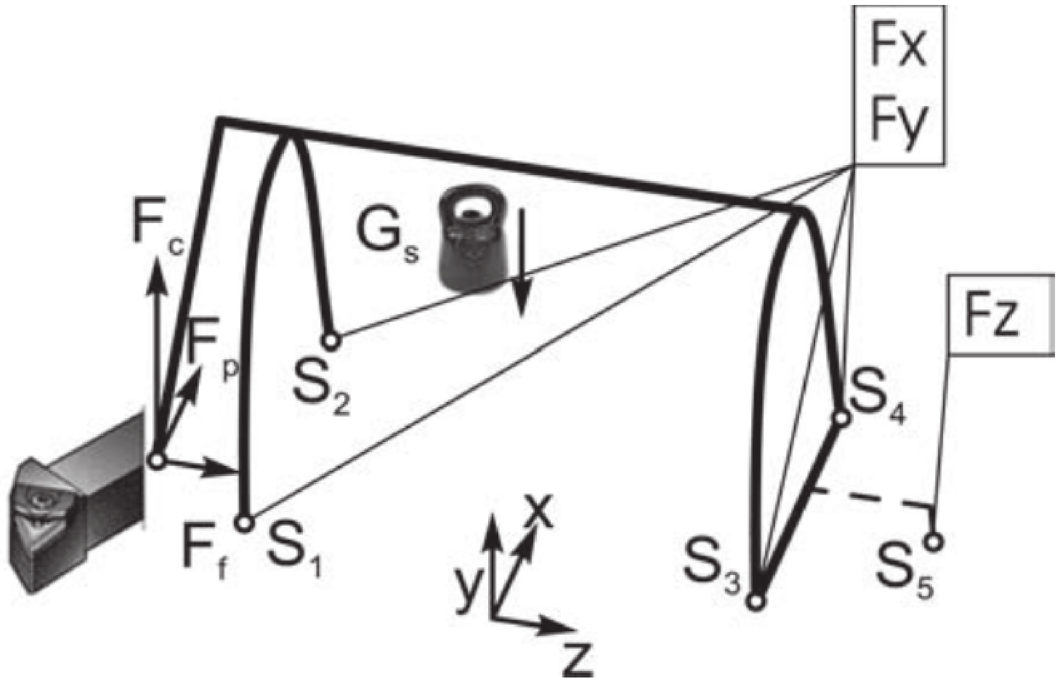

Figure 2.3: A diagram to determine reaction forces and moments of the slide[2]

customer. These actions consider the flow of both information and materials within the overall supply chain. The ultimate goal of VSM is to identify all types of waste in the value stream and to take steps to try and eliminate these. While researchers have developed a number of tools to optimize individual operations within a supply chain, most of these tools fall short in linking and visualizing the nature of the material and information flow throughout the companys entire supply chain. Taking the value stream viewpoint means working on the big picture and not individual processes. VSM creates a common basis for the production process, thus facilitating more thoughtful decisions to improve the value stream.[4]

VSM is a pencil and paper tool, which is created using a predefined set of standardized icons. The first step is to choose a particular product or product family as the target for improvement. The next step is to draw a current state map that is essentially a

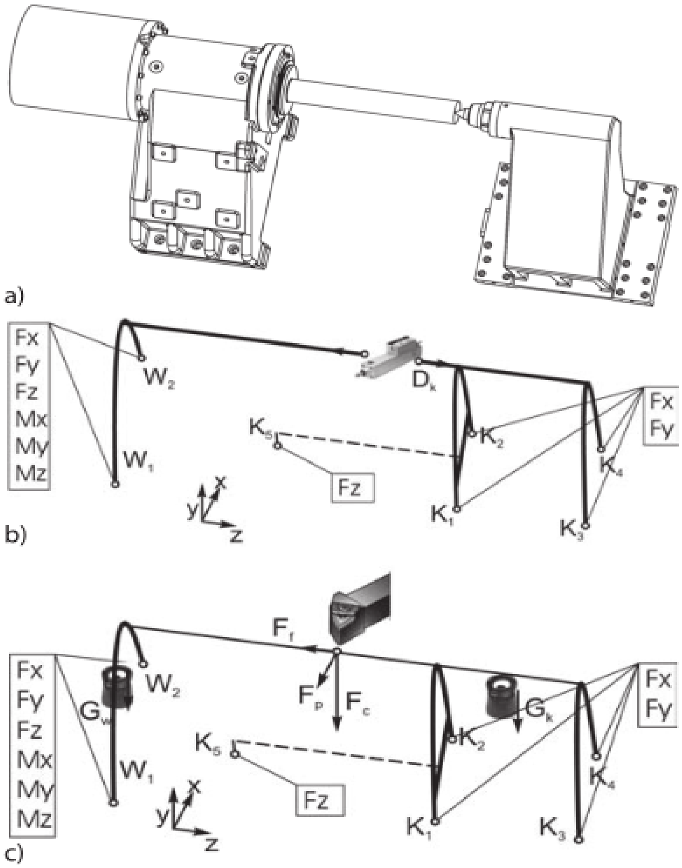

Figure 2.4: A diagram to determine headstock and tailstock force and moment reactions[2]

snapshot capturing how things are currently being done. This is accomplished while walking along the actual process, and provides one with a basis for analyzing the system and identifying its weaknesses. The third step in VSM is to create the future state map, which is a picture of how the system should look after the inefficiencies in it have been removed. Creating a future state map is done by answering a set of questions on issues related to efficiency, and on technical implementation related to the use of lean tools. This map then becomes the basis for making the necessary changes to the system.[4]

2.2.1 Simulation in support of VSM

For organizations that have long relied on traditional approaches to their manufacturing systems, it is often difficult to gain from management the commitment required to implement lean manufacturing. Doing so is hard because of differences in a number of aspects including raw material procurement, inventory management, employee management and production control. For traditional manufacturers, the reluctance to implement many lean ideas arises because their distinctive requirements often make it hard to predict the magnitude of the gains that can be achieved by implementing these. As a result, management decisions on implementing lean manufacturing often come down to their belief in lean manufacturing, reported results of others who have implemented lean techniques, and heuristic rules of thumb on the expected payback. For many managers this is insufficient justification, and lacks the quantifiable evidence needed to convince them to adopt lean.[4]

While in some situations the future state map can be evaluated with relatively modest effort, it is not as easy to do so in many others. For example, predicting inventory levels throughout the production process is usually impossible with only a future state map, because with a static model one can not observe how inventory levels will vary for different scenarios. In general, we need a complementary tool with VSM that can quantify the gains during the early planning and assessment stages. An obvious tool is simulation, which is capable of generating resource requirements and performance statistics whilst remaining flexible to specific organizational details. It can be used to handle uncertainty and create dynamic views of inventory levels, lead-times, and machine utilization for different future state maps. This enables the quantification of payback derived from using the principles of lean manufacturing, and the impact of the latter on the total system. The information provided by the simulation can enable management to compare the expected performance of the lean system relative to that of the existing system it is designed to replace, and assuming that this is significantly superior, it provides a convincing basis for the adoption of lean.[4]

2.3 Optimization

In the tough international competition, companies can only survive if, besides highly innovative power they can provide strongly cost optimized products. Therefore in new procedures like the Simultaneous Engineering, the calculation engineer is already integrated into the concept phase of the product development process. Efficient methods of working require powerful optimization algorithms to be provided in addition to the discrete methods (FEM/BEM) proved worthwhile to support the calculation engineer in the draft and design phase.
In recent years many optimization approaches have been integrated into commercial FE programs. In industrial applications only few optimization methods are established partially. The problems to be solved in industry have to be abstracted dramatically. The further development of the optimization methods regarding application, integration and numeric is necessary and pushed ahead intensively.[5]

2.4 Types of Optimization

- Topology Optimization
- Shape Optimization
- Size Optimization

2.4.1 Topology Optimization

Both for sizing and shape optimization a first design proposal, which is used as the start design, exists. The objective of general structural optimization methods is to compute even this first design proposal. Therefore an area (2D or 3D) with a homogeneous material distribution is used. Subsequently the functionally required boundary conditions are applied. The efforts for the modeling and preparation are extremely low. The optimum structural shape with the appropriate topology is issued as design proposal. The originally homogeneous material distribution becomes highly inhomogeneous. Areas arise with no mass at all or areas which contain high density mass. Compared with the sizing and shape optimization the numerical efforts strongly increase. The number of design variables is typically between 5.0 and 100.0. Therefore large efforts have to be put into the sensitivity analysis up till now. So far no method can be considered as a standard for calculating the optimum Topology due to the above mentioned difficulties regarding the mathematical approaches. Essentially because of the expensive calculation efforts these approaches can handle only extremely simplified models. Commercially only few programs are available and many FE developers work in this field. Because of the development of powerful iterative solvers and the more and more increasing computer capacities, topology optimization will be state-of-the art very soon.

2.4.2 Shape Optimization

a. Shape optimization based on FE models

   The coordinates of the surface nodes are regarded as design variables which will be modified during the optimization. This usually leads to a large number of design variables which might cause considerable mathematical difficulties. Using suitable couplings of node displacements to define basis vectors.

b. Shape optimization based on geometry of models

   Using the shape optimization method based on geometry models, the linkage of an FE model and a geometry model is maintained. As in this case the parameters of the geometry model are the design variables, the geometry model has to be fully parametric.

   Therefore the use of an efficient solid modeler is necessary. Each parameter modification of the geometry model also results in changes of the FE model.

Within each optimization loop the entire FE model has to be set up a new according to the modifications of the geometry model parameters considering the boundary conditions. The selection of the design variables are left to the user. In general they differ due to his experiences and creativity. The results of the optimization essentially depend on the number and selection of the design variables. If free form surfaces are allowed, the selection of the design variables is very difficult.

The main difficulty with shape optimization is to transfer the surface changes to the FE mesh. Most programs avoid this transfer by an automatic re meshing in each optimization loop. Hence the original element topology (meshing) is destroyed and often models with only tetrahedral elements are created.[5]

2.4.3 Size Optimization

Size optimization involves a modification of the cross-section or thickness of finite elements. The optimization is carried out by mathematical optimization algorithms with different objective functions e. g. maximum stiffness or minimum weight. Many programming approaches were tested and implemented in finite element programs or special optimization programs. Due to the easy calculation of sensitivities for size optimization purposes even realistic problems can be handled.[6]

2.5 Structural Optimization

A structure is an assemblage of materials with the purpose to sustain loads. Designing a structure is an iterative process where the design is modified until it complies with the requirements set upon it. This iteration process can be formulated as an optimization problem where the optimal solution is the best of all the designs that fulfils the requirements.

Chapter 3

Forces Developing and Acting in Lathe Machine

3.1 Sources And The Types Of The Forces That Develop In Lathe Machine During Machining

- Cutting forces originating at the cutting point(s)
  - In continuous type machining;
    - Main cutting force, P_Z along the velocity vector, V_C
    - Feed or thrust force, P_X along the feed direction
    - Transverse force, P_Y normal to P_Z P_X plane in turning, boring and similar single point cutting process
    - Torque and thrust force in drilling, counter boring, counter sinking etc.
  - In impact initiated type;
    - Shaping, planing, slotting, gear shaping etc.
  - In intermittent type;

        * Fluctuating forces due to intermittent cutting in milling, hobbing etc.

- Gravitational forces
    - Dead weight of the major and heavy components of the Machine Fixture Tool Work (M F T W) system, e.g., workpiece, headstock, tailstock, saddle, bed and moving tables etc.
- Frictional forces
    - Due to rubbing at the sliding surfaces.
- Inertia forces
    - Due to acceleration and deceleration at the end points of sliding and reciprocating motions of heavy parts like carriage or saddle, turret slide, tool slides, moving beds, reciprocating tables, rams, jobs etc.
- Centrifugal forces
    - Due to high speed rotation of eccentric masses
    - Due to wide run out or eccentric rotation of jobs, machine tool parts, spindle, shafts, tools etc.

3.2 Effects Of The Various Forces On Machining And Lathe Machine

a. Energy or power consumption

b. Increased cutting zone temperature and its detrimental effects

c. Dynamic forces resulting vibration and chatter cause poor surface quality and reduction of life of cutting tools as well as damage of the machine tools

d. Elastic deflection and thermo-elastic deformation of several bodies leading to dimensional inaccuracy

e. Rapid wear and tear at the sliding surfaces

f. Noise and inconvenience

g. Chances of premature mechanical failure of cutting tools and other components due to excessive stresses, thermal fracture, wear, fatigue, resonance etc.

3.3 Purposes Of Analysis And Evaluation Of The Forces Acting In Lathe Machine

It is essentially needed to know or determine the magnitude, location and direction of action and also the nature of the forces that develop and act during machining to enable :

- Estimate the cutting power and total power requirement for selection of type and capacity of the main power sources (motors)
- Design of the machine tool and cutting tool systems and the tool work holding devices
- Design of the machine tool foundations
- Evaluate process capability of the machine tools
- Assess the machinability characteristics of various tool work combinations under different operating conditions of the machine tools
- Determination of the role of the different process, geometrical and environmental parameters on the magnitude and pattern of the forces, which will help their optimal selection for good performance of the M F T W system and overall economy.

- Comprehend the need and way of improvement in design, construction, performance, safety and service life of the machine tools.

3.4 Analysis of forces acting and developing in a center lathe

Center lathes are used for various machining work but mostly for straight turning. Fig.3.1 shows the location and direction of action of the different forces that develop in the headstock and tailstock being originated by the machining forces (components).

- Tangential component, P_Z - main cutting force.
- Axial component, P_X - feed force.
- Transverse component, P_Y - thrust force.

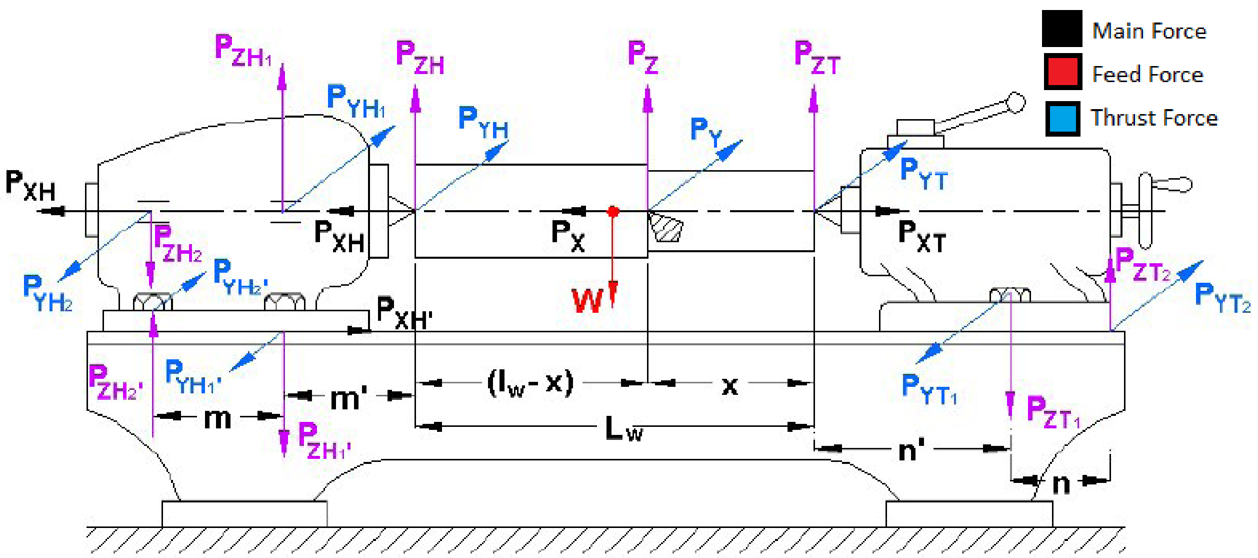

Figure 3.1: Forces acting on lathe[7]

3.4.1 Specification of lathe

The lathe specification are used in calculating maximum force. The Table 3.1 illustrates required specification of lathe.

Table 3.1: Specifications of lathe

Capacity	Unit	Dimensions
Swing over cross slide	mm	200(8")
Distance between center	mm	800(32")
Bed width	mm	242(9.5")
Speeds range	rpm	45-1120
Main motor	kw	2.2
Bed type	-	2V & 2 flat

3.4.2 Parameter selection for maximum power utilization [7]

$$P = 13.2 \times 10^{-3} \times K \times z \times v \times (1.55 \times a \times s)^{\beta} = 2.2kW \tag{3.1}$$

Where,

P = Power in kW,

K = Material factor = 3.44,

β = Machining exponent = 0.803,

z = No of cutting edges in contact = 1,

v = Cutting speed in m/min,

a = Depth of cut in mm,

s = feed in mm/rev.

Following are the hypothetical parameters considered for maximum utilization of main motor power of 2.2kw.

Table 3.2: Turning parameter case-1

Case study 1	
Tool	Single point HSS
Material	Free cutting steel (35 S 11, 17 S 11, 12 S 12, C20, C45)
Brinell Hardness	375-425
Rough Depth of Cut	3 mm
Feed	0.4 mm/rev
Speed	29 m/min

Table 3.3: Turning parameter case-2

Case study 2	
Tool	Single point Brazed Carbide
Material	Free cutting steel (35 S 11, 17 S 11, 12 S 12, C20, C45)
Brinell Hardness	375-425
Rough Depth of Cut	4 mm
Feed	0.4 mm/rev
Speed	53 m/min

Table 3.4: Turning parameter case-3

Case study 3	
Tool	Single point Disposable Carbide
Material	Free cutting steel (35 S 11, 17 S 11, 12 S 12, C20, C45)
Brinell Hardness	375-425
Rough Depth of Cut	4 mm
Feed	0.4 mm/rev
Speed	65-66 m/min

Table 3.5: Turning parameter case-4

Case study 4	
Tool	Single point HSS
Material	Hard Brass, Hard Bronze, Marble
Brinell Hardness	70-160
Rough Depth of Cut	2.5 mm
Feed	0.6 mm/rev
Speed	28 m/min

Table 3.6: Turning parameter case-5

Case study 5	
Tool	Single point Carbide
Material	Hard Brass, Hard Bronze, Marble
Brinell Hardness	70-160
Rough Depth of Cut	3.2 mm
Feed	1.6 mm/rev
Speed	63 m/min

3.4.3 Cutting forces [7]

- Main cutting force P_Z :

$$P_Z = \frac{19.5 \times 10^{-6} \times P}{D \times n} = 4766.65N \tag{3.2}$$

Considering factor of safety for live loads on Cast iron material as 8-12. [21]

$$P_Z = 4766.65 \times 8 = 38133N \tag{3.3}$$

- Feed force P_X

$$P_X = 40\% of P_Z = 15253N \tag{3.4}$$

- Thrust force P_Y

$$P_Y = 40\% of P_Z = 15253N \tag{3.5}$$

3.4.4 Forces acting on the Headstock side [7]

- On the headstock (HT) center:

$$P_{ZH} = P_Z \times (\frac{x}{L_w}) - \frac{W}{2} = 18081N \tag{3.6}$$

where,

W = weight of the workpiece (rod)= 1970 N

L_w = length of the workpiece

x = distance of the cutting tool from the tailstock centre = 400mm

P_Z = Main cutting force

$$P_Z = \frac{19.5 \times 10^{-6} \times P}{D_w \times n} = 38133N \tag{3.7}$$

where,

D_w = maximum diameter of the workpiece = 200mm

$$P_{YH} = P_Y \times (\frac{x}{L_w}) + P_X \times (\frac{D_w}{2L_w}) = 9532N \tag{3.8}$$

where,

P_Y = Thrust force = 40% of P_Z = 15253 N

P_X = Feed force = 40% of P_Z = 15253 N

- At the bearing housings

$$P_{ZH_1} = P_{ZH} \times (\frac{m + m'}{m}) = 27725N \tag{3.9}$$

where,

m = 300mm and m' = 160mm (Refer fig.3.1)

$$P_{ZH_2} = P_{ZH} \times (\frac{m'}{m}) = 9643N \tag{3.10}$$

$$P_{YH_1} = P_{YH} \times (\frac{m + m'}{m}) = 14615N \tag{3.11}$$

$$P_{YH_2} = P_{YH} \times (\frac{m'}{m}) = 5084N \tag{3.12}$$

- At the supports (bolting)

$$P_{ZH'_1} = -P_{ZH_1} = -27725N \tag{3.13}$$

$$P_{ZH'_2} = -P_{ZH_2} = -9643N \tag{3.14}$$

$$P_{YH'_1} = -P_{YH_1} = -14615N \tag{3.15}$$

$$P_{YH'_2} = -P_{YH_2} = -5084N \tag{3.16}$$

3.4.5 Forces acting on Tailstock side [7]

- On the Tailstock center

$$P_{ZT} = P_Z \times (\frac{L_w - x}{L_w}) - (\frac{W}{2}) = 18081N \tag{3.17}$$

$$P_{YT} = P_Y \times (\frac{L_w - x}{L_w}) - P_X \times (\frac{D_w}{2L_w}) = 5720N \tag{3.18}$$

- At the bolting and rear bottom end (heel)

$$P_{ZT_1} = P_{ZT} \times (\frac{n + n'}{n}) = 30135N \tag{3.19}$$

$$P_{ZT_2} = P_{ZT} \times (\frac{n'}{n}) = 12054N \tag{3.20}$$

$$P_{YT_1} = P_{YT} \times (\frac{n+n'}{n}) = 9533.33N \tag{3.21}$$

$$P_{YT_2} = P_{YT} \times (\frac{n'}{n}) = 3814N \tag{3.22}$$

3.5 Forces acting on lathe bed [7]

The lathe bed receives forces through;

- The headstock and tailstock.
- The saddle on which the cutting tool is mounted.

3.5.1 Forces through headstock and tailstock [7]

The headstock is kept fixed by two pairs of bolts or studs on the lathe bed and the tailstock is clamped on the bed by one bolt. The forces acting on the bed through the front and the rear pair of bolts are :

$$P_{V_1} = P_{ZH_1} = 27725N \tag{3.23}$$

$$P_{V_2} = P_{ZH_2} = 9643N \tag{3.24}$$

$$P_{H_1} = P_{YH_1} = 14615N \tag{3.25}$$

$$P_{H_2} = P_{YH_2} = 5084N \tag{3.26}$$

where,
P_{V1} and P_{V2} are vertical forces
P_{H1} and P_{H2} are horizontal forces
Similarly the forces acting on the lathe bed through the tailstock are:

$$P'_{V_1} = -P_{ZT_1} = -30135N \tag{3.27}$$

$$P'_{V_2} = -P_{ZT_2} = -12054N \tag{3.28}$$

$$P'_{H_1} = -P_{YT_1} = 9533.33N \tag{3.29}$$

$$P'_{H_2} = -P_{YT_2} = 3814N \tag{3.30}$$

3.5.2 Forces acting on the lathe bed through the saddle [7]

The cutting tool receives all the forces P_Z, P_X and P_Y but in opposite direction as reaction forces. And those forces are transmitted on the lathe bed through the saddle as indicated in Fig.3.2. The saddle rests on and travels along the lathe bed. All the forces acting on the bed through the saddle are assumed to be concentrated at four salient locations, A, B, C and D within the saddle bed overlapped area as shown. Then from the force diagram in Fig.3.2 the vertical forces (V) and horizontal forces (H) can be roughly determined;

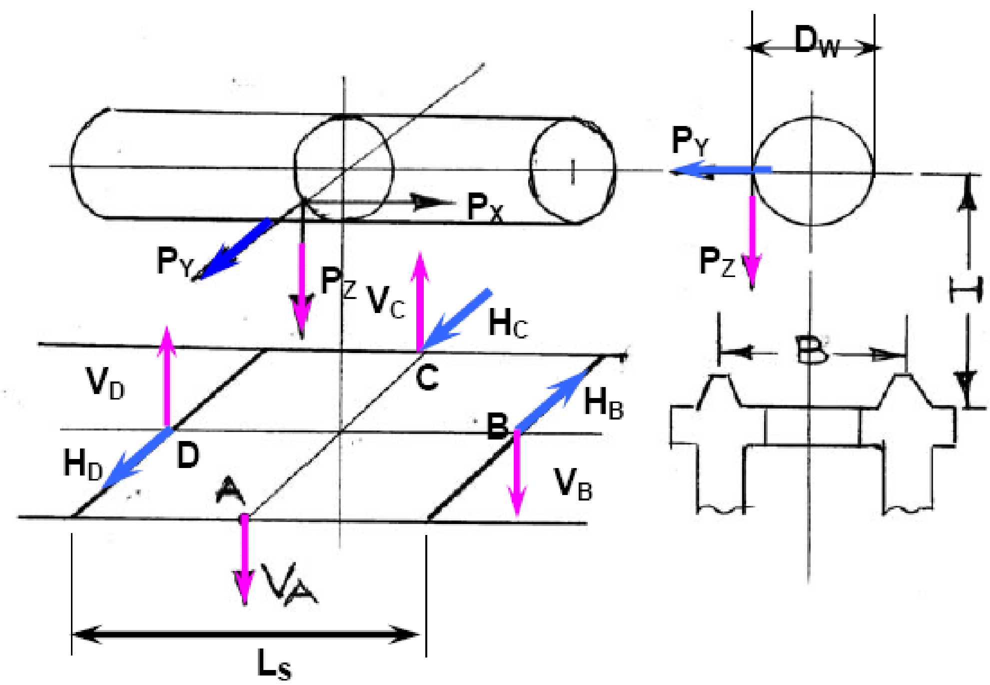

Figure 3.2: Forces acting on lathe bed due to saddle through cutting force[7]

$$V_A = P_Z \times (\frac{B + D_w}{2B}) + P_Y \times (\frac{H}{B}) = 29572N \tag{3.31}$$

$$V_C = P_Z \times (\frac{B - D_w}{2B}) - P_Y \times (\frac{H}{B}) = 26694N \tag{3.32}$$

where,

B = Width of the saddle = 370mm

H = Center height = 200mm

Chapter 4

CAD Modeling and FE Analysis of lathe bed

4.1 Introduction

For carrying the analysis one of the most important tasks is to model the part correctly. It is after modeling the component one can able to calculate the forces and moments that are further used to carry out analysis. Generating Variants of CAD and CAE Models is a State of the Art. An easy and fast generation of variants is an essential requirement to operate an efficient optimization process. Both major areas, CAD as well as CAE, have to be taken into account. In the best case modifications in the CAD model can be transferred to the CAE model and vice versa.

So it is quite obvious from the above discussion and fig 4.1 that parametric modeling of the components is to be carried out. There is a fundamental difference between explicit modeling and parameter based modeling. In the traditional approach of explicit modeling there is no dependence between root geometry and the geometry for additional parts. The explicit modeling is used for styling and design surfaces. New methods are based on parametric modeling and are used for other areas in automotive design. They can be divided into two classes:

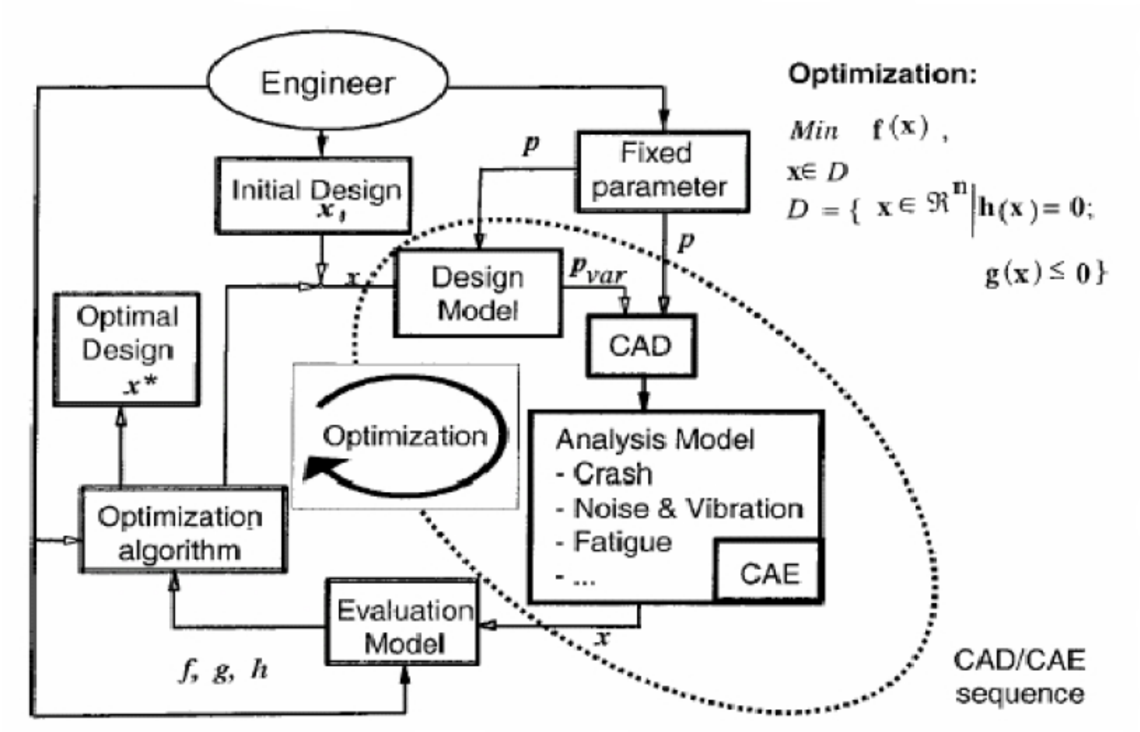

Figure 4.1: CAD/CAE sequence within the optimization process

- Feature based modeling.
- Condition based modeling.

Feature based modeling builds up geometry by combining simple volumes, called volume primitives e.g. cylinder, cubic or sphere, this modeling technique has a decisive disadvantage: the modeling of complex geometry requires a large number volume primitive. This difficulty is overcome with the condition based modeling method. 2D sketches define the basic geometry. Afterwards conditions are assigned to the geometry. Knowledge based engineering techniques help to formulate powerful conditions. The principal steps for the sketch technique are,

- Creation of the fuzzy contour with simple curves and line segments.
- Definition of rules for the description of exact geometry.

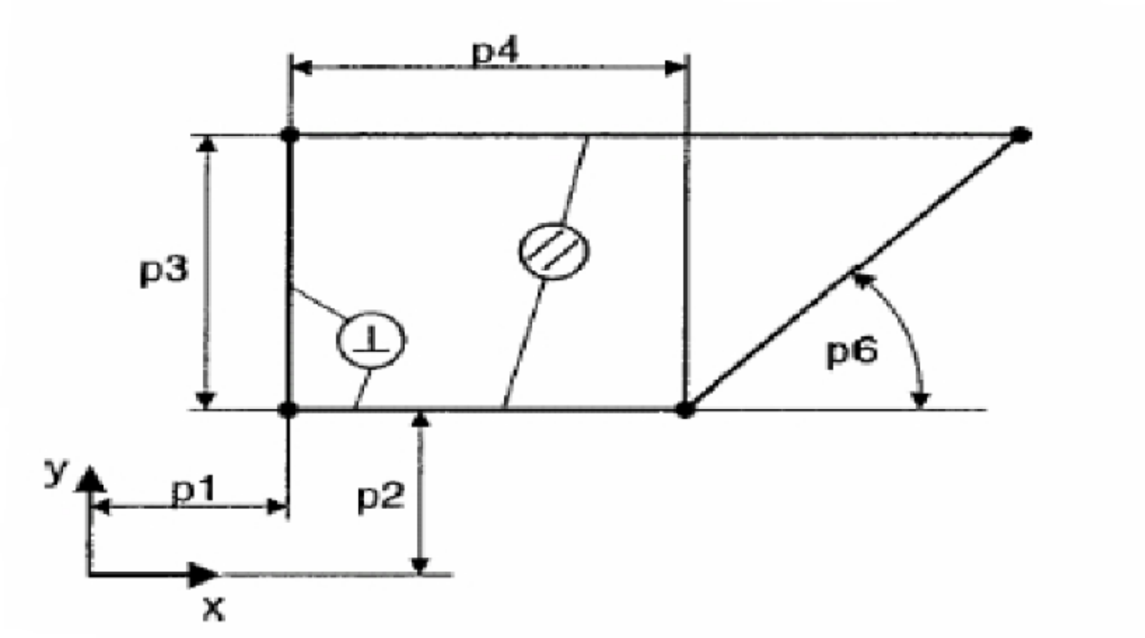

Figure 4.2: Defining Geometry by Parameters

These rules are formulated as algebraic equations by connecting parameters. Fig.4.2 gives a modeling example.

4.2 Solid Modeling

Lean manufacturing concept is applied on conventional center lathe machine of Kirloskar make available in workshop facility of Nirma University campus. For the analysis purpose a solid model of lathe bed is created using CAD software SolidWorks. First the cross-sectional geometry of lathe bed is created using sketch command. In next step by using extrude command the solid model of lathe bed is created. The dimension and geometry are taken from the Kirloskar lathe manual guide. The Fig. 4.3 represent the solid model of existing lathe machine bed.

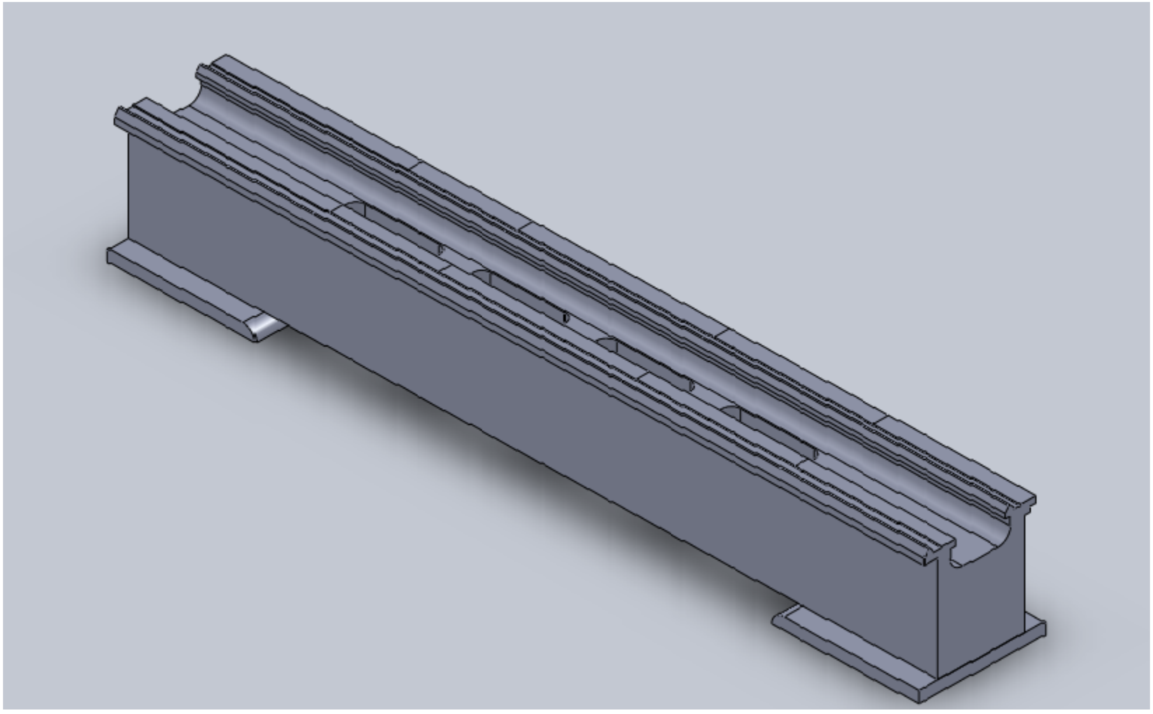

Figure 4.3: Solid modeling of lathe bed

4.3 FEA introduction

Finite Element Analysis (FEA) was first developed in 1943 by R. Courant, who utilized the Ritz method of numerical analysis and minimization of variation calculus to obtain approximate solutions to vibration systems.
Finite-element analysis is getting a bigger role in development projects. One of the reasons is that it helps slash expensive prototype testing. The technology is also seen as another way to improve product integrity. FEA consists of a computer model of a material or design that is loaded and analyzed for specific results.
In finite element analysis, the design is discretion or subdivided into a series of elements that are connected by nodes. Material properties and element properties are specified to represent the physical properties of the model. Boundary conditions and applied loads are then defined to represent the operating environment for which the design is to be subjected and its simulation tool that enables engineers to simulate

the behavior of an entire structure.

Mathematically, the structure to be analyzed is subdivided into a mesh of finite sized elements of simple shape. Within each element, the variation of displacement is assumed to be determined by simple polynomial shape functions and nodal displacements. Equations for the strains and stresses are developed in terms of the unknown nodal displacements. From this, the equations of equilibrium are assembled in a matrix form which can be easily be programmed and solved on a computer. After applying the appropriate boundary conditions, the nodal displacements are found by solving the matrix stiffness equation. Once the nodal displacements are known, element stresses and strains can be calculated.

4.3.1 General steps in finite element method and its interpretation

There are two general approaches associated with the finite element method. One approach is called as the force, or flexibility, method, uses internal force as the unknowns of the problem. To obtain the governing equations, first the equilibrium equations are used. Then necessary additional equations are found by introducing compatibility equations. The result is a set of algebraic equations for determining the redundant or unknown forces.

The second approaches, called the displacement, or stiffness, method, assumes the displacement of the nodes as the unknowns of the problem. For instance, compatibility conditions requiring that elements connected at a common node, along a common edge, or on a common surface before loading remain connected at that node, edge or surface after deformation takes place are initially satisfied. Then the governing equations are expressed in terms of nodal displacement using the equation of equilibrium and an applicable law relating force to displacements.

a. Step: 1 Discretion and select the element types.

   It involves dividing the body into an equivalent system of finite elements with associated nodes and choosing the most appropriate element type of model most closely the actual physical behavior.

b. Step: 2 Select a displacement function.

   It involves choosing a displacement function within each element. The function is defined within the element using the nodal values of the element. Linear, quadratic and cubic polynomials are frequently used functions because they are simpler to work with in finite element formulation. High level polynomial can also be used as per requirement.

c. Step: 3 Define the strain/displacement and stress/strain Relationship.

   Strain/displacement and stress/strain Relationship are necessary for deriving the equation for each finite element. In case of one dimensional we have strain $\varepsilon_X$in the x direction for small strain, also here material defined is assumed as elastic which follows hooks law.

$$\varepsilon_X = (\frac{du}{dx}) \tag{4.1}$$

d. Step: 4 Derive the element stiffness matrix and Equations.

e. Step: 5 Assemble the element equations to obtain the Global or total equations and introduce Boundary conditions.

f. Step: 6 Solve for the unknown Degree of freedom.

g. Step: 7 Solve for the Element strain and stresses

h. Step: 8 interpret the results

4.4 Application of FE analysis

4.4.1 Automotive application

In the automobiles the road loads are transferred to the vehicles and hence the stresses and strain in the body panel are of interest. Hence the FEA tool is used to perform modal analysis, static analysis, torsional analysis, and service load analysis. In addition to this crash analysis is emerging as an important tool in finite element analysis. At the same time FEA is used to determine the temperature distribution through the engine.

4.4.2 Electrical and Electronics engineering applications

FEA can be used for reliability enhancement and optimization of insulation design in high voltage equipment by finding accurately the voltage stress and corresponding withstands. For complex configuration of electrodes and dielectric insulating materials, analytical formulation are inaccurate and extreme difficult, if not impossible. The FEA Can used in such cases. An analysis of eddy currents in structural conducting parts and minimization of stray losses in electrical machines is possible also Using FEM.

4.4.3 Aerospace application

In typical aerospace application, finite element analysis is used for several purposes, viz. structural analysis for natural frequencies, mode shapes, response analysis, aero-servo elastic studies, and aerodynamics.

4.4.4 Manufacturing process simulation

The FEA is used as emerging tool in the field of manufacturing simulation. The FEA analysis is used to study the solidification, thermal field and evolution of stress and factors causing failure. This information is further used to change the processing conditions so as to eliminate these high tensile stresses. As may be evident from the above examples of real life application of finite element analysis, present day engineering design based on CAE tools involves extensive use of finite elements in wide variety of fields. Hence the knowledge of finite element analysis is crucial in significant way to aid intelligent use of commercial software in solving day-to-day problems.

4.5 Types of elements used in FEA analysis

Element contains the following for modeling a wide range of spatial dimensionality.

a. One-dimensional (link) elements

b. Two-dimensional elements

c. Three-dimensional elements

d. Cylindrical elements

e. Axis symmetric elements

4.5.1 One-dimensional (link) elements

One-dimensional heat transfer, coupled thermal or electrical, and acoustic elements are available. In addition, structural link (truss) elements are available. These elements can be used in two-dimensional or three-dimensional space to transmit loads or fluxes along the length of the element.

4.5.2 Two-dimensional elements

There are several different types of two dimensional elements used for different application. For structural applications these include plane stress elements and plane strain elements.

- Plane stress elements

  Plane stress elements can be used when the thickness of a body or domain is small relative to its lateral (in-plane) dimensions. The stresses are functions of planar coordinates alone, and the out-of-plane normal and shear stresses are equal to zero.

  Fig.4.4 Reference cross-section and element in an axis symmetric solid about

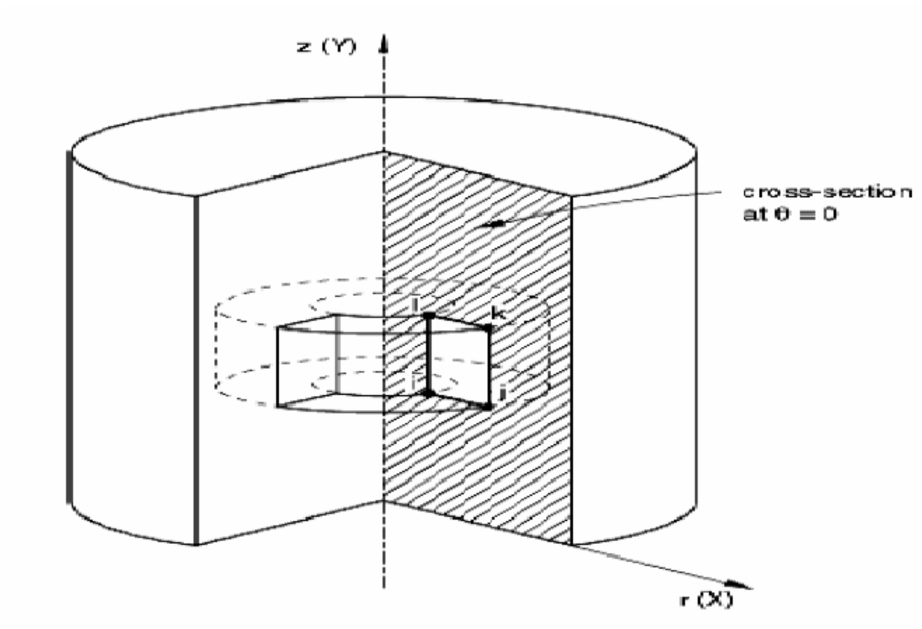

Figure 4.4: Axis symmetric element

an axis (the symmetry axis) and is readily described in cylindrical polar coordinates.

Plane stress elements must be defined in the - plane, and all loading and deformation are also restricted to this plane. This modeling method generally applies to thin, flat bodies. For anisotropic materials the -axis must be a principal material direction.

- Plane strain elements

  Plane strain elements can be used when it can be assumed that the strains in a loaded body or domain are functions of planar coordinates alone and the outofplane normal and shear strains are equal to zero. Plane strain elements must be defined in the - plane, and all loading and deformation are also restricted to this plane. This modeling method is generally used for bodies that are very thick relative to their lateral dimensions, such as shafts, concrete dams, or walls. Plane strain theory might also apply to a typical slice of an underground tunnel that lies along the -axis. For anisotropic materials the -axis must be a principal material direction.

4.5.3 Three-dimensional elements

Three-dimensional elements are defined in the global X, Y space. These elements are used when the geometry and/or the applied loading are too complex for any other element type with fewer spatial dimensions.

4.5.4 Cylindrical elements

Cylindrical elements are three-dimensional elements defined in the global X, Y space. These elements are used to model bodies with circular or axis symmetric geometry subjected to general, non axis symmetric loading.

4.5.5 Axis symmetric elements

Axis symmetric elements provide for the modeling of bodies of revolution under axially symmetric loading conditions. A body of revolution is generated by revolving a plane cross-section Fig. 4.4 shows a typical reference cross section at θ= 0. The radial and axial coordinates of a point on this cross-section are denoted by r and z respectively. At $\theta = 0$, the radial and axial element in an axis symmetric solid coordinates coincide with the global Cartesian X- and Y coordinates. If the loading and material properties are independent of , the solution in any r-z plane completely defines the solution in the body. Consequently, axis symmetric elements can be used to analyze the problem by Discretion the reference cross-section at θ=0. Fig. 4.4 shows an element of an axis symmetric body. The nodes i, j ,k ,l and are actually nodal circles, and the volume of material associated with the element is that of a body of revolution, as shown in the fig. 4.4. The value of a prescribed nodal load or reaction force is the total value on the ring, that is the value integrated around the circumference.

4.6 Present state FE Analysis of lathe bed

4.6.1 Problem Description

Lathe bed is a part of lathe machine which absorbs all forces and thrust developed during machining. Moreover it provides support to entire structure of lathe machine. Lathe bed is made of Gray cast iron material. Analysis requires to implement cutting forces on various areas of lathe bed due to combined effects of headstock, tailstock and carriage. The detail study is required to be carry out to understand vibrational, stress and displacement pattern under static and dynamic loading conditions.

4.6.2 Static structural analysis

Static structural analysis is performed using Ansys 12.1 software. First the geometry is imported in the form of parasolid model (Fig. 4.5). Then material properties are defined in library of software. The GCI material's properties are illustrated in Table 4.1. Then meshing of lathe bed model is carried out using tetrahedral elements. There are 74572 elements and 25484 nodes (Fig. 4.6). Then force generated at tool tip on carriage is transferred to center of the lathe bed. The contact area is defined as per the actual contact between lathe bed and saddle. All the three forces are applied as per Fig. 4.7. Boundary condition is applied to both the support of lathe bed. All degrees of freedom are constrained and fixed to zero (Fig. 4.8).

Table 4.1: GCI Material properties

Material	Gray cast iron
ISI Grade	GCI 20 DIN 1691/GG22
Density	7200 Kg/m^3
Ultimate tensile strength	2.4×10^2 MPa
Young modulus of Elasticity	1.1×10^5 Mpa
Poisson's ratio	0.28

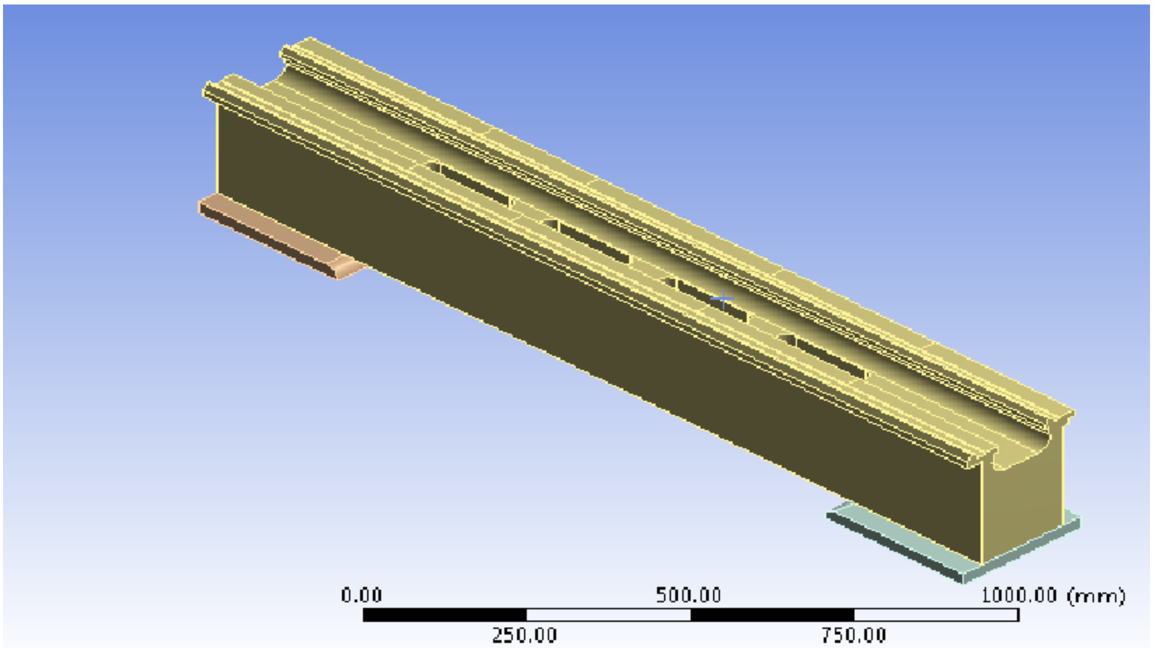

Figure 4.5: Import of solid model in Ansys workbench

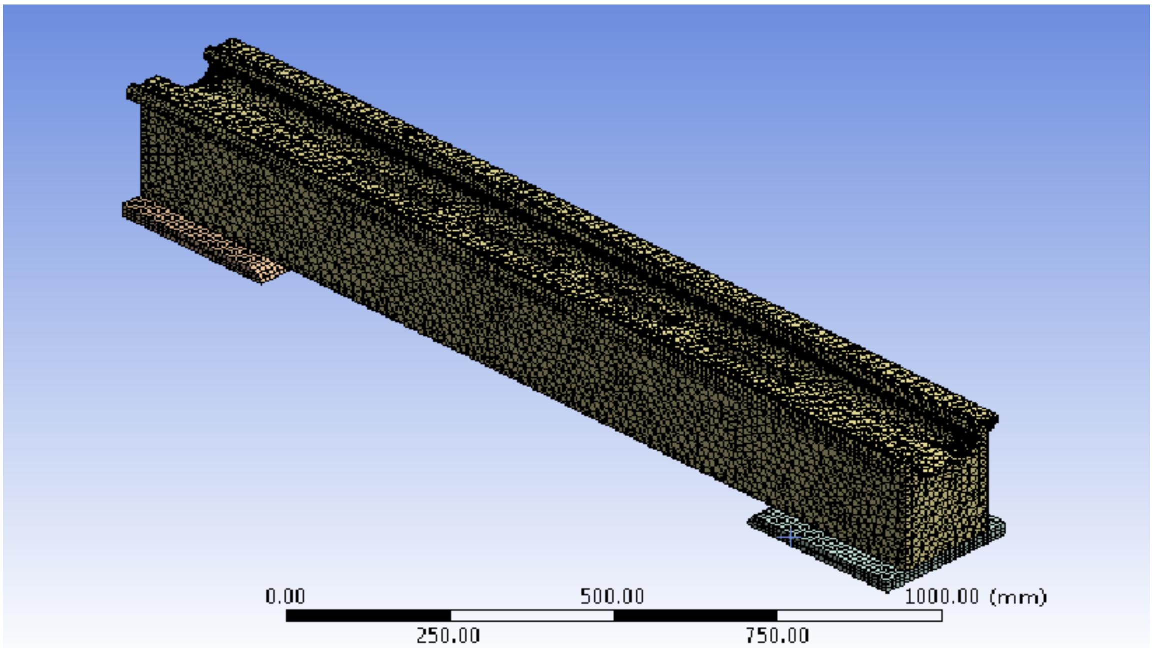

Figure 4.6: Meshed model of lathe bed

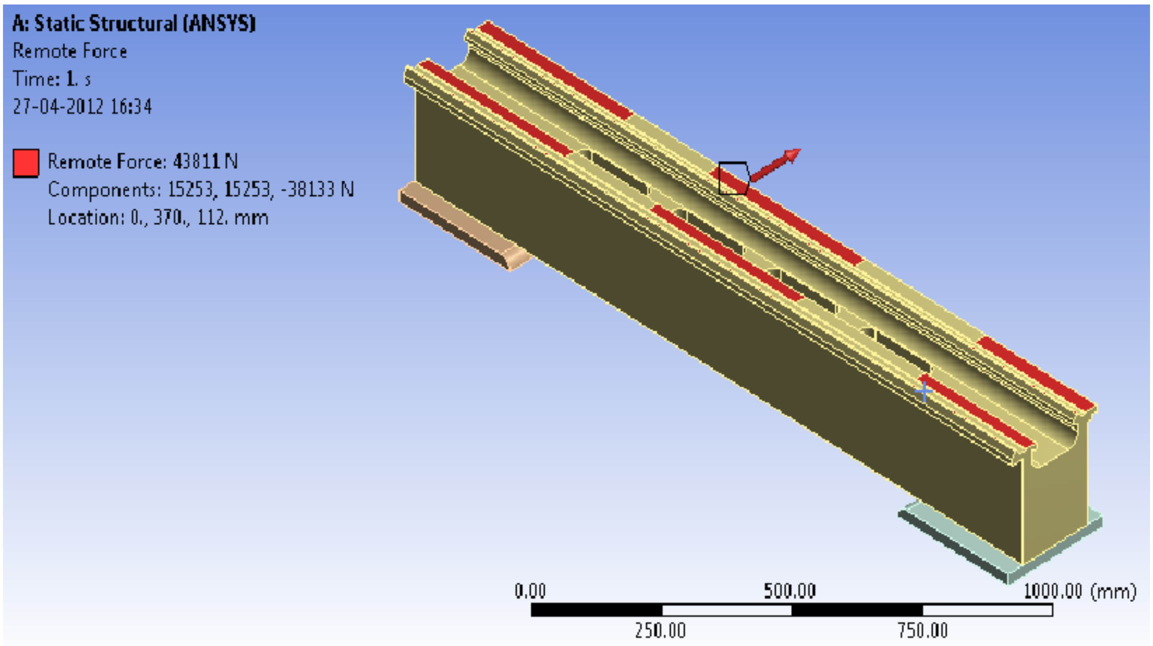

Figure 4.7: Different forces on lathe bed for static analysis

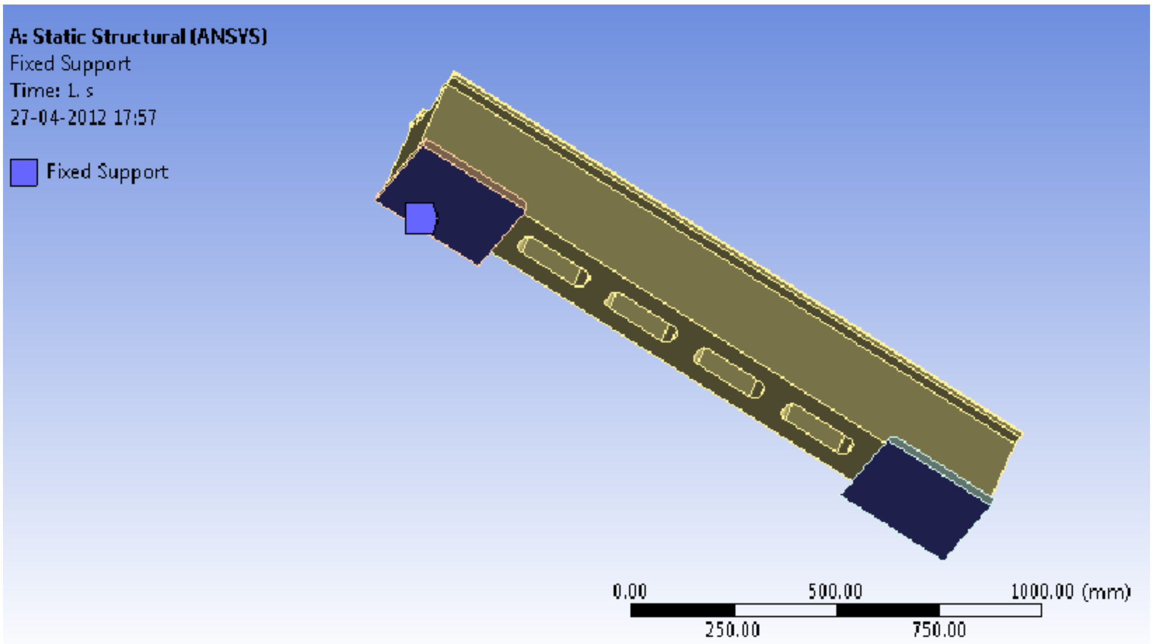

Figure 4.8: Boundary condition applied to support of lathe bed

4.6.3 Modal analysis

Modal analysis is performed to understand vibrational characteristic of machine tool components. It provides results in the form of frequencies at different phase angles. This results are interpreted to understand vibrational behavior of system in steady state conditions. These frequencies are used as reference for comparison with actual frequency generated while machining. The frequency generated during machining must not tune in with the natural frequency of system. If these occurs than it causes failure of component.This analysis does not account forces. First the geometry is imported in the form of parasolid model (Fig. 4.5). Then meshing of lathe bed model is carried out using tetrahedral elements. There are 74572 elements and 25484 nodes (Fig. 4.6). Boundary condition is applied to both the support of lathe bed. All degrees of freedom are constrained and fixed to zero (Fig. 4.8). Results of modal analysis in the form of various modes is carried out. For this case 4 modal analysis solutions are derived. This factor is decided based on the loading condition of the component. Here load is applied perpendicular to the plane of lathe bed. The different mode with their natural frequencies are illustrated in fig. 4.9, 4.10, 4.11 & 4.12. The results of frequencies in tabulated form is presented in Table 4.2.

Table 4.2: Mode Vs Frequency observation

Mode	Frequency (H_Z)
1	429.75
2	657.43
3	819.24
4	960.74

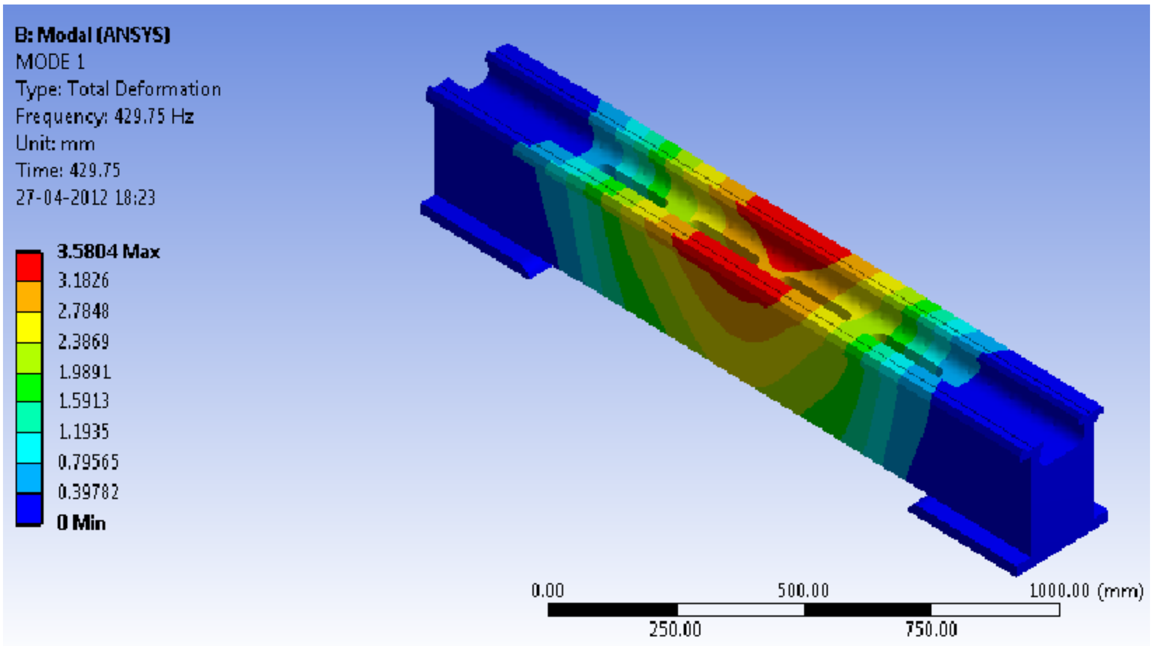

Figure 4.9: Mode-1 of Modal analysis

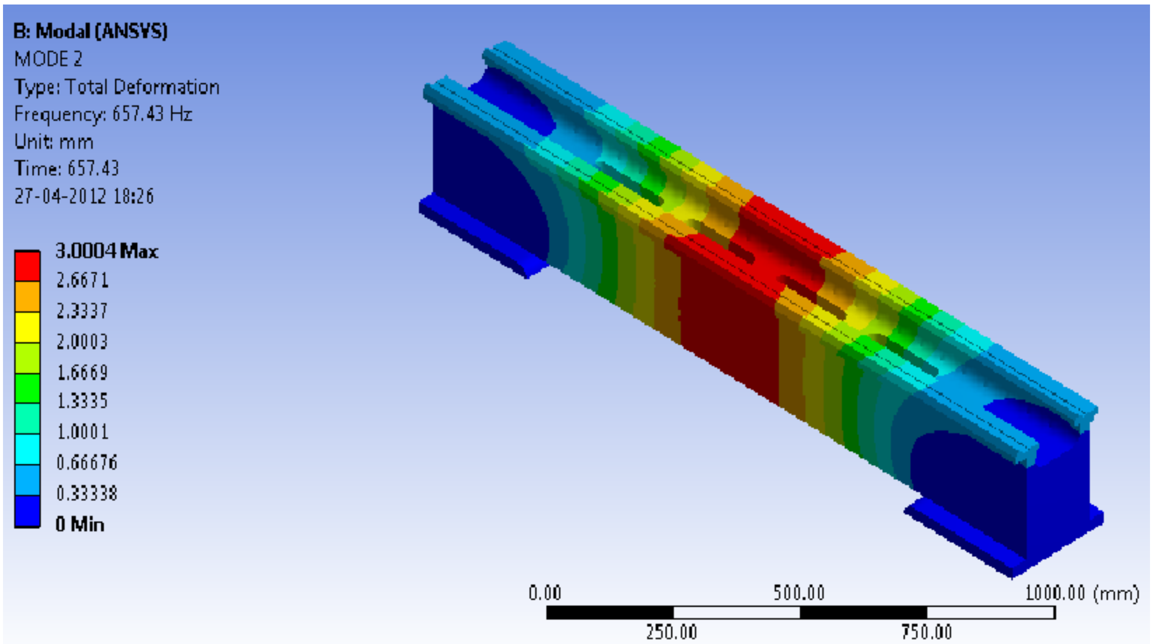

Figure 4.10: Mode-2 of Modal analysis

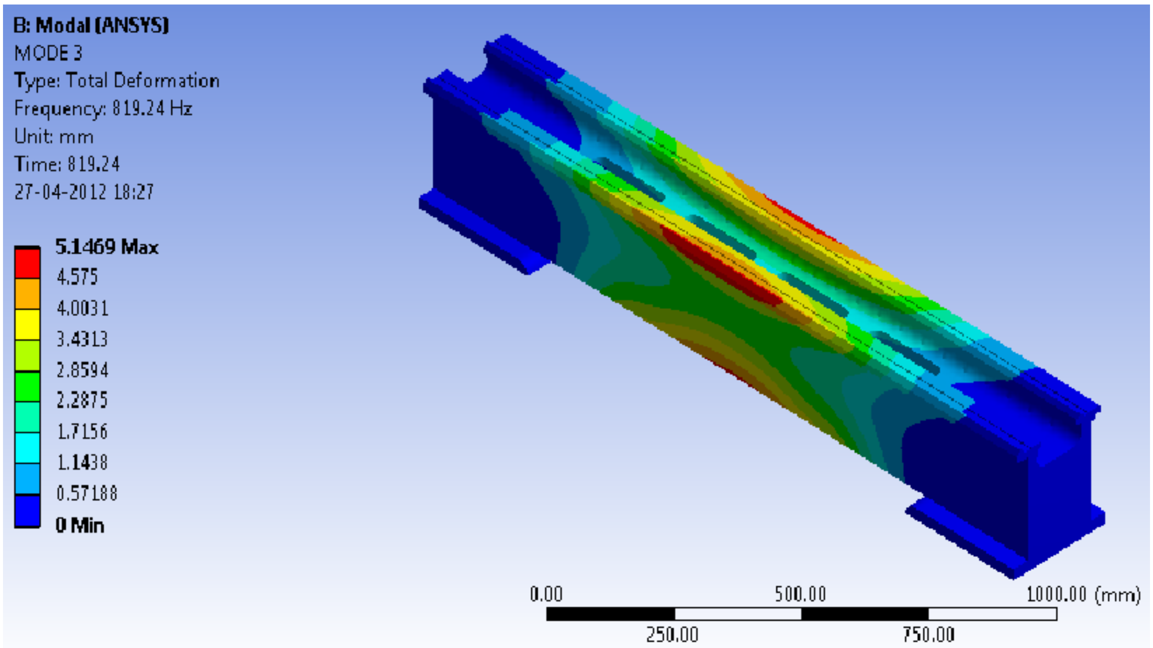

Figure 4.11: Mode-3 of Modal analysis

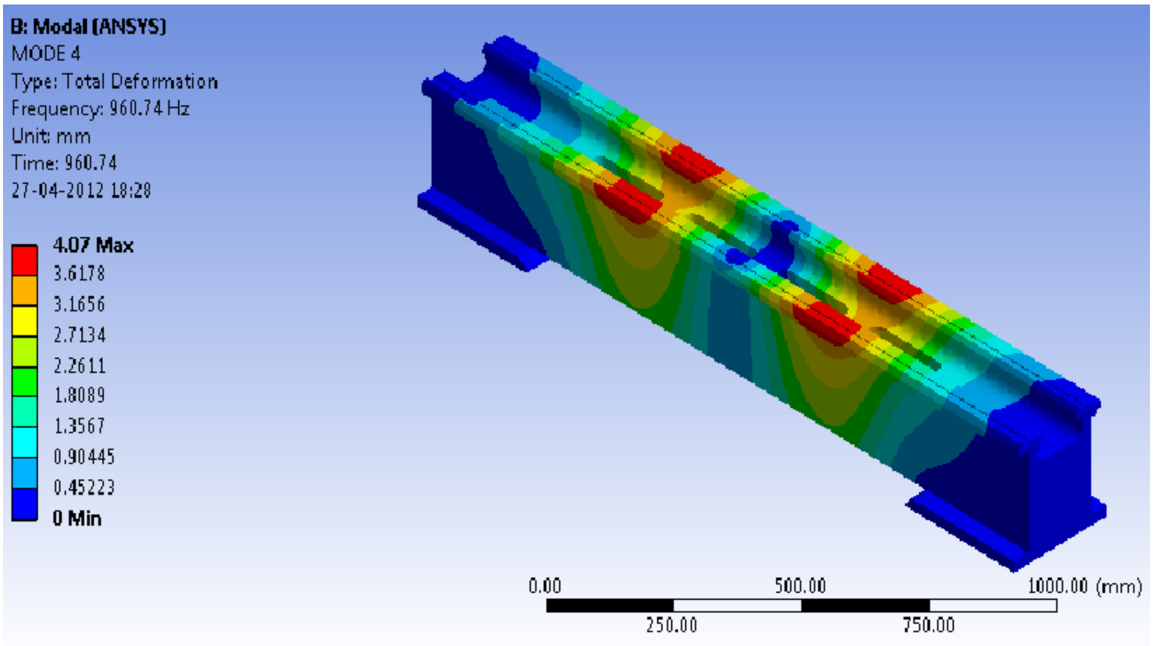

Figure 4.12: Mode-4 of Modal analysis

4.6.4 Transient Dynamic analysis

This analysis is carried out to understand rigidity of lathe bed under dynamic loading condition. The turning process is gradually divided into various numbers of steps and each step is precisely calculated for force generation. These forces are transferred onto lathe bed and its results are observed. First the geometry is imported in the form of parasolid model (Fig. 4.5). Then meshing of lathe bed model is carried out using tetrahedral elements. There are 74572 elements and 25484 nodes (Fig. 4.6). Further in dynamic condition time line is divided into 12 equal number of steps. These steps are applied on lathe bed as per the carriage movement from tailstock towards headstock. The combined effect of loading condition is illustrated in Fig. 4.13.

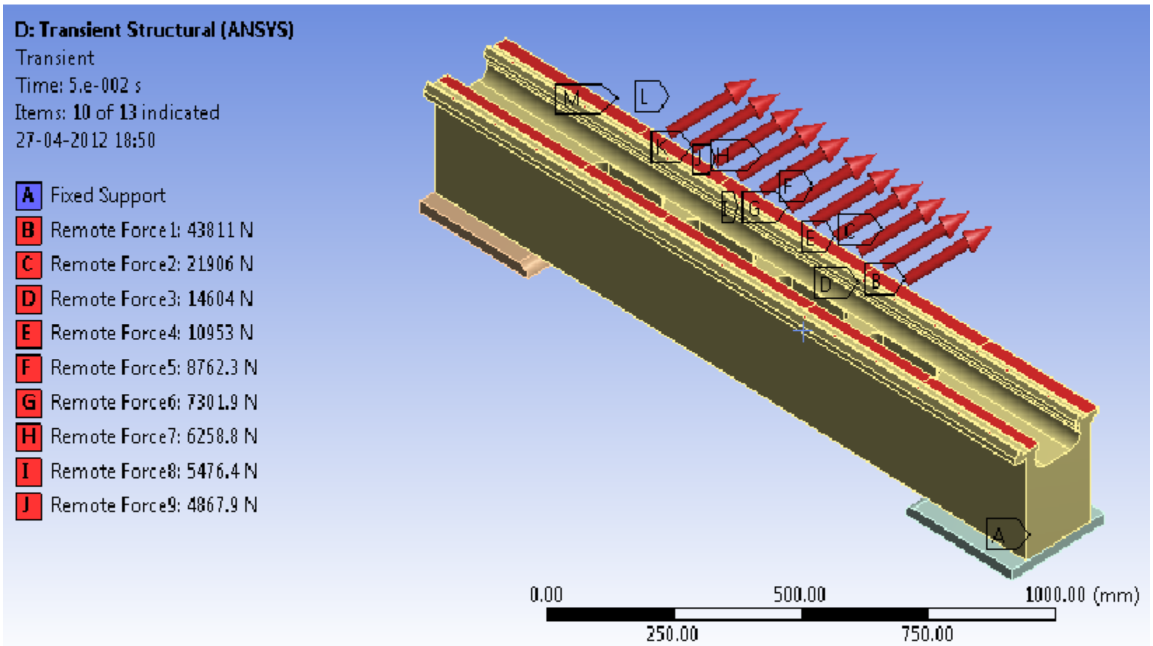

Figure 4.13: Different forces acting on lathe bed for dynamic analysis

4.7 LEAN design of lathe bed

4.7.1 Problem Description

Results of FE analysis using present lathe bed are taken into consideration for further improvement using lean principles. Lean manufacturing ideology leads to two options. The first one is to carry out shape optimization of existing lathe bed, redesign it, again carry out FE analysis and finally compare results with existing analysis. The second suggestion proposes altering Gray cast iron material with Al-Cu alloy metal, perform FE analysis and compare results with present analysis.

4.7.2 Optimization of lathe bed

Optimization of lathe bed is carried out in Ansys 12.1 workbench by shape optimization command. First the geometry is imported in the form of parasolid model (Fig. 4.5). Then meshing of lathe bed model is carried out using tetrahedral elements. There are 74572 elements and 25484 nodes (Fig. 4.6). Then force generated on tool tip of carriage is transferred to center of the lathe bed. The contact area is defined as per the actual contact between lathe bed and saddle. All the three forces are applied as per Fig. 4.7. Boundary condition is applied to both the support of lathe bed. All its degrees of freedom are constrained and fixed to zero (Fig. 4.8). The result of ansys shape optimization is interpreted and implemented for redesign of lathe bed in solidworks. Manufacturing feasibilities and operational feasibilities are considered for redesign purpose. The new redesign lathe bed is shown in Fig. 4.14.

4.7.3 Proposed material for lathe bed

Al-Cu is proposed as alternate material for lathe bed as it has the par level of properties when comparing with Gray cast iron. It is the most second desired material

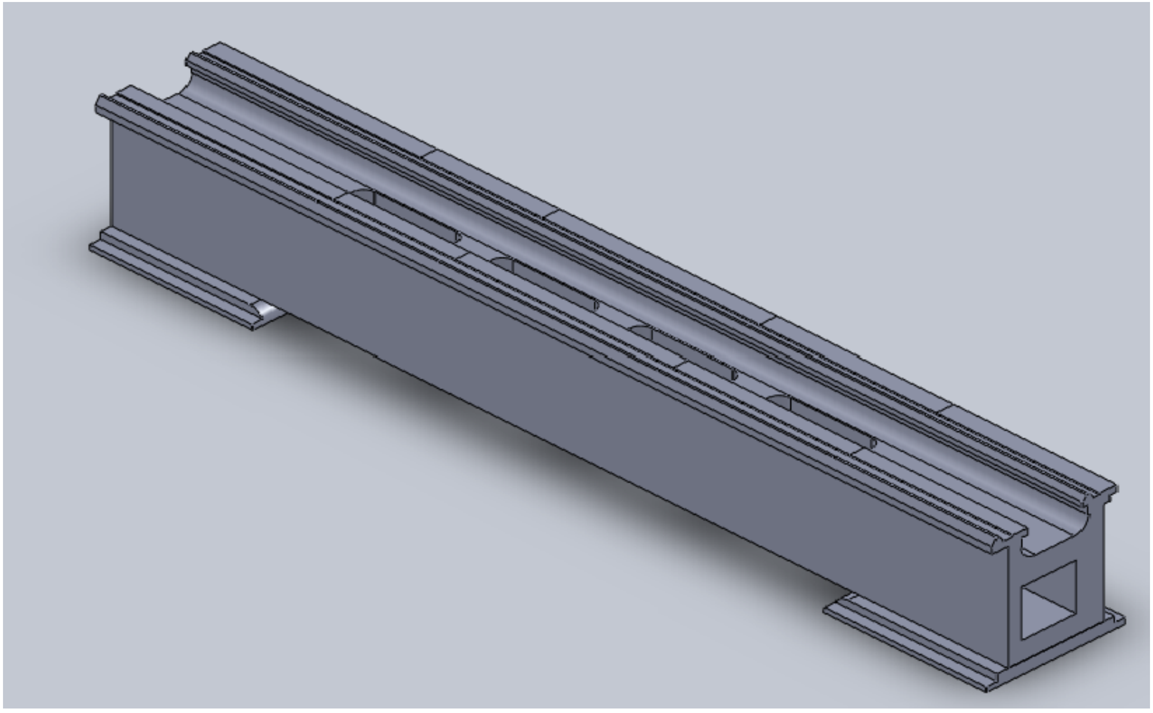

Figure 4.14: New optimized redesigned lathe bed structure

in today's technology to replace GCI. Properties of Al-Cu alloy are illustrated in following Table 4.3.

Table 4.3: Al-Cu Material properties

Material	Al-Cu alloy
ISI Grade	DIN 2547/AC24
Density	2720 Kg/m^3
Ultimate tensile strength	235 MPa
Young modulus of Elasticity	8.12×10^5 Mpa
Poisson's ratio	0.2

4.7.4 Static structural analysis of lathe bed with Al-Cu alloy

Al-Cu static structural analysis is performed using Ansys 12.1 software. First the geometry is imported in the form of parasolid model (Fig. 4.5). Then meshing of lathe bed model is carried out using tetrahedral elements. There are 74572 elements and 25484 nodes (Fig. 4.6). Then force generated at tool tip on carriage is transferred to center of the lathe bed. The contact area is defined as per the actual contact between lathe bed and saddle. All the three forces are applied as per Fig. 4.7. Boundary condition is applied to both the support of lathe bed. All degrees of freedom are constrained and fixed to zero (Fig. 4.8). After all these steps final solution is achieved.

4.7.5 Transient Dynamic analysis of lathe bed with Al-Cu alloy

This analysis is carried out to understand rigidity of Al-Cu lathe bed under dynamic loading. The turning process is gradually divided into various numbers of steps and each step is precisely calculated for force generation. These forces are transferred onto lathe bed and its results are observed. First the geometry is imported in the form of Parasolid model (Fig. 4.5). Then meshing of lathe bed model is carried out using tetrahedral elements. There are 74572 elements and 25484 nodes (Fig. 4.6). Further in dynamic condition time line is divided into 12 equal number of steps. These steps are applied on lathe bed as per the carriage movement from tailstock towards headstock. The combined effect of loading condition is illustrated in Fig. 4.13. After this steps solution is achieved.

Chapter 5

Results & Discussion

5.1 Present state Vs Proposed Optimized state

5.1.1 Comparison of results for shape optimization

FE results for shape optimization reveals that there is scope for 20% optimization as shown in Fig. 5.1. The figure indicates four potential areas named A, B, C & D as possible areas for shape optimization. Considering manufacturing and operational feasibility out off four areas only C and D areas are feasible for shape optimization. Rest A and B areas are not considered for shape optimization as they are on the guide path of lathe bed and hence not feasible for optimization. The modified optimized design has 10 % weight reduction compare to present design. The new optimized design is compared with present design as shown in Fig. 5.2. Results of optimization in form of mass comparison are indicated in Table 5.1.

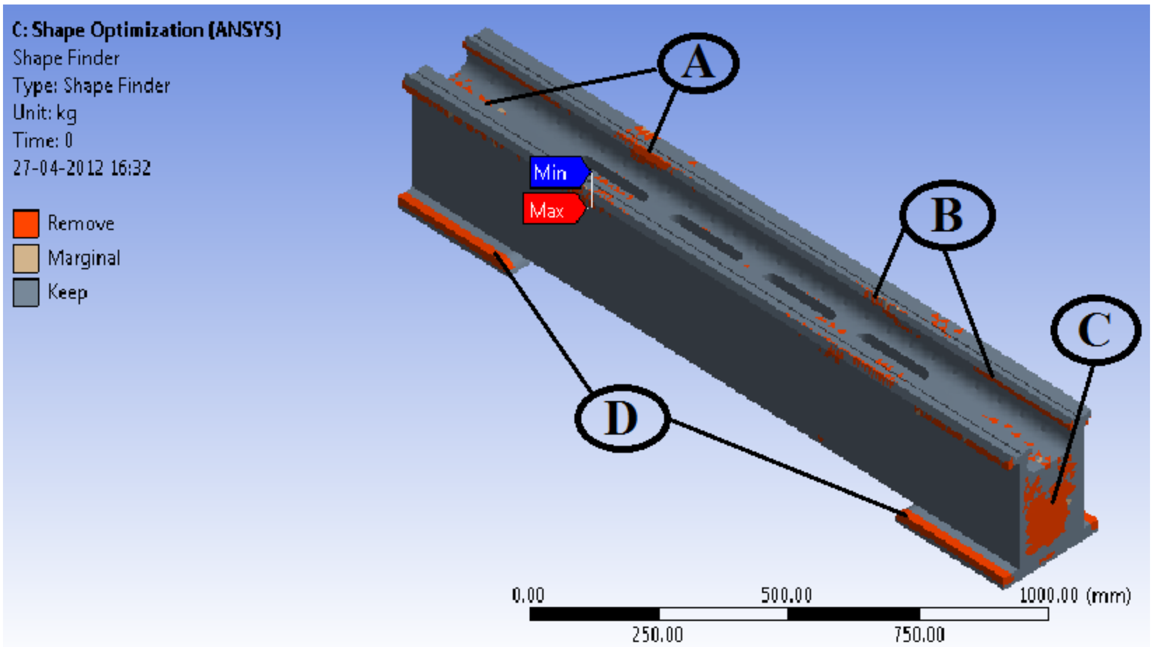

Figure 5.1: Shape optimization result

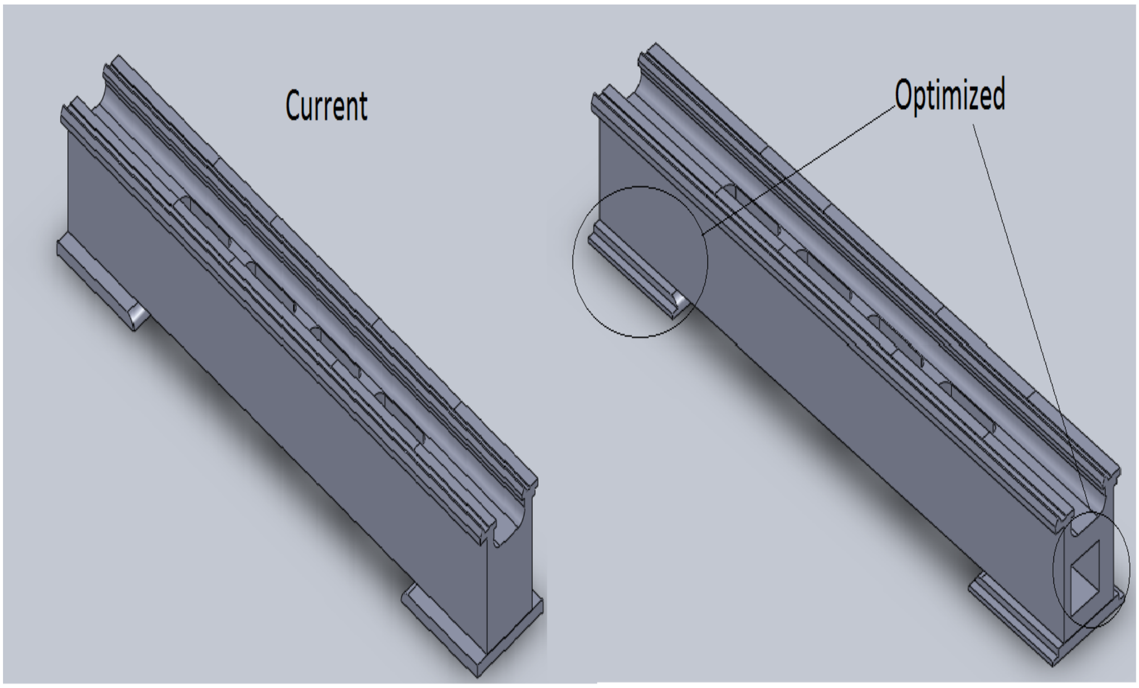

Figure 5.2: Present Vs Proposed design

Table 5.1: Mass comparison for optimized lathe bed

	Present state	Proposed by analysis	Modefied
Mass [Kg]	409.31	334.57	371.68
Mass reduction [%]	-	20	10

5.1.2 Comparison of results for static structural analysis of optimized lathe bed

Comparison of von-Mises static stress for present and proposed optimized lathe bed is shown in Fig. 5.3 & 5.4. Similar comparison of static shear stress and static deformation are indicated in Fig. 5.5, 5.6, 5.7 & 5.8 respectively. The comparison results of static stress are tabulated in Table 5.2. Result reveals a marginal amount of increase in von-Mises stress, Shear stress and displacement. This results are well within tolerance limits of material as per ISO standards.

Table 5.2: Maximum and minimum stress results comparison for Present Vs Proposed optimized state

Type	von-Mises stress		Maximum shear stress		Total deformation	
	Present	Proposed	Present	Proposed	Present	Proposed
Minimum	0.024 MPa	0.054 MPa	0.013 MPa	0.03 MPa	0 mm	0 mm
Maximum	12.41 Mpa	14.99 MPa	6.34 MPa	8 MPa	0.1 mm	0.103 mm

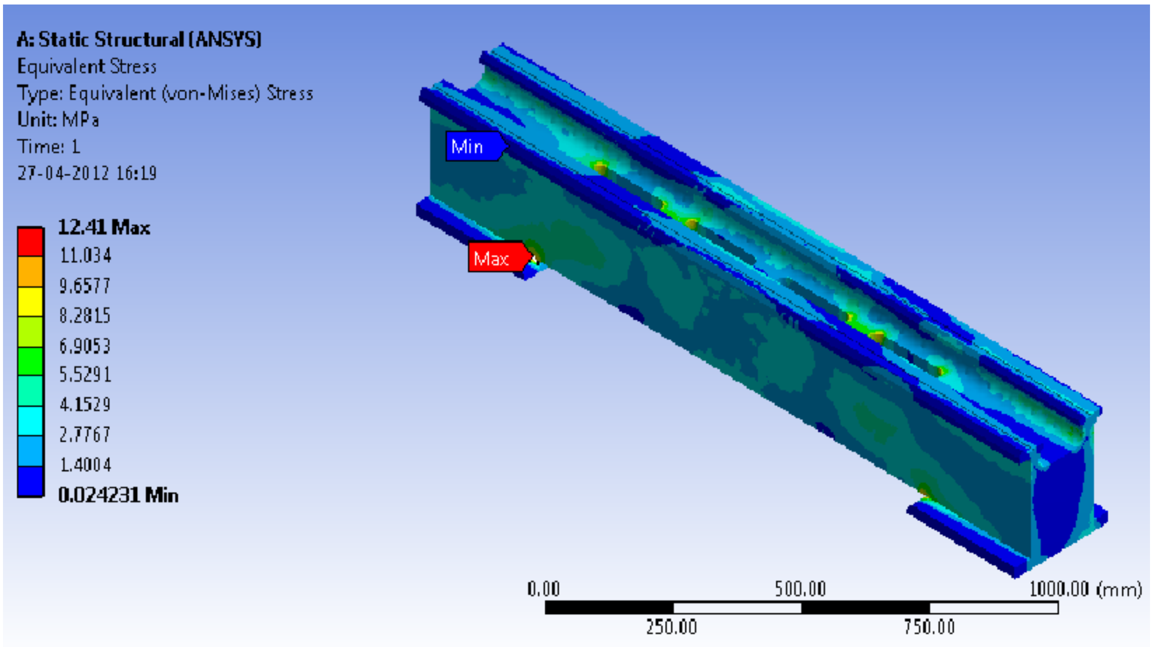

Figure 5.3: Present von-Mises static stress

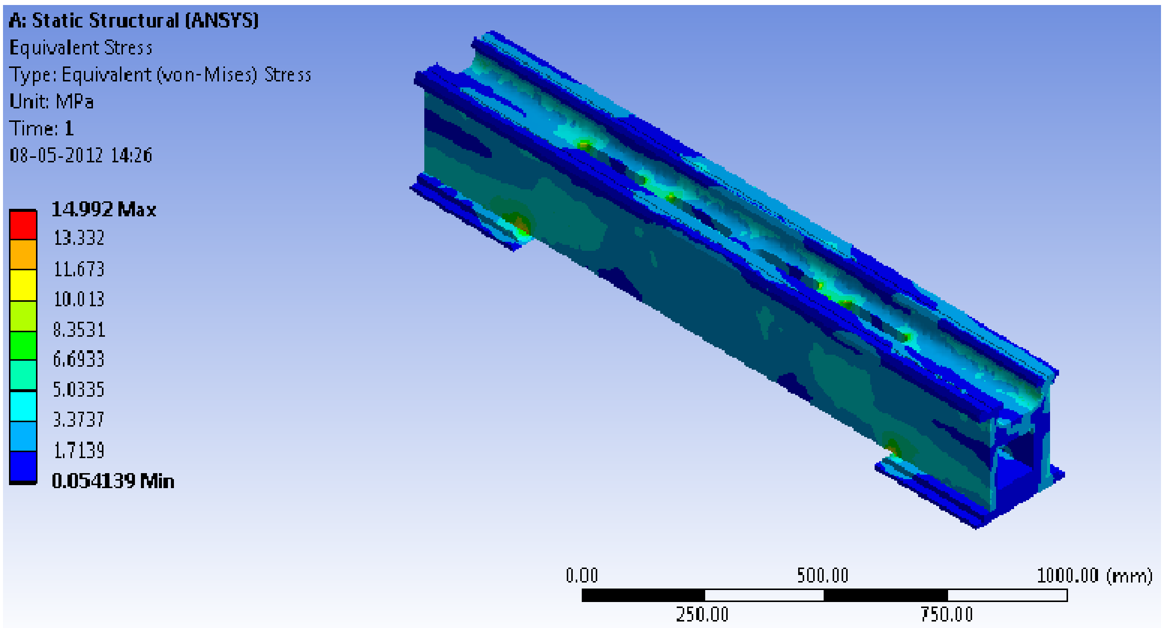

Figure 5.4: Proposed optimized von-Mises static stress

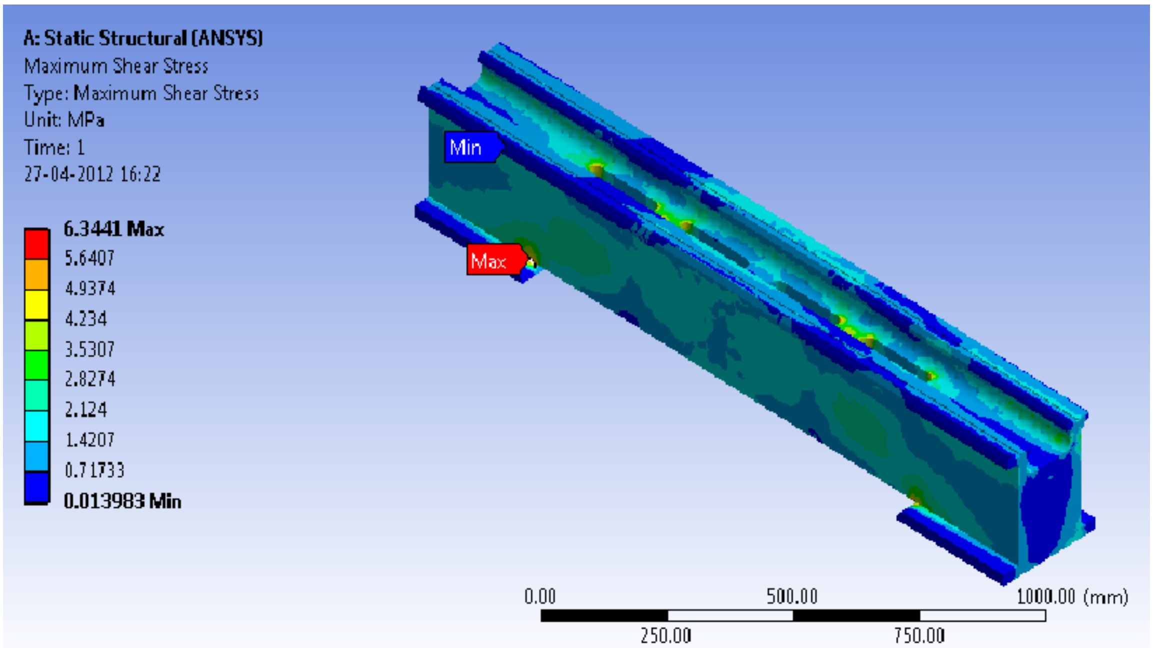

Figure 5.5: Present shear static stress

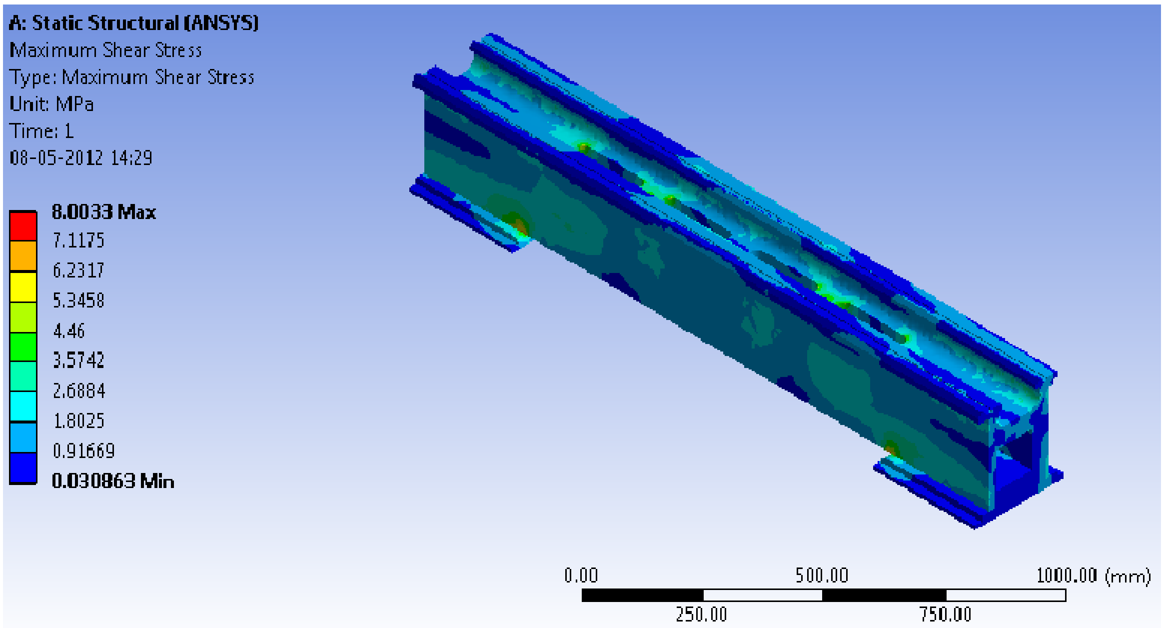

Figure 5.6: Proposed optimized shear static stress

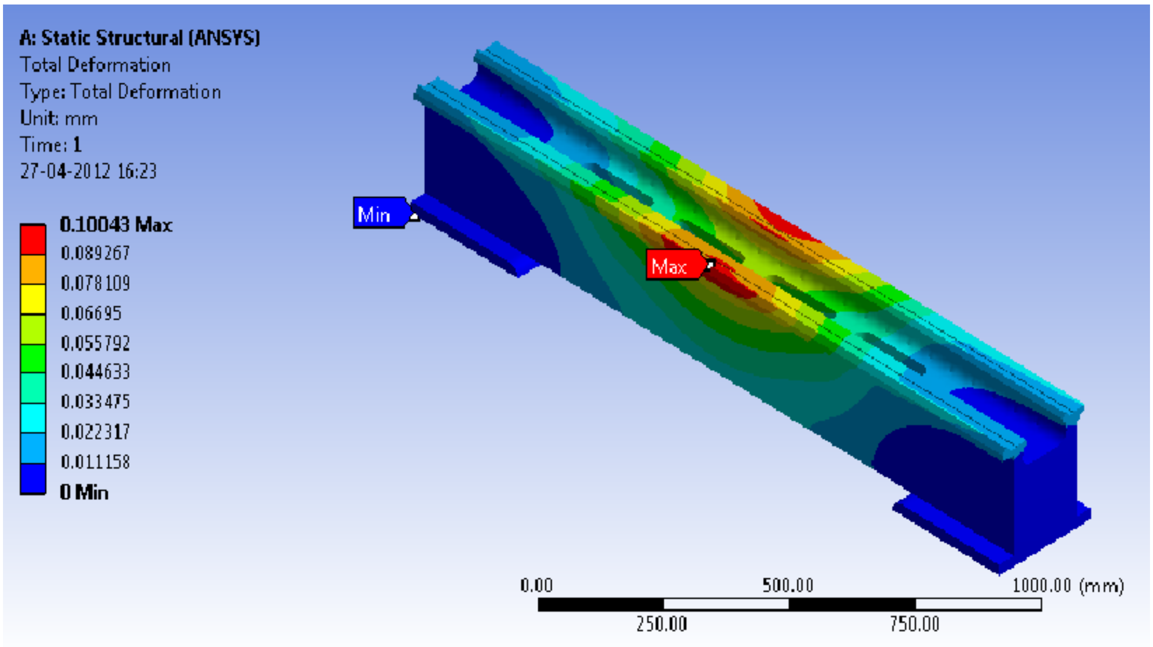

Figure 5.7: Present static deformation

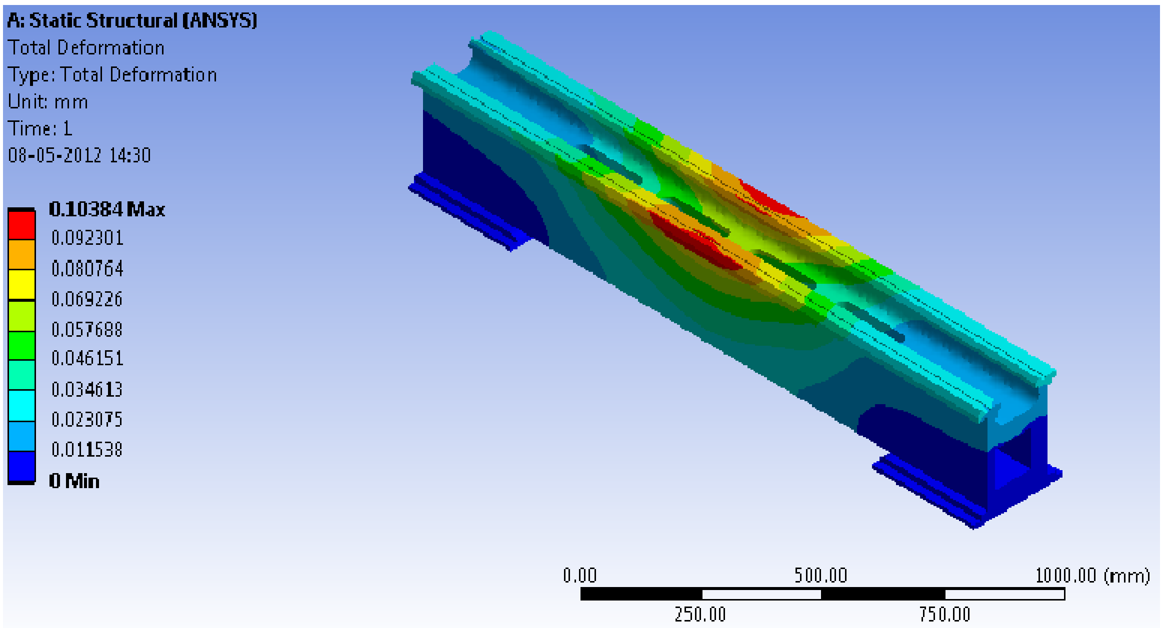

Figure 5.8: Proposed optimized static deformation

5.1.3 Comparison of results for transient structural analysis of optimized lathe bed

Comparison of von-Mises transient stress for present and proposed optimized lathe bed is shown in Fig. 5.9 & 5.10 and results are tabulated in Table 5.3. It is clear from the table that optimized design shows less amount of stress generation compare to present design. The transient results at 0.3 & 0.4 seconds reveals maximum stress generation. These maximum stresses reveals that carriage is passing through center of lathe bed where minimum support is available for load transfer. Similar comparison in transient shear stress and transient deformation are indicated in Fig. 5.11, 5.12, 5.13 & 5.14 respectively and their results are tabulated in Table 5.4 & 5.5 respectively. Result reveals considerable amount of decrease in transient shear stress and marginal increase in deformation. This results are well within tolerance limits of material as per ISO standards.

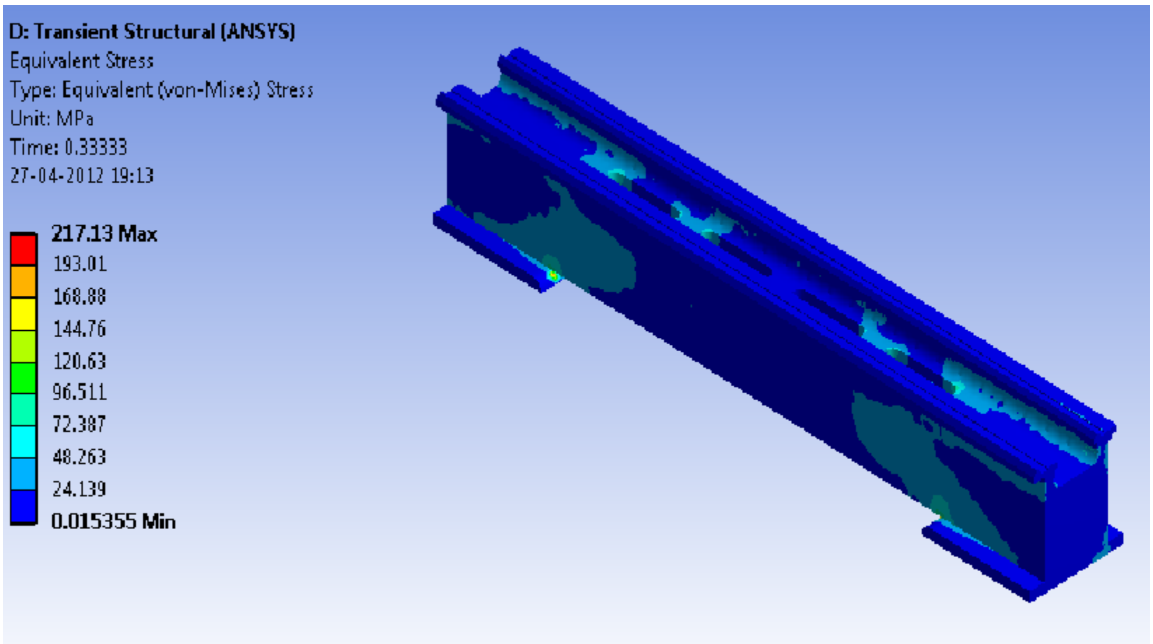

Figure 5.9: Present von-Mises transient stress

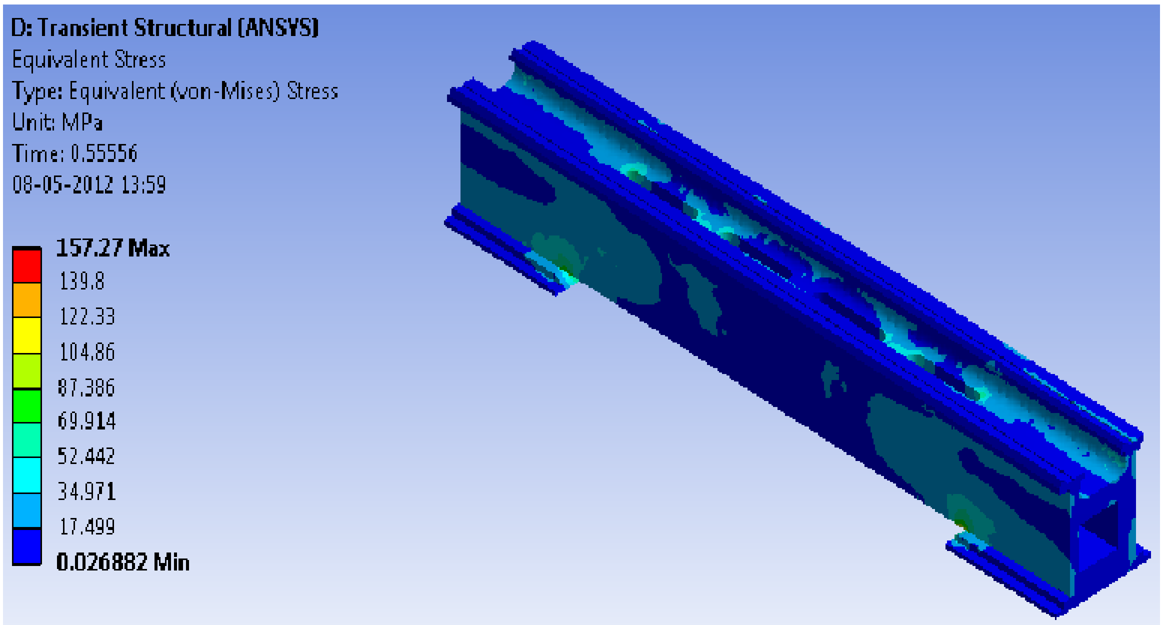

Figure 5.10: Proposed optimized von-Mises transient stress

Table 5.3: von-Mises transient structural stress results comparison for present Vs optimized proposed state

Time [s]	Present von-Mises stress		Optimized von-Mises stress		Maximum allowable stress [MPa]
	Minimum [MPa]	Maximum [MPa]	Minimum [MPa]	Maximum [MPa]	
0	0	0	0	0	
0.1	0.091307	136.91	0.29381	94.901	
0.2	0.14437	196.89	0.30219	137.42	
0.3	0.17658	217.98	0.57283	153.09	
0.4	0.18892	215.44	0.50213	160.16	240
0.5	0.17984	195.03	0.47946	153.66	
0.6	0.15277	160.23	0.4204	129.95	
0.7	0.12404	124.47	0.34668	103.99	
0.8	0.08891	92.713	0.22135	78.063	
0.9	0.051999	60.94	0.14626	52.098	
1	0.015355	29.338	0.02688	26.19	

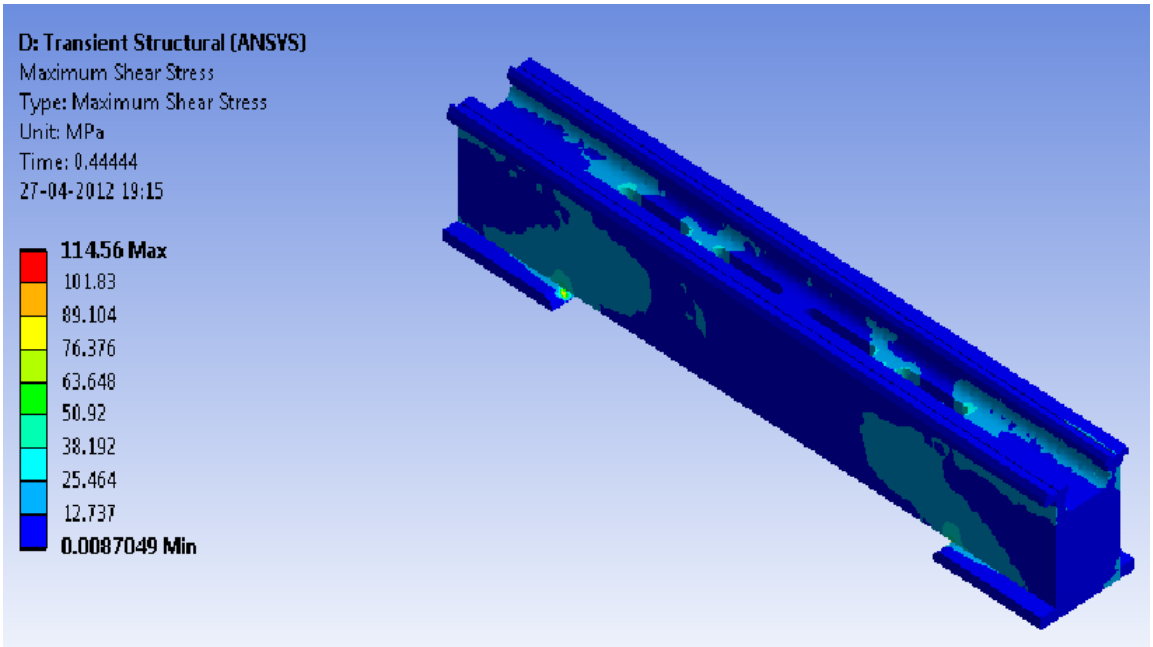

Figure 5.11: Present shear transient stress

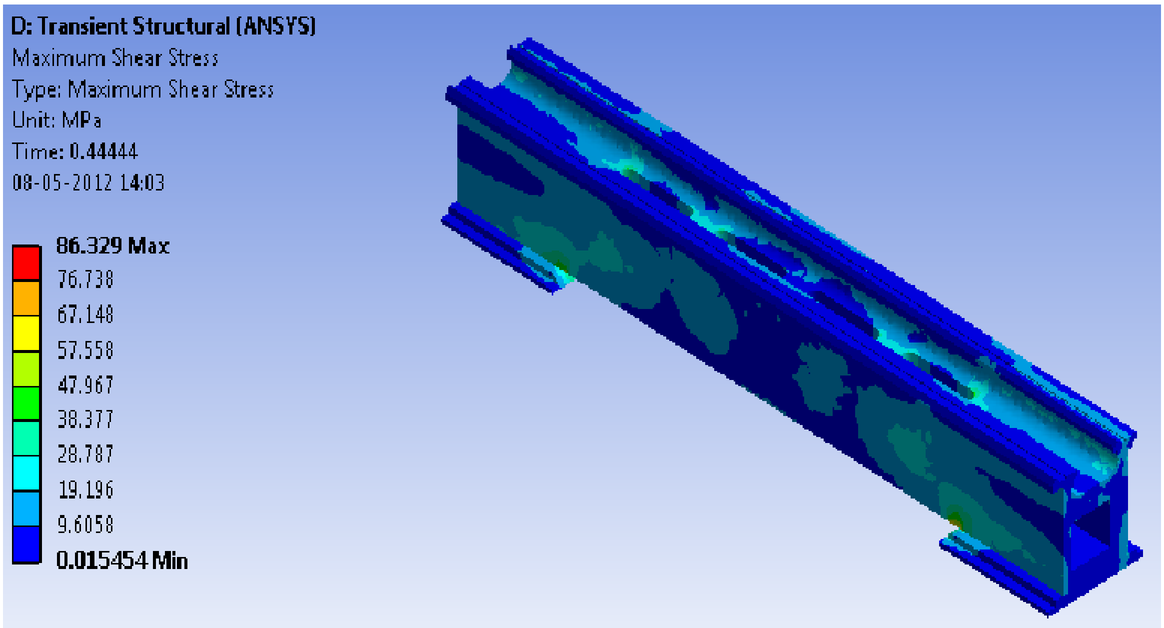

Figure 5.12: Proposed optimized shear transient stress

Table 5.4: Shear transient structural stress results comparison for present Vs optimized proposed state

Time [s]	Present shear stress		Optimized shear stress		Maximum allowable stress [MPa]
	Minimum [MPa]	Maximum [MPa]	Minimum [MPa]	Maximum [MPa]	
0	0	0	0	0	240
0.1	0.052508	72.357	0.167	51.093	
0.2	0.083156	103.95	0.16939	74.047	
0.3	0.10178	115.02	0.32088	82.58	
0.4	0.10895	113.64	0.28252	87.909	
0.5	0.10374	102.84	0.25536	84.354	
0.6	0.088131	84.481	0.22452	71.337	
0.7	0.071566	65.899	0.18881	57.082	
0.8	0.051165	49.092	0.12777	42.844	
0.9	0.029883	32.281	0.083065	28.585	
1	0.0087049	15.584	0.01545	14.359	

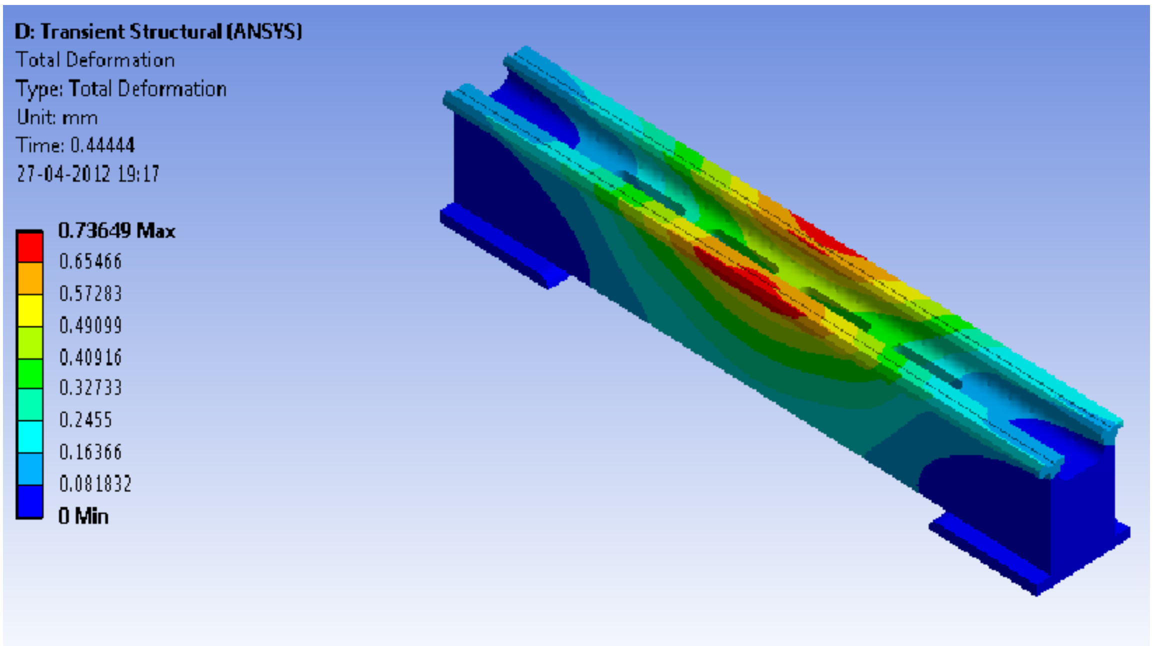

Figure 5.13: Present transient deformation

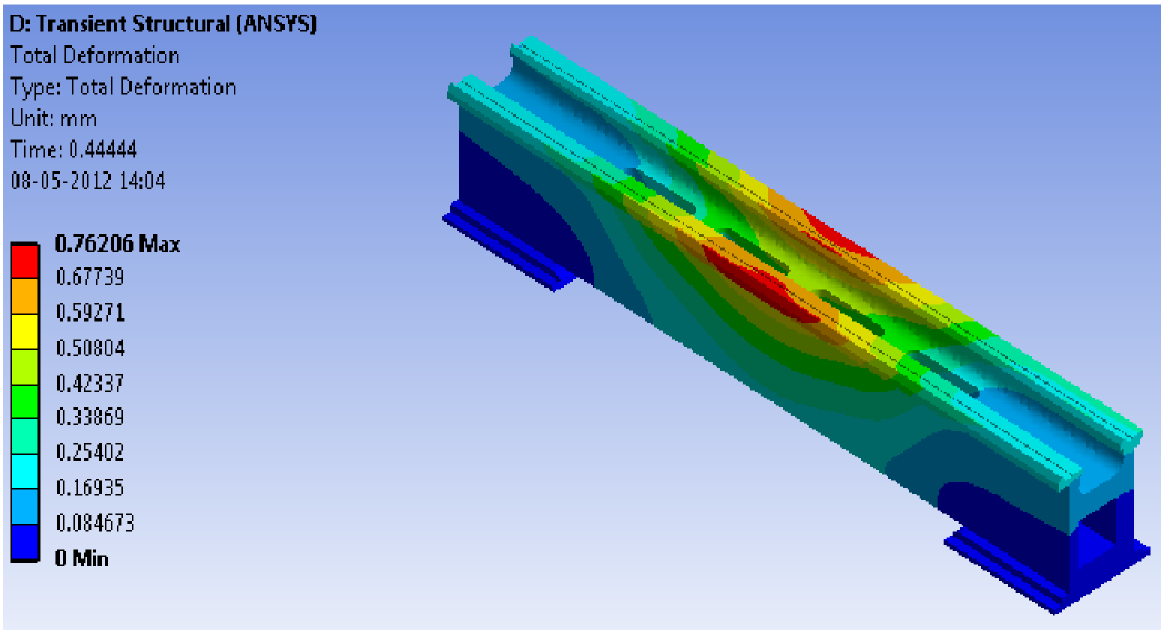

Figure 5.14: Proposed optimized transient deformation

Table 5.5: Transient deformation results comparison for present Vs optimized proposed state

Time [s]	Present deformation Maximum [mm]	Optimized deformation Maximum [mm]
0	0	0
0.1	0.3576	0.37363
0.2	0.591	0.61353
0.3	0.72084	0.74664
0.4	0.75695	0.78336
0.5	0.71117	0.73568
0.6	0.59002	0.61049
0.7	0.45293	0.46899
0.8	0.31625	0.32799
0.9	0.1799	0.18739
1	0.065968	0.070896

5.2 Present state Vs Proposed material state

5.2.1 Comparison of results for static structural analysis of lathe bed with Al-Cu alloy

The results of Al-Cu static structural analysis are compared with present state analysis for feasibility study. von-Mises static stress, shear static stress and static deformation for proposed material are shown in Fig. 5.15, 5.16 & 5.17. Present state comparison is already discussed earlier in section 5.1.2. Comparison of results in static stress is tabulated in Table 5.6.The results reveals that there is marginal increment in all stress and displacement fields results. This results are well within tolerance limits of material as per ISO standards. These results indicates that Al-Cu is suitable as lathe bed material.

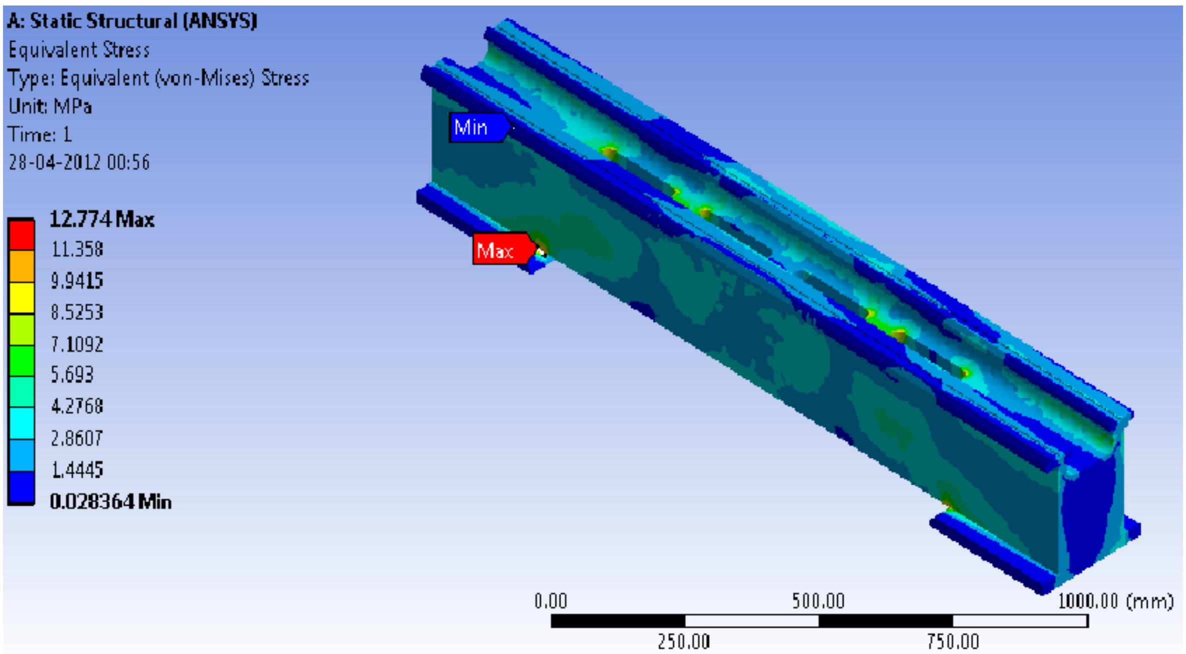

Figure 5.15: Proposed material static von-Mises stress

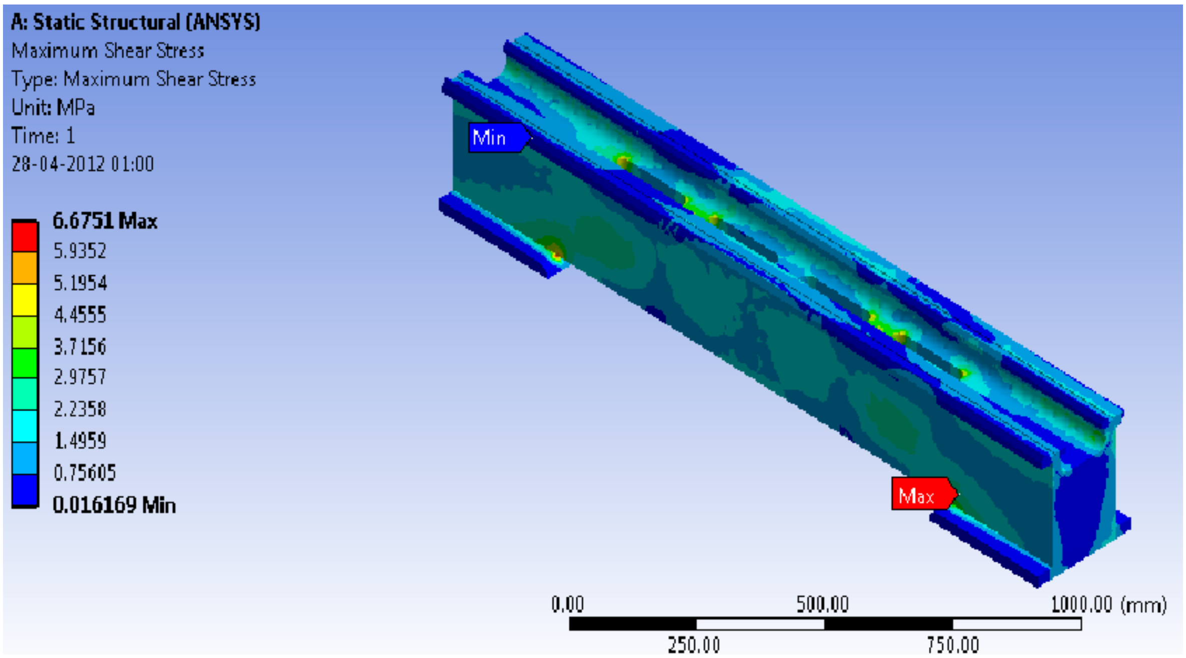

Figure 5.16: Proposed material static shear stress

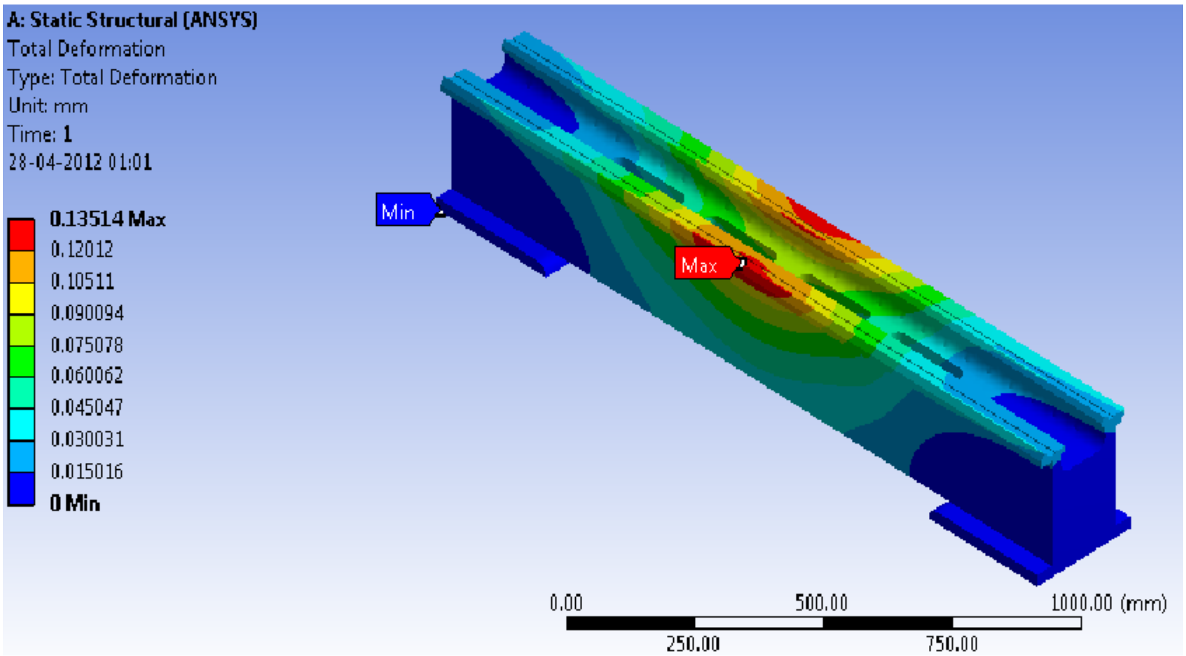

Figure 5.17: Proposed material static deformation

Table 5.6: Maximum and minimum stress results comparison for Present Vs Proposed base metal state

Type	von-Mises stress		Maximum shear stress		Total deformation	
	Present	Proposed	Present	Proposed	Present	Proposed
Minimum	0.024 MPa	0.028 MPa	0.013 MPa	0.016 MPa	0 mm	0 mm
Maximum	12.41 Mpa	12.774 MPa	6.34 MPa	6.67 MPa	0.1 mm	0.13 mm

5.2.2 Comparison of results for proposed transient structural analysis of lathe bed with Al-Cu alloy

The proposed material shows greater amount of flexibility in terms of weight reduction. The proposed material Al-cu alloy weights 63% less than the present material. These reduction in weight is mainly caused by density difference between two materials. The transient structural analysis of proposed material reveals marginal amount of stress increment in lathe bed compare to present state. The maximum amount of stress is generated between 0.3 & 0.4 second of time frame. This indicates that maximum force is generated while carriage reaches towards center of lathe bed. Transient von-Mises stress for lathe bed with Al-Cu alloy results are illustrated in Fig. 5.18 & 5.19. Similarly transient shear stress and deformation results are shown in Fig. 5.20, 5.21, 5.22 & 5.23 respectively. Results of all stresses in the form of table are illustrated in Table 5.7, 5.8 & 5.9.

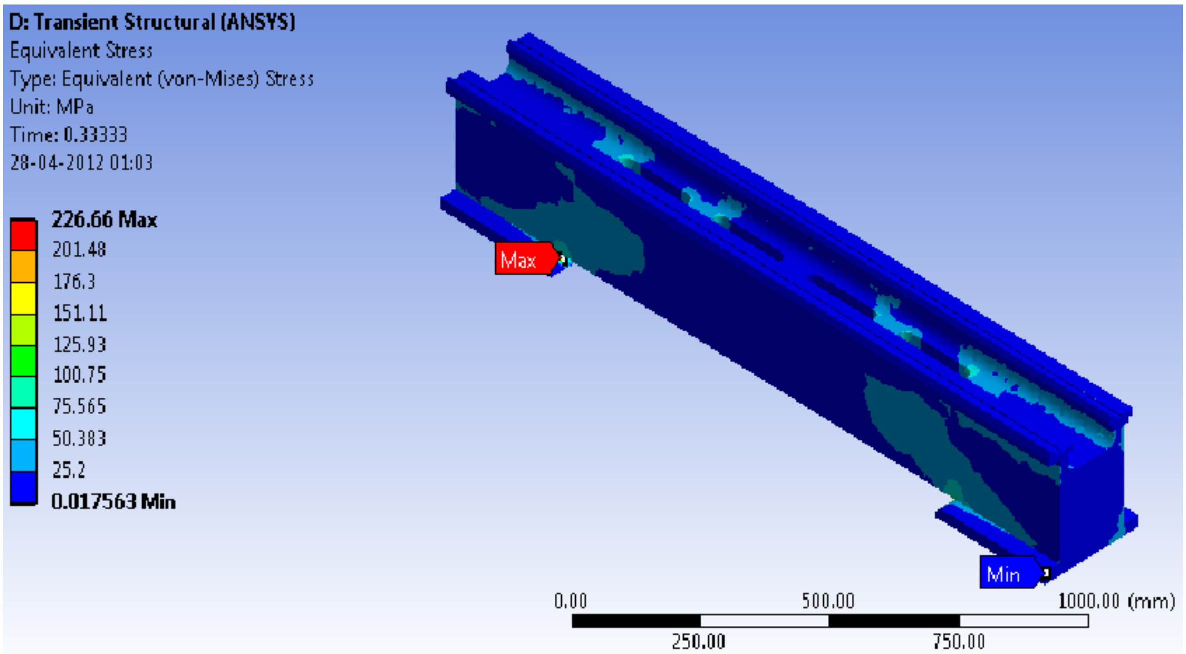

Figure 5.18: Transient von-Mises stress for lathe bed of Al-Cu alloy

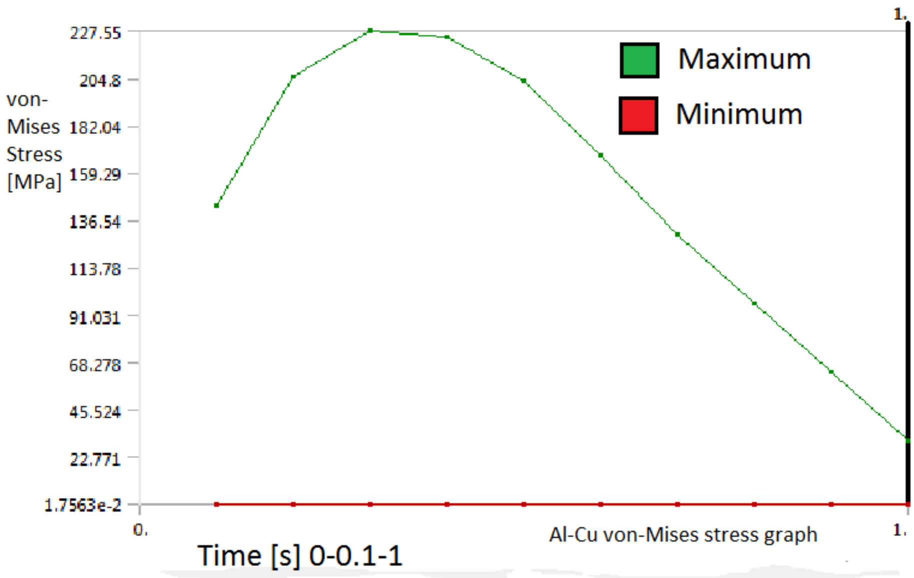

Figure 5.19: Transient von-Mises stress graph for lathe bed of Al-Cu alloy

Table 5.7: von-Mises transient structural stress results comparison for present Vs proposed state

Time [s]	Present von-Mises stress		Proposed von-Mises stress		Maximum allowable stress [MPa]
	Minimum [MPa]	Maximum [MPa]	Minimum [MPa]	Maximum [MPa]	
0	0	0	0	0	240
0.1	0.091307	136.91	0.10643	143.08	
0.2	0.14437	196.89	0.16871	205.57	
0.3	0.17658	217.98	0.20665	227.55	
0.4	0.18892	215.44	0.22132	224.88	
0.5	0.17984	195.03	0.21081	203.57	
0.6	0.15277	160.23	0.17913	167.27	
0.7	0.12404	124.47	0.1455	129.68	
0.8	0.08891	92.713	0.105	96.641	
0.9	0.051999	60.94	0.0611	63.539	
1	0.015355	29.338	0.01756	30.686	

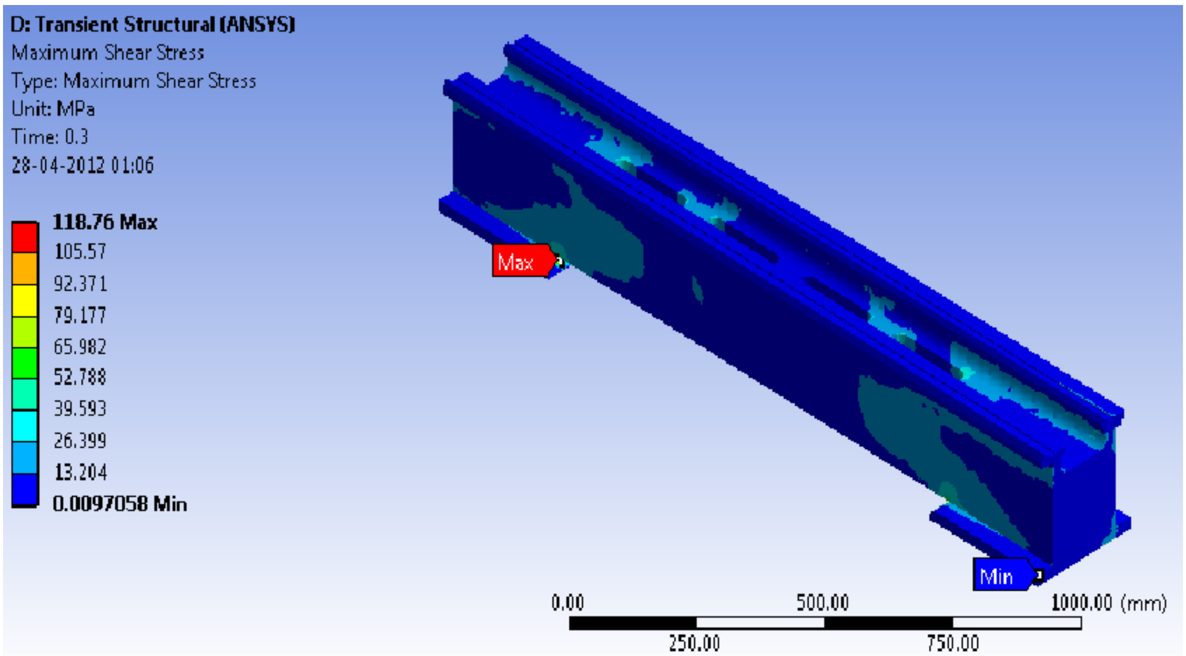

Figure 5.20: Transient shear stress for lathe bed of Al-Cu alloy

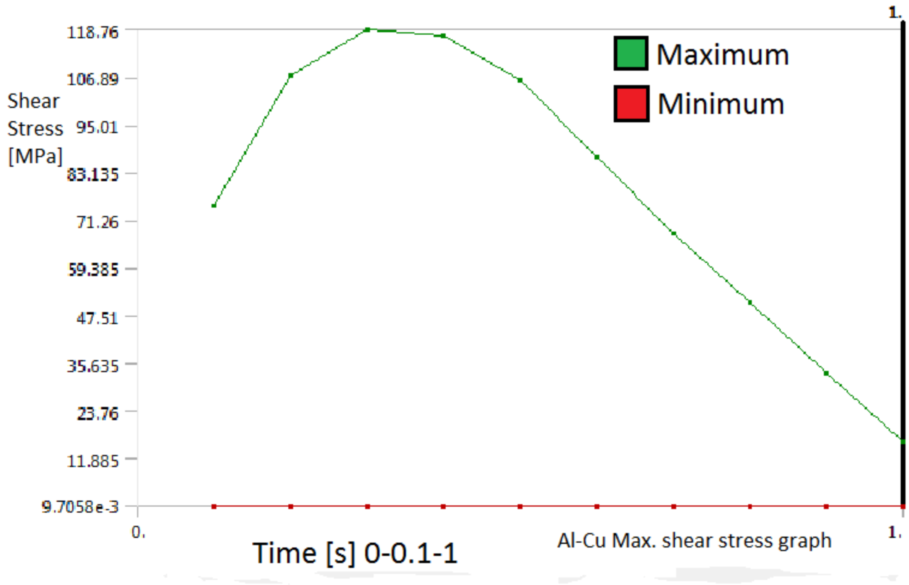

Figure 5.21: Transient shear stress graph for lathe bed of Al-Cu alloy

Table 5.8: Shear transient structural stress results comparison for Present Vs Proposed state

Time [s]	Present shear stress		Proposed shear stress		Maximum allowable stress [MPa]
	Minimum [MPa]	Maximum [MPa]	Minimum [MPa]	Maximum [MPa]	
0	0	0	0	0	240
0.1	0.052508	72.357	0.0602	74.789	
0.2	0.083156	103.95	0.0957	107.35	
0.3	0.10178	115.02	0.11744	118.76	
0.4	0.10895	113.64	0.12592	117.32	
0.5	0.10374	102.84	0.12004	106.17	
0.6	0.088131	84.481	0.10204	87.224	
0.7	0.071566	65.899	0.0829	68.064	
0.8	0.051165	49.092	0.05928	50.72	
0.9	0.029883	32.281	0.03446	33.38	
1	0.0087049	15.584	0.0097	16.145	

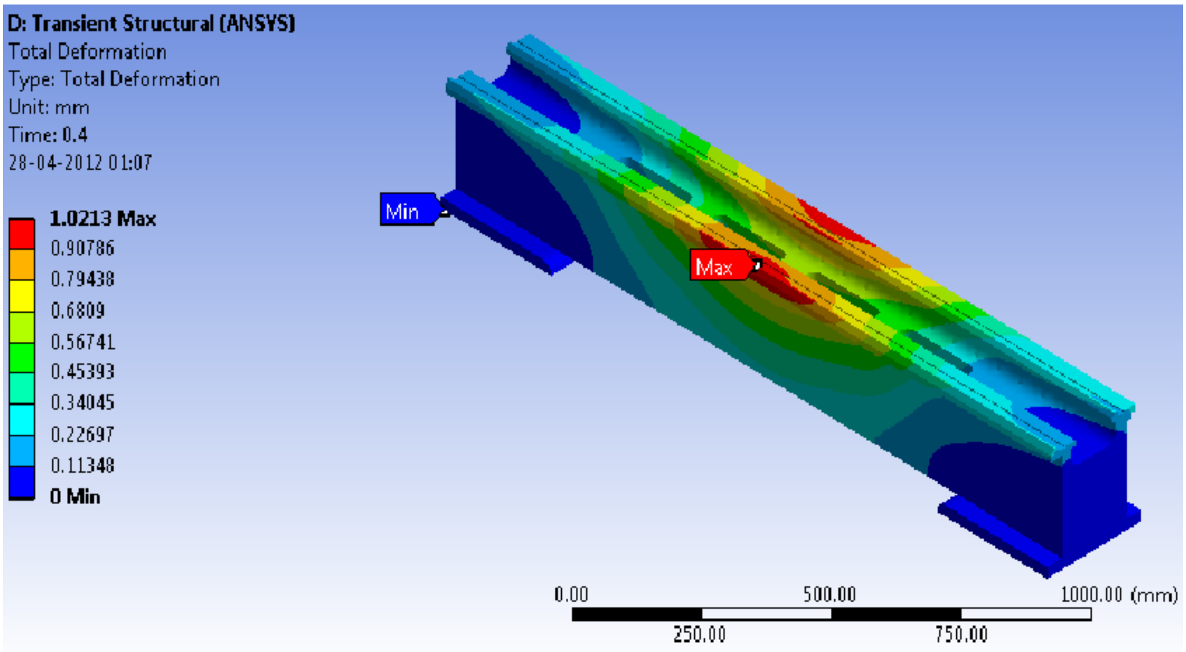

Figure 5.22: Transient deformation for lathe bed of Al-Cu alloy

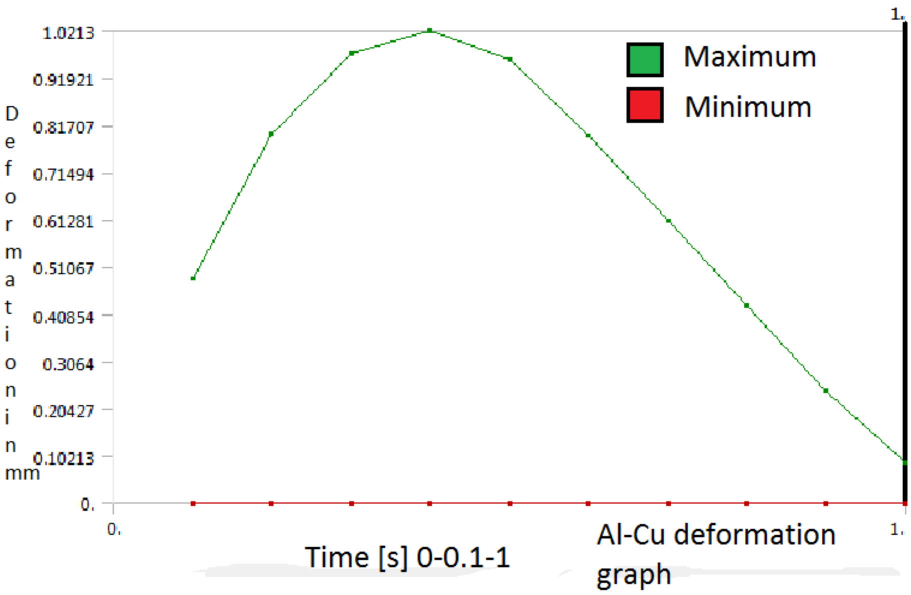

Figure 5.23: Transient deformation graph for lathe bed of Al-Cu alloy

Table 5.9: Transient deformation results comparison for Present Vs Proposed state

Time [s]	Present deformation Maximum [mm]	Proposed deformation Maximum [mm]
0	0	0
0.1	0.3576	0.48469
0.2	0.591	0.79895
0.3	0.72084	0.97325
0.4	0.75695	1.0213
0.5	0.71117	0.9594
0.6	0.59002	0.79601
0.7	0.45293	0.61145
0.8	0.31625	0.42727
0.9	0.1799	0.24364
1	0.065968	0.08954

Chapter 6

Conclusion & Future Work

6.1 Conclusion

From the thesis work following conclusion have been derived,

- Structural analysis of existing bed design has been carried out.
- Design optimization of existing bed has been performed and a lean design has been proposed.
- The stresses and deflection found in proposed design are at par with existing design.
- 10% weight reduction in proposed lean design has been reported.
- Further weight reduction could be achieved by replacing bed. Al-Cu alloy has been proposed to replace existing GCI material. 63% weight reduction is found with proposed material.
- Dynamic analysis was performed for lathe bed for existing design, proposed lean design and with proposed bed material in lean design.
- Modal analysis was performed to find out natural frequencies in different modes.

6.2 Future Work:

The dissertation work can be extended in following directions:

- Lean principle can be implemented for specific purpose of end user and optimum level of performance can be achieved.
- Miniature design of machine tools can be performed using same force calculations.
- Fatigue analysis can be done for more accurate results and to predict the accuracy and life of lathe bed.
- A prototype can be developed based on the outcome of theoretical analysis. Comparison between theoretical and actual results can be measured.

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