

DESIGN AND DEVELOPMENT OF A MISSILE PART FIXTURE

Submitted in partial fulfillment of the requirements for the award of

Bachelor of Engineering Degree in

Mechanical Engineering

By

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SATHYABAMA

INSTITUTE OF SCIENCE AND TECHNOLOGY

(DEEMED TO BE UNIVERSITY)

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This is to certify that this Project Report is the bonafide work of Lakkimsetti Dimanth (Reg. No.38150058), Polavarapu Mitranand (Reg. No. 38150079) who carried out the project entitled "**DESIGN AND DEVELOPMENT OF A MISSILE PART FIXTURE**" under my / our supervision from November 2021 to March 2022.

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This is to certify that this Project Report is the bonafide work of Lakkimsetti Dimanth (38150058) and Polavarapu Mitranand (38150079) who carried out the project entitled “DESIGN AND DEVELOPMENT OF A MISSILE PART FIXTURE” under my supervision from Nov’ 2021 to Mar’ 2022.

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DECLARATION

I LAKKIMSETTI DIMANTH (Reg No.38150058) hereby declare that the Project Report entitled “**DESIGN AND DEVELOPMENT OF A MISSILE PART FIXTURE**” done by me under the guidance of **Dr. G.ARUN KUMAR, M.E.,Ph.D., and A.SUDHEER RAJU, Scientist ‘F’ HOD. Intg, ANSP, DRDL Hyderabad** is submitted in partial fulfillment of the requirements for the award of Bachelor of Engineering in Mechanical Engineering.



DATE: 26/04/2022
PLACE: CHENNAI

SIGNATURE OF THE CANDIDATE

DECLARATION

I POLAVARAPU MITRANAND (Reg No.38150079) hereby declare that the Project Report entitled “**DESIGN AND DEVELOPMENT OF A MISSILE PAR FIXTURE**” done by me under the guidance of **Dr. G.ARUN KUMAR, M.E.,Ph.D., and A.SUDHEER RAJU, Scientist ‘F’ HOD. Intg, ANSP, DRDL Hyderabad** is submitted in partial fulfillment of the requirements for the award of Bachelor of Engineering in Mechanical Engineering.



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ABSTRACT

The objective of this project is to develop an integration fixture to assemble four components of a missile, developed by DRDO (Defence Research and Development Organisation). The mass of the components to be assembled are varying from 400Kg to 1000Kg. The mean working height of the fixture is to be maintained at 2094mm for specific requirement. All the components need to be assembled in the horizontal condition only.

The integration fixture proposed mainly comprises of rigid base assembly with linear motion guide and rail arrangement for longitudinal motion, assembly trolleys to support component-1 and component-4. The Integration fixture support, hold either by gravitational force or by physical fastening and locates the sections to be assembled. It locates the components with respect to the base assembly by means of linear motion guide and rails on handling trolleys.

The fixture has the provision to rotate the components positioned / fastened to it along longitudinal axis and linear motion along three axes. The rotational along longitudinal axis and linear motion along three axes are used to assemble four components of the missile. Design calculations of all load bearing members of integration fixture are presented. All the load bearing members of the fixture are rigid and have enough stability for operational safety.

The handling accessories required to assemble components to their respective handling trolleys are also presented. Design calculations of elements of the fixture and fixture body performed using classical equations supported by FEA results using Ansys wherever possible.

Expected inaccuracy of the assembly fixture and mechanism to correct the inaccuracy are also presented. Acceptance of the assembly fixture for design validation and to prove workmanship is presented in the last topic

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LIST OF SYMBOLS AND ABBEREVIATIONS

CAD	-	Computer-aided design
DRDO	-	Defence Research & Development Organisation
L	-	Length
Mm	-	Millimetre
Kg	-	Kilograms
C.G	-	Centre of gravity
M	-	Metric threads
PCD	-	Pitch Circle Diameter
Φ	-	Diameter

ID	-	Interior diameter
LM	-	Linear motion
2D	-	Two dimensional
3D	-	Three dimensional
Mc/I	-	Maximum bending stress
M	-	Maximum bending moment
I	-	Moment of inertia
$(WL^3/3EI)$	-	Maximum deflection
Cm	-	Centimetre
E	-	Young's modulus
Wt	-	Weight
Z	-	Section modulus
N	-	Newton
N/mm	-	Newton millimetre
F	-	Force
T	-	Torque
FEA	-	Finite element analysis
CAE	-	Computer aided engineering
σ_y	-	Yield stress
σ	-	Working stress
IS 2062 GR B	-	The micro-alloy that helps in manufacturing the micro- alloyed plates, sheets, and sections that are used in the industries.
En9	-	Medium carbon steel grade
HRC	-	Hardness Rockwell

CHAPTER 1

INTRODUCTION

1.1 FUNDAMENTALS OF FIXTURE DESIGN

Fixture design consists of a number of distinct activities: fixture planning, fixture layout design, fixture element design, tool body design, etc. They are listed in figure in their natural sequence, although they may be developed in parallel and not necessarily as a series of isolated activities in actual execution.

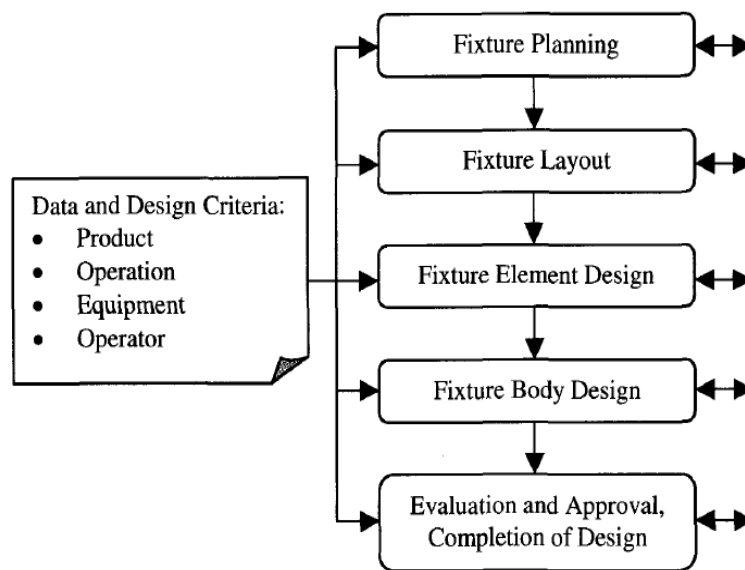


Fig :1.1 Fundamental of fixture desing

1.1.1 Fixture planning

The fixture planning deals with the establishment of the basic fixture concepts. It is to conceptualize a basic fixture configuration through analyzing all the available information regarding the material and geometry of the work piece, assembly operations required, handling equipment for the operations, and the operator. The following outputs are included in the fixture plan. Fixture type and complexity, Number of jobs per fixture, Orientation of work piece within fixture, Locating datum faces, Clamping surfaces & Support

1.1.2 Fixture layout

It is an embodiment of the concepts in the form of a spatial configuration of the fixture. It is to represent the fixture concepts in a physical form. The following outputs

are included in the fixture layout. Positions of locators, Positions of clamps, Positions of supports, Type of locators, Type of clamps, Type of supports Clamping forces and sequence

1.1.3 Fixture Element Design

Fixture element design is concerned with the concrete details of the locators, clamps and supports. It is either to detail the design drawings committed on paper or to create the solid models in a CAD system of the practical embodiment of the conceptual locators, clamps and supports. It is possible to use standard designs or proprietary components. The following outputs are included in the fixture element design. Detailed design of locators, detailed design of clamps, and detailed design of supports, if any

1.1.4 Fixture Body Design

Fixture body design produces a structure combining the fixture elements in the desired spatial relationship with the machine tool. Fixture body design is to produce a rigid structure carrying all the individual fixture elements in their proper places. Apart from the above, the following design criteria must be observed during the procedure of fixture Design. Design specifications, Factory standards, Ease of use and safety & Economy

1. 2 SPECIFICATION FOR FOLLOWING FOUR COMPONENTS

Four components of a particular missile developed by DRDO need to be assembled with the following specifications.

The geometry, weight and Center of gravity details of the components of a Missiles are as follows

Component -1:

Cylindrical geometry with $\Phi 1300\text{mm}$ and length of 450mm.

Mass of the Component is 750 Kg.

Center of Gravity is at center of the Component.

Component -2:

Conical geometry with diameter varying from $\Phi 1290\text{mm}$ to $\Phi 1062\text{mm}$ and length is 162.5 mm.

Mass of the component is 400 Kg.

Center of Gravity is at geometric C.G.

Interface to be used is 24 holes M10 on PCD 1027.

Component -3:

Conical geometry with diameter varying from $\Phi 1062\text{mm}$ to $\Phi 924\text{mm}$ and length is 174mm.

Mass of the component is 400 Kg.

Center of Gravity is at geometric C.G.

Interface to be used is 24 holes $\Phi 10.5$ on PCD 1027 and ID of $\Phi 960$ mm

Component -4:

Cylindrical geometry with diameter $\Phi 700\text{mm}$ and having flange at other end.

Mass of the component is 1000 Kg.

Center of Gravity is 450 mm from aft end.

Interface to be used is 24 holes M8 on PCD 900 with ID $\Phi 860$ mm.

1.3 CONDITIONS TO BE FOLLOWED BY THE ALL COMPONENTS DURING THE ASSEMBLY PROCESS

All the four components need to be assembled in the horizontal condition.

Each component needs to be supported, held and located suitably to a handling trolley. The geometry of the supports should be decided based on the bulk head geometry of the respective components. Components should be supported only on the / utilizing Bulk head features on either side.

The support for component-1 can be given anywhere along the section with Segmental support having 70° at the center.

The component on the handling trolley should be able to move along three axes and should be able to rotate along longitudinal axis for assembly.

The mean center line of the handling trolleys should be maintained at 2094mm.

Handling trolleys of the components should move on a common base platform which should have minimum 2500mm length and it should have leveling screw arrangement to take care of misalignment of the floor. Grouting of the fixture is not permitted.

All the handling trolleys should move on common base using rack and Pinion mechanism for controlled movement. Provision should be there to lock handling trolley at any point.

Self locking mechanism should be adopted for achieving rotation of the Component-4 along longitudinal axis. Component should not rotate back due to eccentricity of mass.

The integration fixture should be designed with safety factor of 4 on yield stress and should have enough rigidity to avoid undue deformation under Load.

The integration fixture should have enough stability for operational safety.

1.4 THE ABOVE REQUIREMENT OF FIXTURE IS DESCRIBED BELOW

The proposed solution of fixture design comprises of following stages.

1.4.1 Fixture planning

The basic fixture configuration has been analyzed using the available information regarding the material and geometry of the work piece, assembly operations required, handling equipment for the operations and the operator

The Result of fixture planning is

Fixture type & complexity: Integration fixture for assembly of four components of a missile.

Components to be handled are heavy, Mass of the components varying from 400 Kg to 1000 Kg, to be handled and assembled only horizontally. Component 2 & 3 should be handled on bulkheads and Component 4 should be handled using mounting flange. Longitudinal c. g of Component 4 is located at an offset of 450mm from mounting flange.

Number of jobs per fixture: four components per fixture.

Orientation of jobs within fixtures: Orientation of components on fixture should be such that 0° orientation is at top.

Locating datum faces: All the components should be located with respect to common base platform.

Clamping surfaces & supports: All the components should be supported in the horizontal condition. Clamping should be either by gravity / fastening arrangement.

1.4.2 Fixture Layout

The fixture concepts are represented in the physical form as shown in figures 2,3 & 4. The following are the output of fixture layout

1.4.2.1 Position and Type of locators

Location is by means of

LM guide and rail bearings at the base assembly level.

Guided roller at cross slide level.

By locating pin at screw jack mounting and cradle roller Mounting level.

1.4.2.2 Position and type of clamps & clamping forces

Clamping and supporting surfaces are same. Clamping force is by means of mechanical fastening using bolted joint for component-4 and by self weight for component-1.

1.4.2.3 Type and position of supports

Radial support for component1.End support for component 4 and hanging support for components 3 & 4.

1.4.2.4 Assembly sequence

Component 2 is assembled to Component 1.

Component 3 is assembled to sub-assembly of Components 1-2.

Component-4 is assembled to Sub-assembly of Components 1-2-3.

1.5 FIXTURE LAYOUT DESCRIPTION

The Proposed assembly fixture mainly comprises of the following

Fixed base frame: It is made from standard fabricated channels, beams and plates. The top side of the frame is machined. On this machined face, LM Rails and rack is fixed. The base frame is provided with screw jacks for leveling on the floor. The length of the base frame is 3500mm.

Movable trolley with cross slide for holding Component -1: A fabricated frame with LM Blocks fixed on bottom frame for movement along longitudinal axis. This motion is given by rack and pinion using extended handle attached to pinion. The fabricated frame will support and hold two no's of frames on top with wheels. These frames can be moved by +/- 50mm with the help of screw and nut along transverse direction.

The fabricated frames with wheels in turn support two pairs of screw jacks on top. The screw jacks can be raised and lowered manually by rotating the nut. The top ends of the screw jacks are connected to one more fabricated frame with pin and bush. On this frame two no's of rollers are fixed. These rollers are mounted for easy rotation.

On these pair of rollers, two no's of saddles are held for supporting and holding the base shroud at two places. The saddles can be rotated about center of the section by +/- 5° Using screw and nut.

Movable trolley for holding Component-4: A fabricated box frame with LM Blocks fixed on bottom frame for movement along longitudinal axis. This motion is given by rack and pinion using extended handle attached to pinion. This fabricated box frame will have a bore through which a shaft is fixed with two bearings. One end of the shaft will have adopter plate fitted to receive and holding Component4. Other end of the shaft will have worm and worm gear arrangement for rotation of the shaft.

The apparatus arranging manages the foundation of the essential installation ideas. It is to conceptualize an essential apparatus arrangement through breaking down all the accessible data with respect to the material and math of the work piece, gathering activities required, taking care of hardware for the tasks, and the administrator. The accompanying results are remembered for the apparatus plan.

Installation type and intricacy, Number of occupations per apparatus, Orientation of work piece inside installation, Locating datum faces, Clamping surfaces and Support

It is an epitome of the ideas as a spatial design of the installation. It is to address the apparatus ideas in an actual structure. The accompanying results are remembered for the installation format. Places of finders, Positions of braces, Positions of supports, Type of finders, Type of cinches, Type of supports Clamping powers and grouping Fixture element design is concerned with the concrete details of the locators, clamps and supports.

It is either to detail the design drawings committed on paper or to create the solid models in a CAD system of the practical embodiment of the conceptual locators, clamps and supports. It is possible to use standard designs or proprietary components. The following outputs are included in the fixture element design. Detailed design of locators, detailed design of clamps, and detailed design of supports, if any Installation body configuration delivers a construction consolidating the apparatus components in the ideal spatial relationship with the machine instrument. Apparatus body configuration is to create an inflexible design conveying all the singular installation components in their legitimate spots. Aside from the abovementioned, the accompanying plan models should be seen during the technique of apparatus design. Plan determinations, Factory guidelines, Ease of purpose and safety and Economy.

Every part should be upheld, held and found reasonably to a taking care of handling trolley. The math of the backings ought to be concluded in view of the mass head calculation of the particular parts. Parts ought to be upheld just on the/using Bulk head highlights on one or the other side.

The help for part 1 can be given anyplace along the segment with Segmental help having 70° at the middle.

The part on the dealing with streetcar ought to have the option to move along three tomahawks and ought to ready to pivot along longitudinal hub for gathering.

The mean community line of the dealing with handling trolley ought to be kept up with at 2094mm.

A created outline with LM Blocks fixed on base casing for development along longitudinal pivot. This movement is given by rack and pinion utilizing stretched out handle connected to pinion. The created casing will support and hold two no's of casings on top with wheels. These edges can be moved by $\pm 50\text{mm}$ with the assistance of screw and nut along cross over course.

The created outlines with wheels thus support two sets of screw jacks on top. The screw jacks can be raised and brought down physically by turning the nut. The top closures of the screw jacks are associated with another manufactured edge with pin and shrub. On this outline two no's of rollers are fixed. These rollers are mounted for simple revolution.

On these sets of rollers, two no's of seats are held for supporting and holding the base cover at two spots. The seats can be turned about focus of the segment by Using screw and nut.

CHAPTER 2

LITERATURE SURVEY

2.1 Trappey J.C., Liu C.R., A literature survey of fixture design automation, International journal of advanced Manufacturing technology [1] Summarizes that modular fixture design adheres to two basic procedures. First the functions and shapes of modular elements are defined and complete versatility of each shape is explored. Second, the right modular elements are selected based upon their geometrical and topological characteristics and specific functions of the fixture assembly. The criteria for component selection include the simplicity for the fixture design, the complete restriction of the work piece, the prevention of work piece bending and cutter collision, the ease of loading/ unloading and setting- up and the machining accuracy. A comprehensive fixture is constructed by selected components at the right locations, based on fixture design principles.

2.2 Carr Lane Mfg co., Modular fixturing Handbook 1991 [2] explains that the tool design process for modular fixtures is basically the same as for conventional fixtures with a few differences. They are generally three ways to design a modular fixture. a) Designing by actually binding the fixture b) By preparing manual drawings and c) using a CAD system .The first method is possible because modular fixture require only assembly of modular components. Thus, once the designer have a rough design concept in mind, can build up the fixture around the sample part. Simply place the sample part on the base plate and add locators and clamps. The second method can be speeded up by using tracing templates. Thus, when no sample part is available, the fixture can be assembled around the template of the part. Templates are also useful to check for interference that occurs when loading / unloading the part. The Third method is the most effective means for modular fixture. A database representing all the elements of the modular fixture system in either 2D or 3D is created by the user using a commercially available CAD system.

2.3 Khurmi, R.S. and Gupta, J.K. (2005) A Textbook of Machine Design. 14th Edition, Eurasia Publishing House (PVT.) Ltd, Ram Nagar, New Delhi [3] In this study, the thermo-elastic effects of frictional heat generation in a disc brake system due to braking actions were simulated. The mathematical model that defined the

problem was developed from the kinetic and potential energies of moving vehicles on the gradient surfaces. This problem was solved for the selected geometry of disc brake and pad with their material properties selected from existing literatures using the finite element method and the computational results were obtained. The thermal deformation obtained was in good agreement with similar literature results. Also, for the same braking period and conditions, the results showed that a vehicle ascending a hill gave a higher temperature rise, Von Mises stress and thermal deformation on brake contact surfaces than when descending hill. Therefore, the braking period required to bring a moving vehicle in ascendent motion to a lower speed is expected to be shorter because of the gravity effect than horizontal motion, while descendent motion requires longer braking period.

2.4 Joseph E Shigley and Larry D Mitchell Mechanical Engineering Design, Journal of Applied Science and Advanced Technology 2019 [4] Some of the damage that occurs in the structure and construction of engine elements that experience dynamic loading at the time of operation is caused by fatigue (fatigue) which occurs without beginning signs so that it causes great damage and loss. Fatigue strength is the process of local changes in the structure that occur permanently in accordance with the period of time caused by conditions where stress or strain occurs which results in cracks or fractures after being subjected to repetitive loads. Surface roughness factors that affect fatigue are applied in the analysis and application of machining, which are finely grinded and rough machined. Analysis was carried out on 6063-TF aluminum alloy material which is widely used for motor vehicles. To find out the effect of the above, the fatigue strength test is carried out with a Rotary Bending Fatigue Test to obtain a cycle value or fatigue age and certain neighbors so that it can make a semi-log S-N diagram with statistical calculations.

2.5 Thirupati Chandra Patla, Belugundu, Finite Element Methods ,PTC,1985. [5] Introduction to Finite Elements in Engineering By Tirupathi R. Chandrupatla, Ashok D. Belegundu- Introduction to Finite Engineering is ideal for senior undergraduate and first-year graduate students and also as a learning resource to practicing engineers. This book provides an integrated approach to finite element methodologies. The development of finite element theory is combined with examples and exercises involving engineering applications. The steps used in the

development of the theory are implemented in complete, self-contained computer programs. While the strategy and philosophy of the previous editions has been retained, the Fourth Edition has been updated and improved to include new material on additional topics.

2.6 ANSYS Guidance manual, ANSYS Inc, 2006 [6] Finite Element Modeling and Simulation with ANSYS Workbench 18, Second Edition, combines finite element theory with real-world practice. Providing an introduction to finite element modeling and analysis for those with no prior experience, and written by authors with a combined experience of 30 years teaching the subject, this text presents FEM formulations integrated with relevant hands-on instructions for using ANSYS Workbench 18. Incorporating the basic theories of FEA, simulation case studies, and the use of ANSYS Workbench in the modeling of engineering problems, the book also establishes the finite element method as a powerful numerical tool in engineering design and analysis.

2.7 Markus A, Markus Z, Farkas J and Filemen J, “Fixture design using Prolog: an expert system”, Robotics and Computer-Integrated Manufacturing, 1(2), pp. 167–172, 1984 [7] A program for designing fixtures to be built out of a set of given elements has been implemented. Its input is a description of the shape of the workpiece, the machining and the co-ordinates of the support and of the clamps. The program generates a sequence of blueprints of the fixtures. We describe the relevant features of Prolog and the appropriate use of logic programming in computer-aided engineering.

2.8 Ohwovoriol M.S and Roth R, “An extension of screw theory”, Journal of Mechanical Design. 103(4), pp. 725–735 1981 [8] It is shown that, if one considers the “total freedom” of instantaneous motion, three distinct types of screw systems arise. Of these only the ones which maintain the kinematic constraints—the so-called reciprocal screw systems—have been studied before. This paper introduces and studies the two new types, which we name “repelling” and “contrary” screws. We present a detailed theoretical study of the repelling screw systems, these as well as the other screw systems are useful in the mathematical modelling of the parts assembly process and the study of direct contact mechanisms.

2.9 Grippo P. M , Gandhi M .V and Thompson B .S , “The computer-aided design of modular fixturing systems”,*International Journal of Advanced Manufacturing Technology*,2(2), pp. 75–88, 1987 [9] The flexibility of modular fixturing systems can be exploited to significantly enhance the productivity of computer integrated manufacturing facilities. A major impediment to the implementation of this philosophy is the absence of viable methodologies for the automated design and assembly of these fixturing systems. This paper addresses this technological void by proposing a CAD/CAM tool for the automated design and robotic assembly of modular fixturing systems which can form the basis for a fully automated design system. The development of automated flexible fixturing technologies that transcend the traditional time-consuming and expensive practice of manual assembly would significantly contribute to the evolution to the factory-of-the-future.

2.10 Miller A.S and Hannam R .G , “Computer aided design using a knowledge base approach and its application to the design of jigs and fixtures”,*Proceedings of the Institution of Mechanical Engineers*,199(B4), pp. 227–234, 1985 [10] The design of jigs and fixtures is generally on the critical path of a new product and thus speed is essential in both their design and subsequent manufacture. They must also be well and economically designed. This paper describes an interactive C ADC AM based jig and fixture design procedure which has been developed using a knowledge base which could subsequently become part of an expert system.

2.11 Hoffman E .G ,*Jig and Fixture Design*, 2nd edition , Delmar Publishers, Albany, NY, 1985 [11] By emphasizing similarities among types and styles, *Jig and Fixture Design*, 5E speeds readers to a complete understanding of the why's and how's of designing and building a variety of different workholders for manufacturing. From simple template and plate-type jigs to complex channel and box-type tooling, this newly revised edition features more than 500 illustrations of tools and applications to spur readers to success

2.12 Colbert J.L, “Workholding technology in flexible manufacturing and the design of a modular fixturing system”, MS thesis, Rensselaer Polytechnic Institute, Troy, NY, 1985 [12] This paper gives a review of fixture-design research, most of it done in the 1980s. The major topics of the review are the fixturing principles (supporting, locating and clamping), automated fixture design (configuration, assembly and verification), and fixture hardware design (dedicated, modular and electric/magnetic types).

2.13 Gordon S.J and Seering W.P, “Reconfigurable assembly fixtures”, Proceedings, Symposium on Manufacturing Systems—Design, Integration, and Control, Manufacturing International Conference, Atlanta, Ga., pp. 91–96, 1988 [13] A fixture is a production tool that locates, and holds a workpiece securely so that the required manufacturing operations can be performed. Fixtures can be classified as dedicated or reconfigurable. Dedicated fixtures generally imply that they have been designed for a specific workpiece geometry and/or manufacturing operation. Most fixturing systems reviewed in this paper had been either developed for a specific purpose or limited in number of applications. Sensor integration has always been considered as the last step, or completely ignored due to the complicated designs of the systems. The objective of our research was therefore set as 'to develop a modular fixturing system (hardware/software) with some built-in flexibility, for robotic assembly (in relatively clean environments)' with the following basic features: modularity, automatic reconfigurability, sensory feedback, and programmability through a computer interface. This paper presents the results of this research.

CHAPTER 3

AIM AND SCOPE OF THE PRESENT INVESTIGATION

Aim

The main aim of this project is to develop and design a missile integration part fixture to assemble four components of a missile, developed by DRDO. The mass of the components to be assembled are varying from 400Kg to 1000Kg. The mean working height of the fixture is to be maintained at 2094mm for specific requirement. All the components need to be assembled in the horizontal condition only.

SCOPE

The scope of this work is to design and develop an integration fixture to assemble components of a missile integration fixture proposed mainly comprises of rigid base assembly with linear motion guide and rail arrangement for longitudinal motion fixture has the provision to rotate the components positioned / fastened to it along longitudinal axis and linear motion along three axes.

Design calculations of elements of the fixture and fixture body performed using classical equations supported by FEA results using Ansys Software wherever possible. Linear motion guide and rails selected for longitudinal motion of assembly trolleys of the fixture are having low coefficient of friction and hence the trolleys can be moved with little effort. Expected inaccuracy of the assembly fixture arising from the dimensional and geometrical tolerances specified on assembly/ supporting trolleys and its correction mechanism is also presented. Acceptance of the assembly fixture for design validation and to prove the workmanship is presented in the last chapter.

CHAPTER 4

MECHANICAL DESIGN CALCULATIONS OF FIXTURE ELEMENTS

4.1 FIXTURE ELEMENT DESIGN

4.1.1 Component-4 Assembly /Handling trolley

4.1.1.1 Shaft supporting Component-4

Component 4 is attached to mounting plate which is in turn is attached to Supporting shaft. The supporting shaft is connected to handling trolley for Component4 by means bearings at either ends.

Mass center of component-4 is acting at a distance of 695 mm from shaft. The mass of mounting plate is 240 Kg acting at a distance of 300 mm from Shaft.

The shaft is assumed cantilever beam with mass of component-4 attached at its free end and mass of mounting plate acting at a

distance of 300mm from Shaft respectively as shown in above figure

Maximum Bending stress occurs at fixed end.

Diameter of the shaft = 130 mm.

Sectional modulus = $\pi d^3/32 = 215690 \text{ mm}^3$

Bending Moment at fixed end = $(1000 \times 695 + 240 \times 300) \times 9.81$

$$= 7524270 \text{ N-mm}$$

Maximum Bending stress = $M/Z = 34.88 \text{ N/mm}^2$

The material of the shaft is En9 in hardened and tempered 30-34 HRC. For En9 material (hardened & tempered cond) yield stress = 490.5 N/mm^2

Factor of safety = $490.5/34.88 = 14$

Point load at the free end of the shaft corresponding to weights of

Component-4 and its mounting plate is given by

Load free end = $(1000 \times 695 + 240 \times 300) / 150 = 5114 \text{ Kg}$.

Maximum Deflection of the shaft at the free end = $(WL^3/ 3EI)_{5114\text{Kg}}$

Moment of inertia of the shaft (I) = $\pi d^4 / 64 = 14019848 \text{ cm}^4$

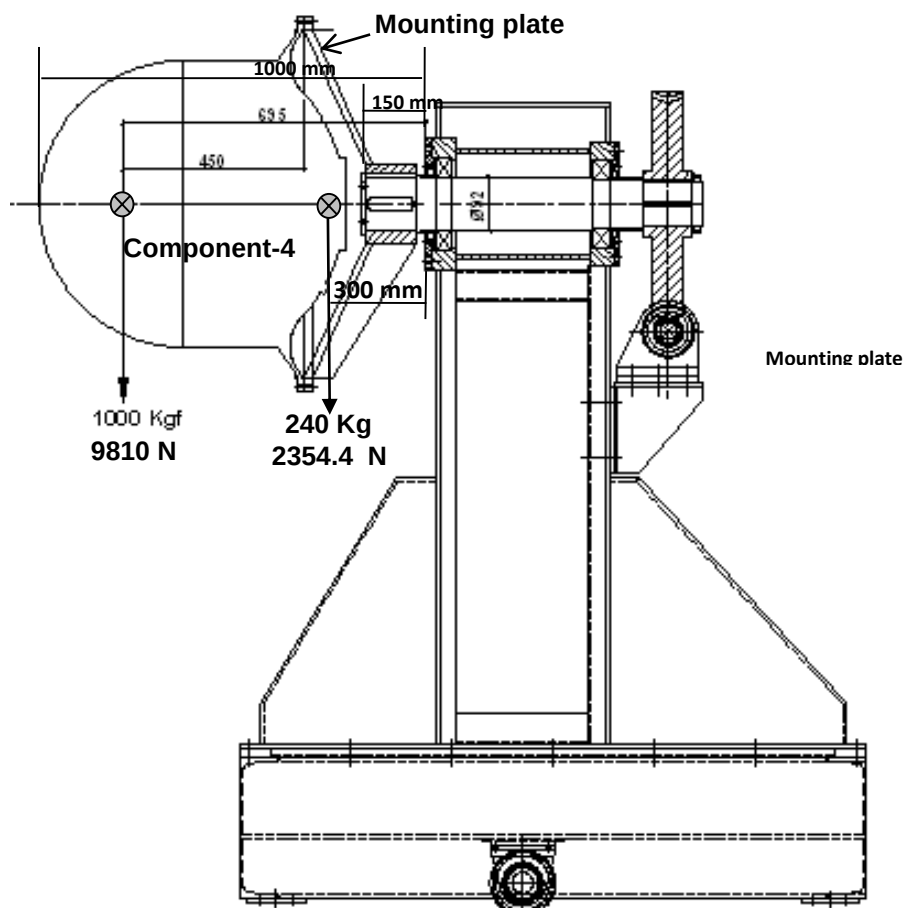
Deflection $Y_{\max} = (5114 \times 9.81 \times 150^3 / (3 \times 2.1 \times 10^5 \times 14019848)) \times 1000 \text{ Kgf}$
 $= 0.01917 \text{ mm.}$

Slope of the shaft due to this deflection at fixed end

$= \tan^{-1}(0.01917/150) = 7.322 \times 10^{-3}^\circ$

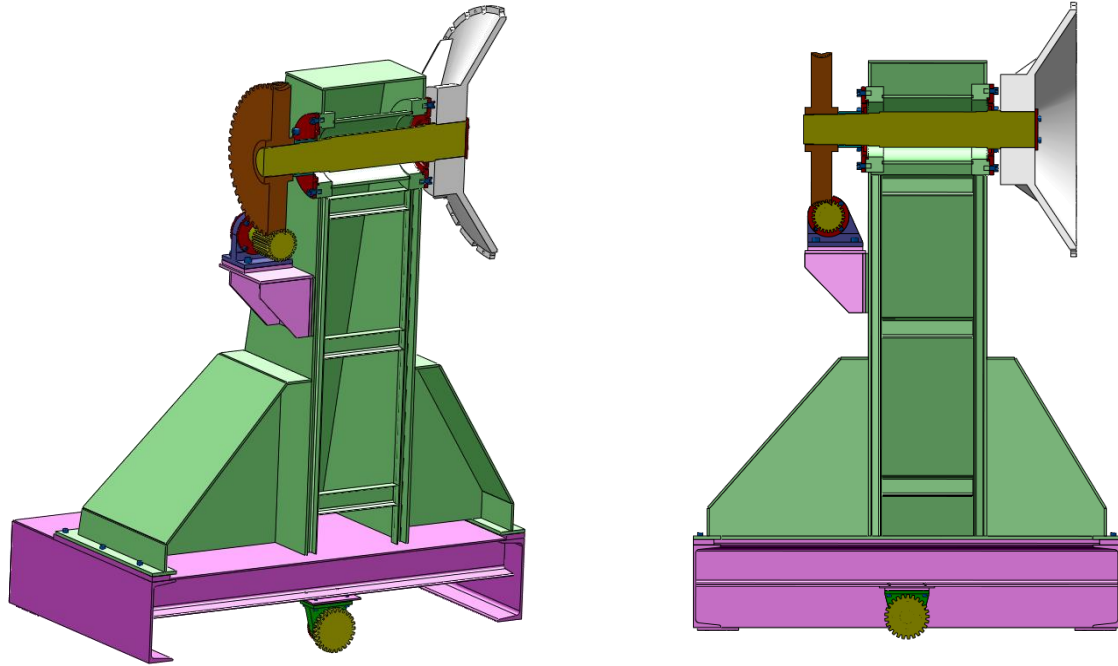
Deflection of Component-4 due to the above slope

$= 1000 \times \tan(7.322 \times 10^{-3}) = 0.1278 \text{ mm.}$



Component-4 Assembly /Handling trolley

Fig:4.1.1.1(a): Shaft supporting component-4



Component-4 Assembly /Handling trolley

Fig:4.1.1.1(b): Shaft supporting component-4

4.1.1.2 Mounting plate of Component-4

Bending of Mounting Plate

Mounting plate is assumed as a cantilever beam with mass of component 4 is acting at a distance of 450mm at its free end

The distance between free end to fixed end is 150 mm.

Load on Mounting plate due to mass of component-4= 1000 Kg

Bending Moment due to external load at fixed end= $1000 \times 570 \times 9.81$

$$(M1) \quad \quad \quad = 5591700 \text{ N-mm}$$

Mass of mounting plate-4 is 240 kg acting at a distance of 100 mm from fixed end

Bending Moment due to self weight at fixed end $(M2) = 240 \times 100 \times 9.81$

$$= 235440 \text{ N-mm}$$

Total bending moment $(M) = M1 + M2 = 5827140 \text{ N-mm}$

Moment of inertia of Mounting plate $(I) = 113.2 \text{ E6 mm}^4$ at the fixed End

Section Modulus of the Mounting plate $(Z) = 905.66 \text{ E3 mm}^3$ ($y=125\text{mm}$)

Bending Stress induced in the Mounting plate = $M/Z = 6.43 \text{ N/mm}^2$

acts at fixed end of the mounting plate.

Factor of safety = yield stress / working stress

Material used is Structural steel IS: 2062 Gr'B' , where

yield stress = $25 \text{ Kg/mm}^2 = 245.25$

Factor of safety = $245.25/6.43 = 38$

Deflection of mounting plate

Mounting plate is assumed as a cantilever beam with component-4 acts at a distance of 600 mm from fixed end

Since load of 9810 N (1000 Kg) acts at a distance of 600mm from fixed end, Corresponding point load at free end is given by

$1000\text{Kg} \times 600\text{mm}/150\text{mm} = 4000 \text{ Kg}$

Self mass of the mounting plate is 240 Kg act at a distance of 100 mm from fixed end.

Maximum deflection of cantilever beam due to point load = $(WL^3/3EI)$

Occurs at free end

Deflection of mounting plate

$= (WL^3/3EI)_{4000\text{Kg}} + (WL^3/3EI)_{240\text{Kg}}$

$= (4000 \times 9.81 \times 150^3 / 3 \times 2.1 \times 10^5 \times 113.2\text{E}6) + (240 \times 9.81 \times 100^3 / 3 \times 2.1 \times 10^5 \times 113.2\text{E}6)$

$= 0.001857 + 0.000033 = 0.00189 \text{ mm}$

Bending Stress in fillet weld

Mounting plate is welded to its hub by means of fillet weld as shown in figure (above)
Bending stress in fillet weld is given

by $4xM / (\pi \times t \times d^2)$.

Where M= Bending Moment, t= Throat thickness & d = Diameter of the shaft

This Bending stress occurs along the leg of the fillet weld.

The maximum bending stress occurs on the throat of the weld which is inclined at 45° to the horizontal plane.

length of throat = size of leg $\times \sin 45^\circ = 0.707s$

Where s= size (leg) of the weld.

Bending Stress in fillet weld

$$\begin{aligned}\text{Bending stress} &= 4M / (\pi \times 0.707s \times d^2) \\ &= 5.66 M / (\pi \times s \times d^2)\end{aligned}$$

Size of weld used for welding mounting plate welding is 8 mm.

$$\begin{aligned}\text{Bending stress} &= 5.66 \times 5827140 / (3.14 \times 8 \times 200^2) \quad \{ \text{Since } M=5827140 \text{ N-mm} \} \\ &= 32.83 \text{ N/mm}^2\end{aligned}$$

Since material is IS: 2062 Grade 'B',

Where yield stress = 25 Kg/mm² = 245.25 N/mm²

Factor of safety = 245.25/32.83 = 7.4

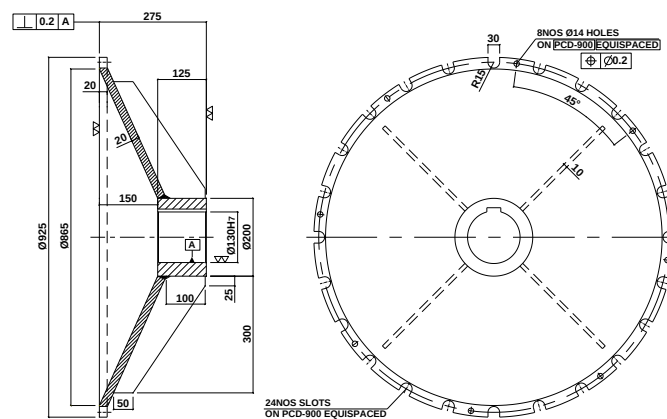


Fig:4.1.1.2(a): Mounting plate of Component-4

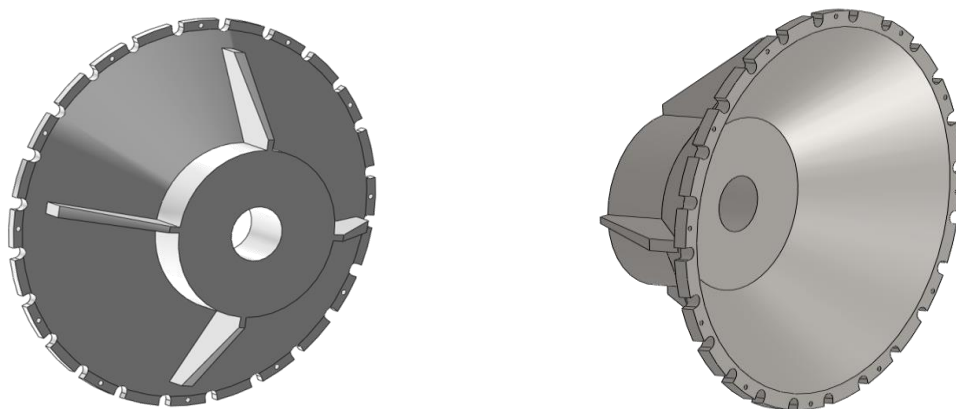


Fig:4.1.1.2(b): Mounting plate of Component-4

4.1.1.3 Bottom frame

Mass of component-4 = 1000 Kg

Mass of support trolley housing and Bottom frame = 1500 kg.

Hence total load on bottom frame= $2500 \times 9.81 = 24525 \text{ N}$.

The bottom portion of the bottom frame is supported on track frame by means of LM guide at four points as shown in above figures

Bottom frame is assumed as a simply supported beam as shown in Figure below

The total load on bottom frame is shared by two sides

Therefore load acting per side of the frame= $2500/2 = 1250 \text{ Kg}$

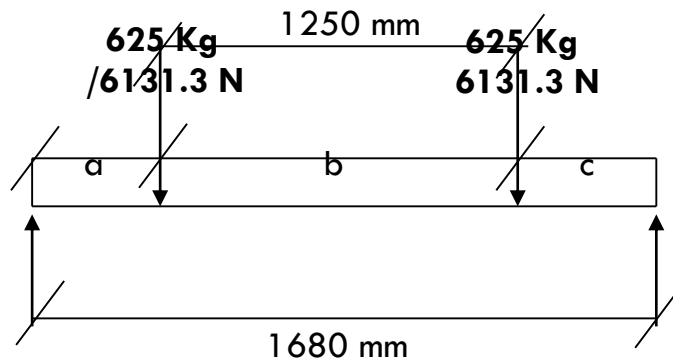
Moment of inertia of channel section (I_{xx}) of MC-350 = 6047.9 cm^4 – taken from design data book.

Section modulus of channel section Z_{xx} of MB-350 = 403.2 cm^3
 $= 403200 \text{ mm}^3$

Bending Stress = $W a / Z$

Bending Stress = $625 \times 9.81 \times 215 / 403200$
 $= 3.34 \text{ N/mm}^2$

Yield Stress = $12 \text{ Kg/mm}^2 = 117.7 \text{ N/mm}^2$ for standard structural steel channels.



Deflection

Maximum deflection occurs at the center of the bottom frame given by $Y_{\max} = \frac{W a (3L^2 - 4a^2)}{24 EI}$

Where $L = 1680 \text{ mm}$, $W = 625 \text{ Kg}$, $a = 215 \text{ mm}$

For MC 350 $I_{xx} = 60479000 \text{ mm}^4$

$Y_{\max} = 625 \times 9.81 \times 215 \times (3 \times 1680^2 - 4 \times 215^2) / (24 \times 2.1 \times 10^5 \times 60479000) = 0.036 \text{ mm}$

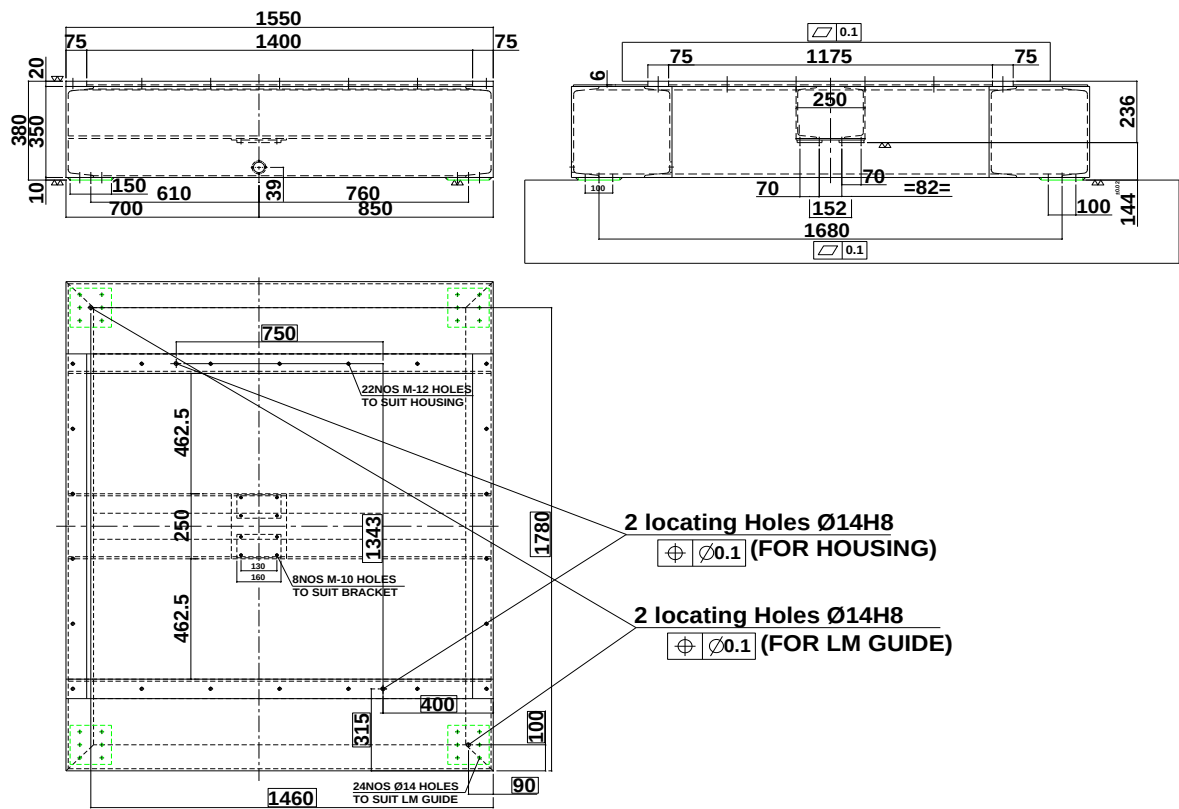


Fig:4.1.1.3(a): Bottom frame

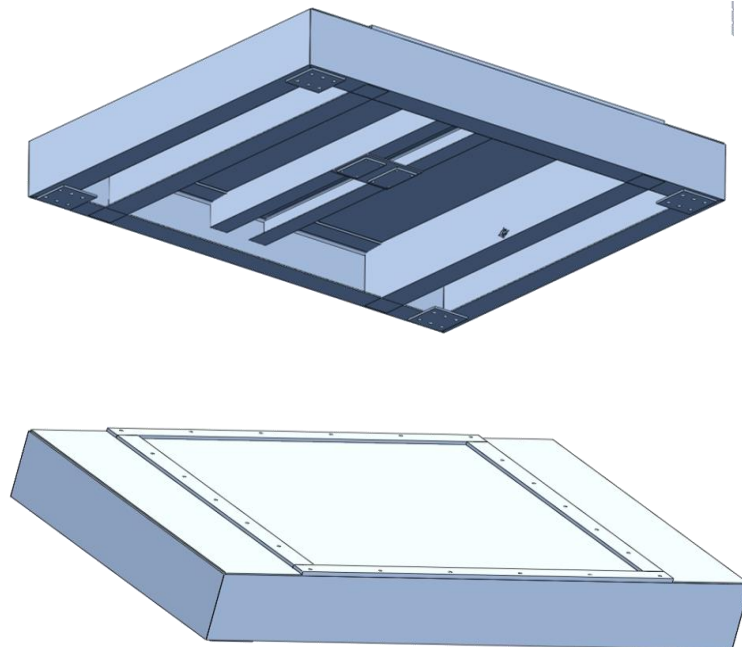


Fig: 4.1.1.3(b): Bottom Frame

4.1.1.4 Housing for assembly trolley

The mass of component 4 is acting at a distance of 695 mm from Housing fore end.

The mass of mounting plate is acting at a distance of 300 mm from fore end.

Mass of Component-4 = 1000 Kg.

Mass of Mounting plate for component-4 = 240 Kg

Moment Generated by the above offset loads = $1000 \times 9.81 \times 695 + 240 \times 9.81 \times 300$

$$(M) = 7524270 \text{ N-mm}$$

This moment will tilt the housing about its fore end.

The above moment will induce bending stress in the housing.

Bending stress will be maximum at the cross section along centerline

Cross section of support housing along Center line

$$\begin{aligned} \text{Moment of inertia} &= (601 \times 430^3) / 12 - (585 \times 410^3) / 12 - (310 \times 10^3) / 12 - \\ &\quad (330 \times 10^3) / 12 + (310 \times 60^3) / 12 + (330 \times 60^3) / 12 - \\ &\quad (2 \times 230 \times 50^3) / 12 + (2 \times 9.5 \times 330^3) / 12 + \\ &\quad (9.5 \times 330) \times (254 + 9.5)^2 - (9.5 \times 330) \times (254 + 9.5)^2 \\ &= 685652083.3 \text{ mm}^4 \end{aligned}$$

$$\begin{aligned} \text{Section modulus (Z)} &= I/y = (2 \times 685652083.3) / 430 \\ &= 3189079.46 \text{ mm}^3. \end{aligned}$$

$$\begin{aligned} \text{Bending stress}_{\max} &= M/Z \\ &= 7524270 / 3189079.46 \\ &= 2.36 \text{ N/mm}^2 \end{aligned}$$

$$\text{Area of cross section} = 29250 \text{ mm}^2$$

$$\begin{aligned} \text{Normal stress} &= \text{Load} / \text{area} = 1240 \times 9.81 / 29250 \\ &= 0.42 \text{ N/mm}^2 \text{ (compressive)} \end{aligned}$$

$$\begin{aligned} \text{Principal stress}_{\max} &= \text{Normal stress} + \text{Bending stress}_{\max} \\ &= 2.78 \text{ N/mm}^2 \text{ (compressive)} \end{aligned}$$

Housing is made up of standard structural steel plates.

Yield Stress = $12 \text{ Kg/mm}^2 = 117.72 \text{ N/mm}^2$ for standard structural steel.

Let f_{b1} , f_{b2} , f_{b3} , f_{b4} , f_{b5} & f_{b6} corresponds to the bolt force induced /unit distance in the bolts at their respective sections.

$$f_b (6 \times 25^2 + 2 \times 275^2 + 2 \times 525^2 + 2 \times 775^2 + 2 \times 1025^2 + 2 \times 1275^2 + 6 \times 1525^2) \\ = 7524270 \text{ N-mm}$$

(Factors 6 and 2 indicates connected by 6 bolts and 2 bolts respectively at that section)

$$f_b = 7524270 / 21213750 = 0.3547 \text{ N /mm.}$$

Maximum bolt force will be induced at the farthest section from tilting edge.

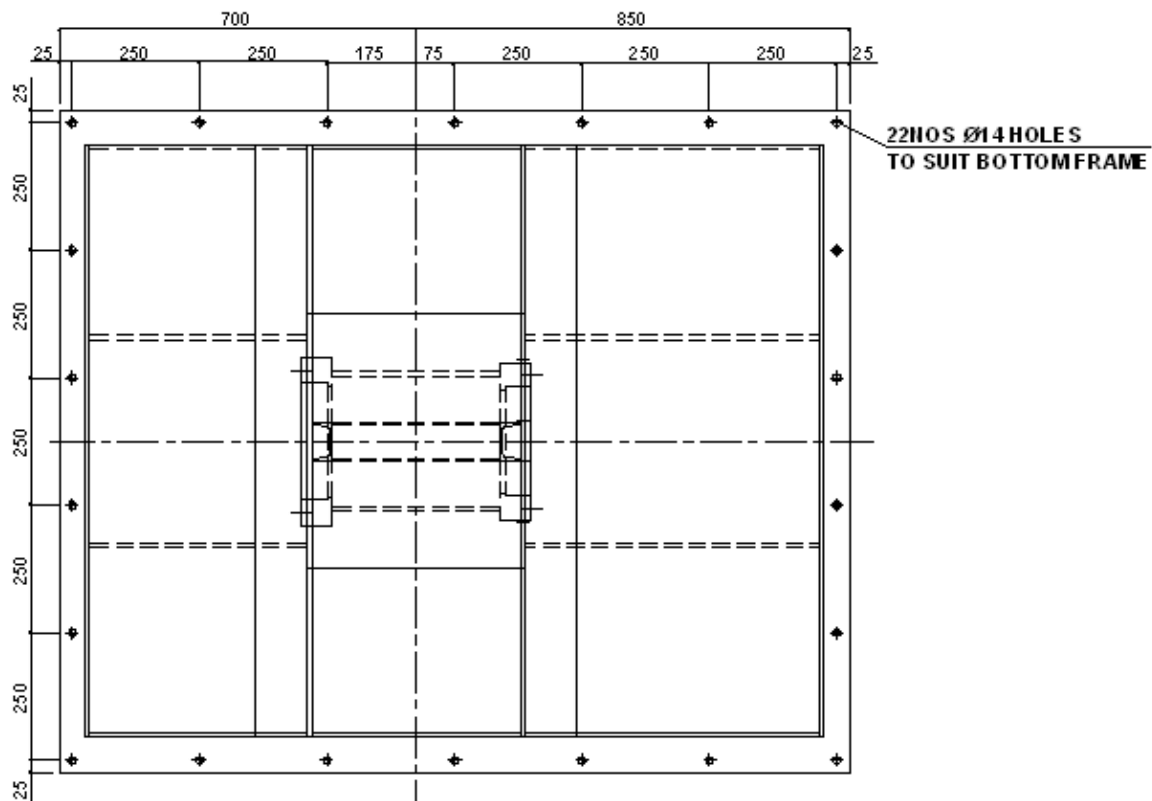


Fig:4.1.1.4(c): Bolted design

$$\text{Max bolt force} = f_b \times 1525 = 541 \text{ N.}$$

Bolt force generated due to preload is given = 30 % of proof load

$$= 0.3 \times 8247 \times 9.81 = 24270 \text{ N.}$$

Total Bolt force = Bolt force due to external Moment + Bolt force due to Preload

$$= 541 + 24270 = 24811 \text{ N.}$$

Factor of safety = Maximum bolt force / actual bolt force

$$= 80903 / 24811 = 3.26.$$

4.1.1.5 Effort required for moving component-4 on track frame

Mass of component-4 = 1000 Kg

Self mass of the trolley with support structure= 1450 Kg

Therefore total mass to be moved = 2450 Kg.

Friction force required to overcome in order to move the support trolley is Friction force = $f \times \text{Weight}$

Friction factor $f = 0.02$ for linear motion rail & guide blocks

Friction force = 481 N.

Torque at rack and pinion

$T = F \times r$ where r = radius of the pinion.

= 481×75 where $r = 75\text{mm}$

Torque = 36075 N-mm

Effort required to move the trolley on LM rails = $T/\text{length of handle}$

= $36075 / 300 = 120.3 \text{ N}$

Where length of the handle used is 300 mm.

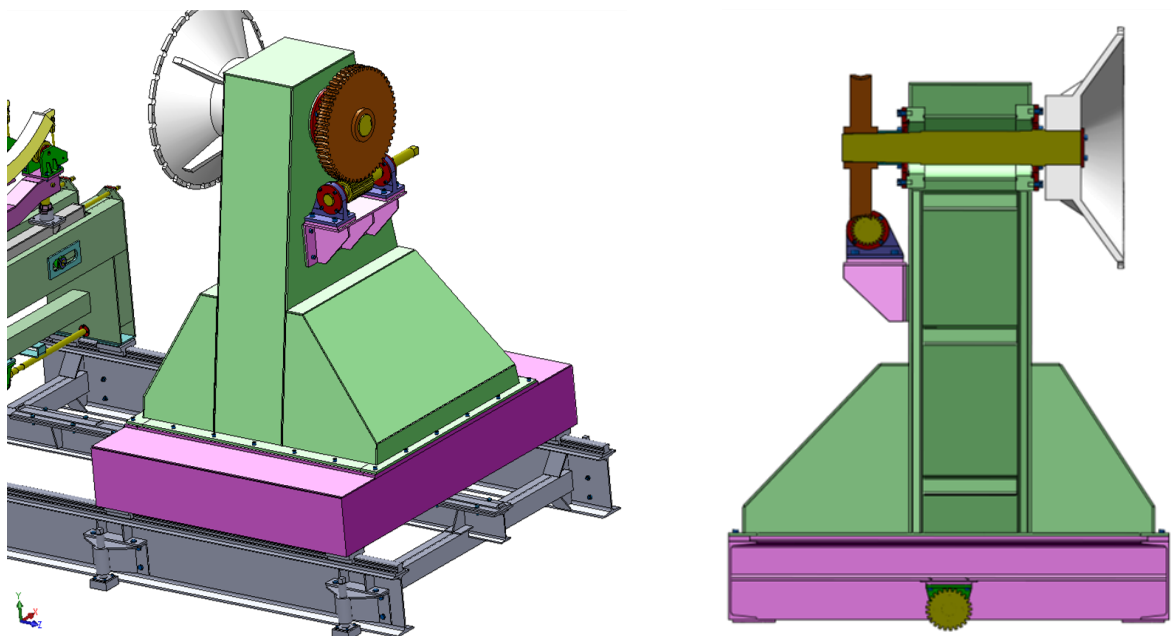


Fig:4.1.1.5: Moving component 4 on track frame.

4.1.2 Component-4 Lifting bracket

4.1.2.1 Bending of the Beam

Mass of component-4= 1000 Kg

The beam portion of the lifting bracket is assumed as a cantilever beam with one end fixed by bolts and load is applied at free end.

Moment Generated by external load = $1000 \times 510 \times 9.81$

$$M = 5003100 \text{ N-mm}$$

Moment of inertia of beam (I) = 932262.75 mm^4 . (Calculated from geometry) Section Modulus of the beam (Z) = $(I/y) = 30869.6 \text{ mm}^3$

Bending Stress of the beam = $M/Z = 162.1 \text{ N/mm}^2$

Factor of safety = yield stress / Bending stress

Material used is steel BS: 970 En24 in Hardened and tempered cond. 30-34 HRC where yield stress = $90 \text{ Kg/mm}^2 = 882.9 \text{ N/mm}^2$

Factor of safety = $882.9/162.1 = 5.45$

4.1.2.2 Bolt Force

The beam is connected to plate by means of 2 sets of 3 bolts M12.

$$M = 2fr_1^2 + 2fr_2^2 + 2fr_3^2$$

Where M= Moment generated by external load = 5003100 N-mm

f= Bolt force per unit length & r_i = radial distance of bolt from bending edge.

$$5003100 \text{ N-mm} = 2f(20^2 + 47^2 + 75^2) \Rightarrow f = 304.11 \text{ N/mm}$$

$$f_{\max} = 304.11 \times 75 = 22808 \text{ N.}$$

3 sets of M12 bolts of class 10.9 are used.

Force due to pretension = $0.3 \times \text{max bolt force} = 24350.4 \text{ N}$.
(Since $f_{\max}$ for M12 bolt = $8247 \times 9.81 \text{ N} = 81168 \text{ N}$)

Bolt force = $F_{\text{pretension}} + F_{\text{externalload}}$

$$= 24350.4 + 22808 = 47158.4 \text{ N}$$

$$\text{Ultimate bolt force} = \pi/4 \times 10.25^2 \times 100 \times 9.81 = 81168 \text{ N}$$

Factor of safety = ultimate bolt force/ actual bolt force

$$= 81168/47158.4 = 1.72$$

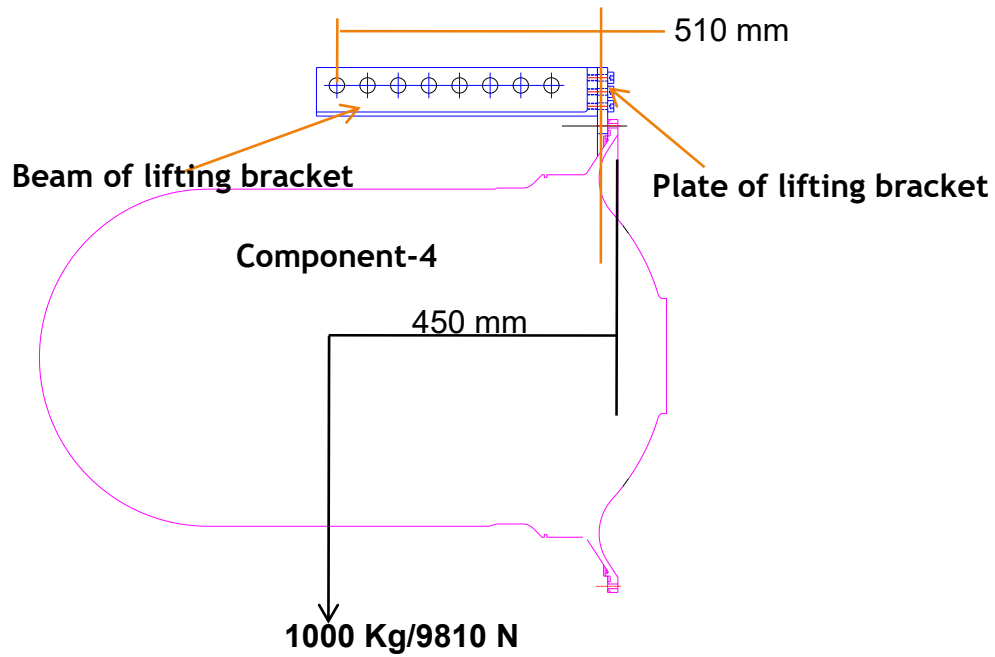


Fig:4.1.2(a): Component 4 lifting bracket

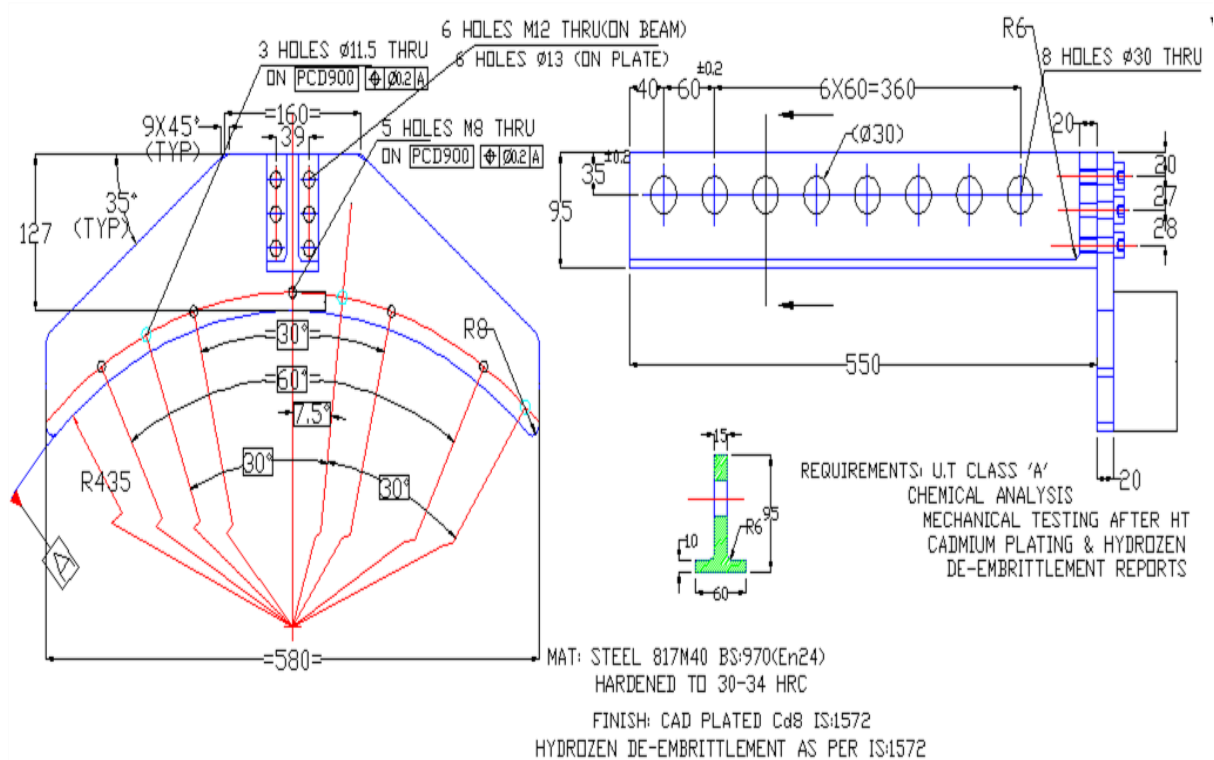


Fig:4.1.2(b): Component 4 lifting bracket

4.1.2.3 FEA Analysis of the lifting bracket for component-4

Lifting bracket is modeled in Ansys FEA software.

Bolted joint between beam and plate portion of lifting bracket is considered as a rigid joint.

Solid 45 element is used for meshing.

Meshing of model carried out using free mesh option.

The following properties are used as material properties

Young's modulus = $2.1 \times 10^5 \text{ N/mm}^2$, Poisson's ratio = 0.3

The following constraints and loading conditions are used for FEA model.

All the eight axial holes on plate are constrained in all directions.

Load (from mass of component-4) is applied at the extreme end of beamhole. The results of FEA as follows

The maximum stress in plate from FEA model = 198 N/mm^2

Factor of safety = $882.9 / 198 = 4.5$

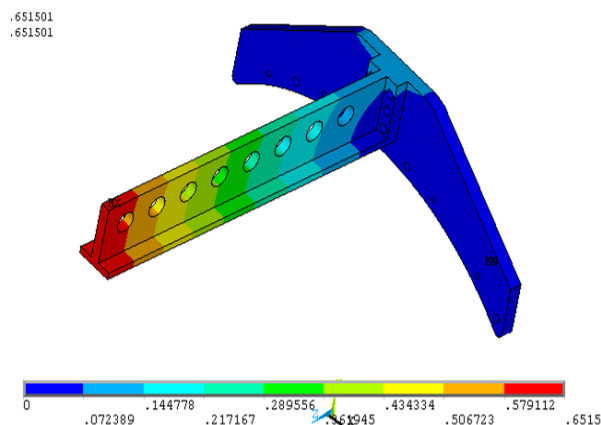


Fig:4.1.2.3(a): FEA Analysis of the lifting bracket

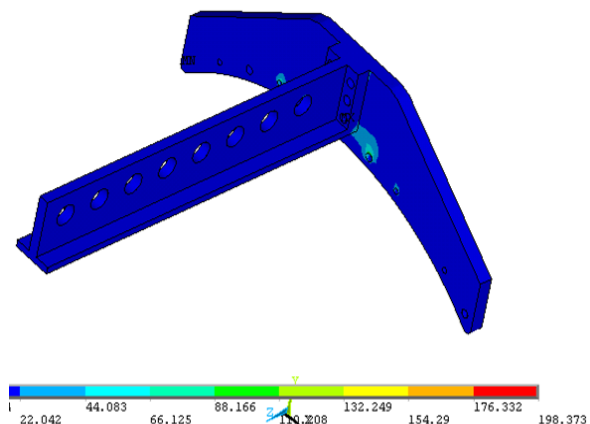


Fig:4.1.2.3(b): FEA Analysis of the lifting bracket

4.1.3 Component-3 Lifting bracket

4.1.3.1 Bending of the Beam

Mass of component-3= 400 Kg.

The beam is assumed as a cantilever beam with one end fixed by bolts and load is applied at free end. Refer below Figure

Moment Generated by external load = $400 \times 9.81 \times 108$

$$M = 423792 \text{ N-mm}$$

Moment of inertia of beam (I) = 137635.94 mm^4 .

Section Modulus of the beam (Z) = 8342 mm^3

Bending Stress of the beam = $M/Z = 50.8 \text{ N/mm}^2$

Factor of safety = yield stress / Working stress

Material used is steel BS: 970 En24 in hardened and tempered condition 30-34 HRC.

Where yield stress = $90 \text{ Kg/mm}^2 = 883 \text{ N/mm}^2$.

Factor of safety = $883/50.8 = 17$

4.1.3.2 Bolt Force

The beam is connected to plate by means of 2 sets of 2 bolts M12

$$M = 2fr_1^2 + 2fr_2^2$$

Where M= Moment generated by external load about tilting edge

f= bolt force per unit length & r_i = radial distance of bolt from bending edge.

$$423792 = 2f(15^2 + 35^2)$$

$$\Rightarrow f = 146.14 \text{ N/mm}^2.$$

$$f_{\max} = 146.14 \times 35 = 5115 \text{ N.}$$

2 sets of M12 bolts of class 10.9 are used.

Force due to pretension = $0.3 \times \text{max bolt force} = 24270.9 \text{ N.}$

(Since $f_{\max}$ for M12 bolt = 80903.1 N).

Bolt force = $F_{\text{pretension}} + F_{\text{external load}} = 24270.9 + 5115 = 29386 \text{ N.}$

Ultimate bolt force = $\pi/4 \times 10.252 \times 100 \times 9.81 = 80903.1 \text{ N.}$

Factor of safety = ultimate bolt force/ actual bolt force

$$= 80903.1/29386 = 2.75.$$

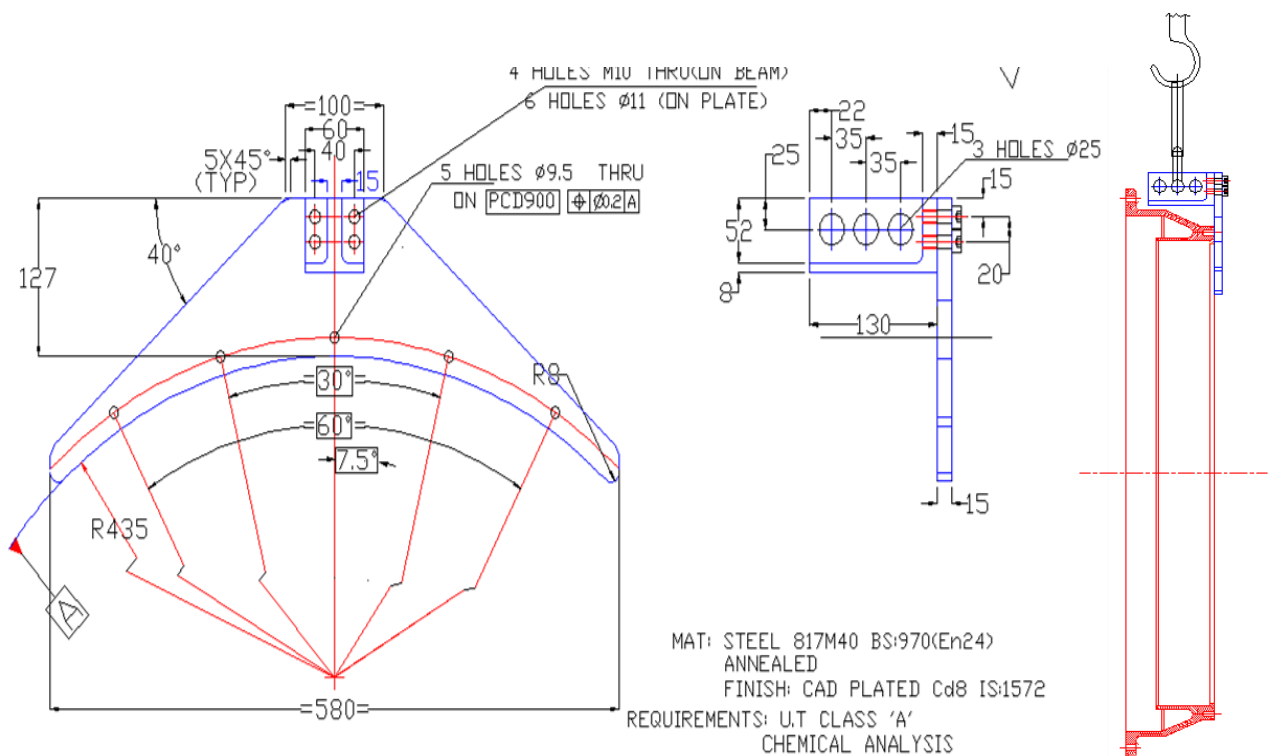


Fig:4.1.3:Lifting bracket for component 3

4.1.3.3 FEA Analysis of the lifting bracket for component-3

Lifting bracket is modeled in Ansys FEA software.

Bolted joint between beam and plate portion of lifting bracket is considered as a rigid joint.

Solid 45 element is used for meshing. Meshing of model carried out using free mesh option. The following properties are used as material properties Young's modulus = $2.1E5 \text{ N/mm}^2$, Poisson's ratio = 0.3. The following constraints and loading conditions are used for FEA model. All the five axial holes on plate are constrained in all directions. Load (from mass of component-4) of 3924 N is applied at the extreme end of Beam hole.

The results of FEA as follows

Maximum stress = 224 N/mm^2

Factor of safety = $883/224 = 3.94$.

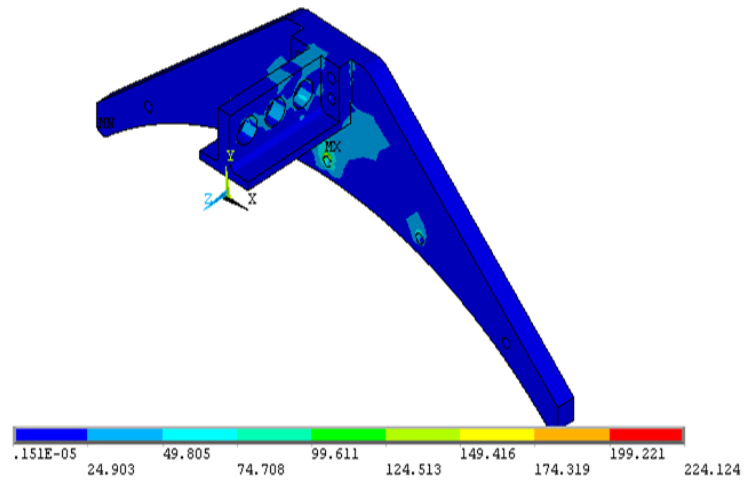


Fig:4.1.3.3(a): FEA Analysis of the lifting bracket

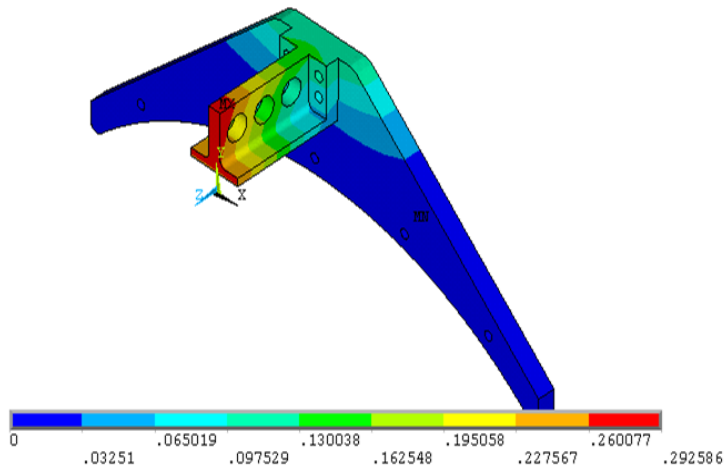


Fig:4.1.3.3(b): FEA Analysis of the lifting bracket

4.1.4 Component-2 Lifting bracket

4.1.4.1 Bending of the Beam

Mass of Component-2= 400 Kg.

The beam is assumed as a cantilever beam with one end fixed by bolts and load is applied at free end.

Moment Generated by external load = $400 \times 9.81 \times 120 = 470880 \text{ N-mm}$

Moment of inertia of beam (I) = 137635.94 mm^4 .

Section Modulus of the beam (Z) = 8342 mm^3

Bending Stress of the beam = $M/Z = 56.45 \text{ N/mm}^2$

Factor of safety = yield stress / Working stress

Material used is steel BS:970 En24 in hardened and tempered condition 30-34 HRC
yield stress= 883 N/mm²

Factor of safety = 883/56.45= 15

4.1.4.2 Bolt Force

The beam is connected to plate by means of 2 sets of 2 bolts M12

$$M = 2fr_1^2 + 2fr_2^2$$

Where M= Moment generated by external load

f= Bolt force per unit length & r_i = radial distance of bolt from bending edge.

$$470880 = 2 f(15^2 + 35^2) \Rightarrow f = 162.37 \text{ N/mm}$$

$$f_{\max} = 162.37 \times 35 = 5683 \text{ N.}$$

2 sets of M12 bolts of class 10.9 are used.

Force due to pretension = 0.3 x max bolt force = 24270.9 N

(Since f_{max} for M12 bolt = 80903.1 N)

Bolt force = F_{pretension} + F_{external moment} = 24270.9 + 5683 = 29954 N

Ultimate bolt force = $\pi/4 \times 10.25^2 \times 100 \times 9.81 = 80903.1 \text{ N}$

Factor of safety = ultimate bolt force/ actual bolt force

$$= 80903.1/29954 = 2.7$$

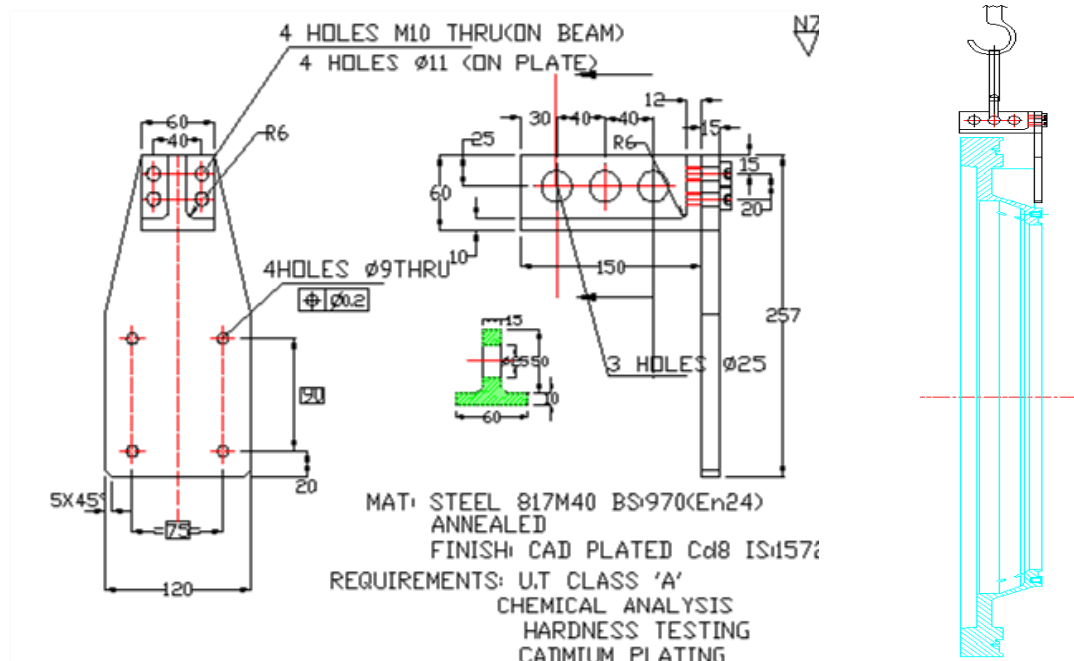


Fig:4.1.4.2: Bolt force

4.1.4.3 FEA Analysis of the lifting bracket for component-2

Lifting bracket is modeled in Ansys FEA software.

Bolted joint between beam and plate portion of lifting bracket is considered as a rigid joint.

Solid 45 element is used for meshing.

Meshing of model carried out using free mesh option.

The following properties are used as material properties

Young's modulus = $2.1 \times 10^5 \text{ N/mm}^2$, Poisson's ratio = 0.3

The following constraints and loading conditions are used for FEA model. All the four axial holes on plate are constrained in all directions. Load (from mass of component-4) of 3924 N is applied at the extreme end of beamhole. The results of FEA as follows

Maximum stress = 250 N/mm^2

Factor of safety = $883/250 = 3.5$

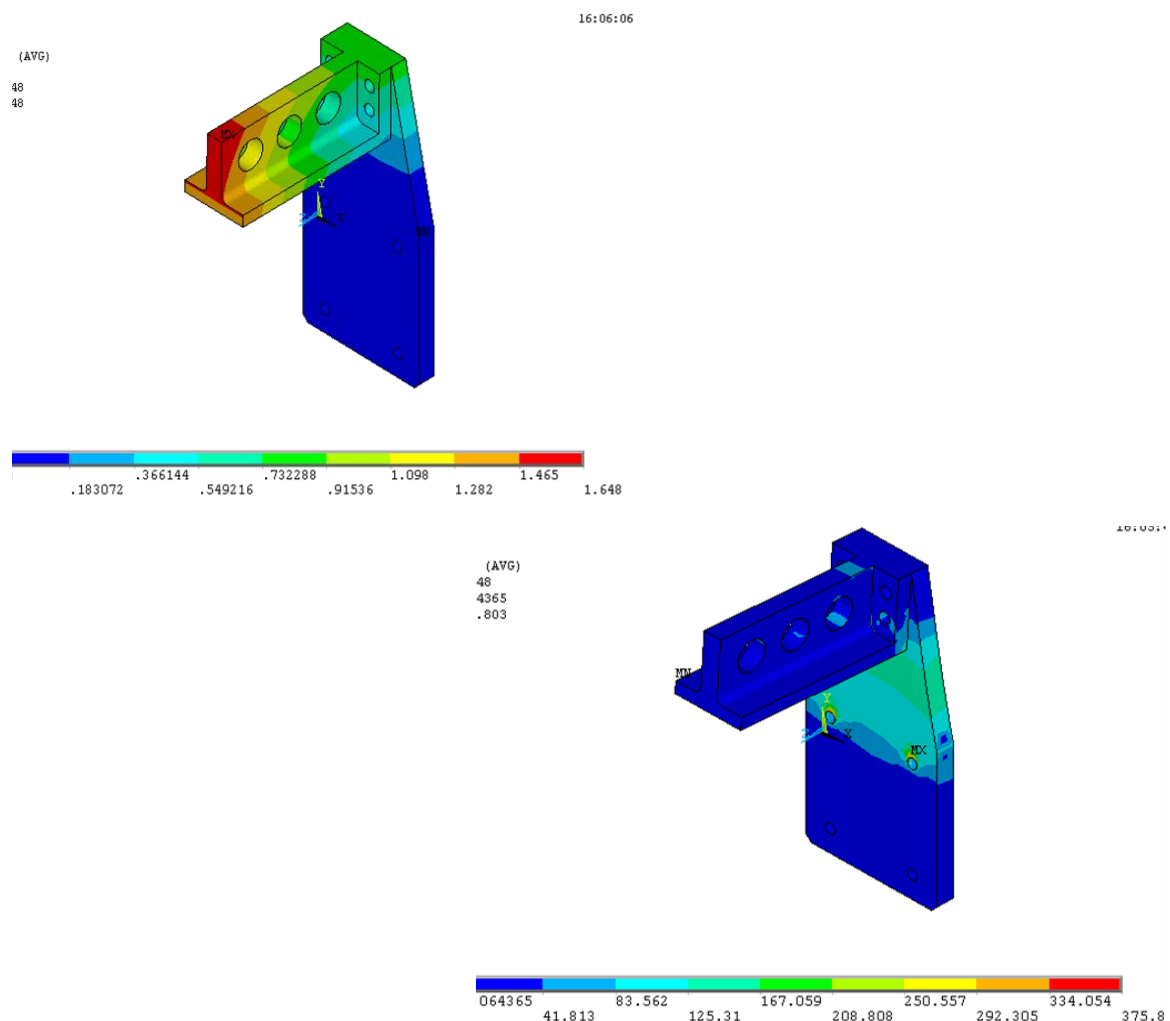


Fig:4.1.4.3:FEA Analysis for lifting bracket

4.1.5 Component-1 Assembly /Handling trolley

4.1.5.1 Bottom frame

The Bottom frame is made up of MC-250 mm channel in Box construction. Mass of component-1 = 750 Kg

Mass of Bottom frame, sliding frame = 750 Kg

top frame, rollers and screws etc.,.

Total load = 14715 N

The total load is shared by two parallel frames as shown in the fig

Hence Load on each bottom frame = 7357.5 N.

No of load acting points = 2

Therefore load on each point = 3678.75 N.

Bending stress $_{\max} = W \times a / Z$

Where $a = 445$ mm and $W = 3678.75$ N

MC- 250 mm channel properties are given by

Sectional modulus $Z_{xx} = 305300 \text{ mm}^3$ taken from design data book.

Since the cross section is made up of two C- channels in Box form

Z_{xx} of cross section = $2 \times 305300 \text{ mm}^3 = 610600 \text{ mm}^3$

Bending Stress $_{\max} = 3678.75 \times 445 / 610600 = 2.68 \text{ N} / \text{mm}^2$

Cross sectional area of Box form (made up of 2 C-channels)= 4884 mm^2

Normal stress = $7357.5 / 4884$

= $1.51 \text{ N} / \text{mm}^2$ (compressive)

Maximum Stress (compression) = Bending Stress $_{\max}$ + Normal stress
= $4.19 \text{ N} / \text{mm}^2$ (Compressive)

For MC- 250 yield stress is $12 \text{ Kgf} / \text{mm}^2 = 117.7 \text{ N} / \text{mm}^2$

(taken from Design data book)

Factor of safety = $117.7 / 4.19 = 28$

Deflection

Maximum deflection will occur at center of bottom frame.

$Y_{\max} = Wa / 24EI \{3L^2 - 4a^2\}$

Where $L = 1680$ mm (length of the bottom frame)

$W =$ Load acting at each loading point = 3678.75 N

a= distance of loading point to its adjacent support.

For MC-250 $I_{xx} = 38160000 \text{ mm}^4$ each

Since the cross section is made up of two C- channels in Box form

I_{xx} of cross section = $2 \times 38160000 \text{ mm}^4 = 76320000 \text{ mm}^4$

$Y_{\max} = 3678.75 \times 445 / (24 \times 2.1 \times 10^5 \times 7632 \times 10^4) \{3 \times 1680^2 - 4 \times 445^2\}$
= 0.033 mm

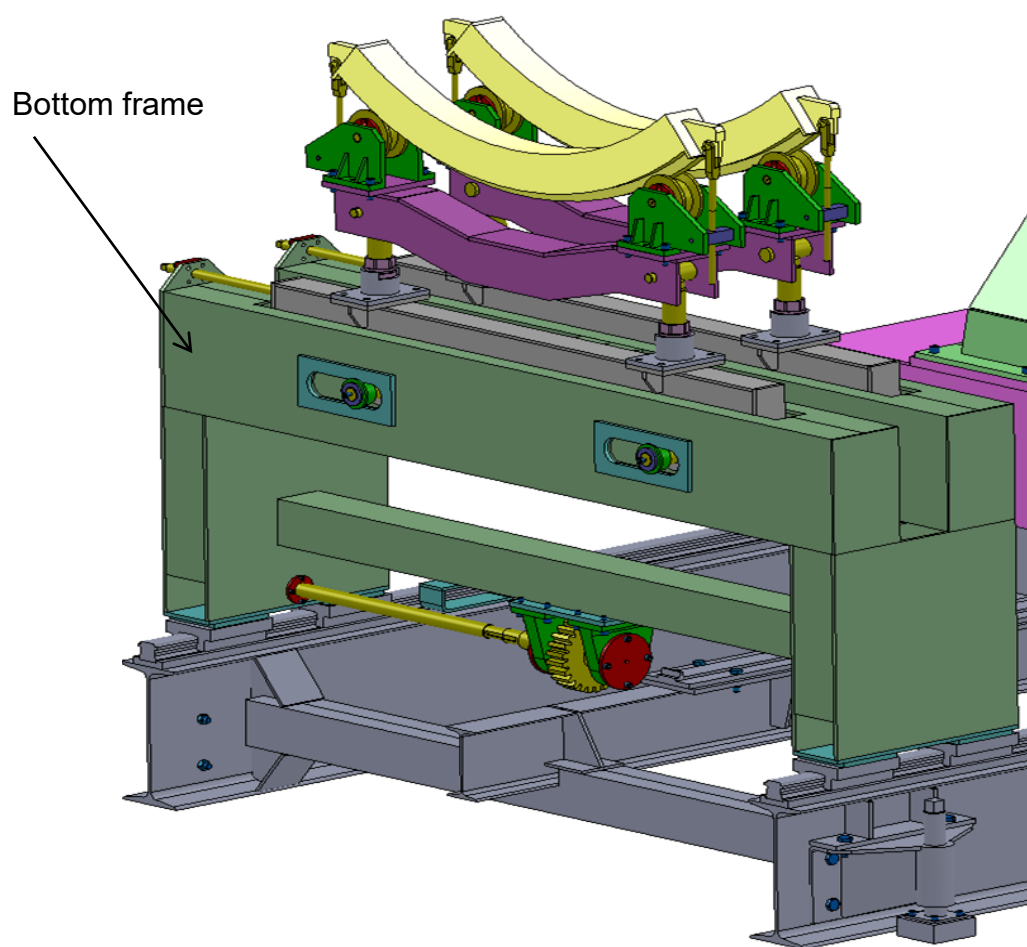


Fig:4.1.5:Assembly / handling trolley

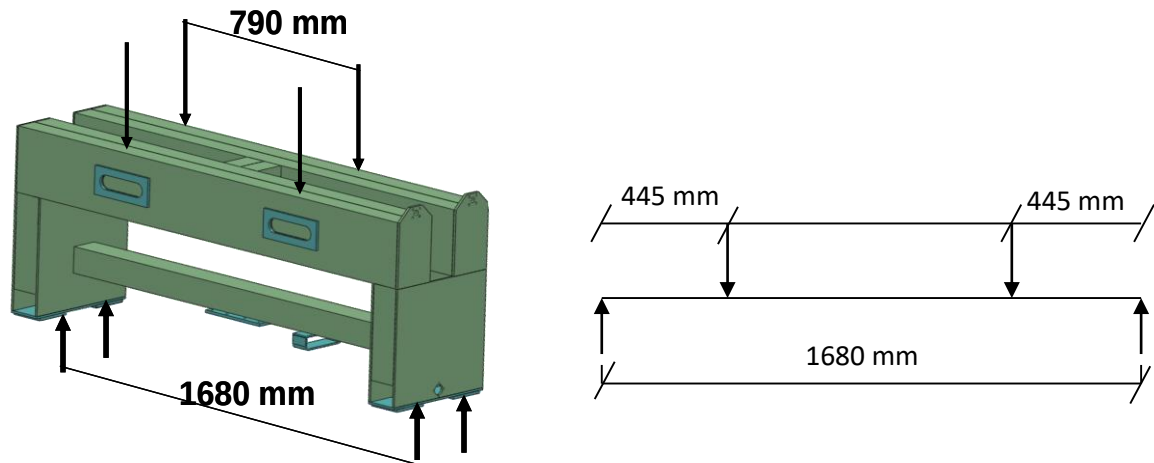


Fig:4.1.5.1: Bottom frame

4.1.5.2 Pin supporting Sliding frame on Bottom frame

Total load of component -1 is supported on 2 sliding frames

In turn each sliding frame is supported on bottom frame using two supporting roller pins.

Load on each pin = (Load of component 4 + load of top frame + rollers + cradle + supporting frame) / 4

$$= (750 + 550) \times 9.81 / 4 = 1300 \times 9.81 / 4 = 3188.3 \text{ N.}$$

$$\text{Bending Moment}_{\max} = (3188.3) / 2 \times 80 - 45.546 \times 35^2 / 2 = 99635.1 \text{ N-mm}$$

occurs at the center of the pin

$$\text{Sectional Modulus } Z = \pi d^3 / 32$$

$$= 3.14 \times 30^3 / 32$$

$$= 2649.38 \text{ mm}^3$$

$$\text{Bending Stress}_{\max} = 99635.1 / 2649.38 = 37.61 \text{ N/mm}^2$$

Shear stress = Load / 2A (pin under double shear)

$$= 3188.3 / (2 \times \pi / 4 \times 30^2)$$

$$= 2.25 \text{ N/mm}^2$$

$$\text{Principal Stress} = \text{Bend. stress} / 2 + \sqrt{(\text{Bend. stress} / 2)^2 + \text{shear stress}^2}$$

$$= 37.74 \text{ N/mm}^2$$

Shaft is made up of En9 Material in hardened & tempered condition 30-34 HRC for which yield stress = 90 Kgf/mm² = 883 N/mm²

Factor of safety = yield stress / principal stress

$$= 883 / 37.74 = 23$$

Maximum Deflection of pin occurs at center

$$\text{Deflection } Y_{\max} = wa/24EI \{L^3 - 2La^2 + a^3\}$$

$$= 4.643 \times 35 / (24 \times 2E4 \times 6283) \{1603 - 2 \times 160 \times 35^2 + 35^3\}$$

$$= 0.202 \text{ mm}$$

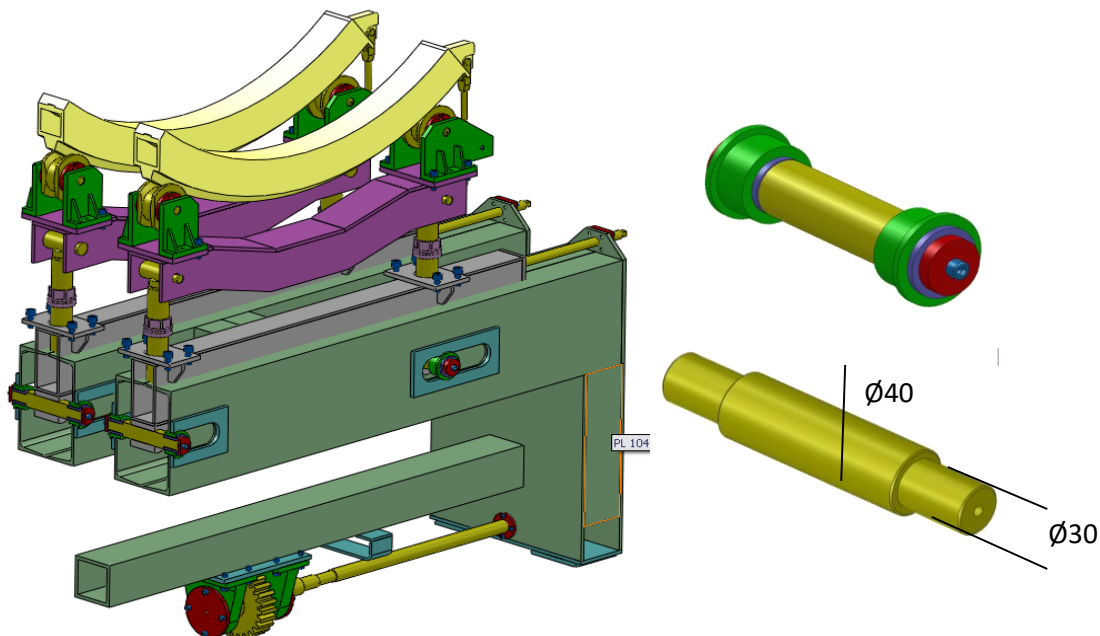
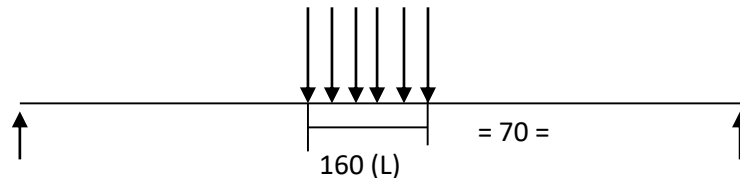


Fig:4.1.5.2 Pin supporting Sliding frame on Bottom frame

4.1.5.3 Lifting screws

Total load of component-1 is supported on two cradles (half rings) which in turn are mounted on top frame using rollers. Top frame is supported on sliding frame using two lifting screws on each of the sliding frame. Lifting screw support the load on its top using pin joint and its bottom is engaged to the nut (constrained longitudinally) on top of sliding frame. Nut can rotate but cannot be allowed to move longitudinally which causes load to be lifted.

$$\text{Load on each lifting screw} = (\text{Mass of component 1} + \text{load of top frame} + \text{rollers} + \text{cradle} + \text{roller assy.})/4$$

$$= (750 + 550) \times 9.81/4 = 3188.3 \text{ N}$$

$$\text{Shear stress} = W / (\pi \times d_c \times n \times t)$$

Where d_c = core diameter of the lifting screw

n = number of threads in engagement (actually sharing the load)

t = thickness of the thread = pitch / 2 = 1.5

$$= 3188.3 / (3.14 \times 41 \times 10 \times 1.5) = 1.65 \text{ N / mm}^2$$

Material of the lifting screw is En – 9 in hardened and tempered condition. Maximum Shear stress allowed = 50 Kgf / mm²

$$= 490.5 \text{ N / mm}^2.$$

Bending stress of the lifting screw

Load on two rollers are given by = (Mass of section-1 / 2 + Mass of the cradle)
= (750 / 2 + 125)

$$= 500 \text{ Kg} = 4905 \text{ N}$$

The above tangential load will bend the lifting screw

Arm length = Unsupported length of the lifting screw + distance between roller center to lifting screw top connection

$$= 210 + 162 \text{ mm}$$

$$= 372 \text{ mm}$$

Bending moment created by the above tangential load = $144.4 \times 9.81 \times 372 = 526961.8 \text{ N-mm}$

Sectional modulus of the lifting screw = $(\pi \times d_c^3 / 32)$

$$= (3.14 \times 41^3 / 32)$$

$$= 6766.3 \text{ mm}^3$$

Bending stress of the lifting screw = M/Z

$$= 526961.8 / 6766.3$$

$$= 77.88 \text{ N / mm}^2$$

Principal Stress = Bend. stress/2 + $\sqrt{(\text{Bend. stress} / 2)^2 + \text{shear stress}^2}$

$$= 77.91 \text{ N / mm}^2$$

Lifting screw is made up of En-9 in hardened and tempered condition

Where yield stress = 90 Kgf/mm² = 883 N/mm²

Factor of safety = $883 / 77.91 = 11.$

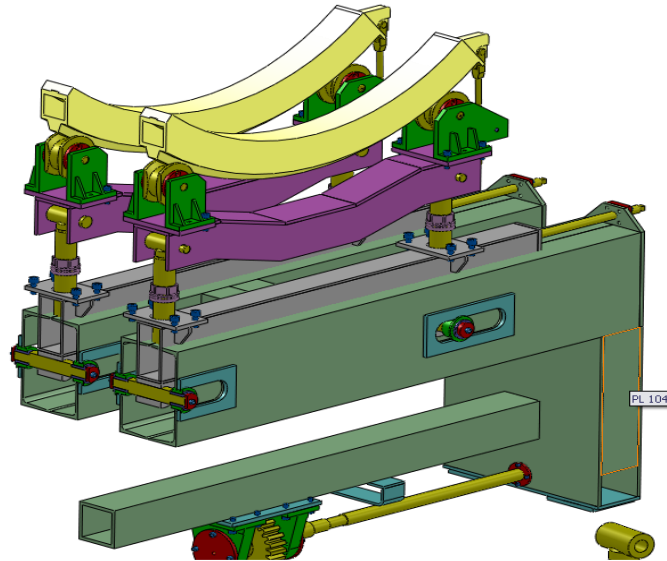


Fig:4.1.5.3(a): Lifting screws

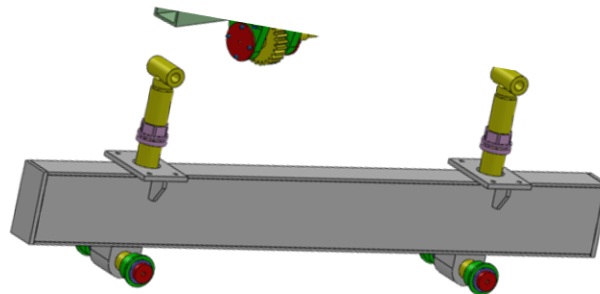


Fig:4.1.5.3(b): Lifting screws

4.1.5.4 Nut for lifting screw

Height of nut = 40mm, Threading used is M44X3.

Number of threads in engagement (n) = 13

Shear stress developed = $W / (\pi \times d_c \times n \times t)$

Where W= load on nut = 3188.3 N

d_c = core diameter of the lifting screw

n = number of threads in engagement (actually sharing the load)

t = thickness of the thread = pitch / 2 = 1.5

Shear stress= $3188.3 / (3.14 \times 41 \times 13 \times 1.5) = 1.275 \text{ N/mm}^2$

(Since load on lifting screw = load on nut= 3188.3 N)

Material of the Nut is gun metal where maximum allowable shear stress is $15 \text{ Kg / mm}^2 = 147 \text{ N / mm}^2$

Factor of Safety = $147 / 1.275 = 115$

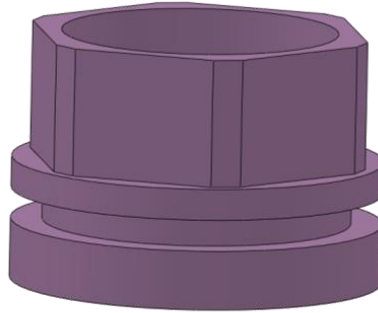


Fig: 4.1.5.4 Nut for lifting screw

4.1.5.5 Top Frame

Bending stress of top frame

Bending of top frame will be due to horizontal component of radial load of half cradle.

This horizontal component will generate bending moment about the point where top frame is connected to lifting screw.

Bending moment (M) = $1416.56 \times 162 = 229485.33 \text{ N-mm}$

The above bending moment will generate maximum bending stress at the center of the top frame.

Top frame is made up of 8 mm Structural steel plates.

At the middle section of top frame, the cross section is box section with side thickness of 8 mm.

Moment of inertia of middle section of top frame (I) = $BD^3/12 - bd^3/12$

$$= 1/12 (100 \times 100^3 - 84 \times 84^3) = 4184405.33 \text{ mm}^4$$

Section modulus at the middle section of top frame = I/y

$$= 4184405.33 / 50 = 83688.11 \text{ mm}^3$$

Maximum Bending stress developed = $M/Z = 2.74 \text{ N/mm}^2$

Normal stress = normal load / Area = $500 \times 9.81 / (100 \times 150 - 84 \times 134)$

$$= 1.31 \text{ N/mm}^2$$

Principal stress = Bending stress + Normal stress = 4.05 N/mm^2

For structural steel plates built up section, yield stress = 12 Kg/mm^2

$$= 117.7 \text{ N/mm}^2$$

Factor of safety = 29.

Deflection of top frame

Deflection of top frame is maximum at the center of the top frame.

Deflection can be found out by double integration method

$$EI \frac{d^2y}{dx^2} = \pm M \Rightarrow EI \frac{dy}{dx} = -Mx + C_1 \Rightarrow EI y = -\frac{Mx^2}{2} + C_1x + C_2$$

Integration constants can be found out by imposing boundary conditions that @ $x=0$ $y=0$ & @ $x=l/2$, $dy/dx=0$

By imposing the above conditions we get $C_1 = Ml/2$ & $C_2 = 0$

$EI y = -\frac{Mx^2}{2} + (Ml/2)x$ since Deflection is maximum at the center

By substituting $x = l/2$, we get $EI y_{\max} = -\frac{Ml^2}{8} + \frac{Ml^2}{4} = \frac{Ml^2}{8}$.

$$\text{Maximum Deflection } y_{\max} = \frac{(229485.33 \times 790^2)}{(8 \times 2 \times 10^5 \times 4184405.33)} = 0.022 \text{ mm}$$

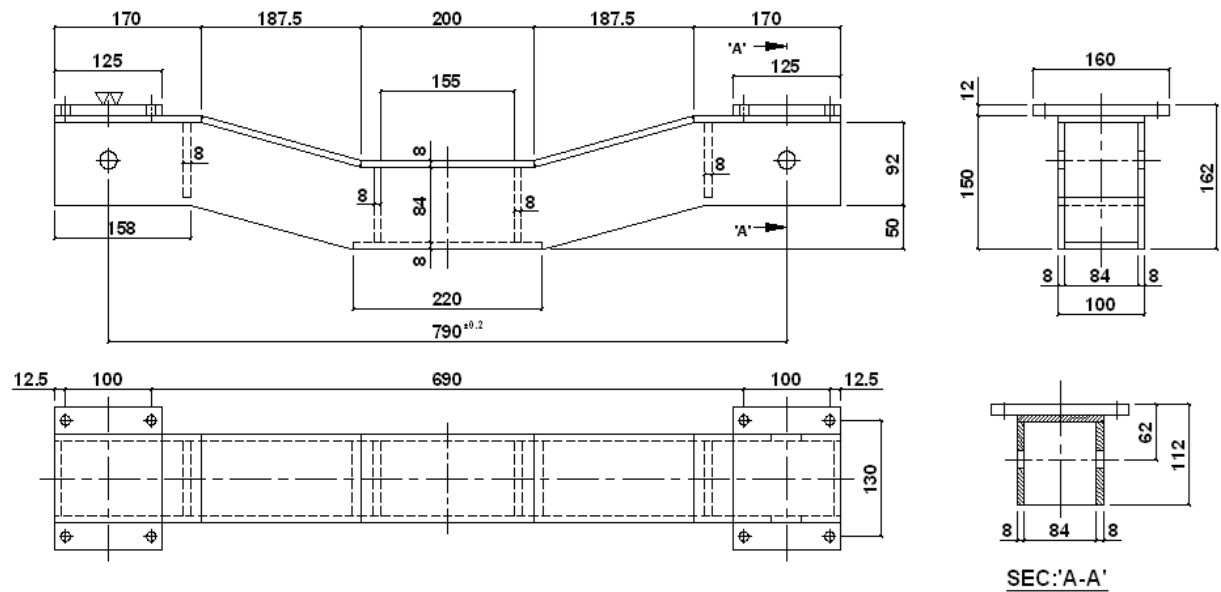


Fig: 4.1.5.5(a): Top Frame

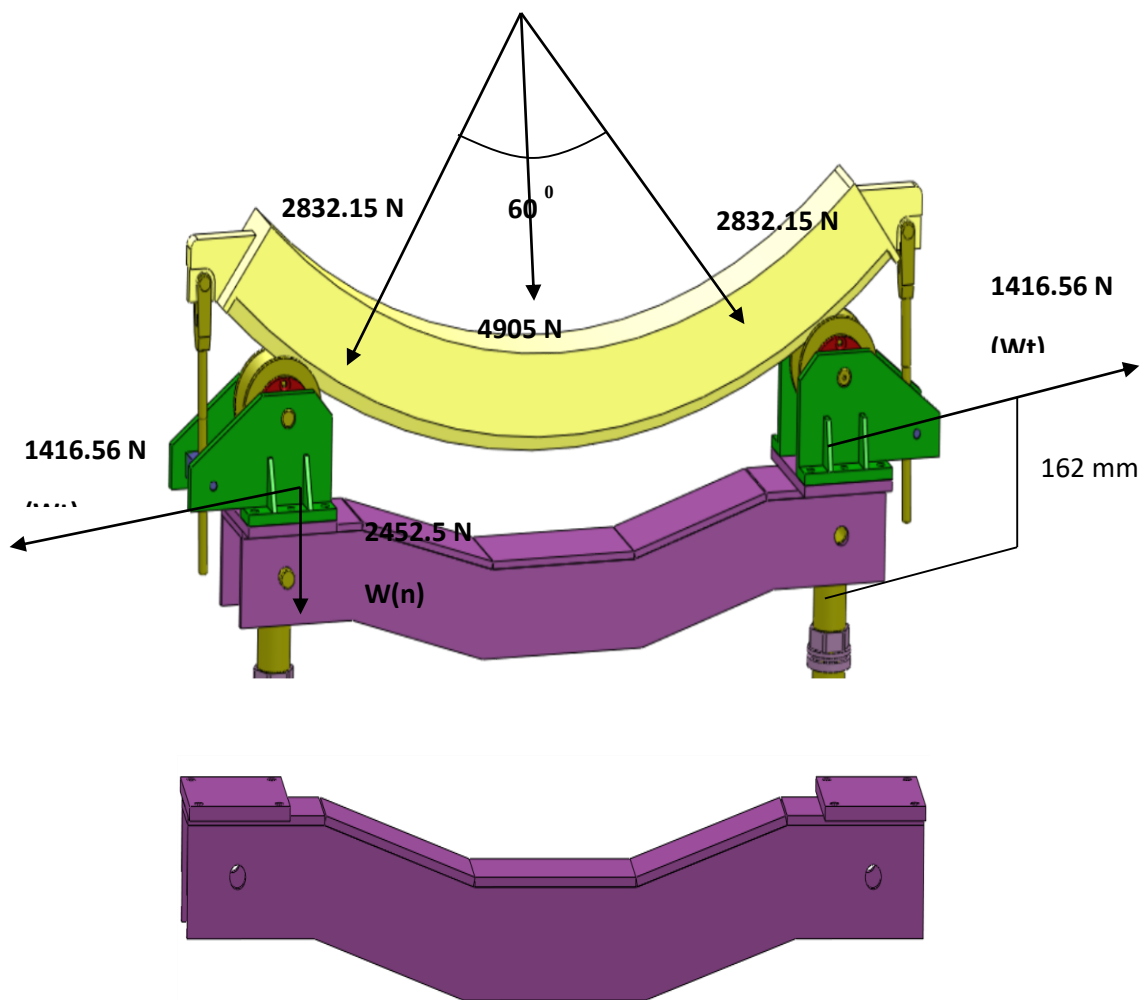


Fig:4.1.5.5(b): Top frame

4.1.5.6 Cradle / Supporting ring

The cross section of cradle / supporting ring is built-up box section with 8mm thickness at each side as shown below. The material of the cradle is aluminum alloy 65032 plates in WP condition.

The ratio $h / R_0 = 65 / 697.5 = 0.093$

Where h & R_0 are depth of the section and mean radius of the section in the plane of bending respectively.

Since ratio $h/R_0 \leq 0.2$, the beam is having small initial curvature and hence straight beam equation can be applied.

The arc length along the neutral axis = 974 mm.

The load on the cradle / supporting ring is assumed as UDL on top surface of the supporting ring.

By applying straight beam equation

$$M/I = F_b / y \quad \text{Bending stress } (F_b) = M/Z$$

$$\begin{aligned} \text{Moment of Inertia of built up box section} &= 1/12(bd^3 - b_1d_1^3) \\ &= 1/12(75 \times 65^3 - 59 \times 49^3) \\ &= 1137965.33 \text{ mm}^4 \end{aligned}$$

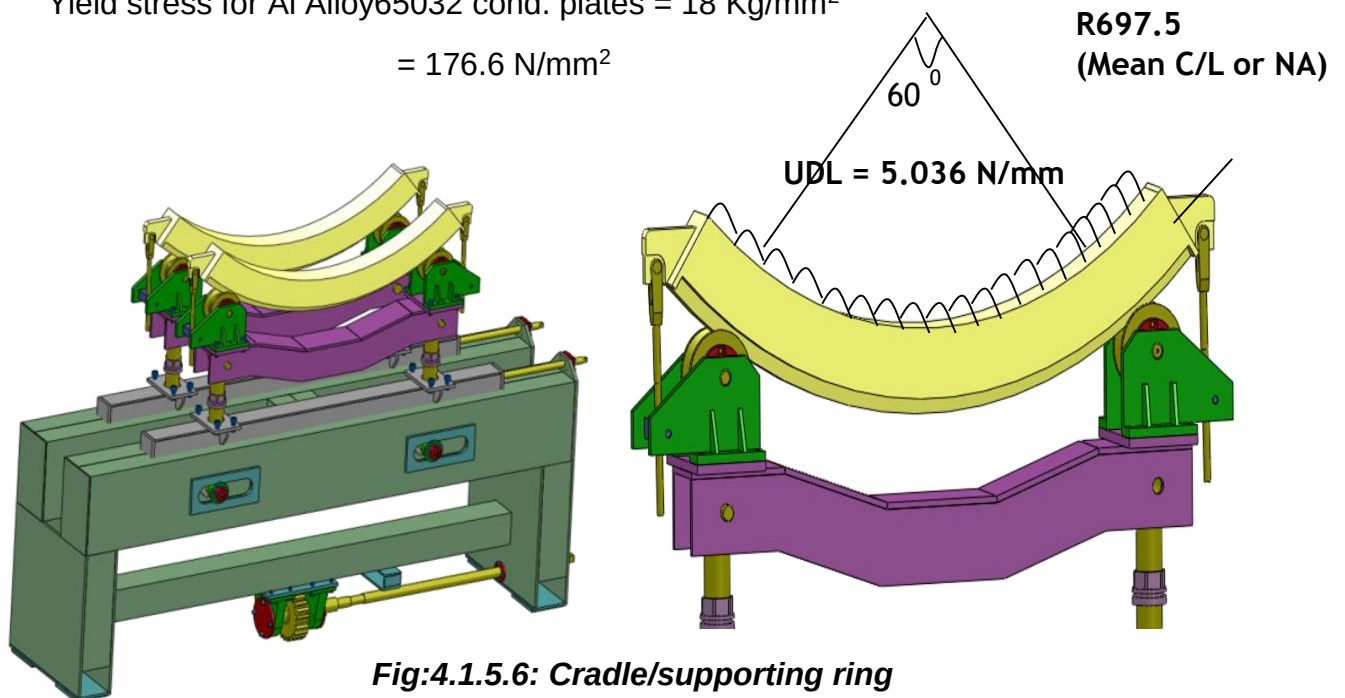
$$\begin{aligned} \text{Section modulus } (Z) &= I/y \\ &= 1137965.33 / 32.5 = 35014.32 \text{ mm}^3 \end{aligned}$$

Since the above beam is a simply supported beam with UDL along its length, maximum bending moment, maximum bending stress and deflection occurs at the center of the beam.

$$\begin{aligned} \text{Maximum bending moment} &= (5.036 \times 487 \times 487 / 2 - 250 \times 9.81 \times 365) \\ &= 297975.3 \text{ N-mm.} \end{aligned}$$

$$\text{Bending stress } (F_b) = M/Z = 297975.3 / 35014.32 = 8.51 \text{ N/mm}^2$$

$$\begin{aligned} \text{Yield stress for Al Alloy 65032 cond. plates} &= 18 \text{ Kg/mm}^2 \\ &= 176.6 \text{ N/mm}^2 \end{aligned}$$



Deflection of the beam = $5/384 \times (wl^4 / EI)$

$$= 5/384 \times (0.714 \times 730^4 / 0.78E5 \times 1137965.33) = 0.214 \text{ mm.}$$

Supporting ring is modeled in Ansys FEA software.

Solid 45 elements is used for meshing.

Meshing of model carried out using free mesh option.

The following material properties are used

Young's modulus = $0.7E5 \text{ N/mm}^2$, Poisson's ratio = 0.3

The following constraints and loading conditions are used for FEA model.

All the nodes attached to the line along 60° radial roller supports are constrained in all directions.

Load (from mass of component-4) of 4905 N is applied on inner surface of supporting ring as a pressure (as a surface load)

The maximum stress in plate from FEA model = 8.32 N/mm^2

The maximum stress obtained by using classical method = 8.51 N/mm^2 obtained by using classical method = 8.51 N/mm^2

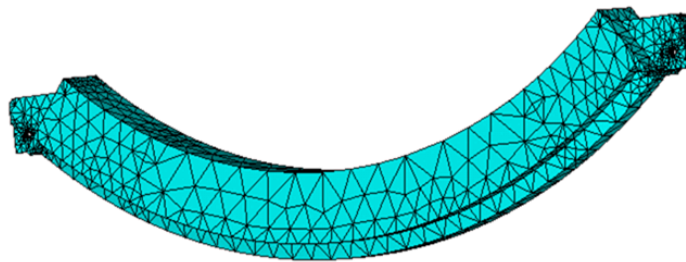


Fig:4.1.5.6(a): Cradle

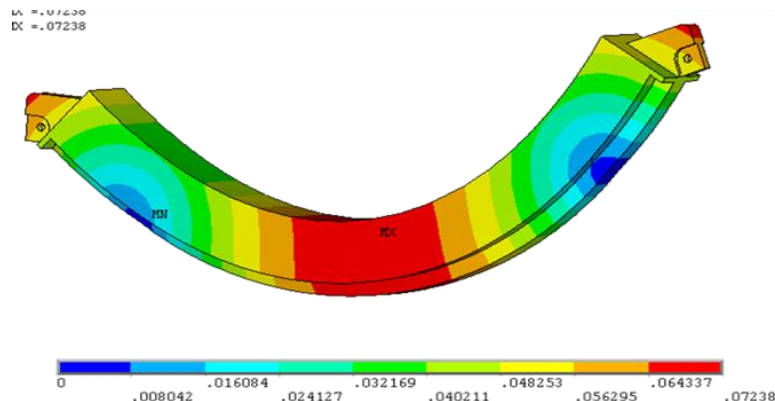


Fig:4.1.5.6(b): Cradle

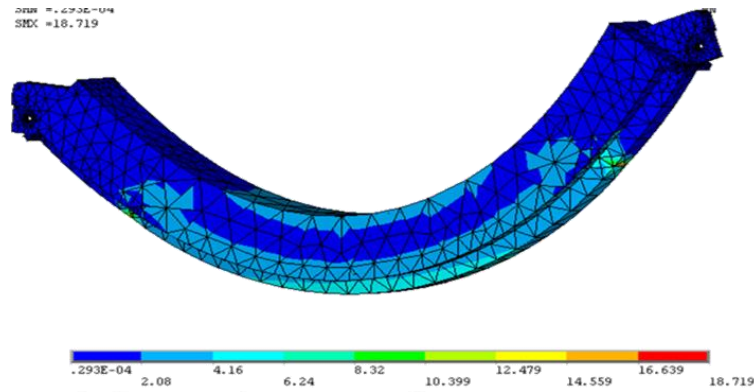


Fig:4.1.5.6(c): Cradle

4.1.5.7 Pin Supporting top frame on lifting screw

Load on each pin = (Load of component 4 + load of top frame + rollers + cradle + supporting frame)/4
 $= (750 + 550)/4$ since 4 pins are sharing load
 $= 1300 / 4 \text{ Kg} = 325 \times 9.81 = 3188.3 \text{ N}.$

The above load is acting as a uniformly distributed load on Pin UDL (w) = $325 \times 9.81 / 92 = 34.655 \text{ N/mm}$

Maximum Bending stress $f_b = M/Z$

Maximum bending moment = $wl^2 / 8 = (35.655 \times 92^2) / 8$
 $= 36669 \text{ N-mm}$

Section Modulus of the Pin = $(\pi \times d^3 / 32) = 785 \text{ mm}^3$

Bending stress $f_b = M/Z = 36669 / 785 = 47.71 \text{ N/mm}^2$

Pin will also undergo under double shear by the presence of above load.

Shear stress developed in the pin is given by

Shear stress = $W / [(\pi/4 \times d^2) \times 2]$
 $= 325 \times 9.81 / [(3.14/4 \times 20^2) \times 2] = 5.082 \text{ N/mm}^2$

Maximum Principal stress = $f_b / 2 + \sqrt{\{(f_b / 2)^2 + \tau^2\}}$
 $= 47.71/2 + \sqrt{\{(47.71/2)^2 + 5.082^2\}} = 48.25 \text{ N/mm}^2$

Material of the pin is Steel En-9 in hardened and tempered condition, Where yield stress = $90 \text{ Kg/mm}^2 = 883 \text{ N/mm}^2$

Factor of safety = yield stress / Principal stress = $883/48.25 = 18$

4.1.5.8 Roller Supporting Bracket

Bending Stress of Roller Supporting Bracket

Tangential / horizontal component of radial load on supporting ring will cause bending of roller supporting bracket about the base of mounting.

Bending stress $f_b = M/Z$ will be maximum at the base of roller.

Section modulus at the base of mounting of roller supporting bracket = $t w^2 / 6 = 12 \times 200^2 / 6 = 80000 \text{ mm}^3$

Bending Moment generated by tangential load = $W_t \times h$

$$(M) = 1416.56 \times 100 = 141656.4 \text{ N-mm}$$

Bending stress $f_b = M/Z = 141656.4 / 80000 = 1.77 \text{ N/mm}^2$

Normal Stress = normal Load / area = $2452.5 / (12 \times 200)$

$$= 1.022 \text{ N/mm}^2 \text{ (Compressive)}$$

Maximum Principal Stress = $f_b + \text{Normal stress} = 2.792 \text{ N/mm}^2 \text{ (Compressive)}$

The material used for Roller supporting bracket is structural steel plates where yield stress = $12 \text{ Kg/mm}^2 = 117.7 \text{ N/mm}^2$

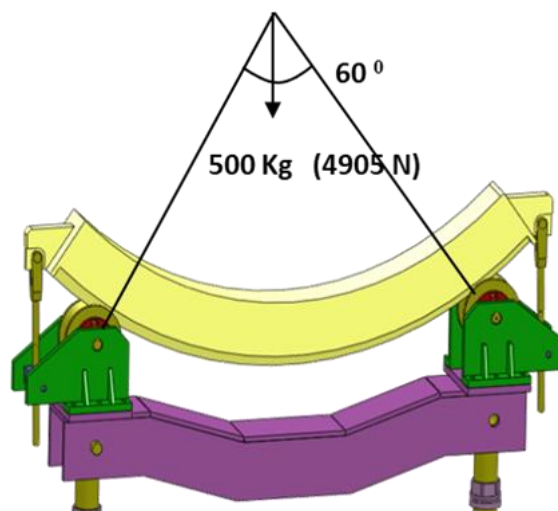


Fig:4.1.5.8(a): Roller supporting bracket

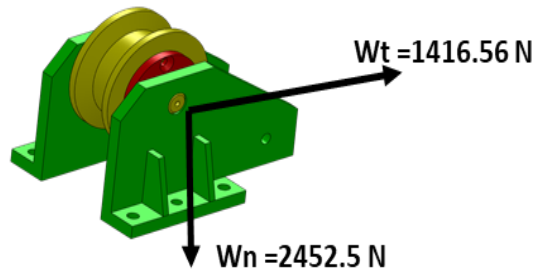


Fig:4.1.5.8(b): Roller supporting bracket

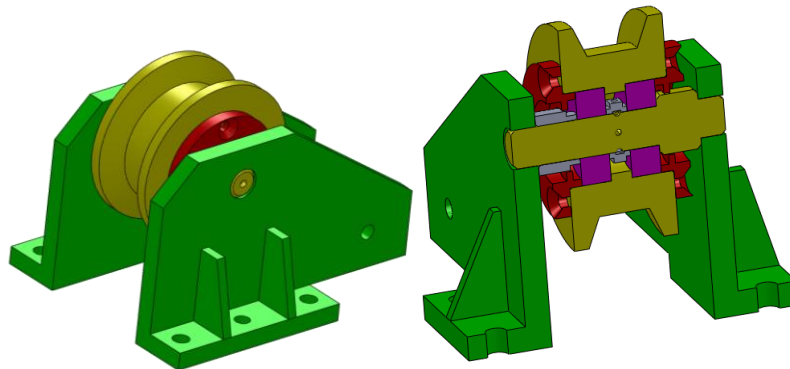


Fig:4.1.5.8(c): Roller supporting bracket

4.1.5.9 Support Bracket for lifting screw

Support bracket for lifting screw locates the nut in its groove and will not allow the nut for lifting screw to move longitudinally. Support bracket allows the nut rotation about its axis. Thus support bracket imparts longitudinal motion to lifting screw when nut is rotated.

The support bracket for lifting screw is attached to sliding frame using 4 bolts of M10.

Loading cases:

Case a: When the component-1 on support trolley is lowered by rotating the nut. In this case the support bracket base rest on sliding frame and the bracket will be under pure compression.

Case b: When the component-1 on support trolley is raised by rotating the nut. In this case the support bracket upward motion is constrained by the bolts through which it is attached to sliding frame. Entire load is taken by the groove internal segment in tension.

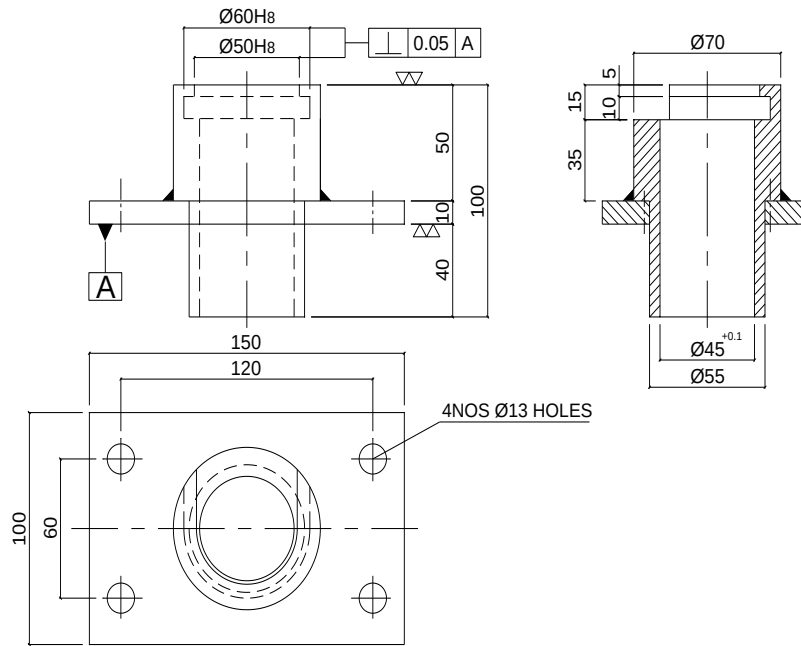


Fig:4.1.5.9: support bracket for lifting screw

4.1.5.9.1 FEA analysis

The support bracket for lifting screw is modeled in Ansys.

Solid 45 element is used for meshing using free mesh option.

The Following material properties are used Young's modulus $E = 2E5 \text{ N/mm}^2$ and poissons ratio=0.3

All the four base mounting holes are constrained.

A load of 3188.25 N (325 Kg) is applied on internal segment grove area along the axial direction on nodes attached to that area.

The results of Analysis.

Maximum stress of 66.6 N/mm^2

Maximum deflection found is 0.012mm at segment grove area

The material of support bracket is IS: 2062

Gr'B' where yield stress = 25 Kg/mm^2
 $= 245.25 \text{ N/mm}^2$.

Factor of safety = yield stress / working stress = $245.25/66.6 = 3.7$.

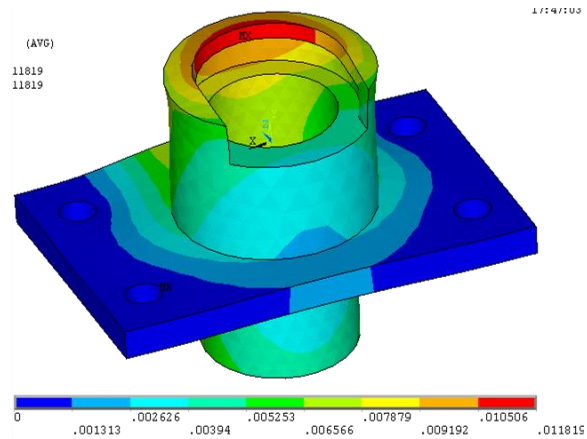


Fig:4.1.5.9(a): FEA analysis of lifting screw

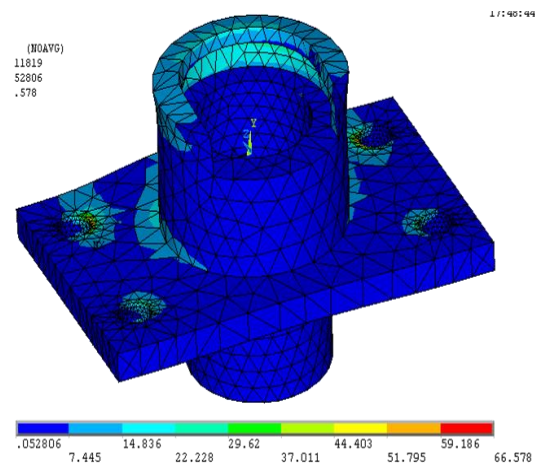


Fig:4.1.5.9(b): FEA analysis of lifting screw

4.1.5.10 Sliding frame

The sliding frame is constrained by guided roller pin and groove. Hence the hole which receives supporting pin is constrained in all directions at both locations.

The sliding frame is modeled in Ansys package.

A load of 3188.3 N (325 Kg) is applied on the surface which receives support bracket

Meshing of the model is carried using solid 45 element using free mesh option.

Following properties are used as material properties.

Young's modulus $E = 2.1E5 / \text{mm}^2$ and Poisson's ratio $= 0.3$

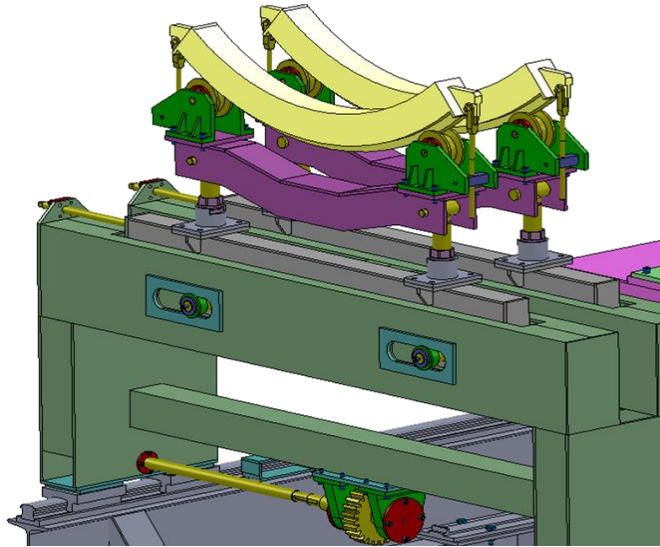


Fig:4.1.5.10(a): Sliding frame

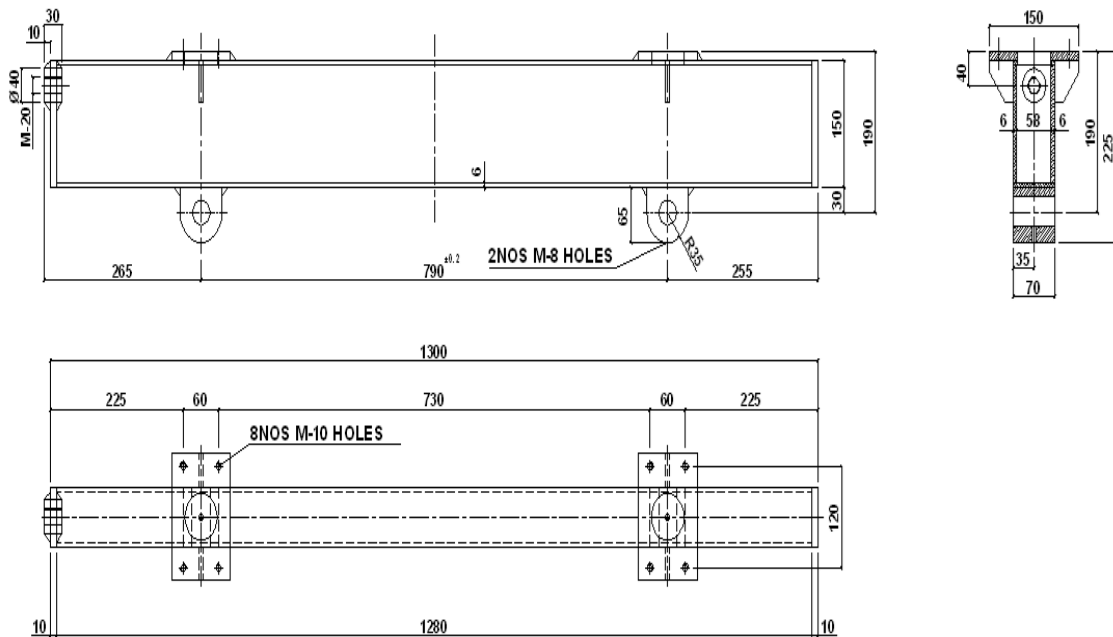


Fig:4.1.5.10(b): Sliding frame

The component-1 is supported on sliding frame which in turn is supported on bottom housing in guided roller groove as shown in model above. There are two loading cases arise

Case a: When the component-1 is lowered, the mass of component-1 act as a compressive load on support bracket and sliding frame

Case b: When the component-1 is raised, the mass of component-1 acts as a tensile load on support bracket.

The maximum stress found on sliding frame = 31.3 N/mm^2

The maximum deflection found on sliding frame = 0.021 mm

Since the cross section of sliding frame is made from 8mm structural steel plates IS:2062 Gr'B' by welding route.

The yield stress of structural steel plates = 25 Kg/mm^2

$$= 245.25 \text{ N/mm}^2$$

Factor of safety = $245.25 / 31.3 = 7.8$.

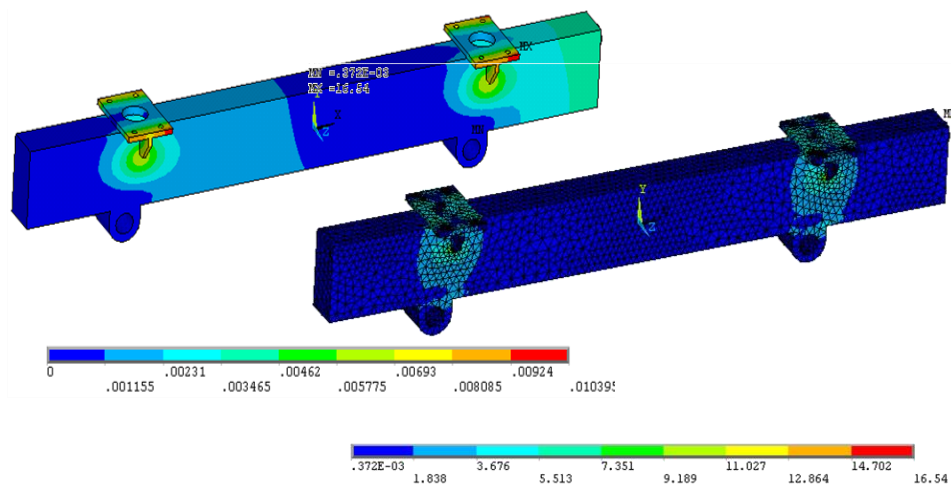


Fig:4.1.5.10(c): Sliding frame

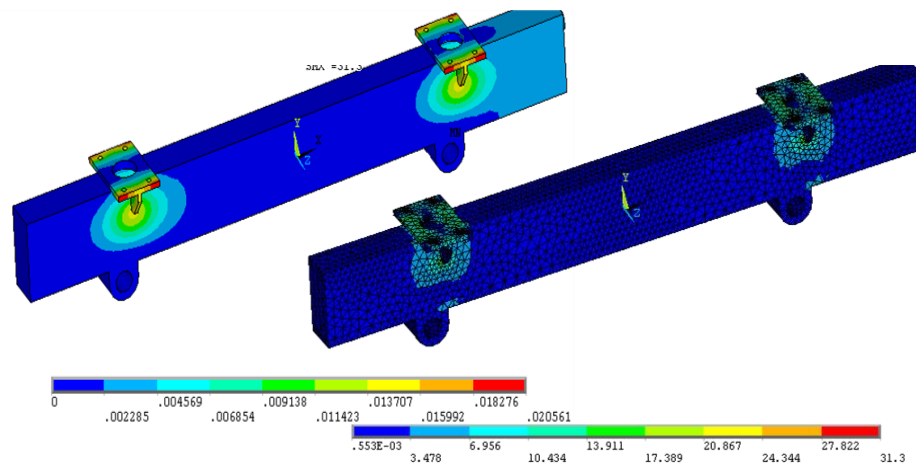


Fig:4.1.5.10(d): Sliding frame

4.2 FIXTURE BODY DESIGN

4.2.1 Track Frame

The track frame assembly is a rigid base structure which supports the assembly/handling trolleys of component-1 & component-4. It is made up of one set standard I beams (of 300mm height) connected crosswise by means of box sections as shown in the above figure. Each I –beam is supported on the ground by means of three screw jack assemblies. The top flange of each of the I-beams are welded with structural steel plate on top and machined to receive the LM guide. The assembly / handling trolleys will move on the LM guides using LM rails attached to the assembly trolleys. The track frame assembly is assumed as a simply supported beam with load of assembly trolley with component positioned acting on top it.

Analyzing the loading cases acting on the track frame, it can be concluded that worst loading case is when supporting trolley of component-4 with components-4, 3, 2 and Component1 assembled and fastened to mounting plate of support trolley. Hence this particular loading case is considered for designing the track frame.

Mass of components 1,2,3 &4 = 2550 Kg.

Mass of support trolley of component-4= 1450 Kg

Total load on track frame = 39240 N.

This load is assumed to be acting along the vertical center line of support trolley of component-4.

The above total load is shared by two base I beams.

Load on each I-beam = 19620 N.

I-beam selected is ISMB 300mm for which $I = 129502000 \text{ mm}^4$ &

Area = 4808 mm^2 and Mass = 37.6 Kg/mm

In the above figure, R_D and R_E are the reactions from track frame through LM Guides. At the above two locations, support trolley is supported on track frame using LM guides.

By calculating the reactions at points

D & E $R_D = 10884.2 \text{ N}$ (1109.5 Kg)

& $R_E = 8735.8 \text{ N}$ (890.5 Kg)

Each of the longitudinal I-beam of track frame assembly acts as a simply supported beam with three jack supports acts a pin joints of beam. The screw jack support allows rotation of the beam at its support point on the ground as the base end is made sphere inserted on a receiving spherical grove on bottom pad.

The above beam is a statically indeterminate beam where in the number of unknowns is more than number of equations. However reactions (R_A , R_B & R_C) can be calculated by using principle of superposition.

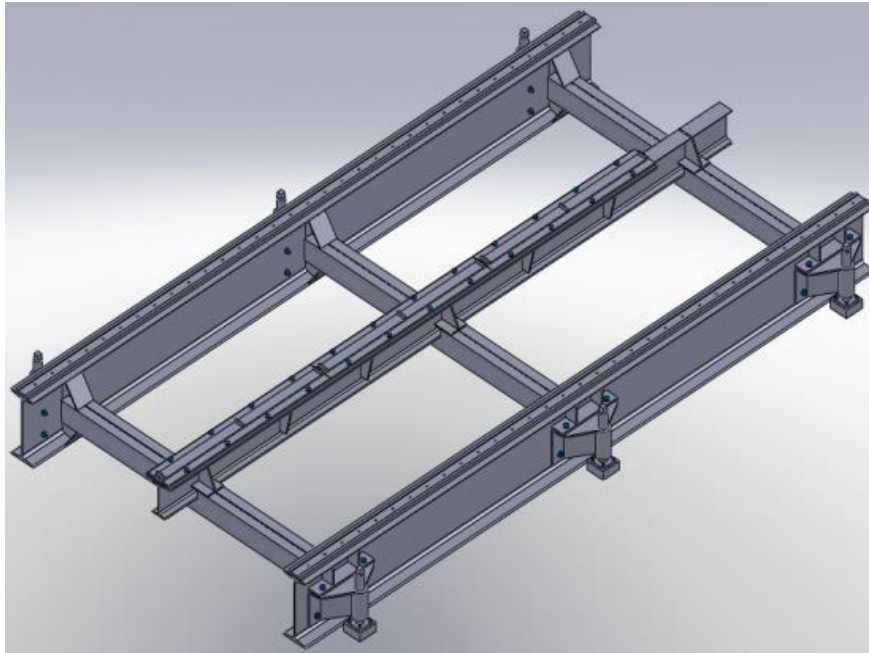


Fig:4.2.1(a): Track frame

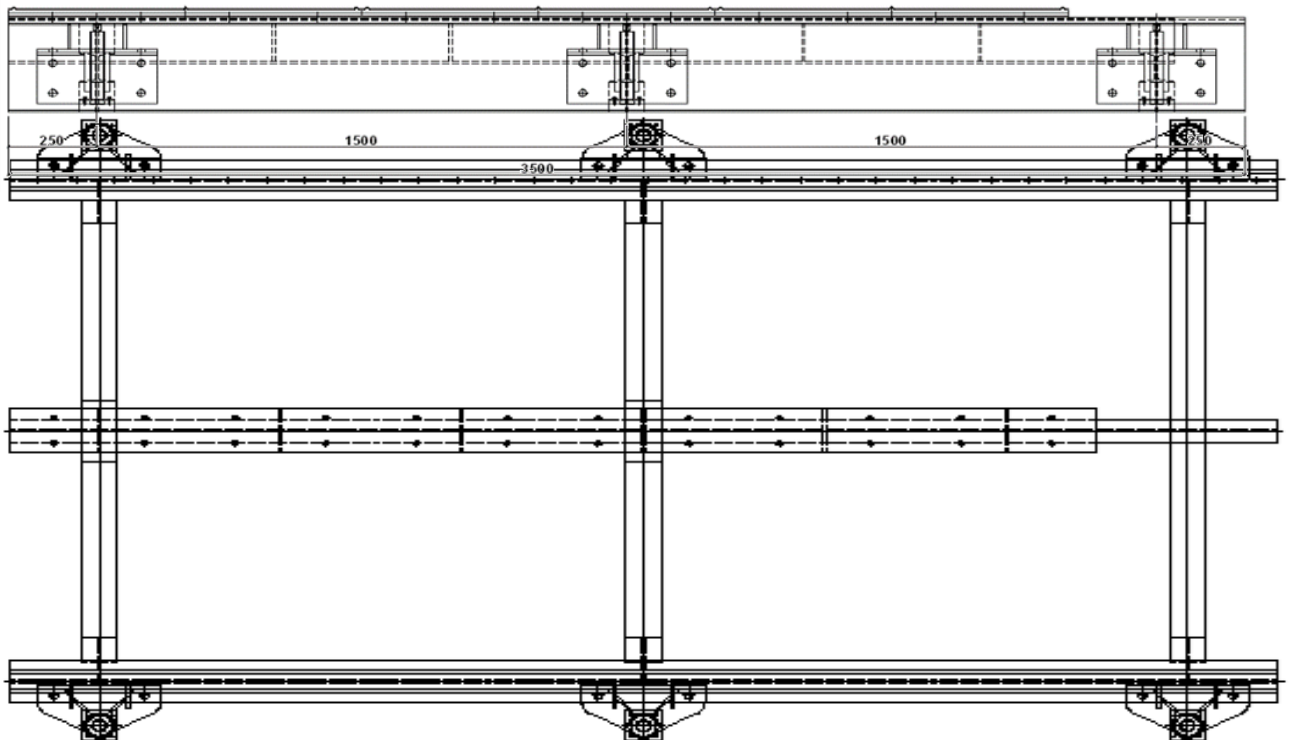


Fig:4.2.1(b): Track frame

The solving procedure is as follows.

Remove the middle support at B and consider the simply supported beam that spans between the two outer supports. The applied load acts on this simply supported beam. Calculate the reactions.

Compute the deflection at B using the appropriate deflection formula.

Consider the simply supported beam that spans between the two outer supports subjected to unknown upward force (reaction) Z that acts at the middle support. Calculate the reactions at the supports.

Compute the beam deflection at B expressed in terms of unknown force Z

The deflection computed in '2' and '4' should be made equal to compute force 'Z' which is equal to reactions R_B

The remaining reactions R_A and R_C can be computed by using force and moment equations of the beam.

By adopting the above procedure, the reactions forces of supports are determined

$$R_A = -314.85\text{N}, R_B = 11420.8\text{N}, R_C = 8514.1\text{N}$$

Bending stress of track frame

The maximum bending stress and deflection of the beam occurs between supports B & C.

To compute the maximum bending stress, it is required to find the distance from one of the supports at which maximum deflection of the beam occurs.

The maximum deflection of the beam can be found by double integration method

$$EI \frac{d^2y}{dx^2} = \pm M$$

$$EI \frac{d^2y}{dx^2} = -R_B x \mid + 1109.5(x-65) \mid + 890.5(x-1435)$$

By integrating the above equation with respect to x.

$$EI \frac{dy}{dx} = -1164.2(x^2/2) + C_1 \mid + 1109.5/2(x-65)^2 \mid + 890.5/2(x-1435)^2$$

By integrating the above equation with respect to x.

$$EI y = -1164.2(x^3/6) + C_1 x + C_2 \mid + 1109.5/6(x-65)^3 \mid + 890.5/6(x-1435)^3$$

Where C_1 & C_2 are integration constants which are solved using boundary conditions. @ $x=0$ & $x=1500\text{mm}$, $y=0$

$$C_2=0 \text{ \& } C_1=72263488.8$$

$$EI y = -1164.2(x^3/6) + 72263488.8x \mid + 1109.5/6(x-65)^3 \mid + 890.5/6(x-1435)^3$$

$$EI \frac{dy}{dx} = -1164.2(x^2/2) + 72263488.8 \mid + 1109.5/2(x-65)^2 \mid + 890.5/2(x-1435)^2$$

Deflection y is maximum at x where $dy/dx|_x=0$, since maximum deflection occurs between the two loads, first two portions of the above equation should be considered

$$0 = -1164.2(x^2/2) + 72263488.8 | + 1109.5/2(x-65)^2$$

Solving the above equation on excel , we get $x = 794.8963 \text{ mm}$

By substituting $x = 794.8963 \text{ mm}$ in the bending moment equation, Maximum bending moment can be calculated

$$M = EI \frac{d^2y}{dx^2} = -1164.2x | + 1109.5(x-65)$$

$$M_{\max} = 115598.3276 \text{ Kg-mm} = 1134019.6 \text{ N-mm.}$$

By substituting $x = 794.8963 \text{ mm}$ in the deflection equation, Maximum deflection can be calculated

$$Ely = -1164.2(794.8963/6) + 72263488.8x \ 794.8963 | + 1109.5/6 x (794.8963-65)^3 \text{ where } E = 2.1E5 \text{ N/mm}^2 \text{ \& } I = 129502000 \text{ mm}^4$$

$$y_{\max} = 0.012 \text{ mm. Bending stress of track frame} = M_{\max}/Z$$

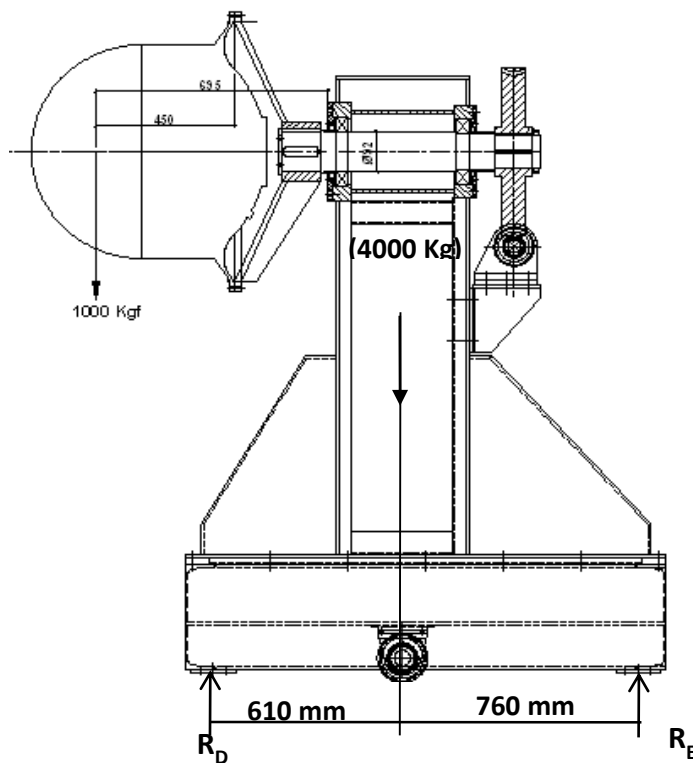
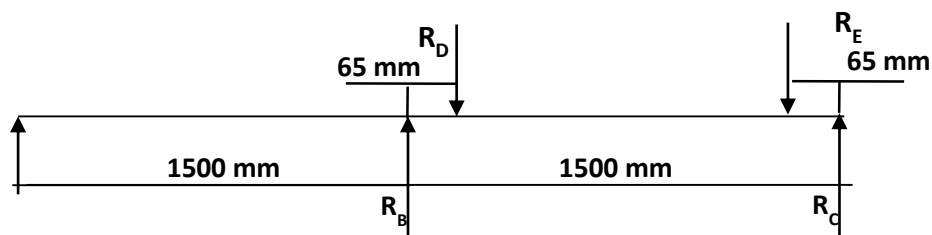


Fig:4.2.1(c):Track frame



Where M is bending moment & Z is section modulus

For ISMB 300, $I = 129502000 \text{ mm}^4$ & $y = 300/2$

Section modulus $Z = I/y = 2 \times 129502000 / 300 = 863346.67 \text{ mm}^3$

Bending stress = $M_{\max} / Z = 1134019.6 / 863346.67 = 1.314 \text{ N/mm}^2$

Yield stress of I-beams (Indian standard medium beam) which is made of standard structural steel = $12 \text{ Kg/mm}^2 = 117.7 \text{ N/mm}^2$

4.2.2 Track Frame Screw jack bracket

Track frame screw jack bracket supports the screw jack of track frame assembly as shown in the above assembly model. Each of the longitudinal I-beams of track frame assembly has three screw jack assemblies (in turn three screw jack brackets) through which the track frame assembly rests on the ground.

Analyzing the loading cases, the worst loading case is when support trolley of component-4 with component-4, 3, 2 & 1 assembled and fastened to trolley supports on one set of track frame screw jack assemblies.

Mass of assembly trolley of component-4 = 1450 Kg

Mass of components 4,3,2 & 1 = 2550 Kg

Total mass of the components & support trolley = 4000 Kg

The above load is shared by two screw jack assemblies' in turn two screw jack brackets

Mass of track frame assembly = 320 Kg

The above load is shared by six screw jack assemblies' in turn six screw jack brackets.

Load acting on screw jack bracket = $(4000/2 + 320/6) \times 9.81 = 20150 \text{ N}$

Material considered for track frame jack bracket is steel 817M40 BS: 970 En24 in Hardened and tempered (30-34 HRC) condition.

Yield stress of the material considered = $85 \text{ Kg/mm}^2 = 834 \text{ N/mm}^2$

Maximum Stress induced in the bracket = 245 N/mm^2

Factor of safety = yield stress / working stress = $834/245 = 3.4$

Bolt joint of Screw jack bracket to track frame

$$M = 2fr_1^2 + 2fr_2^2$$

Due to the presence of offset load of 20150N, the bracket will bend about the top right edge (about fillet edge)

$$M = 20150 \times 150 = 3022500 \text{ N-mm.}$$

The above bending moment will create bolt force for all mounting fasteners.

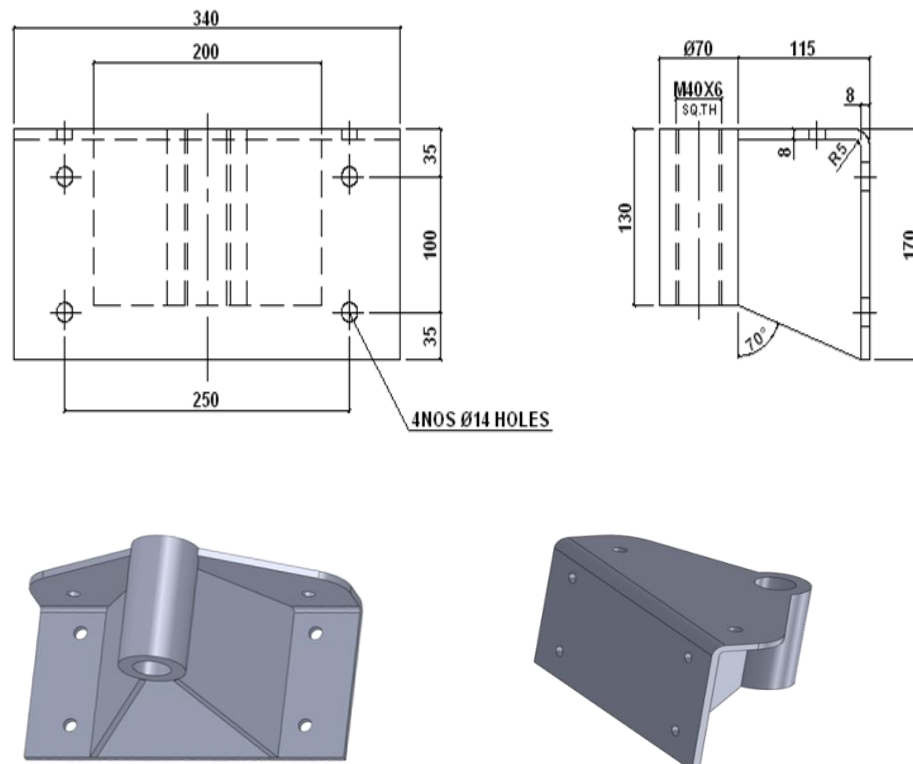


Fig:4.2.2: Track Frame Screw jack bracket

Bolt force corresponding to the above moment = $2f(35^2 + 135^2)$

$$3022500 = 2f(35^2 + 135^2) \Rightarrow f = 77.7 \text{ N/mm}$$

Maximum bolt force $f_{\max} = 77.7 \times 135 = 10490 \text{ N}$, 2 sets of M12 bolts of class 10.9 are used. Force due to pretension = $0.3 \times \text{max bolt force} = 24270 \text{ N}$

(Since $f_{\max}$ for M12 bolt = 80903 N)

Actual Bolt force = $F_{\text{pretension}} + F_{\text{external moment}} = 24270 + 10490 = 34760 \text{ N}$

Factor of safety = ultimate bolt force/ actual bolt force = $80903/34760 = 2.33$

4.2.3 Stability of Handling trolley

Supporting / Handling trolley of component-4 will tilt about the first LM block due to offset mass of component-4 being fastened to fore end of the assembly trolley

Moment generated by the offset mass of component-4 about Front LM block to topple assembly trolley = $1000 \times 9.81 \times 308 = 3021480 \text{ N-mm}$

Restoring Moment generated by the self weight of the Assembly trolley about front LM block = $1450 \times 9.81 \times 765 = 10881742.5 \text{ N-mm}$

Since restoring moment is more than moment generated by offset mass of component 4, assembly/ handling trolley will not topple. Force required to topple the assembly trolley of component-4

Toppling force = Restoring Moment / 308 = 35330.3 N.

If the offset mass increases from 1000 Kg to 3601.5 Kg

(35330.3N), Assembly / handling trolley of component-4 will topple about front LM guide.

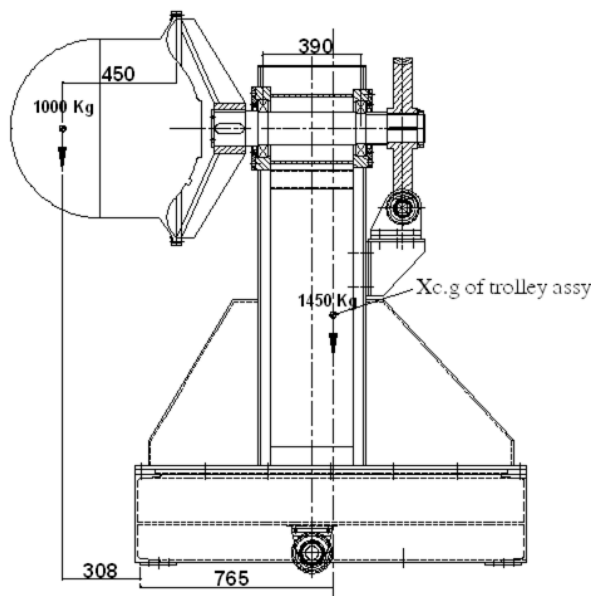


Fig:4.2.3: Stability of Handling trolley

CHAPTER 5

DESIGN VALIDATION AND PERFORMANCE ANALYSIS

5.1 EXPECTED INACCURACY OF FIXTURE

Table:5.1.1: Assembly/ Supporting trolley of component-1

Component / sub-assembly	Misalignment contribution	
	Along Y- axis	Along Z-axis
Bottom frame	0.4 mm	0.2 mm
Sliding frame		0.2 mm
Lifting screw, Nut and their resting bracket	0.45 mm	
Top frame		0.1 mm
Roller supporting bracket and Roller assembly		0.4 mm
Overall expected deviation	0.85mm	0.9 mm

Table:5.1.2: Assembly/ Supporting trolley of component-4

Component / sub-assembly	Misalignment contribution	
	Along Y- axis	Along Z-axis
Housing for support trolley	0.3 mm	0.3 mm
Mounting plate		0.282 mm
Overall expected deviation	0.3 mm	0.582 mm

5.2 CORRECTION OF INACCURACY OF FIXTURE

5.2.1 Correction of inaccuracy of supporting trolley-1

The misalignment / in-accuracy of the supporting trolley for component-1 can be corrected in the following way.

For movement of the supporting trolley along Y-axis, pitch of the screw being used is M20x2.5mm. Hence by rotating the screw by one third of full rotation will correct the misalignment along Y-axis.

For movement of the supporting trolley along Z-axis, pitch of the lifting screw being used is M44x3. Hence by rotating the nut for lifting screw by one third of full rotation will correct the misalignment along Z-axis.

5.2.2 Correction of inaccuracy of supporting trolley-2

There is no feature available on support trolley of component-4 to correct the misalignment / in-accuracy. However the required correction can be performed on the support trolley of component-1 during assembly stage.

For movement of the supporting trolley of component-1 along Y-axis, pitch of the screw being used is M20x2.5mm. Hence by rotating the screw by $\sim 1/8^{\text{th}}$ of full rotation will correct the misalignment along Y-axis.

For movement of the supporting trolley of component-1 along Z-axis, pitch of the lifting screw being used is M44x3. Hence by rotating the nut for lifting screw by $1/5^{\text{th}}$ of full rotation will correct the misalignment along Z-axis.

5.3 ACCEPTANCE OF THE FIXTURE

The following procedure is to be adopted for acceptance of the above fixture after fabrication.

5.3.1 Load test for Design Validation and workmanship

The fixture after fabrication is to be load tested for design validation and prove the workmanship. The load to be used for load testing should be twice the functional load and should confirm to geometry, size and center of gravity requirements of actual components to be assembled.

5.3.1.1 Load testing of assembly trolley of Component-4

Configure the load for load testing of assembly / support trolley of component-4. The load to be used should be 2000 Kg. Center of gravity, Size of load, geometry and assembly interface should confirm to actual component-4 specifications.

Attach the load to component-4 mounting plate using fasteners. The load should be held for 15 minutes and check for deflections, permanent set of members and cracks developed in critical load bearing weld joints. The assembly trolley should not be moved while carrying out this test

Check for crack development in the fillet weld joint of mounting plate to its HUB.

Deflection of extreme end of the load with respect to center of shaft Hole.

Check for stability of assembly trolley by applying force using one hand at the top most point of the trolley

Deflection of track frame screw jack bracket.

Acceptance criteria for the above checks is no cracks are permitted. No deflection is permitted. Should not become unstable. Deflection to the extent of 0.2 mm is permitted

5.3.1.2 Load testing of assembly trolley of Component-1

Configure the load for load testing of assembly / support trolley of component-1. The load to be used should be 1500 Kg. Center of gravity, Size of load, geometry and assembly interface should confirm to actual component-4 specifications.

Place the load on two support cradles of component-1 assembly trolley. The load should be kept in position for 15 minutes and check for deflections, permanent set of members and cracks developed in critical load bearing weld joints. The assembly trolley should not be moved while carrying out this test

Deflection of Bottom frame

Deflection / permanent set of Guided roller supporting pin

Deflection of Lifting screw support bracket

Deflection of Top frame, Roller support bracket

Deflection/ permanent set of Roller supporting pin & Support cradle

Since all the above elements are having large margin of safety, no deflection / deformation are permitted.

5.3.2 Functional test at no-load (Using Empty Components)

Empty component-4 should be fastened to its lifting bracket and Using crane attach the component-4 to its mounting plate of support trolley. Attach component -1 to its lifting bracket and using crane place component-1 on its assembly trolley support cradles. Ensure that the sliding frame of support trolley of component-1 is at the center of the fixture (this can be adjusted using motion along Y-axis using screw nut mechanism). The center line of component-4 should be maintained at a distance of 2094 mm from ground (this can be ensured using motion along Z-axis using lifting screws arrangement). Attach component-3 to its lifting bracket and lift using crane. Assemble rear end of component-3 to fore end of component-4 using slow motions of crane. Attach component-2 to its lifting bracket and lift using crane. Assemble fore end of component-2 to aft end of component-1 using slow motions of crane.

Move the support trolley of component-4 with component-3 & 4 assembled towards component-1 support trolley using LM guide and rail arrangement. When component-4 is about to enter into aft end of component-2 (Component-1 & 2 assembled) check the alignment along Y & Z –axis. Measure radial gap between the OD of component-4 and ID of component-3 and ensure gap is same at two diagonally opposite locations. If required use motion of jig along Y-axis and Z-axis using screw nut and lifting screw arrangement respectively to make it concentric. Slowly move support trolley of component-4 till flange of component-4 butt against the face of Component-2. This test proves that the fixture is free from manufacturing and assembly errors and fit for using as a assembly fixture.

5.3.3 Functional test at full load (using Fully Equipped Components)

Repeat the same procedure as adopted in the above functional using fully equipped components instead of empty components. Components should get assembled with not hindrance. This test proves the functionality of the jig at load.

The assembly fixture should qualify through the above mentioned three tests to use it as a assembly fixture.

5.4 FINAL FIXTURE DESIGN

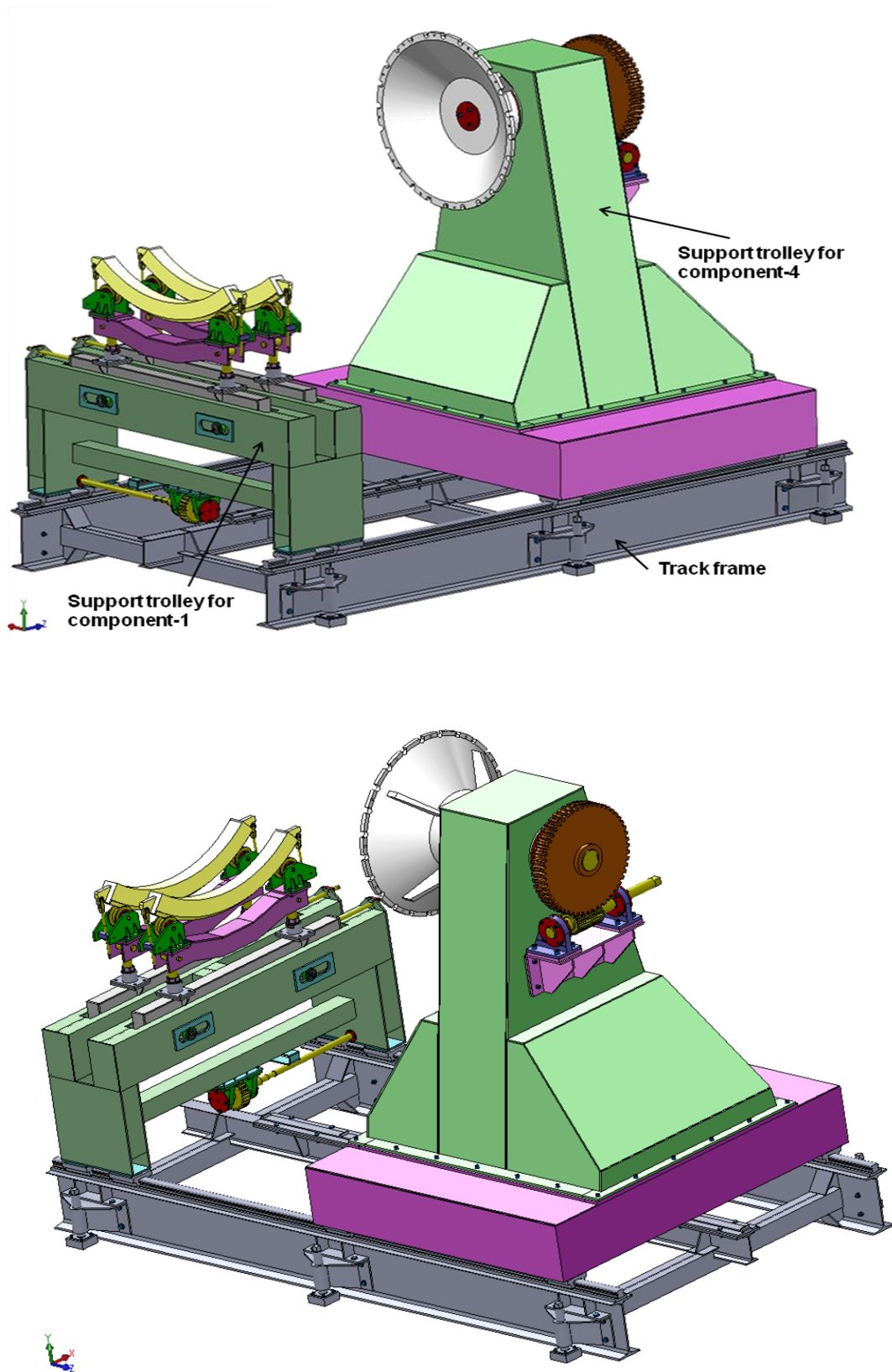


Fig:5.4 Fixture Design

CHAPTER 6

SUMMARY AND CONCLUSIONS

The main aim of this project is to establish an integration sequence and develop an integration fixture to assemble four components of a missile developed by DRDO.

The integration fixture proposed is based on integration methodology to be followed for assembly of four components of a missile. The proposed integration fixture mainly comprises of Base / track frame with linear motion guide / rail and screw jack arrangement, supporting / assembly trolleys for component 1 & component-4. No assembly trolleys are planned for component-2 & 3 due to space constraint during assembly. Although Lifting brackets are not part of the fixture, they are presented in the development of the fixture as they are required for assembly of components.

The integration fixture satisfies the functional requirements of the integration fixture placed under problem definition. It has got factor safety of 4 on yield stress for all major load bearing members and it has good stiffness and enough stability for operational safety.

All the motions of the assembly fixture should be given manually and automation is not incorporated due to operational and handling safety of explosive components (component-4) to be assembled.

Design calculations are presented for components of the assembly fixture using classical equations and the same are supported by FEA software Ansys results where ever possible. The geometry and size of the components are considered from fixture layout and checked for design calculations.

The mass of support trolley of component-4 is made heavy for stability to counter the toppling moment created by offset mass of component-4. Although the option of counter balance scheme can also be used, that is not being adopted due to high working height.

Linear motion guide and rails selected for longitudinal motion of assembly trolleys of the fixture are having low coefficient of friction and hence the trolleys can be moved with little effort. Although the cost of linear motion rail and guides is high,

since the integration fixture is utilized for several assemblies, the cost is distributed and hence the cost increase is marginal.

Expected inaccuracy of the assembly fixture arising from the dimensional and geometrical tolerances specified on assembly/ supporting trolleys and its correction mechanism is also presented.

Acceptance of the assembly fixture for design validation and to prove the workmanship is presented in the last chapter.

CHAPTER 7

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