Design & Fabrication of Electrical Control Multipurpose Hydraulic Press Machine: A Case of Wonji /Shoa Sugar Factory

Demelash Jirru Feyisa

A Thesis Submitted to

The Department of Mechanical Engineering

School of Mechanical, Chemical and Materials Engineering

Presented in Partial Fulfillment of the Requirements for the Award of degree of Master of Science in Automotive Engineering

Office of Graduate Studies

Adama Science and Technology University

March, 2023 G.C

Adama, Ethiopia

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CANDIDATE'S DECLARATION

I hereby declare that this Master Thesis entitled "**Design and fabrication of electrical control multipurpose hydraulic press machine"**: is my original work a case of Wonji/Shoa sugar factory, Ethiopia. That is, it has not been submitted for the award of any academic degree, diploma or certificate in any other university. All sources of materials used for this thesis have been duly acknowledged through appropriate citations.

Demelash Jirru Feyisa	----------------------------	-------------------------
Name of Student	Signature	Date

RECOMMENDATION

we, the advisors of this thesis, hereby certify that we have read the revised version of the thesis entitled Design and fabrication of electrical control multipurpose hydraulic press machine: A case of Wonji/Shoa sugar factory" prepared under our guidance by Demelash Jirru submitted in partial fulfillment of the requirements for the degree of Master of Science in Automotive Engineering. Therefore, we recommend the submission of revised version of the thesis to the department following the applicable procedures.

Major Advisor	Signature	Date
Co-advisor	Signature	Date

APPROVAL OF BOARD OF EXAMINERS

We, the advisors of the thesis entitled "Design & Fabrication of Electrical Control Multipurpose Hydraulic Press Machine: A Case of Wonji /Shoa Sugar Factory" and developed by Demelash Jirru, hereby certify that the recommendation and suggestions made by the board of examiners are appropriately incorporated into the final version of the thesis.

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Co-advisor	Signature	Date

We, the undersigned, members of the Board of Examiners of the thesis by Demelash Jirru have read and evaluated the thesis entitled "Design & Fabrication of Electrical Control Multipurpose Hydraulic Press Machine: A Case of Wonji /Shoa Sugar Factory" and examined the candidate during open defense. This is, therefore, to certify that the thesis is accepted for partial fulfillment of the requirement of the degree of Master of Science in Automotive Engineering.

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Finally, approval and acceptance of the thesis is contingent upon submission of its final copy to the Office of Postgraduate Studies (OPGS) through the Department Graduate Council (DGC) and School Graduate Committee (SGC).

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ABSTRACT

ECMHPM is an electrically controlled multipurpose hydraulic press machine used to operate an electrical joystick and both pulling and pushing hydraulic cylinders. Most hydraulic press machines use a main cylinder to generate a compressive force. Hydraulic systems use Pascal's law, which focuses on pressure applied to an enclosed fluid being transmitted without a change in magnitude to every point of the fluid. In hydraulic presses, the same principle is used. Presses are classified into three principal categories, such as hydraulic presses, screw presses, and mechanical presses. The performance of a hydraulic press depends largely upon the behavior of its structure during operation. This thesis is focused on solving a tangible problem observed in the case industry, Wonji/Shoa Sugar Factory, which is concerned with the lack of hydraulic press machines, specifically in the workshop where the researcher is working. Thus, the main goal of this thesis is to design and fabricate an electrically controlled multipurpose hydraulic press machine to alleviate the problems mentioned above, reduce high human effort, reduce unnecessary costs, and use hydraulic press machines for any work that is close to the case's workshop. The methodologies followed were data collection, design, and fabrication; modeling; analysis using the ANSYS 2021R1 and 2022R2 programs; and testing of the performance of ECMHPM. The objective of this thesis was to get the newly designed press machine's parameters, such as stress distribution and total deflection, to be in the safe region of a design standard while also having an economically cost-effective hydraulic press machine. Finally, the design calculation results were compared with the simulation results for validation and found to be matching and in a safe region. The design and testing of the fabricated ECMHPM showed that the output of the electrical pump and the capacity of the cylinder have remarkable performances that minimize human labor as well as cost. As a result, these works, which required solving the industry's problem, achieved their goals. Finally, it can be recommended that problems of similar nature in different industries be solved by following the same methods.

Keywords*: Dissolved materials, electrical control, hydraulic press machine, human labor and Performance.*

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ACRONYMS

AP	Auxiliary Piston
BDC	Bottom Dead Center
GP	Guide Pin
HA	Holding Adapter
HPM	Hydraulic Press Machine
ECMHPM	Electrical Control Multipurpose Hydraulic Press machine
ECP	Electrical Control power
EHP	Electrical Hydraulic Press
FESD	Flied Equipment Service Department
MHP	Multipurpose Hydraulic Press
MT	Movable Table/Horizontal bed/
SAE	Society of American Engineers
SR	Sliding Roller
SST	Special Service Tools
TDC	Top Dead Center

CHAPTER ONE

INTRODUCTION

1.1. Background of study

A hydraulic press is a device using a hydraulic cylinder to generate a compressive force. It uses the hydraulic equivalent of a mechanical lever and was also known as a Bramah press after the inventor, Joseph Bramah, of England. He invented and was issued a patent on this press in 1795 (Muni Prabaharan and V.Amarnath, 2011).

press can be classified into three principal categories as hydraulic presses which operate on the principles of hydrostatic pressure, screw presses which use power screws to transmit power and mechanical presses which utilize kinematic linkage of elements to transmit power Most A hydraulic press is a machine using a hydraulic cylinder to generate a compressive force. Frame, hydraulic cylinder, and press table are the main components of the hydraulic press (Sumaila, M., 2002). Hence a hydraulic press is a machine that makes use of the pressure exerted on the fluids to crush, remove, straighten, or mold.

The concept of the hydraulic press is based on Pascal's theory, which states that when pressure is applied to fluids in an enclosed system, the pressure throughout the system always remains constant. In the hydraulic press, the force generation, transmission, and amplification are achieved using fluid under pressure. The liquid system exhibits the characteristics of a solid and provides a very positive and rigid medium of power transmission and Amplification. In a simple application, a smaller piston transfers fluid under high pressure to a cylinder having a larger piston area, thus amplifying the force (Nawale Sagar et. al, 2016).

This research done under Wonji / Shoa Sugar Factory for field equipment service department. The factory is found at Oromiya region near Adama City at 110 Kilo Meters from Addis Ababa. Wonji Sugar Factory is the oldest and the pioneer in the history of Ethiopia’s sugar industry. Commencing production in 1954. Then, followed Shoa Sugar Factory in 1962; is the second oldest and both, being obsolete, have stopped production since July, 2012 and July, 2013 respectively. The two factories constructed by the Holland Company known as H.V.A had a capacity of producing 75 thousand tons of sugar a year. The sugarcane plantation land of these two factories was 7,000 hectares out of which 1,000 had been planted by out growers. In a bid to replace these two oldest factories with a new and modern one, Accordingly, the newly built and modern Wonji/ Shoa Sugar Factory has currently a design capacity of crushing 6,250 tons of cane a day and producing 174,946 tons of sugar per annum which with further expansion work will reach up to 12,500 TCD maximizing its production to 220,700 tons of sugar a year. The new plant, when having an ethanol plant, will initially have a capacity of producing 10,000 meter cube ethanol per annum that with further expansion will reach 12,800meter cube. The Factory is currently contributing 20 megawatt electric powers to the national grid in addition to satisfying its owndemand which is around11Megawatt.(https://etsugar.com/neww/2019/09/10/).practical view of wonji/shoa sugar factory in figure 1.1

Figure 1. 1. Partial view of Wonji/ Shoa Sugar Factory

Under Wonji/ Shoa Sugar Factory, there is one maintenance department built up to maintain different machineries. Which is locating Wonji town name is known as filed equipment service department (FESD). There are different kinds of machinery available there. For instance, light duty, heavy duty, and earth moving machineries Up to date different kind technical problems available on there. One of the problems is that it is difficult to remove press-fitting material. So, these are the steps to design and fabricate an electrically controlled multipurpose hydraulic press machine for different activities. This is electrically operated and uses liquefying material. This will not only help to recover the excise lost in the form of foreign exchange but also improve the level of our local technology.

1.2 Statement of the Problem

Heavy farm machineries of different types are employed for various duties in Wonji/Shoa Sugar Factory. Since the working environment for the machineries is very difficult, they are mostly subjected to failures that need corrective maintenance. The size of the machineries and their components are large. Disassembling and reassembling of these big components need hydraulic press machines but these machines vary far from central garage (FESD) and need a lot of energy. So, the working environment is very tiresome and unsafe. There are number of maintenance activities. Such as; Disassembling and reassembling of torque rod bushing, lateral rod bushing, hub bearing and so on... Therefore, solving this problem of paramount importance to reducing machinery down time, human labor and rising the application of hydraulic press machines to increase profitability and make maintenance fast, easy and safe. To achieve these benefits the researcher intended to design & fabricate an electrically controlled multipurpose hydraulic press machine using scraped available materials in the work shop with reduced cost as compared to market price.

1.3 Objectives

1.3.1 General Objective

The general objective of this thesis is to design & fabricate electrically controlled multipurpose hydraulic press machine.

1.3.2 Specific Objectives

The specific objectives of this study are to;

- Design and model the components of press machine;
- Analyzed the components using ANSYS software and check for validation;
- Fabricate the components and assemble;
- Test the functionality of the newly produced machine and compare with existing one;

1.4 Significance of the study

The results of this thesis, by lowering human labor and enhancing application HPM, have the potential to considerably help people, researchers, and manufacturers involved in ECMHPM design. To reduce costs, it is easily converted from dissolved material. The main beneficiaries are: industrial automotive workshops for assembling and disassembling parts; workshops for the automotive industry's maintenance; individual users as sources of income and for more Ethiopia Sugar Corporation (Wonji/Shoa Sugar Factory); and researchers to further explore this area.

1.5 Scope and Delimitation

This study is limited to the selection of appropriate material from the scrap collection area and the design of a new electrically controlled multipurpose hydraulic press machine, followed by checking for the design's safety using comparative theoretical analysis and ANSYS software to make validation.

1.6 Limitations

The researcher is limited to the unavailability of materials and the high speed of the computer.

1.7 Organization of the thesis

This thesis consists of five chapters. The **first chapter** is an introduction to the study. It includes: background of the study; statement of the problem; objectives of the study; significance of the study; scope and delimitations of the study; and organization of the thesis. **Chapter Two** focuses on the literature review based on the previous related works. Chapter three describes the materials and methods used to complete this work. **Chapter four** consists of results and discussion. **Chapter five** consists of conclusions, recommendations, and future work areas.

CHAPTER TWO

LITERATURE REVIEW

2.1 Types of Presses machine

There are many brand sorts of hydraulic presses categories to the various requirements of industries. According to (Mr.Umesh.C.Rajmane et.al., 2016) some of them are Arbor presses, laminating presses, C- frame presses; Pneumatic presses Power presses. So; the following are described as under.

2.1.1 Arbor presses

These presses are commonly used whilst the paintings concerned are not of a heavy-obligation nature. These presses come in a spread of sizes and specs. But in comparison to different hydraulic presses, they do now not compress big amounts of strain required to generate more output. Arbor presses are utilized in techniques like piercing holes into metals, stamping, pulling down metals, tearing, marking inscriptions, and so forth.

2.1.2 Laminating presses

Unlike other hydraulic presses that are operated robotically, these presses employ guide labor. Laminating presses have openings that can be known as plates. One is used for heating while the opposite is used for cooling. This makes the lamination method comparatively quicker. Through those presses, materials like polymer can be laminated onto paper and metallic. In the case of laminating presses, the plates are normally heated with oil or via power. A laminating press is likewise used for common uses along with laminating identification playing cards, certificates, and even book covers. In this way, laminating presses facilitate fast and easy lamination for commercial and home needs.

2.1.3 C- frame presses

These presses have a 'C' like form, that's particularly designed to maximize the floor area for the workers a good way to pass round without difficulty at the place of business. Unlike other presses that have multi processes, the C- frame presses only encompass a unmarried press application. Its software consists of straightening, drawing and more often than not includes assembling work. C- Frame presses are available a variety of weights. The C-frame presses are also to be had with greater capabilities together with wheel stands and strain gauges.

2.1.4 Pneumatic presses

These presses are the maximum simple hydraulic presses utilized in industries due to the fact they compress the air to create strain so that you can gain motion. The benefit of pneumatic presses is that the operations are executed speedy whereas the drawback of this press is that it cannot create extremely high pressures, as other hydraulic presses are capable of create. The pneumatic presses are frequently used in automobile and plane brakes system. The business uses of pneumatic presses would encompass assembling, drawing, punching, and so on. A pneumatic press normally calls for a full-time operator and for the sake of his protection; additional safety add-ons such as electrical safety devices are also blanketed.

2.1.5 Power presses

These presses are utilized in massive industries that call for the use of heavy equipment and device. There are 2 sorts of electricity presses on the basis of the form of grab used. They are complete revolution and component revolution grasp. In the case of a complete revolution snatch, the size can't be disrupted until and unless the crankshaft makes a full revolution. In the case of part revolution, the clutch can be disrupted at any time, before or after the overall revolution. Power presses contain numerous risks due to the heavy operations related to them. A lot of safety measures are taken whilst the usage of energy presses.

2.2 Categorization of Hydraulic Presses as per column design

Hydraulic Presses device Categorization Depending upon their shapes, layout and they could be divided into six extensive businesses. According to (S. Raja Shekar et al., 2018) some of them are; Round column press, fabricated column press, Close-body press, C-Frame press, fabricated chamber press, Nonconventional press.

2.2.1 Round Column Press

The important capabilities of those varieties of presses are their spherical columns. In these forms of presses pinnacle and backside, the platen is made-up and machined individually and then held jointly with the aid of spherical columns and nuts. As all of the mechanisms might be machined independently and exactly, therefore, these kinds of presses are the maximum accurate sorts of presses, compared to all different varieties of presses. (Columns also are referred to as pillars). Round column presses might be in addition sub-divided into 3 categories. Two columns press, three columns press, and 4 columns press.

Figure 2. 1. Various types of hydraulic press frames (a) Two Column Press, (b) Three Column Press (c) Four Column Press (d) Fabricated Four Column Press (e) Fabricated Two Column Press or H-Frame Press (f) C-Frame Press (g) Close-Frame Press (h) Fabricated Chamber Press**.**

2.2.2 Two Column Press

In the case of two columns press, crest, and base platens are tightened jointly via only two spherical columns. Hydraulic presses, which might be critical to be very accurate and urgent load, always stay at the principal axis of the principal cylinder, and additionally there may be no opportunity of taking any eccentric load, then two column forms of presses are used.

2.2.3 Three Column Press

In 3-column presses, pinnacle and bottom plates are tightened collectively by means of three round columns. Old aluminum and copper extrusion presses of higher potential were calculated in this manner. These presses had been horizontal, excessive velocity, and made for mass manufacture. A three-column design is stronger than a -column design. The advantage of this type of layout inside the case of extrusion press is that adequate area is to be had for shearing the rejected billet and loading red warm billets in a box by using an overhead crane.

2.2.4 Four columns press

The top and bottom platen are tightened together by using 4 round columns. These varieties of presses are broadly utilized in industry in comparison to other kinds in which accuracy is more important among Dies. For example, powder compacting presses, plastic injection molding machines, and so forth.

2.3 Fabricated Type of Column Press

In this kind of presses top and bottom platens are completely welded collectively with the help of fabricated columns. Made-up column presses are sturdier, low-cost, and feature much less deflection underneath load in comparison to round column presses. Fabricated column press additionally could be divided into two categories, Fabricated Four-column press & Fabricated Two-column press or H-Frame press.

2.3.1 Fabricated Four Column Press

The high capability of hydraulic presses with large-size tables is manufactured on the layout of the fabricated four-column press. As it is sturdier and gives sufficient area to work and take a look at the urgent operation from all sides, in an evaluation too, -column press. For instance, deep drawing press for automobile body, punching, and blanking of the huge size of Steel, and many others.

2.3.2 Fabricated Two Column Press or H-Frame Press

Medium and low ability and competitively priced presses are manufactured on the layout of fabricated -column or H s. Frame press. In low potential, presses rolled metal. Channel or I-segment is used as a faceted column and inside the case of the medium capacity press; its miles fabricated from a metal plate. Rubber mounding, variable-day-light presses used in garages are for example of H-Frame press.

2.4 Close-Frame Press

In the case of the close-body press, the overall structure of the click is in an outline to rectangular ring. In the case of the small size of presses, they may be made with the aid of reducing home windows in metal plates and assembling collectively two or extra such plates to make a press body. In this type, minimum welding is needed as pinnacle, bottom, and facet columns are all internal. In the case of huge presses required for fashionable fabrication, including dishing, plate-bending, straightening, and pre-pinching of plates for rolling’s top, backside platens, and side columns are fabricated one at a time then welded together.

2.5 Fabricated Chamber Press

In this form of hydraulic press, a metallic fabricated container structure or box shape the principal frame of the hydraulic press. Major cylinder, various doors, feeding association auxiliary cylinder are hooked up in this fabricated field-structure as according to the necessity of production and operation. Fabricated chamber act as a load-bearing member of the press. Sometimes in addition they act on boxes for material to be compressed. Bailing presses and extrusion presses are some examples of those types of presses. C. Finite Element Analysis of Hydraulic Press Frame Structure The major goal of a finite detail analysis is to calculate the effects of loading situations on the frame structure. Usually, it is used to decide the displacement, stresses, traces, and force reactions in structures or additives whilst subjected to hundreds that don't set off damping and inertia effects. Assumptions are taken into consideration for loading and reaction conditions i.e., the hundreds and structures response is believed to be static independent of time.

2.6 Previous researcher arts

Below are referred various literatures of previous researchers and are described as under.

2.6.1 Cost & weight reduced

Mr.Asim M.Kamate et al., (2016), made Experiment to analyze and optimize the prevailing press with the help of design, development, and analysis of a 20-ton hydraulic press machine using finite element software. After the design and analysis of the new hydraulic press, it was found that the weight of the new hydraulic press is decreased as much as 50%, and the cost reduced up to 53 to 48 %. These were achieved by changing the design of the existing hydraulic press and by using the usual additives to fabricate. The stresses precipitated inside the new hydraulic press inside the restriction. So; it was concluded that the new hydraulic press is secure to manufacture and use in industries but the main focus area of researchers was only on weight reduction.

Ankit H Parmar and et al., (2014), optimized a hydraulic press structure to lower the total mass of the structure whilst assuring good enough stiffness. A technique of structure optimization for the hydraulic press turned into proposed on the way to reduce mass even as assuring adequate stiffness. Key geometric parameters of plates that have rather larger influences on mass and stiffness have been extracted as layout variables. In order to analyze the relationship among stiffness, mass, and design variables, an ordinary batch file was changed into constructed by means of CREO, and analysis became achieved in

ANSYS. Top plate, movable plate, and column layout and analysis had been executed. A discount of weight was 42 % preserving the stresses & deflection within the restrict. The researcher focused on optimization of the hydraulic press machine structure according to stiffness using ANSYS. It more focuses on stress and defalcation over all frames but the gap is the farm not detachable and it not real performances.

Akshay Vaishnav et.al., (2016), studied on Experiment design of the hydraulic pressure plate through FEA, the hydraulic press operates under continuous load. Due to this continuous loading, tensile as well as compressive stresses are produced in various parts of the machine. These stresses cause permanent deformations in some components of the machine. This work consists on the optimization of a hydraulic press of four pillar and 250 tons capacity, taking into account limitations such as design, weight and cost. The work focuses on the design and optimization of the top platen. The upper is most critical part of hydraulic press machine because it carries cylinder. The project is based on the size optimization method and the results are validated using the finite element method with the appropriate boundary conditions. CAD modeling was done in CREO and the ANSYS software is used for FEA. The attempt was made to limit the total deformation of less than 0.3 mm/m. This is done by changing the height of the plate.

2.6.2 Structural analysis

M.O. Olaleke et al., (2020), made Design, Fabrication and Structural Analysis of a 5 Tons Hydraulic Press and Mold Machine for Crucible Production. They follow method are first Design and analysis of the hydraulic press and Mold machine, second Structural simulation using Solid work application 2018 and third Fabrication of the machine and testing. Final Conclusion that the developed machine was put to test to guarantee its conformability to design objectives and serviceability. The performance of the machine was found to be satisfactory /acceptable at a test load of 50 KN. The theoretically predicted results and those obtained by analytical software have been compared with the design goal and both showed that the design is safe and it will not fail under the designed load. Also, the performance evaluation of the machine affirmed its suitability for compaction of any type of clay into crucible form. When seen the gap it confuses farm constriction, it used single acting cylinder indicate that difficult to return original position and expose for leakage.

Mr.DessaiYusufali et al., (2014), focused on design and optimization of a 30-ton hydraulic forming press gadget. They have studied the evaluation of the body structure in terms of its cloth, geometry, and stress-prompted in it. Metal forming is one in every one of the producing processes which nearly chip less. In this paper, the writer targeted the causes of structural failure problems inside the gadget because the hydraulic press continuously offers with the strain that may be compressive or tensile for that press system usually works underneath impact load conditions. And due to the effect load, the hydraulic press constantly skilled non-stop stress. They studied that extraordinary component of the machine are subjected to special forms of loading conditions and are analyzed through using the FEM device ANSYS. Weight optimization of the click body and the top head is done, which in turn reduces in thickness of the body structure and material. It more analysis on non-stop stress due in excess of operation hydraulic press machine by reducing head plate but, the gap not only focus stress but also deflection is the main problem the farm more of not detachable and it not real show .

Karel Raz et al., (2011), Conducted experiment on hydraulic pressing machine for dynamic behavior for forging operation with 180 SPM speed. Their goal was to study different frame designs of a press machines for forging operation. Evaluation is determination of suitability for each structure for supporting of special purpose operation. FEM modal analysis is made to compare results. It is recommended not to use 2 column press machines for high end operations Four column press design with bottom drive was recommended because of it indicate that low stress value and good performable than two 2 columns .

H.N.Chauhan & M.P.Bambhania, (2013), Designed and analyzed the body of a 63-tonne electricity press gadget with the use of the Finite Element Method. Due to the impact loading at the end of the bolster plate, there was the improvement of a crack at the nook, and stress generated turned into greater because of non-stop loading and stress concentration. Modifications were achieved by using introducing the fillets of proper size. Also, plate thickness changed into reduced which stored fabric.

Santoshkumar S. Malipatil et al., (2014), focused on an industrial utility project which includes mass minimization of H-frame type hydraulic press. ANSYS software program was used for this evaluation the principal goal became to reduce the cost of the Hydraulic presses without compromising the satisfaction of the output. With concerning layout specification, pressure distribution, deflection, and cost were focused on optimized design. The method accompanied by this work changed into an assessment of stresses caused in machines for exclusive thicknesses used for construction of frame and column of the H-frame kind hydraulic press but they are the central gap is single acting force on main cylinder, structural detachable and fixed type of horizontal bed used.

Bhavesh Khichadia and Dipeshkumar Chauhan, (2014), made an assessment of the design and analysis of mechanical press frames. They deal with the analytical method and corresponding layout and evaluation of the mechanical press frame. Modeling of press body has been created via the CAD software and in keeping with the modeling shape failure analysis done via FEA device. Stress distributions in the press body were determined out by using the analytical and simulation methods. Reference information was used for the new layout for modification of a brand-new press shape. With regard to design specification, strain distribution, deflection, optimization, ergonomics, stiffness, and pressure was focused on recent design and development in the press body obtained from structural additives of the click system frame. These researchers focus on stress distribution using cad software and it is not realistic (practical application) and only focus on farms of mechanical press.

Mr.Umesh.C.Rajmane et.al., (2016), focused on design and development of the 100-ton press forging operation. The proposed work consisting of the modification of existence configuration of the single layered forging press machine into the double-forging press configuration by performing structural analysis and stress distribution analysis on the base platen and on the lower die. It shows a comparative study of the stress distribution in the parts and the modifications of the project. An experimental system will be developed to achieve the established objective, which mainly includes the examination of the tensions in all the main parts of the forging press but focused on stress distribution on base plate but it succeeds the performance this project.

Mehmet Aydin and Yasin Kisioglu, (2013), studied Hydraulic press design under different loading conditions using finite element analysis hydraulic press with a capacity of 250 KN is designed with the different types of press columns and heads to minimize the stresses and displacements under different loading conditions. The stresses and displacements are calculated using both analytical and FEA method. In the numerical FEA calculations, finite element method is employed successfully along with both beam and plane elements in the models. Different types of loads are applied successfully in axial, eccentric, and oblique directions to the developed models. Appropriate boundary conditions are also applied to the models. The stresses and displacements of the hydraulic presses are calculated depending basically on the head types. The stresses and displacements on the press head are mostly obtained higher than the press columns. A good agreement is found between the analytical and FEA calculations considering the minimum stress values of the press models. Based on these calculations, the minimum stresses are calculated using press model having T- type head and S4 or S6 columns. Therefore, it is recommended to press manufacturers that the press with T type head and S4 column is the best design. The design only focused on the head of frame of hydraulic press machine.

Nawale Sagar et al., (2016), studied design of compact hydraulic press machine for rock drill components a 300bar hydraulic press was designed, manufactured, and calibrated. The machine was tested to ensure conformability to design objectives and serviceability. The machine was found to be satisfactory at a test load of 10 kn. thus here a hydraulic system is used to develop a press. The press will be useful for mass production of rock drill components. This may increase the productivity and increases the accuracy of the production. Even the press can be completely atomized by using the concept of electro-hydraulics. Direction control valve can be solenoid acted to make the system close loop which may lead to higher production rate. These studied only used for rock dill.

Mengdi Gao et al., (2016), made Design and Optimization of the Slide Guide System of Hydraulic Press Based on Energy Loss Analysis. Sliders the leakage and friction in a slide guide system. Based on the principle of minimum overall energy loss, a method for calculation of optimum clearances was proposed, for which the maximum loss, a method for calculation of optimum clearances was proposed, for which the maximum allowable lateral load may be calculated, which determines the maximum allowable side force that a hydraulic press may undergo. A stiffness and wear check method for the system under eccentric loading was presented. This method allows for the redesign of the press pillars. Based on an allowable wear constraint associated with the movable crossbeam and pillars, a procedure for calculating of the minimum guide length was proposed. A case study was considered for a hydraulic press (RZU2000HM). The methods proposed in this paper were used to determine optimum clearances of approximately 0.4 mm. The energy loss was estimated to increase by approximately 83% when subjected to an eccentric loading condition. Using the relations developed herein, the press’ four pillars were redesigned and their weight was reduced by nearly 20% to 80.69%. The length of the guide plates was shown to be reasonable through a wear check. It is believed that the methods developed in this paper may serve as an excellent reference for the optimization of the slide guide system for a hydraulic press.

Neville Saches, (2005), made a study on the causes of failure for understanding mechanical failure in machines components. Paper gives the brief introductions about the ductile fracture, fatigue failure and stress concentration. Paper also discussed the type of load is responsible to failure i.e., internal pressure, bending, torsion or a combination.

P.E. Uys, K. Jarmai, J. Farkas, (2012), Studied to find the most cost-effective design of a multipurpose hoisting device that can be easily mounted on and removed from a regular farm vehicle, cost optimization including both material and manufacturing expenditure, is performed on the main frame supporting the device. The optimization is constrained by local and global buckling and fatigue conditions. The optimization problem is solved by means of the leap-frog algorithm for constrained optimization of Snyman. This gradient-based method, requiring no explicit line searches, is a proven robust and reliable method, being relatively insensitive to local inaccuracies and discontinuities in the gradients.

Mohamad M. Saleh, (2016), studied the design of a heavy-duty hydraulic machine using finite element techniques. The machine is designed by ENERPAC without any measurement or variable hydraulic system. The investigation dealt the theoretical and experimental model of the machine to establish the accurately optimal design analysis and further development of the present machine at minimum time and lower cost. The applicability of the existing PC based FE package as a computer aided design tool is also investigated. A comparison has been made between the experimental and theoretically predicted results. Both the results are found to be in good agreement with each other.

Sinha and Murarka, (2012), Made Experimental study on hydraulic presses and represented a 3- D complex structure. It is found that an exact analytical method of stress and deformation analysis is cumbersome and time-consuming. In order to reduce core memory requirement and the cost of computation, a simplified plane stress (PS) FEM model for a hydraulic press structure (welded frame) has been identified for its analysis. On the basis of this investigation, certain significant guidelines have been obtained for the design of press frames. Such a model has resulted in savings in computational time, core memory requirement and cost of analysis.

M.Bapat and DessaiYusufali, (2012), investigated on the design and optimization of a 30 ton hydraulic forming press machine. They have studied that analysis of the frame structure in terms of its material, geometry and stressed induced in it. Metal forming is one of the manufacturing processes which are almost chip less. In this paper the author focused on the causes of structural failure problem in the machine because hydraulic press continuously deals with the stress that may be compressive or tensile for that press machine always works under impact load condition. And because of impact load the hydraulic press always experienced continuous stress. It is studied that different components of the machine are subjected to different types of loading condition and are analyzed by using FEM tool ANSYS. Weight optimization of press frame and upper head is done, which in turn reduces in thickness of the frame structure and material.

B.parthiban et al., (2014), investigated the design and analysis of C-type hydraulic press structure and cylinder. It represented C-type structure. Frame and cylinder are the main components of the hydraulic press. It is studied that the structural design of the frame depended on the pressing force this determined for the required rigidity, the dimensions of dies influencing the size of the tool area, work area accessibility that determined by the shape of the press frame. The performance of a hydraulic press depended on largely upon the behavior of its structure during operation. The analysis of hydraulic press structure is needed to increase their performance and productivity. The frame and cylinder are modeled by using modeling software CATIA. Structural analysis has been applied on c-frame by using analyzing software ANSYS. By using ANSYS software, structural performance and stress, strain distributions are plotted for verification. According to the structural values the dimensions of the frame and cylinder are modified to perform the functions satisfactorily as well as to reduce the weight for cost savings.

Pedro G. Coelhoaet, (2015), made a structural analysis and optimization of hydraulic press brakes. It has been found that the source of deflection parallelism errors and minimize them through a structural optimization methodology. Based on the model of the bending process in press brakes defined by Timoshenko theory of beams, it has been not possible to design a machine that achieved uniform bending angles for every bending length because no optimal shapes or dimensions of the bed and ram that lead to parallel deflections for all bending lengths. Workpice Bending errors have been derived by considering the influence of shape, dimensions and initial deformation of the machine structural components. Shape optimization, dimensional optimization has been performed to achieve uniform bending angles for every bending length. And optimal initial deformation was performed in decreasing the bending error.

H.N.Chauhan and M.P.Bambhania, (2012), made a FE Analysis applied on a frame of 63 ton power press machine. Analysis of this frame is done by the simulation software. It is said that Instead of sharp corner of the c - frame, fillet is provided to reduce failure in the structure. Amount of fillet depends on load condition experienced by frame and it is analyzed by using FEM Tool. It is concluded that simulation software is the powerful tool for prediction of plate required for a given load. This analysis given the result of reduction in thickness of plate of frame structure, material saving and as well as cost benefits.

Ankit H Parmar et. al. (2014), made Design and Modification of Elements of Hydraulic Press Machine. Analysis of Top plate, movable plate and column was design using ANSYSE software. Finally, conclusion comparison between result of old bottom platen and modified final bottom platen we get weight reduce that 2263 Kg to 1303 Kg. Deflection increases 0.055 mm to 0.22 mm but it is in permissible limit. It is not affected to component. Von-misses stresses increases 104 M pa to 141 M pa and it also doesn't cross permissible limit and our bottom plate is safe under working condition.

2.6.3 Hydraulic Press and Pull Machine

Mr.Fisayo Adesina et al., (2018), studied fitting liners pressure within the engine blocks. He achieves to the advanced device will convey launch to small and medium scaled Workshops and artisans. By way of reducing the time and stress associated with set up and elimination of bearings, and other sorts of pressure fit inside the system however the gap is it needs more attempt to boost up materials from the floor to the head of the table, it is difficult to hold the stableness like engine all through putting time additionally uncovered to war page and coincidence. But subsequently, it succeeds.

Mr. Bethrand N. Nwankwojike et al, (2017), made a study on Manually Operated Hydraulic Press and Pull Machine and best awareness to check the overall performance of the developed system, seven bearings of bore diameters 24, 30, forty, 50, 65, eighty and a hundred mm respectively have been geared up into seven corresponding shafts with shaft deviation of ±0.001 mm and dismantled later on. The mounting and dismounting of the numerous bearing-shaft assemblies have executed the use of three different comparing strategies: conventional hammering, the usage of the developed guide hydraulic press and pull device, and the use of already present electrically powered hydraulic press in every case. The time taken to mount and dismount the various assemblies for the 3 methods used has been recorded. Even though the researcher comparing pulling and pushing manual hydraulic press machine with electrical and conventional type they only focus inserting bearing and different bore diameter depend on time taking. But they achieve some advantage and disadvantage. Finally electrical hydraulic press machine more performable than manual type but it activates only electrical power and most of structural farms are not separable.

Mr. Shivasheesh Kaushik, (2016), studied the Design and Fabrication of a Special Purpose Hydraulic Press Performing Punching Operation, but he specializes in hydraulic punching and bending machines is successfully enrolled, and is calculated well. In this, we've got brought a special purpose hydraulic model of the Punching and Bending gadget and we have shown theoretically. How is the hydraulic version of a special cause punching machine? Economically and exactly works. First of all, the researcher survey of the literature assessment of the gadget, and marketplace survey for the operating and constructional info. Then we have calculated the formulation for the layout of the exclusive parts of the hydraulic cylinder in order that he can calculate the ability of the hydraulic punching and bending device and sooner or later fabricate the gadget to carry out the operation on the aluminum sheet to produce the very last product. This gadget is used for multipurpose Punching at a single time of stoke of ram/plunger. We have seen how Hydraulic structures allow customers to accurately wield big amounts of electricity with little enter force. And with the aggregate die two operations are completed simultaneously. So he decided that the hydraulic device of unique cause punching and bending machine is quite reasonable as compared to the alternative opportunity but his make interesting machine according to mobility from place to place but when we focus on the gap is that, it manual operate, more of the frames are fixed and accessible to oil leakage and expose to accident.

Joshi Ameya1, et al. (2018), investigated Design of Special Purpose Hydraulic Press Machine the project is developed with the motive of press fitting the two components of muffler used in silencer viz. shell and stuffing. The researchers follow that Design of special purpose hydraulic power press machine of capacity 15 Ton, according to customer requirement, Modeling of hydraulic press using suitable CAD software and Fabrication of hydraulic press. The machine is decided around the provided tonnage (15 Ton). Base plate, guide pillars, and cylinder specification are calculated from the same. It is found that calculated values and design of hydraulic press machine meets the requirements of process of press fitting.

Said M. A. Ibrahim (2019), made a study on Design, Manufacturing and testing of a Hydraulic Press to Produce Oil from Egyptian Jatropha Seeds. The main focus manufacturing this machine is extraction Jatropha oil using hydraulic pressing was investigated. The main goal of this research is to design, manufacture, and test a hydraulic press to produce higher yield and optimum physical and chemical properties of oil extracted from Egyptian jatropha seeds. The present results led to these conclusions: Time of oil extraction process from jatropha seeds is 720 minutes at a yield of 11%, higher extraction temperature caused darker color of the oil because of oxidation. Jatropha oil extracted by hydraulic pressing gave lighter oil color because of the lower FFA of 5% for the produced oil. This method is characterized for higher oil production. The density and viscosity of the extracted oil decreased with temperature increase. The deduced correlations of viscosity and density in relation with temperatures can be correctly used in predicting the viscosity and density for the extracted oil from Egyptian Jatropha seeds. Flash point of jatropha oil extracted by hydraulic press is higher than fossil diesel oil, so, handling and storage of the oil is relatively less hazardous as compared to diesel fuel. The calorific value of jatropha oil extracted by the hydraulic process was 39201 kJ/kg; Jatropha oil contains oleic, linoleic, palmitic and stearic fatty acids. Jatropha biodiesel oxidation instability increased for hydraulic extraction due to its content of oleic and linoleic acids. Theses researchers more focused on extraction of from Jatropha Seeds. The Design that much not consider for moisture, environmental pollution.

Sumaila, M., (2002), focused on the Design and Manufacture of a 30-ton Hydraulic Press device. The Performance Test Result It is an everyday exercise to difficulty engineering products to check(s) after manufacture. This is a first-rate step within the production method. Under tests, the product is checked to look. If purposeful requirements are satisfied, become aware of manufacturing problems, ascertain monetary viability, and so on. Testing is therefore hired to prove the effectiveness of the product. For the hydraulic press, the take a look at for leakages is the maximum vast test. The test began with the initial priming of the pump. After which the fluid becomes pumped. This was accomplished underneath a no-load condition. The system was left to stand in this function for two hours. The gadget changed into then subjected to a load of 10 KN furnished by using two compressions springs of regular 9 N/mm every arranged in parallel between the platens. The springs have been then compressed axially to duration of 100 mm. This arrangement turned into left to face for 2hrs and turned into discovered for leakages.

Finally, Leakage within the device becomes no longer indicated because the decreased platen did now not fall from its initial function.

Bethrand N. Nwankwojike, et al., (2017), designed and fabricated a hand operated hydraulic press and pull machine. It was produced using domestically sourced substances. The energy of this system turned into analyzed via simulating a load of 1000N and the corresponding values of most shear pressure, maximum bending second, and most displacements were decided as 74.9491N, 6.6335N-m, and 4.367e3mm. Also, maximum axial strain, bending stress, and tensional pressure have been determined as 0.00359, 701998, and 0.00653N/m^2 respectively. The developed machine will carry comfort to small and medium scaled workshops and artisans by lowering the time and stress associated with set up and removal of bearings, and different types of force suit in system assemblies. So regionally sourced substances are more worthwhile to fabricate hydraulic press machines in keeping with value minimization, load resistance.

Mr. Ganesh M Mudennavar et. al.,(2018), Made Design and Analysis of 12 Ton Hydraulic Pressing Machine the comparison among C-type and 4-Column type hydraulic press Machine. Hence Analysis software is used to compute the Results In this work the four-Column hydraulic press and C-type press is Designed and Analyzed with standards. The most stresses caused inside the four-Column hydraulic press and C-kind press system are much less than the permissible stress of the fabric. Further decrease of the fabric may also affect the design and gives high deformation and stresses. Which are not perfect for the device, as this device is used for compressive pressing operation in sheet metallic industries wherein the deformation of the click has to be permissible. Pressing accuracy performs an essential role to keep near tolerance and growing productiveness. As the analysis suggests maximum deformation observed is lesser than 1 mm that is tolerable for such hydraulic pressing operations. In this design, the decrease in body and bed thickness of C-Frame is done so there may be a lower weight. It is a multi-reason machine this is used for performing pressing and forging duties. By converting the die one-of-a-kind operations like blanking, sheet steel, and bending, and so on. Can be accomplished on a hydraulic press system through various inputs to the cylinder that's restricted by the output of the pump.

Suraj P Jagtap et.al., (2017), Made Design and Fabrication of Electromagnetic Press machines. This device used electronic devices such as; electromagnet, AC electricity-variety of coil, shank, punching plate, and so forth. This machine is used for the industry to growth performance with the aid of decreasing manpower involvement. After making the real working version we got the subsequent list of conclusions: this machine is useful to boom performance through reducing manpower involvement. It is compact so much less fool area is needed for installation. It calls for low protection due to the fact there is no leakage trouble of the working medium. Easy operation with a excessive rate of manufacturing. The semi-professional operators can without problems perform the gadget. This undertaking is effectively finished with the development of an automated press device. It is n-numbers of electrical device is used to punching capacity depend up on variation of coil but the main gap it can't determine tonnage because of coil variation, it expose for damage, accident according to operation condition. Other way that they more focused on reduction of man power.

Levent Paral, et .al., (2017), Investigated hydraulic press system-based PLC with the inclusion of local piezoelectric sensor. In this study, the local PZT composite sensors are produced by PLC controlled hydraulic pressure system, and unique pressure system is designed. It can produce sensors of various dimension using by pressure system, in addition, by making transformations in the pressure system, the series of production can be performed.

Bhavesh N. Khichadia1, et.al., (2014), Studies of design and analysis of mechanical press frame structures using supporting experimental data, the following conclusions can be drawn: Finally, they explain that, Mechanical press failure most of the sharp corner of press frame and crack initiate at junction of two plates. Analytical method through get where is stress concentration more and deflection of frame. Using FEA Analysis the best solution method for stress analysis and shape optimization of frame structure. FEA analysis through get accurate result. Increasing rigidity and stiffness increasing by the pre-tensioning of press frame. Optimization technique through decreasing material cost and weight. It is difficult to identified type frame but when we came to our aim the mechanical press machine is old's machine for all it is not that much accepted at these monuments according to the Wight, operation, man consumption and so on. But, these analyses only focused stress and deflection on frames and it not detachable one to another.

Mr.k.shravan kumar et.al, (2017), Designed and fabricated of hydraulic press of a multi-purpose machine as it can be used for performing different tasks. This researcher more focuses on frames but using for inserting 10T hydraulic jack. By changing the die different operation like bending, blanking etc. can be performed on a hydraulic press machine. the design has main focus on reducing operator fatigue and increase safety, improving the flexibility and makes operation more convenient, and to achieve dimensional and positional accuracy. Components of press are designed to avoid bending failure due to applied load. mild steel is selected as material based on its properties such as high bending & tensile strength, it compatibility with operation like machining, welding, finishing, cutting etc. and cost as economic factor the main gap when we consider that researcher use fixed horizontal bed, frames not detachable, easily effort is needed and expose to accident.

2.7 Principles of operation hydraulic machine

The pressure throughout a closed system is constant. One part of the system is a piston acting as a pump, with modest mechanical force acting on a small cross-sectional area; the other part is a piston with a larger area which generates a correspondingly large mechanical force. Only small diameter tubing (which more easily resists pressure) is needed if the pump is separated from the press cylinder. Pascal's law: Pressure on a confined fluid is transmitted undiminished and acts with equal force on equal areas and at 90 degrees to the container wall A fluid, such as oil, is displaced when either piston is pushed inward. Since the fluid is incompressible, the volume that the small piston displaces is equal to the volume displaced by the large piston. This causes a difference in the length of displacement, which is proportional to the ratio of areas of the heads of the pistons given that Volume = Area X Length. Therefore, the small piston must be moved a large distance to get the large piston to move significantly. The distance the large piston will move is the distance that the small piston is moved divided by the ratio of the areas of the heads of the pistons. This is how energy, in the form of work in this case, is conserved and the Law of Conservation of Energy is satisfied. Work is force applied over a distance, and since the force is increased on the larger piston, the distance the force is applied over must be decreased,(Nawale Sagar et. al., (2016).concept of the Pascal low is shows in figure 2.2

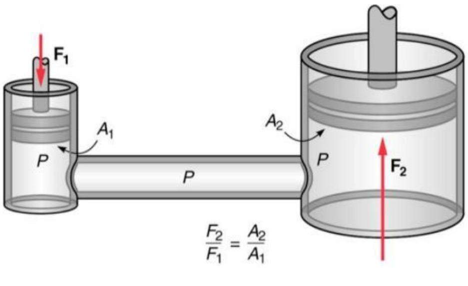


Figure 2. 2.Concept of pascal's law

2.8 Short Summary for related literature review

This section included summarized literature. The most related papers are focused which are pertinent to intended work .The majority of Multipurpose hydraulic press machines used the Action of main cylinder. These activities occur only on the main cylinder; it is applied by the manual pumping mechanism. Horizontal beds start to lift up or down using human effort. According to some researchers during inserted or removed liners, the engine block is easily exposed for warping at the sump side of the engine block and simply exposed for accident. Also it is difficult to transport from place to place because the structure is not detachable; for instance (Mr. Bethrand N. Nwankwojike et al, 2017) they investigate the Development of a Manually Operated Hydraulic Press and Pull Machine. additionally (Mr.Fisayo Adesina et al.,2018) studies manual operation, pulling and pushing action occurs on main cylinder but simple construction of frames and used to insert and remove different axle bearings. It needs human effort for pumping hydraulic fluid and fixed horizontal bed use. These researchers decided that electoral operation more prefect activity occurs than Manually Operated hydraulic press machines. Last researcher (Mr. Shivasheesh Kaushik, 2016) studied the Design and Fabrication of a Special Purpose Hydraulic Press machine which is Performing Punching Operation, but he specializes in hydraulic punching and bending. These researchers use local and dissolved maternal milled steel. Hydraulic oil driven by electrical pump but controlled by mechanical handle lever (spool valve) and the machine performing for both Punching & bending on workpiece but it is exposed to leakage(not used relief valve), environmental pollution and cause of accident but unknown horizontal bed. Finally conclude that no one has literature to increase the application of hydraulic press machines.

2.9 Research gaps of the review

The gaps are noted for the following research: Most hydraulic press machines use only one main cylinder, which provides action throughout the application of the downward stroke on the work piece (compression; assembly or disassembly); they are difficult to transport from one location to another because the frames are not detachable. Human effort is required to move the horizontal bed up and down. As a result, this thesis work seeks to close this gap by design and fabricate a brand-new ECMHPM for our organization, which requires human effort for aligning or feeding the work to the press machine.

CHAPTER THREE

MATERIALS AND METHODS

3.1 Description of the study area

The factory is found at Oromia Regional State at 110 Kilometers from Addis Ababa and 14 km South of Adama town. Commencing production in 1954 it is the oldest and the pioneer in the history of Ethiopia's sugar industry at 80 27'N latitude and 30020'E longitude. Geographical position of wonji/show sugar factory is shown in figure 3.1. .

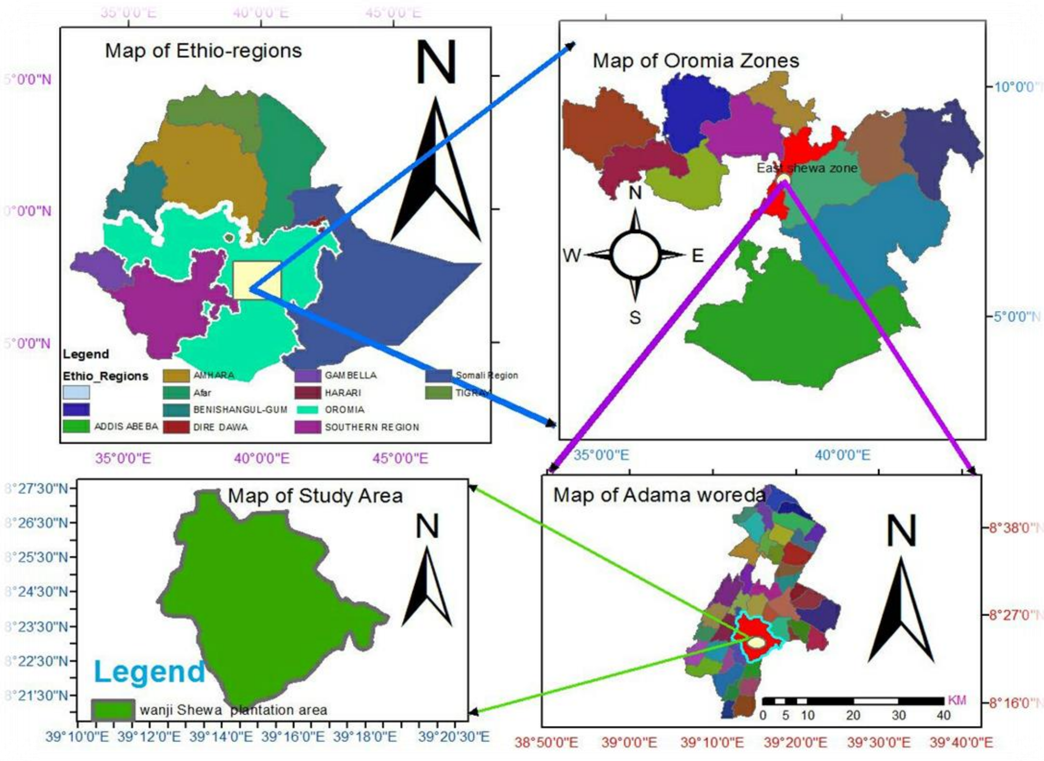


Figure 3. 1.Geographical position of the Wonji/Shoa Sugar Factory

3.2 Materials used to fabricate ECMHPM

Lists of structural materials, lists of essential tools, lists of essential machines, and lists of essential software are given in tables 3.1, 3.2, 3.2, 3.3, and 3.4, respectively.

3.2.1 List of structural materials

Material selection is basic for the fabrication of ECMHPM machine. Most materials are collected from different scrap yards, and most of the material is repaired according to our activities. The following structural materials are clearly shown in Table 3.1.

Table 3. 1.List of structural materials

Type of material	Qut.	Base Thickens (mm)	Width (mm)	Height (mm)	Diameter Internal (mm)	External (mm)
Bolts/32M/	36	-	-	29	29.5	30
Center pins /steles steel/	04	-	-	300	39.5	40
I-channel	02	10	100	2200	39.5	-
Head plate plate/12mm/	02	12	1300	500	29.5	-
Horizontal bed plate	02	16	1420	300	350	-
main piston	01	-	-	800	100	150
auxiliary pistons	02	-	-	1100	50	90
Hydraulics steel pipe	04	15	-	1700	0.22	15
Hydraulics rubber hose	10	24	-	900	0.35	24
Side steel pipe	02	15	-	940	0.22	15
Type oil (15 liter)		SAE10w				
Maximum out put pressure of hydraulic pump		3050psi				
Direct current control voltage		24v solenoids				
Type of press		H-type/vertical operation/				
Transformer power		Input voltage 220/380v output voltage 25v				

3.2.2 List of essential tools

Essential tools are any tools, keys, equipment, or other material or equipment required to facilitate the performance of duties associated with fabrication of machine. So, these essential tools must be in compliance with this work. Properly shown in Table 3.2.

Table 3. 2.List of Essential tools

Measuring tools
Meter
Multi meter
Level gage
Vernal caliper
S.S.T hydraulic presser gages

3.2.3 List of essential machines

Basic fabrication machines for cutting, drilling, shaping, and welding are considered essential devices. Consequently, these crucial tools must be compatible with this project. Shown correctly in Table 3.3

Tabel 3. 3.List of Essential machines

machines
Grinder machine
Carbide and oxygen
Lathe machine
Electrical arc welding machine

3.2.4 List of essential software and special device

Devices are objects or systems that have a specific purpose or intention. This easily isolates the fabrication machine problem. The following are basic software Table 3.4

Tabel 3. 4.List of Essential software

Software
Ansys Software
Digital cameras

3.3 Methods

3.3.1 Data source and collection methods

The necessary data were collected by using primary or secondary data collection through visiting various websites and literatures. To achieve the objectives, both primary and secondary sources were used.

a) The Primary sources

b) The secondary sources

3.3.1.1 Primary sources

i. **Visual Operation and structural observation of conventional H-type hydraulic machine press**

Based on observation, the majority of typical H-type hydraulic presses produce compressive force utilizing hydraulic cylinders. The H-type hydraulic press's primary parts are the head frame, horizontal bed, column, and cylinder. Using cables, a horizontal bed can be raised and lowered vertically, although it frequently leaks from the outside, and the frames are difficult to separate. The typical H-type conventional hydraulic press is made up of a mechanical pump that transforms hydraulic energy into useful work at the point of load resistance, a hand pump (which requires human effort) that drives the fluid, the fluid itself, which requires less power to transmit through hydraulic pipes and connectors. This project involves designing an H-type hydraulic press machine of head frame, a horizontal bed, and a column. Section 4.1.1. explains this more fully.

ii. Visual observation of appropriate materials useful for fabrication in the factory scraps collections

The majority of the supplies for this project are gathered from scrap yards. Somewhere during the new performance, as stated in table 3.1, these utilized materials are applied. The failure of the materials is attributable to ignorance, incompetence, and negligence. As a result, the scrap yard is being filled with a lot of useless stuff. The simple criteria listed below helps in choosing the right material.

a) Visual inspection of corrosion, quality
b) Measuring Thickness/dimension
c) Check for banding
d) Characteristics of Load carrying capacity and so on...

3.3.1.2 Secondary sources

i. Using software such as ANSYS for simulating the system

The hydraulic press's head frame, column, and horizontal bed have all undergone structural analysis using the analyzing program ANSYS. To confirm the structural performance stress and total deformation distributions are shown using ANSYS software, a standard method has been created. The dimensions of the head frame, column, and horizontal bed are altered in accordance with the structural value to effectively carry out the functions. Section 4.1.2.explains this more fully.

3.3.2 Literature review and problem identification

Previous research has been conducted on the following topics: cost and weight reduction, structural analysis, hydraulic press application, and electrically operated hydraulic press machines. Study refers to what others have done and identified, where limitations do exist, so; that this work will be based on the research gap.

3.3.3 Flowchart of the thesis

Flowchart of the thesis is the chart that shows the progress and method that used to develop the required study. Flowchart for this study is shown in Figure 3.2.as under.

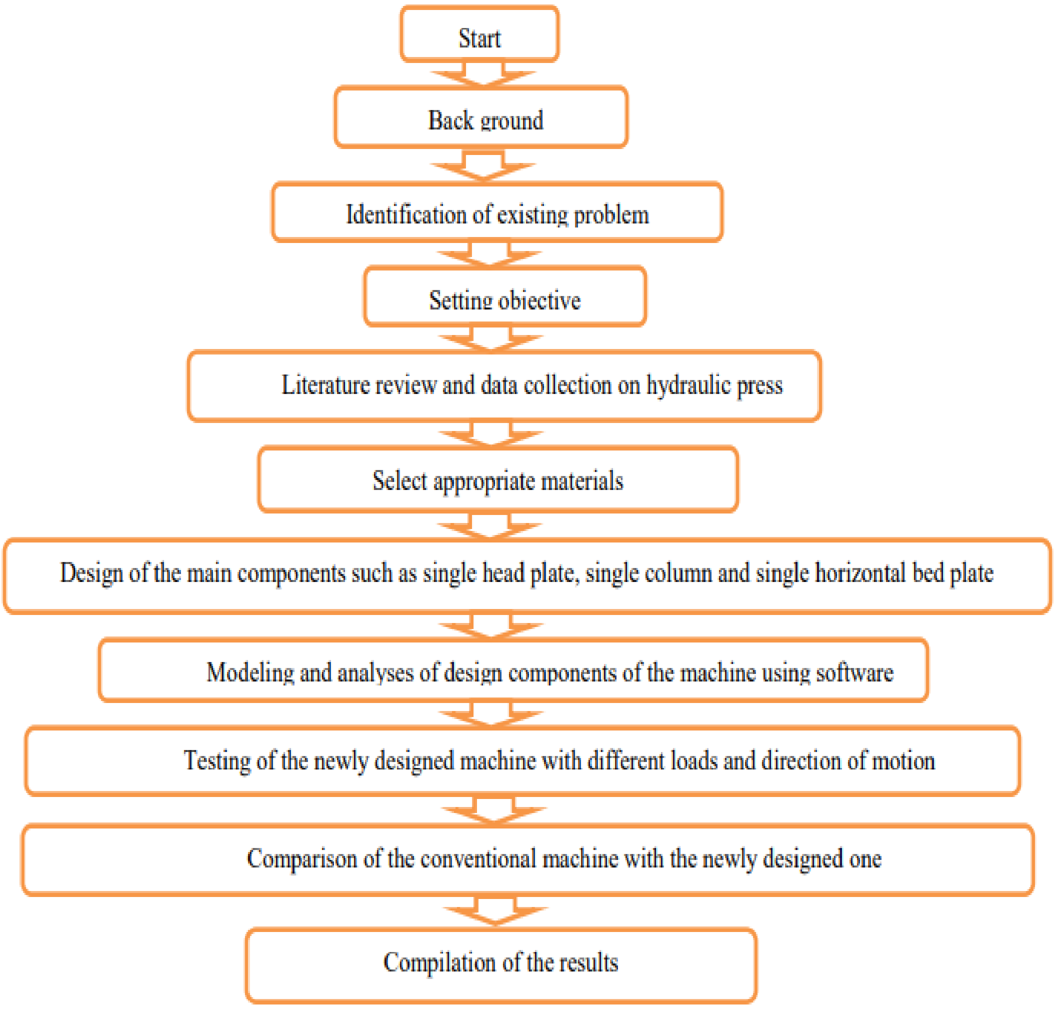


Figure 3. 2.Flowchart for this study

3.3.4 Material selection

For the ECMHPM the structural steel was selected because for fabricating the machine structural steel is easily accessible in the case organization and locally available material; furthermore due to the fact that it has high strength, rigidity, good machinability and less cost it had been selected. During the construction of the new Wonji sugar factory (case organization) most of the s. steel was imported from India. Therefore, Indian standard was used for the designation of the material. For the accomplishment of this work two types of selection were done.

i. Raw material selection

This selection was done so that the raw material was used for fabrication of main structural parts such as single head plate, single column and single horizontal bed plate. Clearly shown on table 4.4.

ii. Standard parts selection

This selection of parts was done from the scrap yard of the case organization; thus, from these collected scraps standard components which were appropriate for the work such as main cylinders, auxiliary cylinders, hydraulic hoses, hydraulic pipes, hydraulic pump assembles and other electrical unit were selected.

3.3.5 Description of selected standard parts

5.3.5.1. Main cylinder:

Double action cylinders are ones that can accept a power stroke in both the forward and backward directions. Such a cylinder has the ability to pull and press at the same time. Pumping oil under controlled pressure and flow conditions from the cylinder's oil ports allows for both forward and backward strokes. The main cylinder was taken from the grape loader equipment. These types of cylinders are used in all general-purpose and standard presses. Hydraulic cylinders transfer the pressure energy in hydraulic fluid into useful pressing force. It is used to hoist large objects from the ground to the head and is fastened to machinery at the top. Its maximum holding pressure is 4050 Psi (27.81MPa).

5.3.5.2. Auxiliary cylinder:

Was taken from an XCMG grader and has a maximum capacity of 4050 Psi (27.81mpa). Its purpose was to lift up and down the grader blade

5.3.5.3. Hydraulic pump assembly:

It was taken from a tire repair machine. It generates a maximum pressure of 3050 psi (20.94MPa).

5.3.5.4. Hydraulic hose:

It is positioned at the top of ECMHPM and is connected directly to relief valve and metallic pipe’ It has a maximum capacity of 36MPa.

5.3.5.5. Relief valve:

It was taken from XCMG grader and has a maximum pressure of 3000Psi (20.6MPa).

5.3.5.6. Joystick:

A work piece may shift positions depending on the circumstances of the working activity because of some exhaustion. You can move in any direction and turn on and off fluid flow with the joystick application. During the condition, less work is required and it is more comfortable. There is an electronic joystick that offers three positions

i. inward and Outward of main cylinder
ii. Helps to move the auxiliary cylinders.
iii. In and out slides of the roller

5.3.5.7. Directional control valves (solenoid):

Directional control valves are very important for all fluid power systems. The hydraulic systems require control valves to direct and regulate the fluid flow from the pump to the various load devices. The valves consist of electromagnetic solenoids. The flow passages can be changed according to the application of the joystick.

5.3.5.7. Control panel:

Control panel governs over-all operation and performance of a press machine and it is electrical device holder. Typically made from metal box and plastic molding. This contains important electrical components that control and monitor a number of mechanical processes. Operators will interact with the controls of the panel to operate and control the overall operational condition of the machine. The main components are Transformers, rectifiers, alternating current (AC), direct current (DC) voltage, Circuit Breakers, Terminal wires, Relays, Fuses, and motor control Breaker center (mcc panel).

4.3.5.9. Hydraulic tank:

All hydraulic systems should have their own oil reservoir. Oil used is stored in a reservoir to which it is turned after use. It has oil, oil filter, inlet and outlet pipe, and oil level gauge. Maximum holding capacity is 40litt.

5.3.5.10. Filters:

One of the main reasons for hydraulic failures is contamination. Fluid or solid oils may be employed in hydraulic systems with a lot of foreign objects. They are typically in liquid form, such as water, acids, paints, etc., but occasionally in solid form, such as dirt, dust, worn-out hydraulic cylinder seals or piston parts, etc. A hydraulic filter is used to stop both types of foreign objects from getting into the system's delicate components.

5.3.5.10.1. Steel pipe and tube:

It located between directional valve and hydraulic rubber hose. The main Function is the way of fluid flow to the system. The fluid flow efficiency is influenced by the piping line. Steel tubes are more widely used because of holding capacity hydraulic pressure and it avoids Leaking .maximum holding capacity is 36MPa.

5.3.5.10.1. Hydraulic Hoses:

It receives fluid from the hydraulic pump that feeds the cylinders at the top of the ECMHPM. It is more adaptable in terms of position and tendency, with a holding pressure of up to 36Mpsi.

CHAPTER FOUR

RESULTS AND DISCUSSION

4.1 Results

Most of existing hydraulic press machines are fabricate after designing. These chapter focused on over all structure of ECMHPM basically include numerical and statically analysis of single components by taking from assembles, fabrication process and tasting under different application of load. Under these chapter validations important to determine the general performance of assemble under different operation condition. The performance of machine depends up on the quality and tendency of material. These material which select for these project is structural steel/ miled steel/ and these material easily available, less cost and easily machinability material. So, under this chapter the following sections are included.

i. Design of structural parts of ECMHPM
ii. Fabrication of newly designed machine
iii. Assembling procedure of newly designed machine
iv. Testing of hydraulic press machine under different loads

a) Specifications of standard cylindrical parts

The specifications of the standard parts are given in Table 4.1 as under

Table 4. 1.standard cylindrical parts specifications

Input					
			Diameter		
No	**Component**	**Pressure (Max) (N/mm^2)**	**Inner (mm)**	**Outer (mm)**	**Remark**
1	Main cylinder	3000psi (20.6Mpa)	Φ100	Φ150	
2	Main cylinder rim	3000psi (20.6Mpa)	Φ45	-	
3	Auxiliary cylinder	1500psi(10.3Mpa)	Φ50	Φ90	Divide
4	Auxiliary cylinder rim	1500psi(10.3Mpa)	35	-	

Source: *Technical Specification of 120 Motor Grader (www.cat.com.)*

b) Maximum Output Force of standard cylindrical

The maximum output forces of cylinder for each main and auxiliary cylinder are given below in table 4.2.

Table 4. 2.Maximum Output Force on ECMHP

No	Component	Force(down) (kN)	Force(up) (kN)	Maximum tonnage
1	Main cylinder	162	129	17tons(down)
2	Auxiliary cylinder	20	10	

c) Mechanical properties of Structural parts

Structural parts' mechanical properties are tabulated as under in Table 4.3.

Table 4. 3.Mechanical properties of structural parts R.s. khurmi j.k. gupt, (2005)

No	Structural part	Material	Density (kg/m^3)	Yield strength (MPa)	Young's modulus (MPa)	Poisson's ratio
1	Single Head plate	S. steel	7850	320	200	0.30
2	Single Column	S. steel	7850	320	200	0.30
3	Single Horizontal bed plate	S. steel	7850	340-345	200	0.30

d) Assumed dimensions of structural parts

Structural parts dimensions are assumed by the designer and given below in Table 4.4.

Table 4.4.Dimensional of Structural parts

No	Structural part	Dimensions			
		Lengt h(mm)	Width (mm)	Height (mm)	Thickness (mm)
1	Single Head plate	1300	12	200	12
2	Single Column	100	240	2200	10
3	Single Horizontal bed plate	1420	16	300	16

4.1.1 Design of structural parts of ECMHPM

4.1.1.1 Calculation of single head plate

The simply-supported beam in figure 4.1 has concentrated force 81kN and 64kN acting vertically downward and upward at its midpoint of single head plate respectively. The distance between the supports is labeled L = 1300mm, so the each force is located half the distance (1300mm /2) = 650mm from each end support. $A_x = 0$ alone horizontal axis.
(Thomas H. Brown, 2005).

The following assumptions have been taken into consideration;

i) Single head plate thickness equal to 12mm and cylinder applies force on single point of head plate and considers maximum applied force;

ii) Weight of the head plate is neglected;

iii) The material of the head plate is perfectly homogeneous and isotropic;

iv) the Young's modulus (E) is the same in tension and compression;

v) the area of the head plate is equal to the sum of the rectangular and trapezoidal Sections

Calculations are as follows;

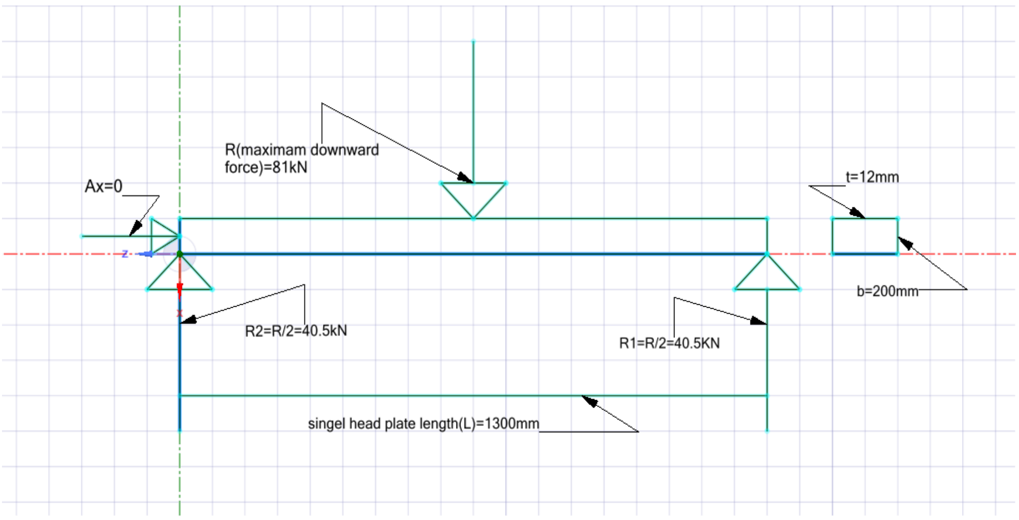


Figure 4. 1. Free body diagram of single head plate

Where – R_1 –simple support due to reaction

R_2 - Ball end support due to reaction

R - Applied maximum reaction force due to compression down (main cylinder)

L - Horizontal length head of plate

t -Thickness of head plate

Note: Take Maximum force on single head plate R = 81kN

Shear force

From FBD: take; $\uparrow + \sum F_Y = 0$ $\qquad R_A + R_B = 81kN;$

From FBD: take: $\sum M = 0$ $\qquad 81kN\ (650mm) - R_B(1300mm) = 0;$

$$R_A = R_B = 40.5kN$$

Details shear force based on fixed supporting single head plate;

$Point\ b/N\ A\ \&\ B:\ (0 \leq x < 650mm);$

$$40.5kN - V = 0; V = 40.5kN$$

$Point\ b/N\ B\ \&C:\ (650mm < x \leq 1300mm);$

$$V + 40.5kN = 0; V = -40.5kN$$

Shear Force Diagram

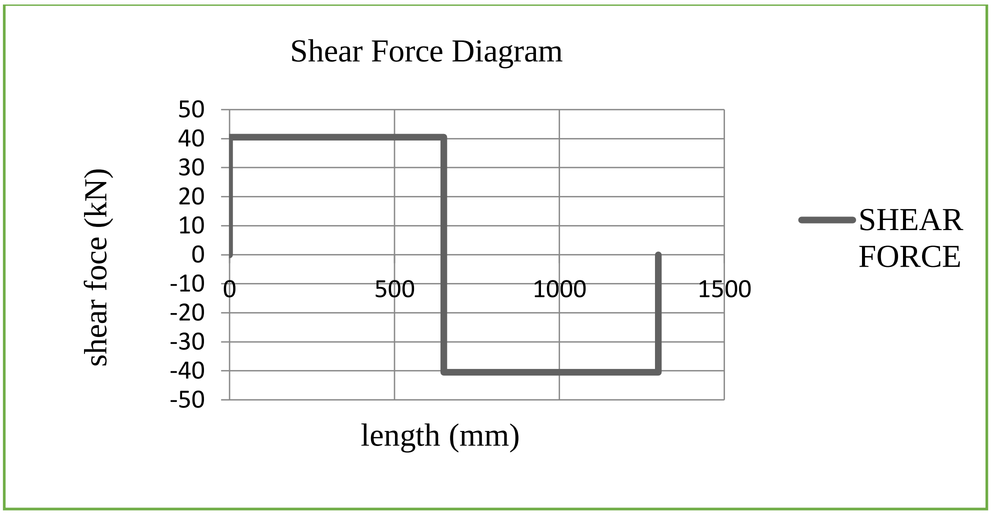


Figure 4. 2. .Shear force diagram single head plate

Note: that the shear force (V) is a positive (R_1= R/2) from the left end of the beam to the midpoint, and a negative (R_2= R/2) from the midpoint to the right end of the beam. So there is a discontinuity in the shear force at the midpoint of the beam, of magnitude is R= 81kN, where the concentrated force is applied. The maximum shear force (V_{max}) is therefore

Take only: Maximum downward force on head plate R = 81kN,

So; Maximum shear force (V_{max}) is therefore,

$$v_{\max} = \frac{R}{2}; \quad V_{\max} = 40.5KN \tag{4.1}$$

Maximum bending moment

Detail bending moment based on fixed supporting single head plate

$$Point\ b/N\ A\ \&\ B:\ (0 \le x < 650mm);$$

$$From\ FBD:\ take;\ when:\ x = 325mm;$$

$$40.5kN\ (325mm) - M = 0$$

$$M = 13162.5kNmm$$

$$Point\ b/N\ B\ \&C:\ (650mm < x \le 1300mm);\ when\ x = 975mm.$$

$$From\ FBD:\ take;\ when\ x = 975mm.$$

$$40.5kN\ (1300mm - x) - M = 0$$

$$M = 13162.5kNmm$$

Bending Momentum Diagram

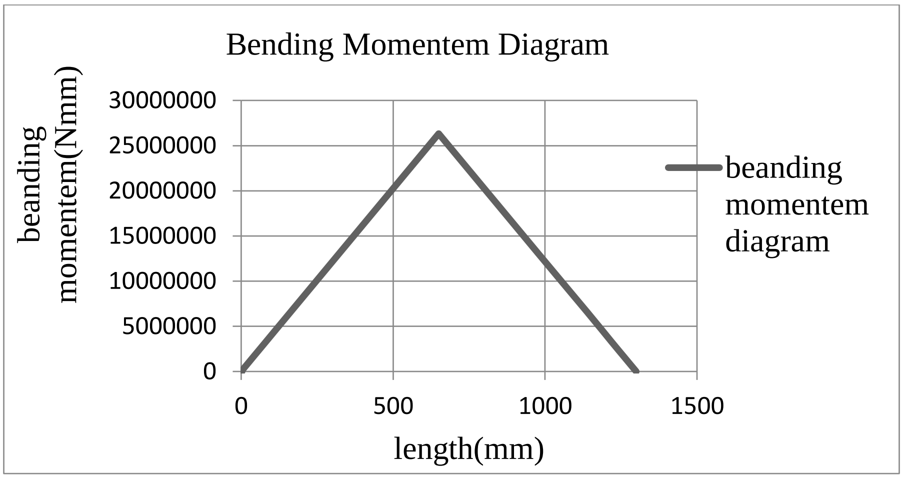


Figure 4. 3.bending moment diagram of single head plate

It is observed from the figure 4.3 that the bending moment (M) is zero at both ends, and increases linearly to a maximum at the midpoint (650mm). From the midpoint, the bending moment decreases linearly back to zero. The maximum bending moment (M_{max} = 26325kNmm) occurs at the midpoint of the beam and is given by:

$$M_{\max} = \frac{FL}{4};$$
$$M_{\max} = 26325 kNmm \tag{4.2}$$

$$stress(\sigma) = \frac{F_{app}}{A_{total}};$$
$$stress(\sigma) = \frac{81000}{532.89} MPa \tag{4.3}$$
$$stress(\sigma) = 152 MPa$$

So, the bending equation is given by

$$\frac{M}{I} = \frac{F_b}{y} = \frac{E}{R} \tag{4.4}$$

Where - M = Bending moment at the given section

f_b = Bending stress

I = Moment of inertia of the cross-section about the neutral axis

Y = Distance from the neutral axis (centroid axis)

E= Young's modulus of the single head plat

R = Radius of curvature of the beam

Z = Section modulus

From above equation bending stress is given by

$$F_b = \frac{M}{I} Y;$$
$$Bending\ stress\ (F_b) = 26325k\ N\ mmx234mm / 125x10^6 mm^4 \tag{4.5}$$
$$Bending\ stress\ (F_b) = 72 N / mm^2$$

$$M_{maximum} = \frac{FL}{4};$$
$$M_{maximum} = 26325 KNmm \tag{4.6}$$

$$I = \frac{tb^3}{12};$$
$$I = 125x10^6 mm^4 \tag{4.7}$$

$$\acute{Y} = A_R \acute{Y}_R + A_T \acute{Y}_T / A_R + A_T;$$
$$\acute{Y} = 347mm \tag{4.8}$$

$$A_{total} = hb + 1/2\ (a + b) xh;$$
$$A_{total} = 532.89 mm^2 \tag{4.9}$$

Calculation of Total deflection (δ) the single head plat due to acting Main cylinder: when pin at one end and roller at the other.

$$Total deflection(\delta) = (Pl^3)/48EI;$$
$$Total deflection(\delta) = (81kN(1300mm)^3)/(48x200kN/mm^2x125X10^6mm^4) \quad (4.10)$$
$$Total\ deflection\ (\delta) = 0.32530mm$$

Where- p= applied load,

L = length of single head plat

E = Young's modulus of the single head plat

I= Moment of inertia of the cross-section about the neutral axis

Factor of safety for head plate

$$F.O.S = \frac{\sigma_y}{\sigma_a};$$
$$F.O.S = 320MPa/169MPa \quad (4.11)$$
$$F.O.S = 1.9 > 1$$

Where- σ_y = yield stress

σ_a = allowable stress

4.1.1.2 Calculation of Single Columns Buckling

From showing below figure 4.4 that, An I-section 240mm ×100mm × 10 mm and 2200mm long is used as a fix at bottom and free at top side. So, to find Euler's crippling load take Young's modulus for the material of the section as 200kN/mm^2. (R.s. khurmi j.k. gupt, 2005)

Assumptions in Euler's Column Theory the following simplifying assumptions are made in Euler's column theory:

i. The cross-section of the column is uniform throughout its length;
ii. The column material is perfectly elastic, homogeneous and isotropic, and thus obeys Hooke's law;
iii. The failure of column occurs due to buckling alone and Take maximum applied force;
iv. The weight of the column itself is neglected;

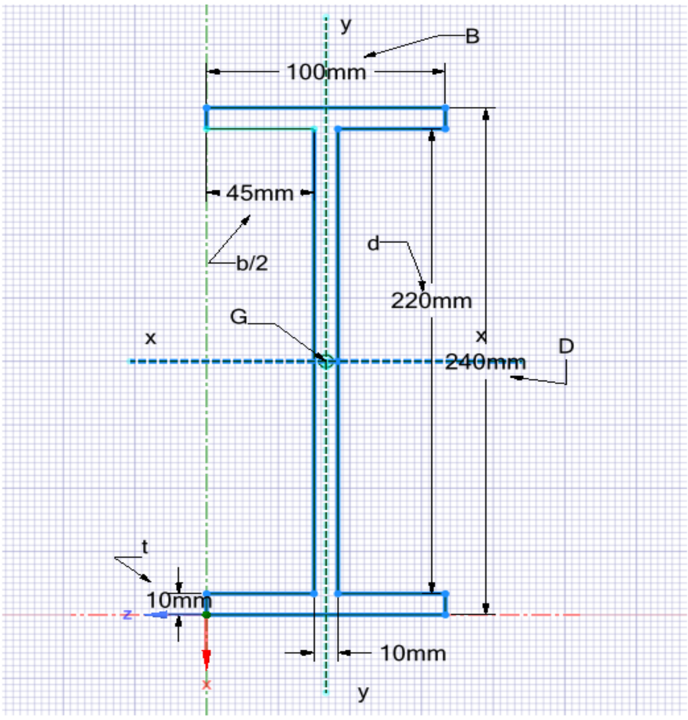


Figure 4. 4.Top Dimensions of single I –column

$$Stress\ (\sigma)\ = F_{app.}\ /\ A_{cross}\ ;$$
$$Stress\ (\sigma)\ = 81KN\ /\ \ 4200mm^2 \qquad (4.12)$$
$$Stress\ (\sigma)\ = 19.28Mpa$$

When- A $_{total}$ = C/S Area of I- Column

F $_{app}$= tensile force acting on I-Column

The moment of inertia I of the I section, relative to centroid y-y axis, is found by:

$$I_{XX} = (BD^3)/12 - (bd^3)/12;$$
$$I_{XX} =\ \ 7527x10^5 mm^4 \,(maximum) \qquad (4.13)$$

$$I_{YY} = 2(tB^3)/12) + (dt^3)/12;$$
$$I_{YY} = 1685X10^3 mm^4 \,(minimum) \qquad (4.14)$$

Where- h= the section height

w= the width of the flanges

t $_f$ =the thickness of the flanges

t $_w$ =the thickness of the web

Since the value of I $_{YY}$ is less than the value of I $_{XX}$ .therefore the section will tend to buckle about alone y-y axis.so take I as I_{YY}=1685 X 10^3 mm^4 (minimum).

Since the column is one end fixed and other end free, therefore equivalent length,

L_E = 2xL =2(2200mm) = 4400mm.

$$F_{(critical-1)} = ((\pi^2 xExI_{yy})/L^2);$$
$$F_{(critical-1)} = (((3.14)^2(200x10^3)(1685000mm^4))/[2(2200mm)]^2) \quad (4.15)$$
$$F_{critical-1} = 182KN$$

Where- E = Young's modulus of the material of the column

I_{YY} = Moment of inertia (minimum) of the cross-section about the y-axis

I_{XX} = Moment of inertia (maximum) of the cross-section about the x-axis

L = 2 x length of column (fixed-free column)

Column stability

So, Column stable condition because of the value of applied load greater than critical load ($F_{applied} < F_{critical}$).

= 81kN < 182kN (buckling not occur).

Calculation of column total deflection

$$Total\ deflection(\delta) = \frac{F_t L}{AE};$$
$$Total\ deflection(\delta) = 81kN\ x\ 2200mm/4200mm^2 x\ 200KN/mm^2 \quad (4.16)$$
$$Total\ deflection(\delta) = 0.21214mm$$

Where – F_t = tensile force acting on I-Column

L = length of I-Column

A = C/S Area of I- Column ($hb-h_1b_1$)

E= Young's modulus

Factor of safety of the column

$$F.O.S = \frac{\sigma_y}{\sigma_a};$$
$$F.O.S = 320N/mm^2/19N/mm^2 \quad (4.17)$$
$$F.O.S = 17 > 1$$

Where- σ_y = yield stress

σ_a = allowable stress

4.1.1.3 Calculation of Single Horizontal Bed

a. **Single horizontal bed plate (at: L= 1220mm)**

The simply-supported beam in figure 4.5 has concentrated force 81kN acting vertically downward at its midpoint of horizontal bed. The distance between the supports is labeled =1220mm, so the force 81kN is located half the distance (1220mm /2) =610mm from each end support. A_X =0 alone horizontal axis (Thomas H. Brown, 2005).

The following assumptions have been taken into consideration;

i. Single horizontal bed plat thickness equal to 16mm and cylinder applies force on single point of horizontal bed plat and considers maximum applied force;
ii. Weight of the Single horizontal bed plat is neglected;
iii. The material of the Single horizontal bed plat is perfectly homogeneous and isotropic;
iv. the Young's modulus (E) is the same in tension and compression;
v. horizontal bed plat is rectangular shape;

Calculations are as follows

Free body diagram of Single horizontal bed plate: at L= 1220mm

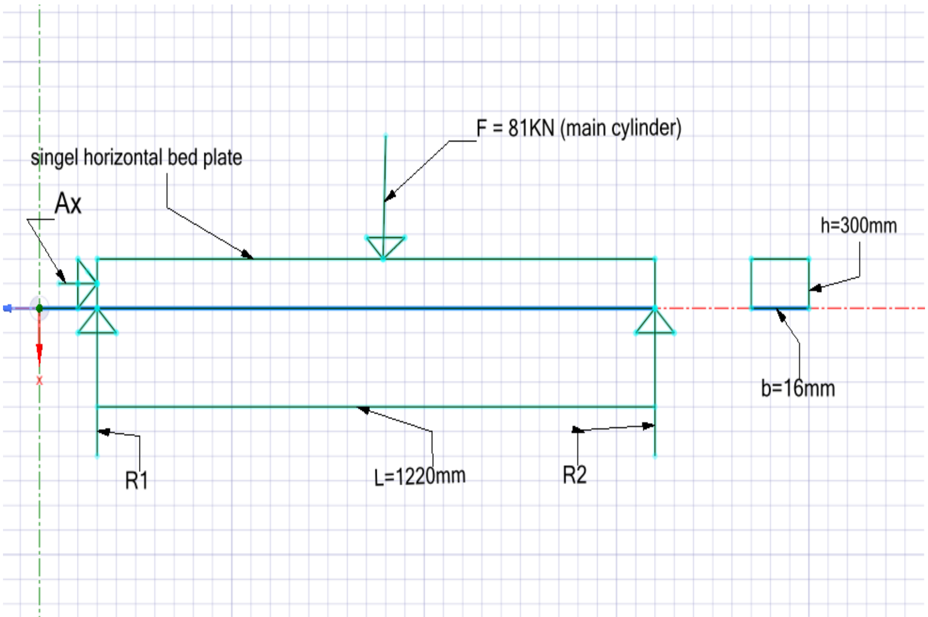


Figure 4. 5.FBD of Single horizontal bed plat :(at L= 1220mm)

Where – R_1 - fixed end support due to reaction

R_2 - Ball end support due to reaction

F- Applied maximum action force

h – Height single horizontal bed plate

b- Thickness of single horizontal bed plate

L - Horizontal length of horizontal bed plate

A_X – horizontal acting force; 0.

Maximum shear force:

$$From\ FBD:\ take; +\uparrow \sum F_Y = 0 \qquad R_1 + R_2 = 81kN;$$

$$From\ FBD:\ take; \sum M = 0; \qquad 81kN\ (610mm) - R_2(1220mm) = 0;$$

$$R_1 = R_2 = 40.5kN$$

Detail shear force based on simply supported horizontal bed plate

$$Point\ b/N\ A\ \&\ B:\ (0 \le x < 610mm);$$

$$40.5kN - V = 0;$$

$$V = 40.5kN$$

$$Point\ b/N\ B\ \&C:\ (610mm < x \le 1220);$$

$$V + 40.5kN = 0$$

$$V = -40.5kN$$

Shear Force Diagram

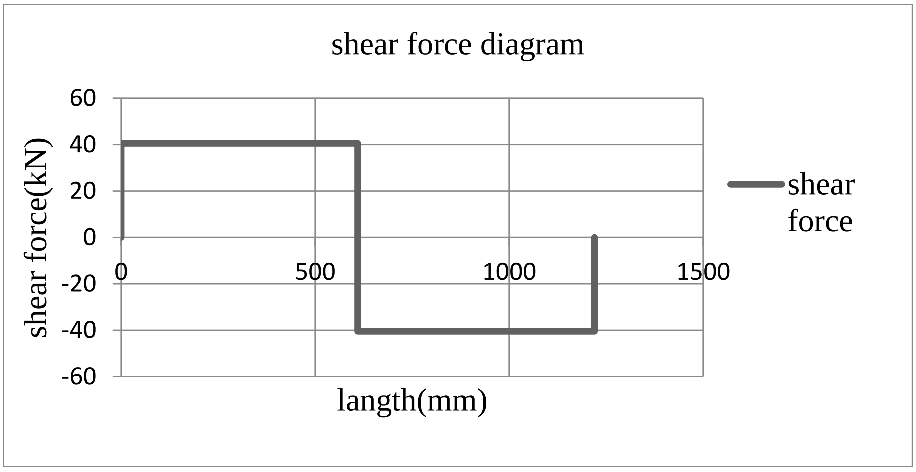


Figure 4. 6.Shear Force Diagram of Horizontal Bed Plate (at L= 1220mm)

<u>Note</u>: the shear force (V) is a positive (R_2= R/2) from the left end of the beam to the midpoint, and a negative (R_1=R/2) from the midpoint to the right end of the beam. So there is a discontinuity in the shear force at the midpoint of the beam, of magnitude is 40.5kN, where the concentrated force is applied. The maximum shear force (V_{max}) is therefore,

$$V_{maximum} = R/2\ ; \qquad V_{maximum} = 40.5kN \tag{4.18}$$

Bending moment:

Detail bending moment based on simple supporting horizontal bed plate

$Point\ b/N\ A\ \&\ B:\ (0 \le x < 610mm);$

$$40.5kN\ (305mm) - M = 0;\ when:\ x = 305mm;$$

$$M = 12352.5kNmm$$

$Point\ b/N\ B\ \&C:\ (610mm < x \le 1220);$

$$40.5kN\ (1220mm - x) - M = 0;\ when\ x = 915mm.$$

$$M = 12352.5kNmm.$$

Bending moment diagram

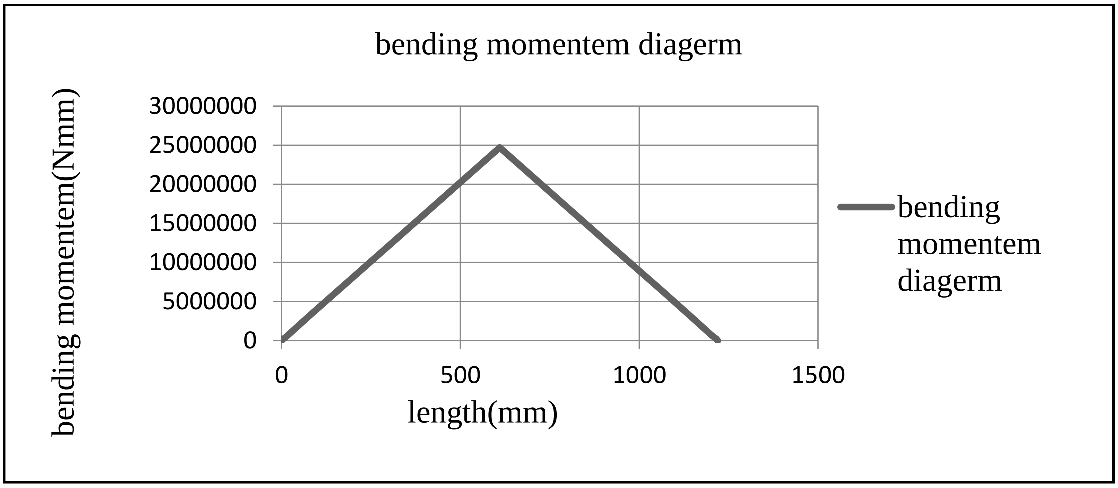


Figure 4. 7.Bending Moment Diagram of Horizontal Bed Plate :(at L= 1220mm)

Note: the bending moment is zero at both ends, and increases linearly to a maximum at the midpoint of 610mm. From the midpoint, the bending moment decreases linearly back to zero. The Maximum bending moment (M_{max}) occurs at the midpoint of the beam and is given by

$$M_{Maximum} = FL/4\ ;$$

$$\text{So;}\quad M_{Maximum} = 81kNx1220mm/4 \tag{4.19}$$

$$M_{Maximum} = 24,705kNmm$$

$$stress(\sigma) = \frac{F_{app.}}{A_{cross}};$$
$$stress(\sigma) = 81000N / 300mm^2 \tag{4.20}$$
$$stress(\sigma) = 270N / mm^2$$

Bending equation is given by

$$\frac{M}{I} = \frac{F_b}{y} = \frac{E}{R} \tag{4.21}$$

Where - M = Bending moment at the given section

F $_b$ = Bending stress

I = Moment of inertia of the cross-section about the neutral axis

Y = Distance from the neutral surface to the extreme fiber

E = Young's modulus of the material of the beam

Z = Section modulus

$$F_b = \frac{M}{I} Y;$$
$$Bending\ stress\ (F_b) = 24,705kNmmx150mm / 36\ x\ 10^6 mm^2 \tag{4.22}$$
$$Bending\ stress\ (F_b) = 102.9375N / mm^2$$

$$M = FL / 4;$$
$$M = 24,705kNmm \tag{4.23}$$

$$I = (tb^3)/12$$
$$I = 36x10^6 mm^4 \tag{4.24}$$

$$Y = \frac{b}{2};$$
$$Y = 150mm \tag{4.25}$$

Calculation of total deflection (δ) single horizontal bed plate due to acting downward Main cylinder: when pin at one end and roller at the other At L=1220mm (simply supporting beam) :

$$Total deflection(\delta) = (Rl^3) / 48EI;$$
$$Total deflection(\delta) = (81kN(1220mm)^3) / (48(200kN / mm^2)36x10^6 mm^4) \tag{4.26}$$
$$Total\ deflection(\delta) = 0.425589mm$$

Where - p = maximum applied load on single horizontal bed plate;

L = length of single horizontal bed plate;

E = Young's modulus of the material of the single horizontal bed;

I = Moment of inertia of the cross-section about the neutral axis;

Factor of safety horizontal bed

$$F.O.S = \frac{\sigma_y}{\sigma_a} \;;$$

$$F.O.S = (366N/mm^2)/(270N/mm^2) \tag{4.27}$$

$$F.O.S = 1.4 > 1$$

Where- σ_y = yield stress

σ_a = allowable stress

b. Calculation of Single horizontal bed plate (at: L= 1220mm, C=100mm)

Double Overhang: Concentrated Forces at Free Ends the simply-supported beam in figure 4.8 has double overhangs with concentrated forces, each of magnitude (10kN), acting directly downward at the free ends: points of horizontal bed plat. The distance between the supports is labeled (L=1220mm), and the length of each overhang is labeled (C=100mm). Therefore, the total length of the horizontal bed plat (L_{total}=1420mm), measured from the left end, is (L + 2C) =1420mm. (Thomas H. Brown, 2005).

The following assumptions have been taken into consideration;

i. The cylinder applies force to a single point on 16mm horizontal bed plat and takes into account the maximum applied force.
ii. Weight of the Single horizontal bed plat is neglected;
iii. The material of the Single horizontal bed plat is perfectly homogeneous and isotropic;
iv. the Young's modulus (E) is the same in tension and compression;
v. horizontal bed plat is rectangular shape;

Calculations are as follows

Free body diagram of Single horizontal bed plat (double overhanging-concentrated force at free ends);

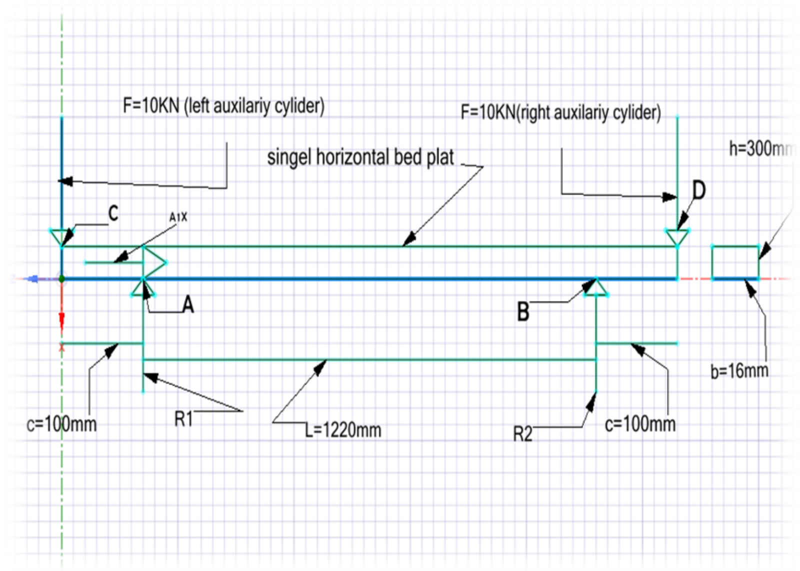


Figure 4. 8.FBD of Single horizontal bed plat Twin concentrated forces when pin at one end and roller at the other (at: L= 1220mm, C=100mm)

Where – R_1- simple support (reaction force);

R_2- Ball end support (reaction force);

R - Applied auxiliary cylinders force;

h- Height horizontal bed plate;

b – Thickness of single horizontal bed plate;

C - Distance between R_1, R_2 &Applied auxiliary cylinders force;

L - Distance between R_1 & R_2 of single horizontal bed plate;

<u>Reactions</u>: Figure 4.8, the balanced free-body diagram, depicts the reactions at the supports. The vertical reactions (R_1 and R_2) are equal, each with a magnitude of 10kN. As both forces are acting directly downward, the horizontal reaction (R_{1x}) is zero. From above, FBD represented the following:

Maximum shear force:

$From\ FBD:\ take; +\uparrow \sum Fy = 0;\quad R_1 + R_2 - 10kN - 10kN = 0;$ But the value of $R_1=R_2$ b/c equal magnitude of force & distance on both side of horizontal bed plate.

$$2R_1 = 20kN;$$
$$so,\ R_1 = R_2 = 10kN$$

Detail shear force based on simple supporting horizontal bed plat

$Point\ c;$

$S.f_{left} = 0;$ $\quad S.f_{right} = 10kN;$

$Point\ A;$

$S.f_{left} = 10kN;$ $\quad S.f_{right} = -10kN + 10kN = 0;$

$Point\ B;$

$S.f_{left} = -10kN;$ $\quad S.f_{right} = -10kN + 10kN - 10kN = -10kN;$

$Point\ D;$

$S.f_{left} = -10kN - 10kN + 10kN = -10kN;$ $S.f_{right} = -10kN + 10kN + 10kN - 10kN = 0;$

Shear Force Diagram

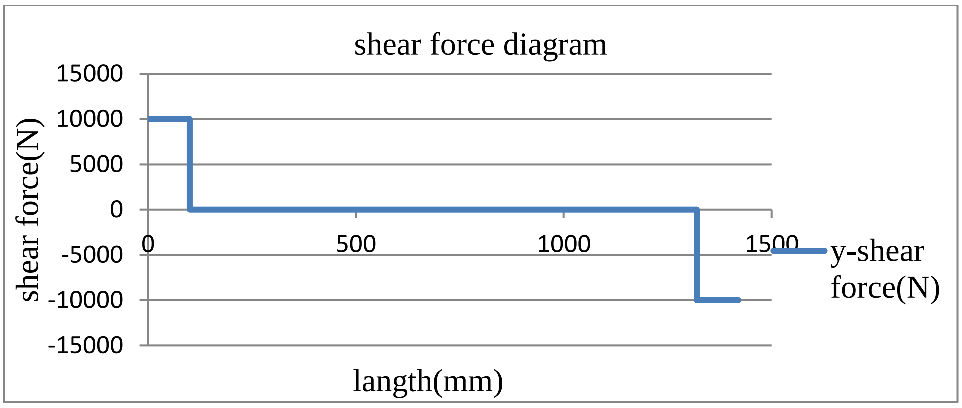


Figure 4. 9.Maximum shear force diagram: Twin concentrated forces when pin at one end and roller at the other (At: L= 1220mm, C=100mm).

From the above figure 4.9 is the maximum shear force (V_{max}) occurs in two regions, one from the left end of the horizontal bed plate to the left support, and the other from the right support to the right end of the horizontal bed plate. So, the maximum shear force (down) is

$$V_{maximum} = F\ ; \qquad V_{maximum} = 10kN \tag{4.28}$$

Bending moment;

$From\ FBD:\ take; \sum M = 0;$

Detail Bending moment on simple supporting horizontal bed plat

$Point\ c;$

$B.M_C = 0;$

$Point\ A;$

$B.M_A = (-10kN)\ \text{x}\,(100mm) = -1000kNmm;$

$Point\ B;$

$B.M_B = (-10kN)\ \text{x}\,(100mm + 1220mm) + 10kN\ (1220mm) = -1000kNmm;$

$Point\ \text{D};$

$B.M_D = 0;$

Bending moment diagram

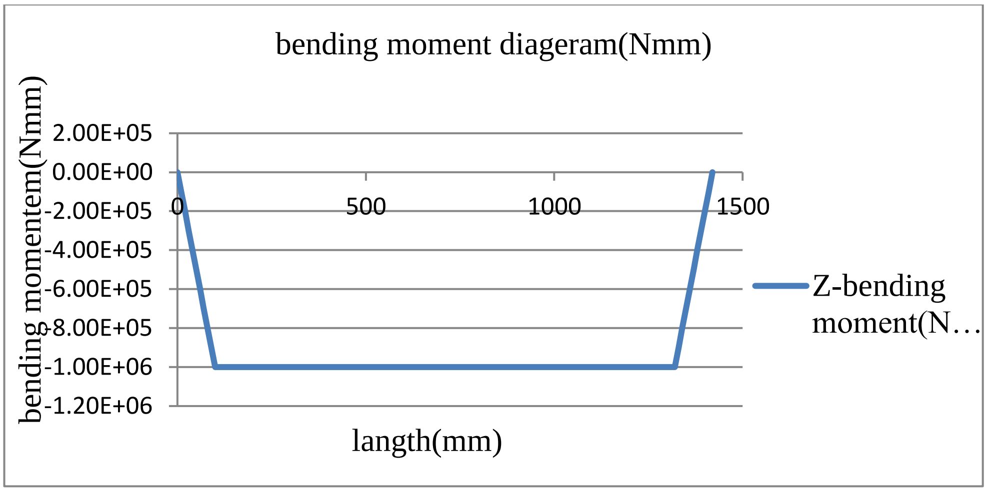


Figure 4. 10.Bending moment diagram: twin concentrated forces when pin at one end and roller at the other (at: L= 1220mm, C=100mm).

<u>**Note**</u>: - from above figure 4.10 the bending moment (M) decreases linearly from zero at the left end of the beam to a value (−1000kNmm) at the left support, stays a constant (−1000kNmm) between the supports, then increases linearly back to zero at the right end. The maximum bending moment (M_{max}) that is always a positive quantity occurs between the supports and is given by

$$M_{max} = Fc$$
$$M_{max.} = 1000kN, \tag{4.29}$$

So, the bending equation is given by

$$\frac{M}{I} = \frac{F_b}{y} = \frac{E}{R} \tag{4.30}$$

Where - M = Bending moment at the given section,

F_b = Bending stress

I = Moment of inertia of the cross-section about the neutral axis;

Y = Distance from the neutral surface.

E = Young's modulus of the material of the beam.

$$stress(\sigma) = \frac{F_{app.}}{A_{cross}}$$
$$stress(\sigma) = 10000N / 480mm^2 20.8N / mm^2 \quad (4.31)$$
$$stress(\sigma) = 20.8N / mm^2$$

Bending stress (double overhanging-concentrated force at free ends).

$$Bendingstress(F_b) = M_{max} y_{max} / I$$
$$Bendingstress(F_b) = 1000kNx150mm / (36x10^6 mm^2) \quad (4.32)$$
$$Bendingstress(F_b) = 4.166N / mm^2$$

$$I = (ab^3) / 12$$
$$I = 36x10^{(6)} mm^4 \quad (4.33)$$

$$Y_{max} = \frac{b}{2}$$
$$Y_{max} = \frac{300mm}{2} = 150mm \quad (4.34)$$

Total Deflection (δ) of single horizontal bed plate due to acting auxiliary cylinder down ward acting: when pin at one end and roller at the other.

Auxiliary cylinder down P $_{down =}$ 10kN,

$$TotalDeflection(\delta) = (Fl^2 c) / 8EI$$
$$TotalDeflection(\delta) = (10kN)(1220mm)2(100mm) / 8(200kN)(36x106mm^4) \quad (4.35)$$
$$TotalDeflection(\delta) = 0.0259mm$$

Where - δ – Deflection of beam

R - Applied auxiliary cylinders force

L - Distance between R_1 & R_2 of single horizontal bed plate

C - Distance between R_1, R_2 &Applied auxiliary cylinders force

E - Modulus of elasticity of single horizontal bed plat

I - moment of inertia of cross-sectional area

Factor of safety horizontal bed

$$F.O.S = \frac{\sigma_y}{\sigma_a} \;;$$

$$F.O.S = (345N/mm^2)/(21N/mm^2) \tag{4.36}$$

$$F.O.S = 16 > 1$$

Where- σ_y = yield stress

σ_a = allowable stress

c. Calculation of Single horizontal bed plate (at L= 1420mm)

Concentrated Forces at Free Ends the simply-supported beam in figure 4.11 has double overhangs with concentrated forces, each R_1&R_2 of magnitude (5kN), acting directly upward at the free ends: points of horizontal bed plat. The distance between the supports is labeled (L=1420mm) (Thomas H. Brown, 2005).

The following assumptions have been taken into consideration

i. The cylinder applies force to a single point on 16mm horizontal bed plat and takes into account the maximum applied force

ii. Weight of the Single horizontal bed plat is neglected

iii. The material of the Single horizontal bed plat is perfectly homogeneous and isotropic

iv. The Young's modulus (E) is the same in tension and compression

v. horizontal bed plat is rectangular shape

Calculations are as follows

Free body diagram of Single horizontal bed plate: at L= 1420mm

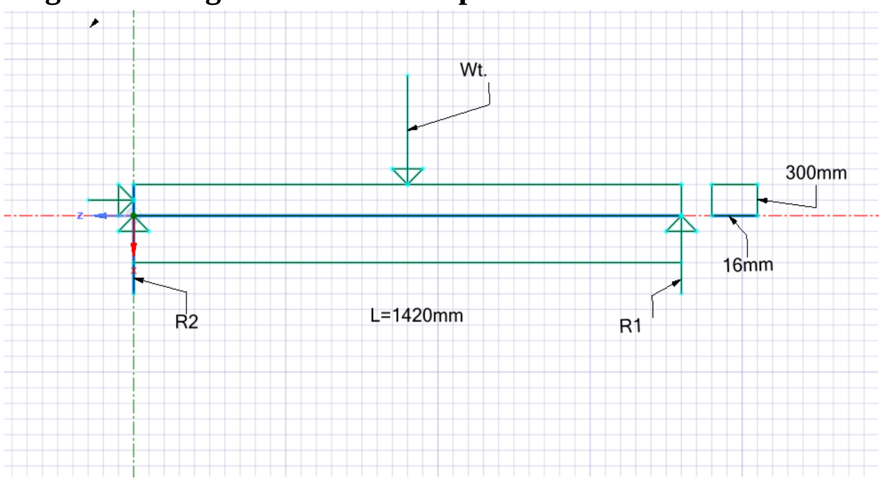


Figure 4. 11.FBD Single horizontal bed plate diagram: Twin concentrated forces when pin at one end and roller at the other: (at L= 1420mm)

Where – R_1 - fixed end support due to reaction of auxiliary cylinder

R_2 - Ball end support due to reaction acting auxiliary cylinder

R -maximum lifting load

h- Height horizontal bed plate

b – Thickness of single horizontal bed plate

L - Horizontal length of horizontal bed plate

A_X – horizontal acting force

Shear force

$$From\ FBD:\ take;\ +\uparrow\sum F_Y = 0; R_1 + R_2 = 5kN$$

$$From\ FBD:\ take;\sum M = 0; 5kN\ (710mm) - R_2(1420mm) = 0;$$

$$R_1 = R_2 = 2.5kN$$

Detail shear force based on simple supporting horizontal bed plate

$$Point\ b/N\ A\ \&\ B:\ (0 \le x < 710mm);$$

$$From\ FBD:\ take; +\uparrow\sum Fy = 0;$$

$$2.5kN - V = 0;$$
$$V = 2.5kN$$

$$Point\ b/N\ B\ \& C:\ (710mm < x \le 1420mm);$$

$$From\ FBD:\ take + \uparrow\sum Fy = 0$$

$$V + 2.5kN = 0$$
$$V = -2.5kN$$

Shear force diagram

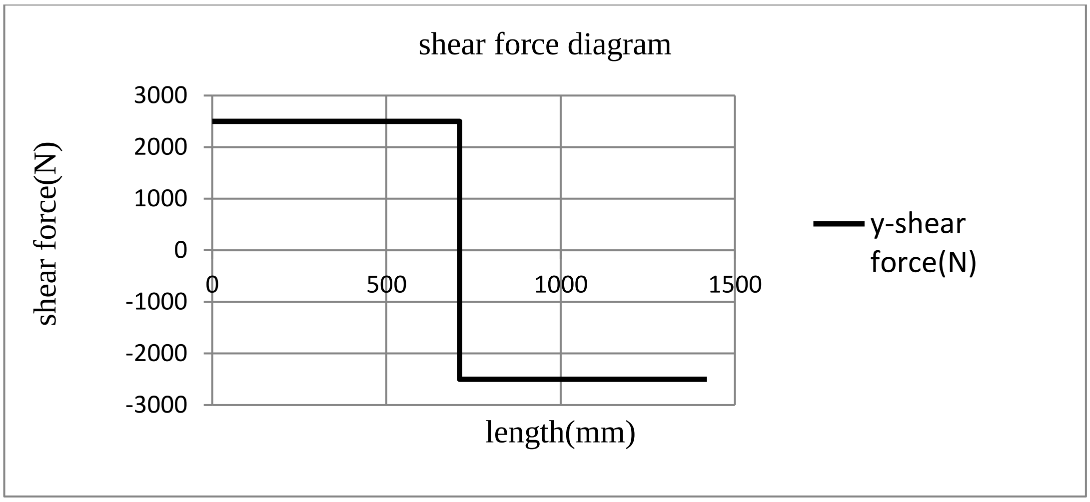


Figure 4. 12.Shear Force Diagram Single Horizontal Bed Plate :(at L= 1420mm)

Note: the shear force (V) is a positive (R_2= R/2) from the left end of the beam to the midpoint, and a negative (R_1=R/2) from the midpoint to the right end of the beam. So there is a discontinuity in the shear force at the midpoint of the beam, of magnitude is 2.5kN, where the concentrated force is applied. The maximum shear force (V_{max}) is therefore,

$$V_{maximum} = R/2\ ; \qquad V_{maximum} = 2.5kN \tag{4.37}$$

Bending moment:

Detail bending moment based on simple supporting horizontal bed plate

$Point\ b/N\ A\ \&\ B:\ (0 \le x < 710mm);$

$$\begin{aligned} &From\ FBD:\ take; \sum M = 0; When:\ x = 355mm; \\ &2.5kN\ (355mm) - M = 0 \\ &M = 887.5kNmm \end{aligned}$$

$Point\ b/N\ B\ \&C:\ (710mm \le x \le 1420mm);$

$$\begin{aligned} &From\ FBD:\ take; \sum M = 0; when\ x = 1065mm; \\ &2.5kN\ (1420mm - x) - M = 0 \\ &M = 1012.5kNmm. \end{aligned}$$

Bending moment Diagram

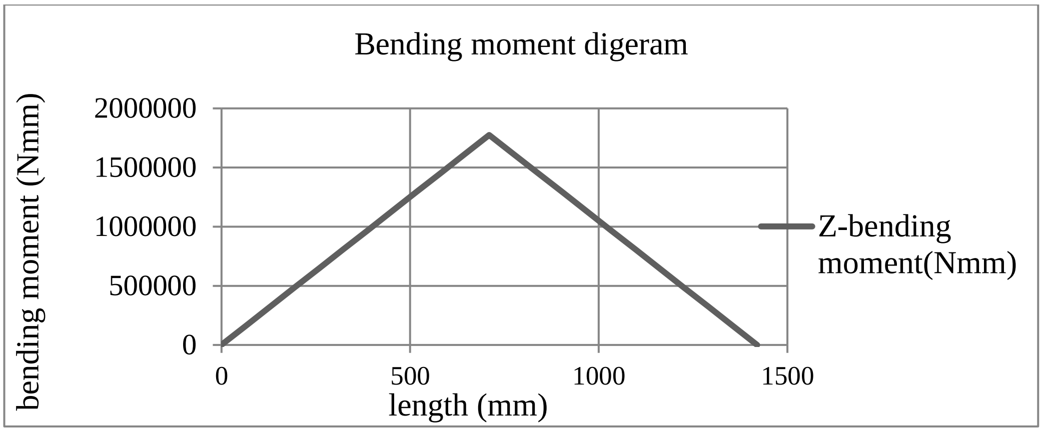


Figure 4. 13.Bending moment diagram for single horizontal bed plate :(at L=1420mm)

Note: the bending moment is zero at both ends, and increases linearly to a maximum at the midpoint of 710mm. From the midpoint, the bending moment decreases linearly back to zero. The Maximum bending moment (M_{max}) occurs at the midpoint of the beam and is given by

$$\begin{aligned} M_{Maximum} &= FL/4 \\ M_{Maximum} &= 5kN\ x\ 1420mm/4 \\ M_{Maximum} &= 1775kNmm \end{aligned} \tag{4.38}$$

$$\begin{aligned} stress(\sigma) &= \frac{F_{app.}}{A_{cross}} \\ stress(\sigma) &= 5000N/16mm^2 \\ stress(\sigma) &= 312.5N/mm^2 \end{aligned} \tag{4.39}$$

Bending equation is given by

$$\frac{M}{I} = \frac{F_b}{y} = \frac{E}{R} \tag{4.40}$$

Where - M = Bending moment at the given section

F_b = Bending stress

I = Moment of inertia of the cross-section about the neutral axis

Y = Distance from the neutral surface to the extreme fiber

E = Young’s modulus of the material of the beam

Z = Section modulus

$$Bending\ stress\ (F_b) = MY / I;$$
$$Bending\ stress\ (F_b) = 1775kNmm\ x150mm / 36\ x\ 10^6 mm^2 \tag{4.41}$$
$$Bending\ stress\ (F_b) = 7.3958N / mm$$

$$I = (tb^3)/12; \quad I = 36x10^6 mm^4 \tag{4.42}$$

$$Y = b/2; \quad Y = 150mm \tag{4.43}$$

Total Deflection (δ)of single horizontal bed plate due to acting upward auxiliary cylinder: when pin at one end and roller at the other At L=1420mm (simply supported beam) :

$$TotalDeflection(\delta) = (Fl^3)/48EI;$$
$$TotalDeflection(\delta) = (5kN(1420mm)^3)/(48(200kN/mm^2)36x10^6 mm^4) \tag{4.44}$$
$$Total\ Deflection\ (\delta) = 0.04142mm$$

Where - p = maximum applied load on single horizontal bed plate;

L = length of single horizontal bed plate

E = Young's modulus of the material of the single horizontal bed

I = Moment of inertia of the cross-section about the neutral axis

Factor of safety horizontal bed

$$F.O.S = \frac{\sigma_y}{\sigma_a};$$
$$F.O.S = 345N/mm^2 / 313\ N/mm^2 \tag{4.45}$$
$$F.O.S = 1.1 > 1$$

Where- σ_y = yield stress

σ_a = allowable stress

4.1.2 Modeling and simulation analysis

Using the ANSYS 2021R1 and 2022R2 program, component analyses are described and simulated. Employed in creating a model of a suitable hydraulic press machine as well. Software that uses pretreatment, processing, and post processing to accomplish static structural evaluation. The models are built on the space claim platform and imported into the ANSYS workbench. Mechanical APDL is used to mesh models of press machine assembly. Apply the load at the appropriate location, and then note the post-processing outcomes. In order to provide correct findings, the determination of the maximum von Misses stress and total deflection rely on the use of proper element sizes and model quality. Steps followed are the given below.

i. Create model on space claim platform & export to ANSYS design modular.
ii. Selection of element and define meshing /Apply Mesh/.
iii. Appling boundary condition & loading conditions.
iv. Analyzing the geometry for its maximum and minimum value of stress and deflection.

Head frame assembly: It is located at the head of ECMHPM and it is organized from; head plates, supporter, flange, holes for bolts, and passing space of the main cylinder. It moves up and down the action due to the action of the main cylinder. The assemble model is shown in figure 4.14

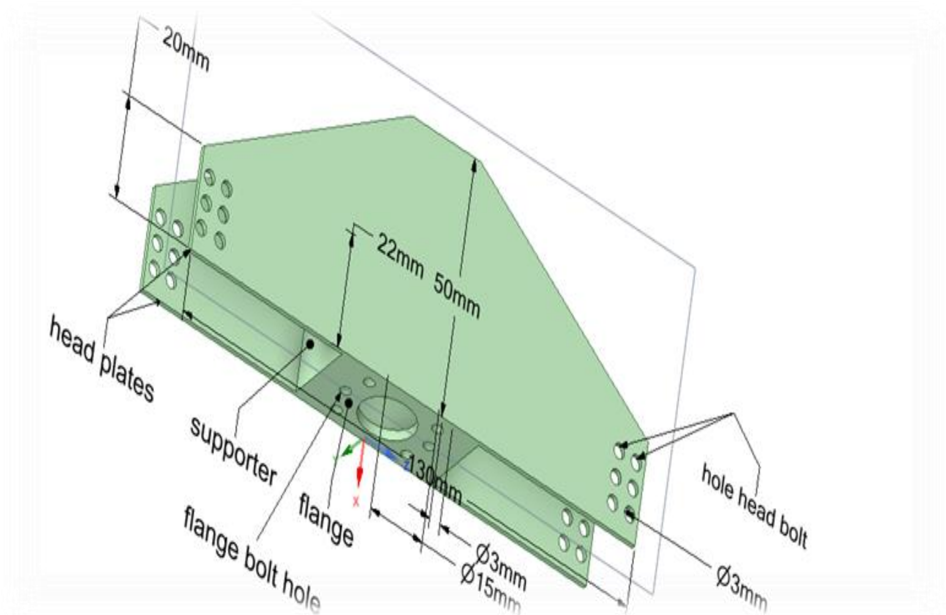
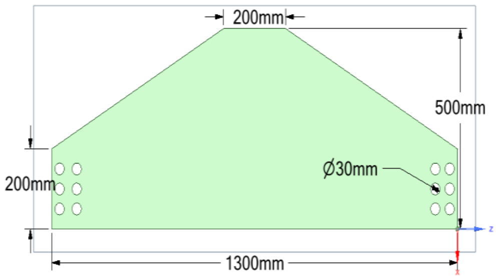


Figure 4. 14.position of Single head plate on head frame assembly

4.1.2.1 Single head plate (Model-1)

Creation of single head plate in space claim platform & exporting to ANSYS design modular. It is possible to create the geometry of the single head plate using Design Modeler in ANSYS Workbench. Before attaching the geometry, it specifies several options to determine the characteristics of the geometry to import from Design Modeler. These are the following options: solid bodies, surface bodies, line bodies, parameters, attributes, named selections, material properties; analysis type 2D or 3D, importing coordinate systems Import work points are only available in the design modeler more application. Then the browsed model of the single head plate is done on spaceclaime ANSYS 2022R2 software. The model is shown in figure 4.15.



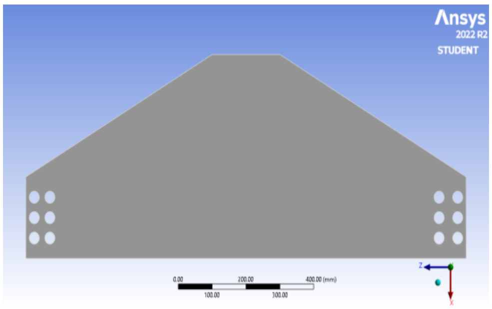


Figure 4. 15.Imported 3d model from spaceclaim to design modeler of ansys 2022R2 work bench (model-1).

i. Defining meshing & Selection of element /Apply Mesh/

Provide an adequate mesh density on contact surfaces to allow contact stresses to be distributed in a smooth fashion. Likewise, provide a mesh density adequate for resolving stresses; areas where stresses or strains are of interest require a relatively fine mesh compared to that needed for displacement resolution. The meshed model of the single head plate then looks like in the figure 4.16.with mesh element size of 10mm, 56392 nodes, and 9962 elements.

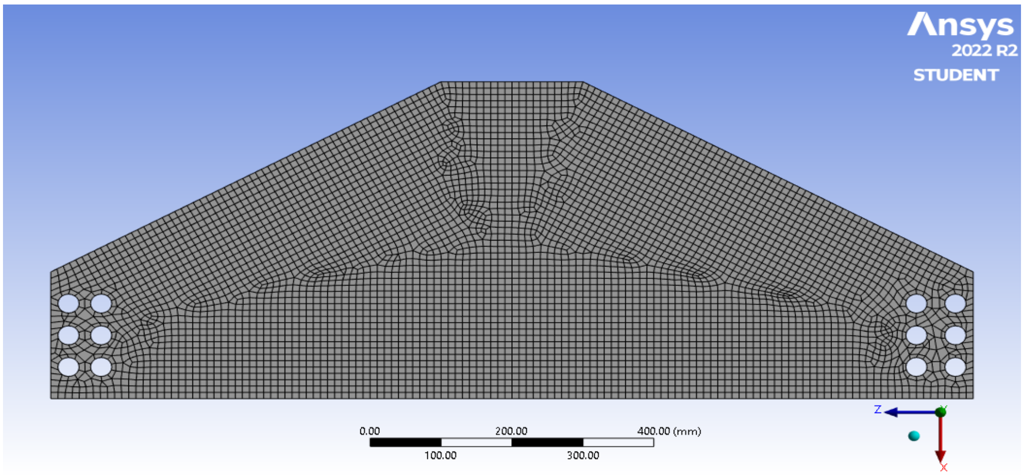


Figure 4. 16.Meshed model of single head plate (model-1).

ii. Applying boundary condition & loading condition

Force constraint: When observing the single head plate mounting position concerning the exerting load of the main cylinder, the right and left sides of the single head plate are connected by the columns. According to this characteristic, the load is applied at the center of the single head plate. The right and left sides of the single head plate are linked by bolts. It acts as the static motion of the main cylinder. So, statics Stress and total deformation analysis in a single head plate; acting 81kN (simple supporting beam).The force application at the center of a single head plate when fixed and displaced at the edge. The maximum von-Mises stress and total deformations so, equivalent (Von Misses) stress is a value used to determine if a given material will yield or fracture as shown in figure 4.17.

Displacement constraint: By considering the actual behavior of a single head plate on the ECMHPM, one end of the single head plate is resting on the column and the other end is on the other column. So one end of a single head plate is fixed support in the x-axis and displacement alone in the x and Z-axes. The other ends of the y-component are free.

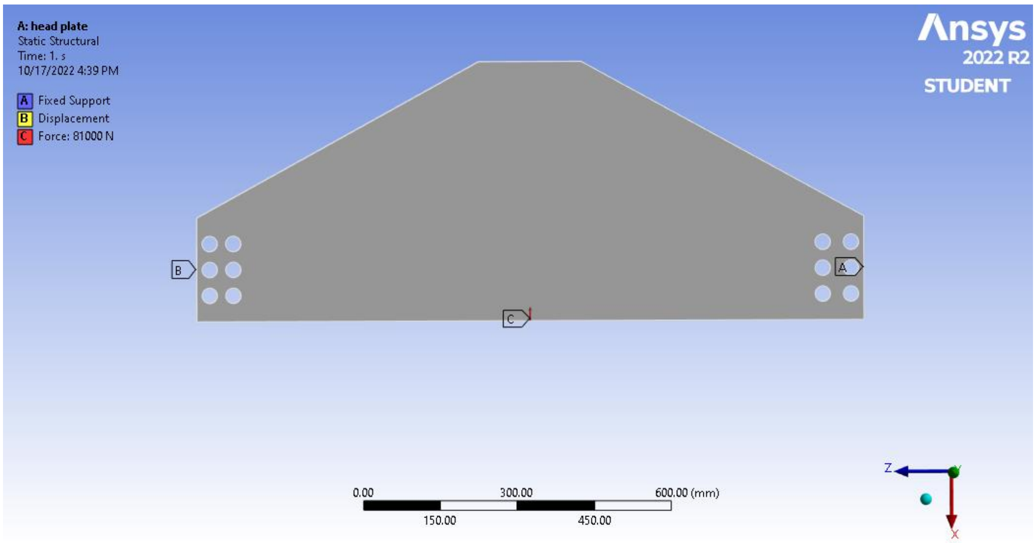


Figure 4. 17.Applied load and boundary conditions of single head plate (model-1)

iii. Model Is Analysed Using ANSYS Software for Stresses and Deflections

The highest value of total deformation is indicated as 0.35276mm in the analysis result for Model-1, and the maximum value is 152MPa. The main cylinder's position, where the maximum deformation occurs, and the canter side of the head plates, where the maximum stress first started, indicates that the critical zone is highlighted in red. The lowest values of stress and deformation are 3.3407Mpa and 0 mm, respectively. It designates the safe area as being blue. The results obtained from theoretical analysis and soft-wear analyses are given in Table 4.5.

Table 4.5. Theoretical and Computer aided analysis single head plate

Analysis	Theoretical Results	Computer aided analysis
Maximum von- Mises stress	152MPa	152.28MPa
Total deflection	0.32530mm	0.35276mm

Validation: it is seen from Table 4.5, the theoretical stress of 152.2MPa and the maximum stress of 152MPa obtained with computer analysis are almost equal and are less than the yield strength of the selected material of structural steel, which is 320MPa. The maximum total deflection is 0.35276mm. The factor of safety is 1.9, which is greater than 1, so the head plate is safe under operation.

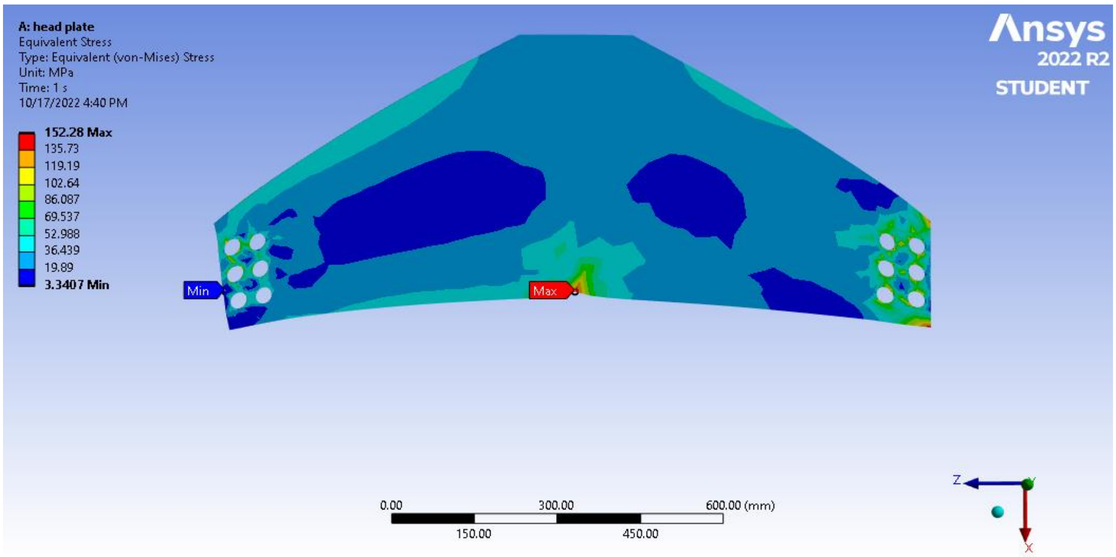


Figure 4. 18.Stress analysis of single head plat with pin at one end and roller at the other

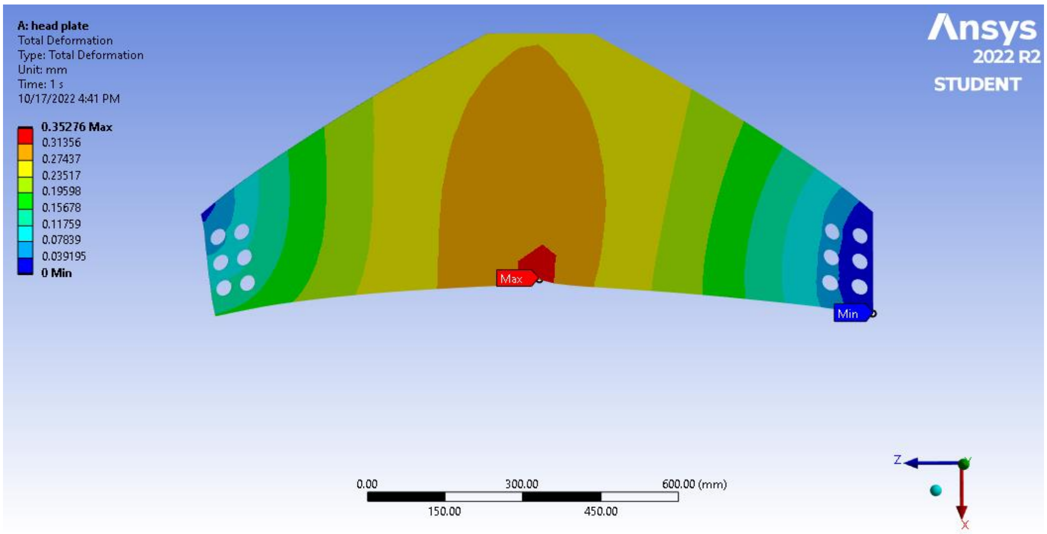


Figure 4. 19.Total deflection single head plate with pin at one end and roller at the other

Column assembly: It is tightened on the head frame assembly and foot plats with 30mm bolts, and the horizontal bed plat assembly is suspended and slid up and down with the help of an auxiliary cylinder. The column is constructed from I-columns, ground plat, foot bolts, pin holes, guide plates, and an auxiliary cylinder supporter is attached on the side of the I-columns, but the auxiliary hydraulic cylinders are tight on the side of the column. The assemble model is shown in figure 4.20.

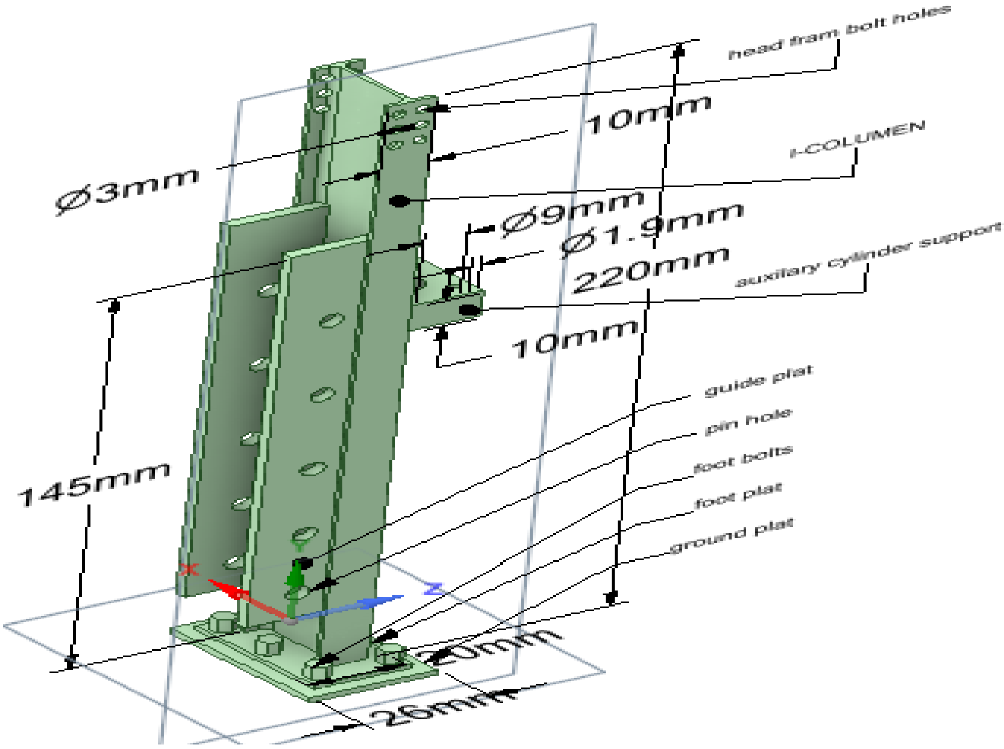


Figure 4. 20.Position of Single column on Single column assembly.

4.1.2.2 Single column (Model-2)

Create model single column on space claim platform & export to ANSYS design modular ANSYS Workbench's Design Modeler, which can export a file that is supported by ANSYS Workbench, can be used to construct the geometry of a single column. It gives a number of choices to decide the properties of the geometry to import from Design Modeler before attaching it. These capabilities, which are exclusively available in the Design Modeler program, include solid bodies, surface bodies, line bodies, parameters, attributes, named selections, material characteristics, using 2D or 3D analysis types, importing coordinate systems, and importing work points. The single-column browsed model is then completed using the Space climate ANSYS 2022R2 program. More shown in figure 4.21.

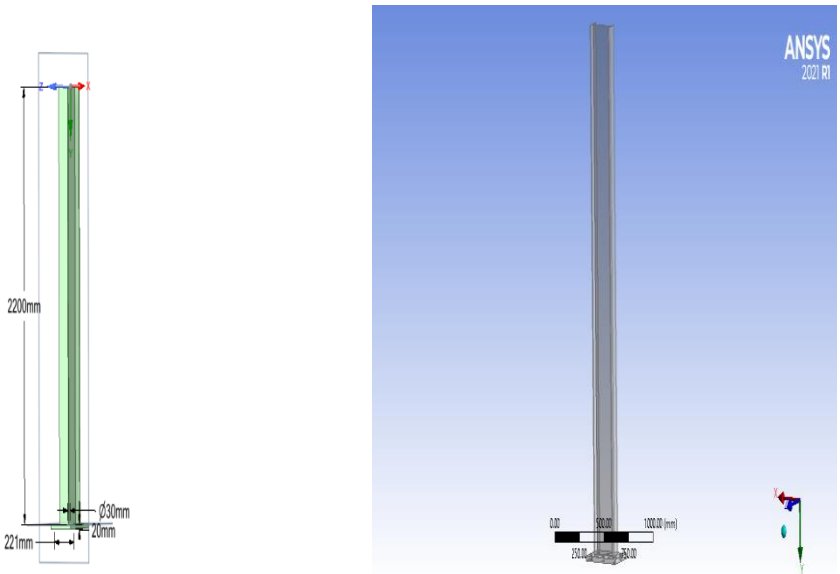


Figure 4. 21.Imported 3D model from spaceclaim to Design Modeler of ANSYS 2022R2 work bench (Model-2).

i. Defining meshing & Selection of element /Apply Mesh/

To enable the smooth distribution of contact stresses, provide a sufficient mesh density on the contact surfaces. Additionally, make sure that the mesh density is sufficient to resolve 53 stresses; regions where stresses or strains are important call for a finer mesh than what is required for resolution of displacements. The single column mesh model then appears as seen in the Figure 4.22; mesh element size 10mm was used, along with 46360 nodes and 22589 numbers of elements.

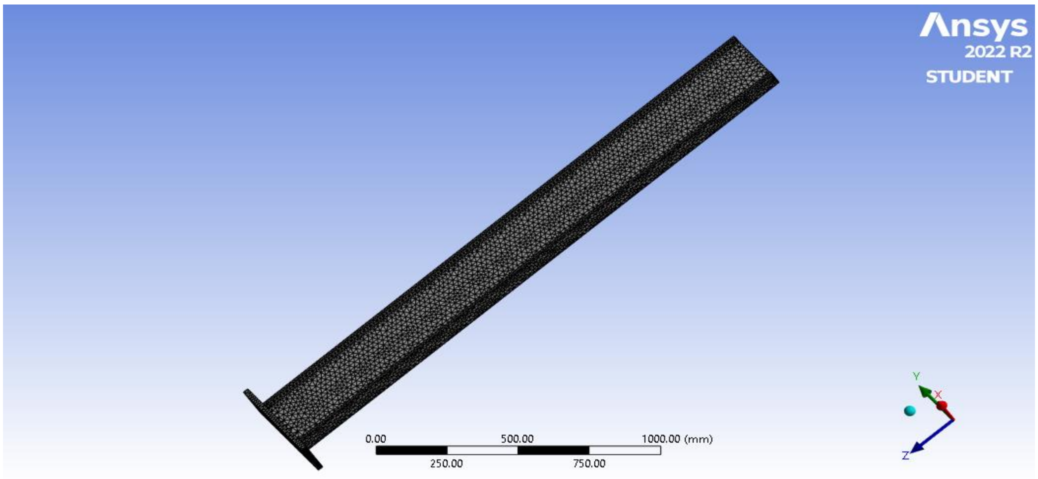


Figure 4. 22.Meshed model of single column (Model-2)

ii. Application of boundary & loading condition.

Force constraint: When the main and auxiliary cylinders' exerting loads are considered while installing in a single column Using 30mm bolts, the top of the single column is joined to one side of the head frame that is built using links. The characteristic columns serve as the static motion by transferring compressive or tensile stress from the loads supplied at the center of the main and auxiliary cylinders to the foot plate. Hence, statics: 81kN of acting tensile load; analysis of stress and total deformation in a single column. The power when it is fixed to the foot plate and moves the top of one column, it is called an application. So, when determining whether a certain material would yield or fracture, the equivalent Von Misses stress is applied. More shown in the Figure 4.23.

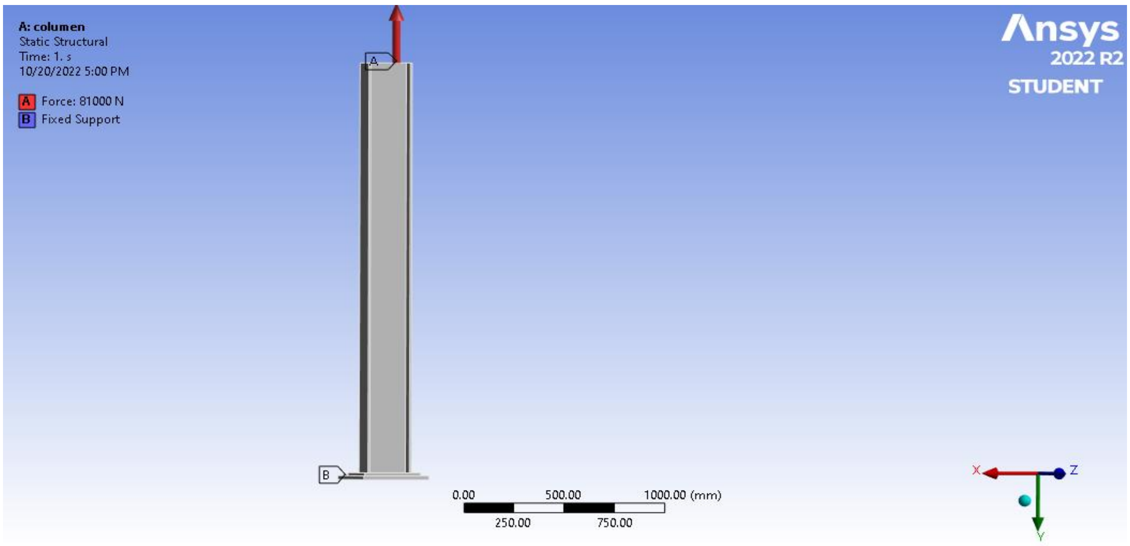


Figure 4. 23.Applied load and boundary conditions of single column (Model-2).

iii. Model is analysed using ANSYS software for stresses and deflections

The newly developed model-2's study yielded a load multiplier value of 188KN and a von-Mises stress of 22MPa. The entire deformation's maximum value is 0.21205mm. It demonstrates that the crucial region is highlighted in red and that the minimal amount of von- Mises stress and deformation is 0.00032MPa and 0MPa, respectively. It denotes that a safe area is designated by the color blue. The two are clearly contrasted in the following: theoretical analysis and computer-assisted analysis are given in Table 4.6.

Table 4.6. Theoretical and Computer aided analysis single singe column

Analysis	Theoretical Results	Computer aided analysis
von- Mises stress	19MPa	22Mpa
buckling	182KN	188KN
Total deflection	0.21214mm	0.21205mm

Validation: The Table 4.6 demonstrates that the theoretical stress of 19MPa is the yield strength of the frame, which is 320MPa because its material is structural steel. It experiences a maximum stress of 19MPa (see Figure 4.24), a maximum displacement of 0.21214 mm, and a strain value of 0.000096. These results show that the design of the single column is safe. Additionally, the value of applied force (81kN) is less than the value of buckling (182kN). So the machine will be safe under operation.

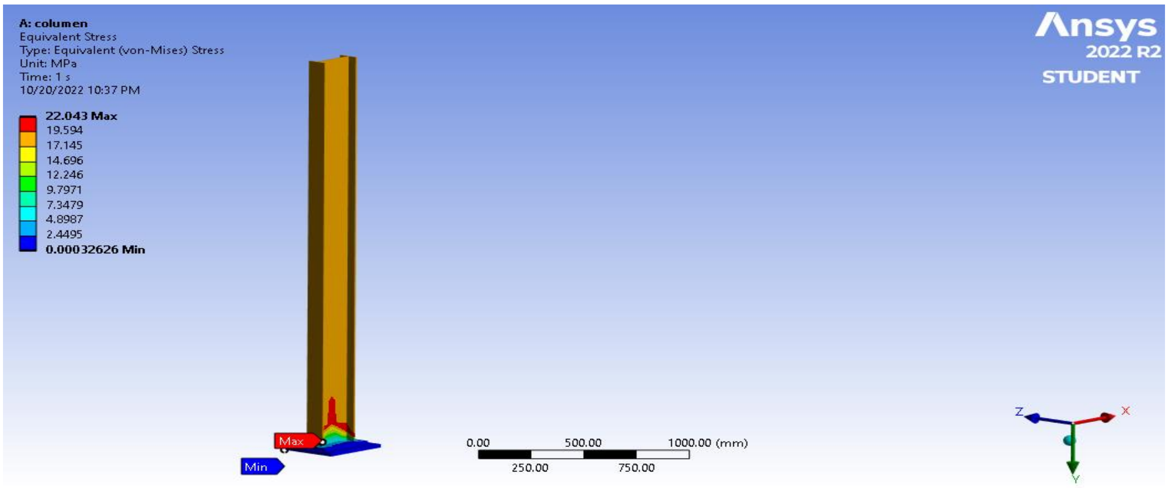


Figure 4. 24.Stress analysis of Single column

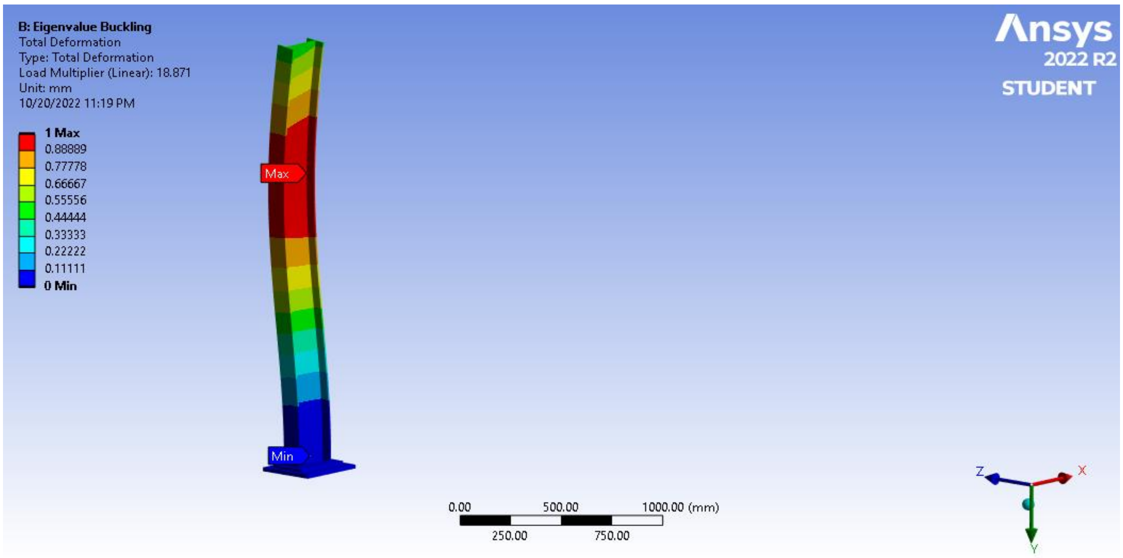


Figure 4. 25.Buckling analysis of Single column

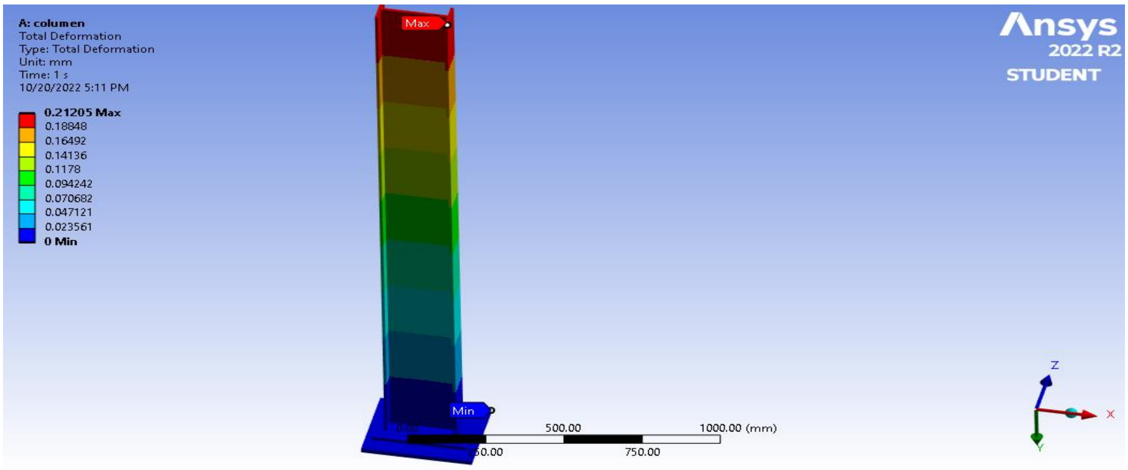


Figure 4. 26.Analysis of total deflection on single Column

Horizontal bed assembly: It is suspended by two auxiliary hydraulic cylinders in the center of the vertical columns. The auxiliary cylinder flange aids in the horizontal bed's up-and-down movement by being fastened to a supporter at the side of the I-columns. Three situations are taken into account:

a) Compression at the center of a single horizontal bed plate: L = 1220mm
b) Holding engine block during extraction of engine liner: L = 1220mm, C = 100mm
c) Lifting heavy weight from the ground: L = 1420mm

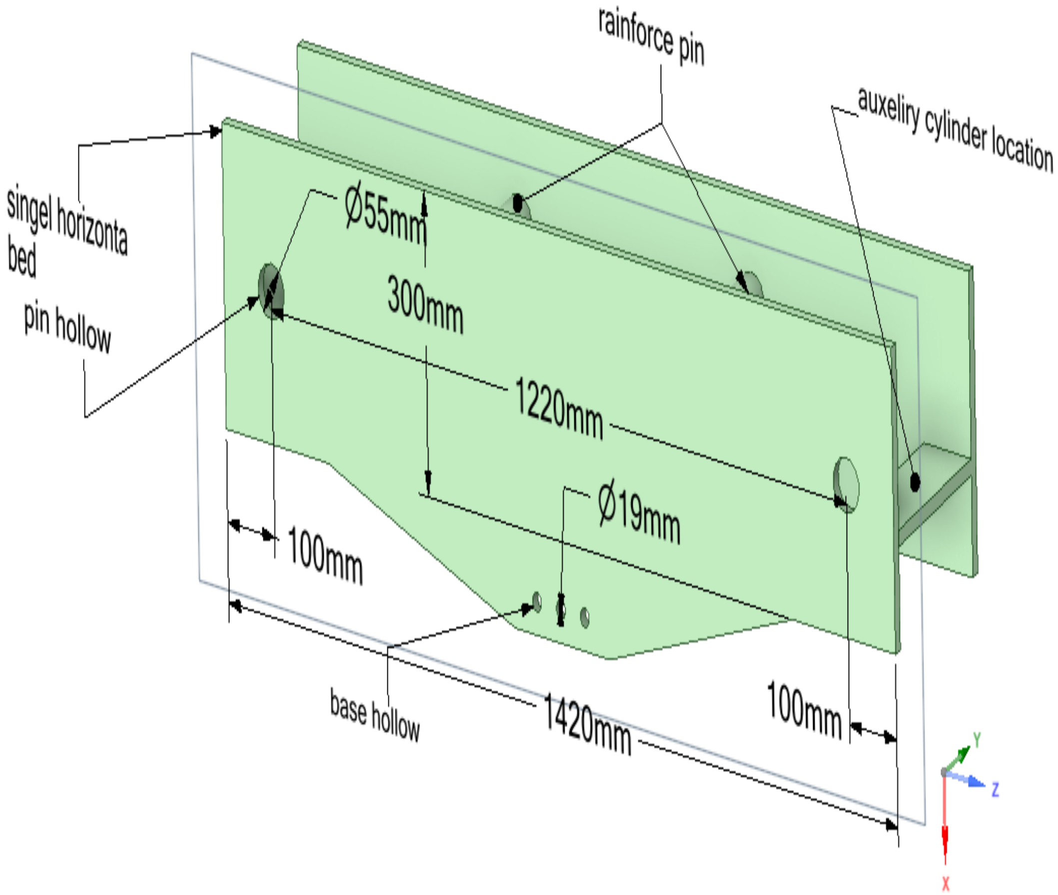


Figure 4. 27.View of 3D model spaceclaim (horizontal bed assembly).

4.1.2.3 Single Horizontal bed (Model-3)

a) Single Horizontal bed (at L= 1220 mm)

Model-3 is created using space claim plat form and & export to ANSYS design modular (at: l =1220mm).Utilizing the space climate platform, the geometry of a single horizontal bed plate at L= 1220 mm can be designed and exported as a file compatible with ANSYS Workbench. Before attaching and importing geometry from Design Modeler, choose the geometry's attributes from a number of options. Options including solid bodies, surface bodies, line bodies, parameters, attributes, named selections, material characteristics, analysis type 2D or 3D, importing coordinate systems, and importing work points are only available in the design modeler's program. The finished model, which makes use of the space climate platform, is clearly displayed in Figure 4.28 and represents a single horizontal bed plate.

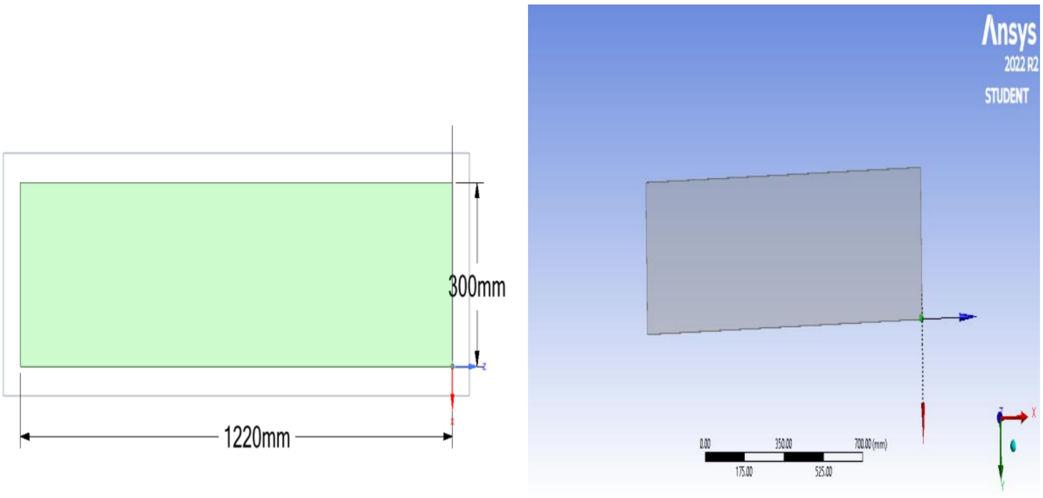


Figure 4. 28.3D model of single horizontal bed plate (At: L =1220mm)

i. Selection of element and define meshing

To enable the smooth distribution of contact stresses, provide a sufficient mesh density on the contact surfaces. Make sure that the mesh density is sufficient to resolve more stresses; regions where stresses or strains are important call for a finer mesh than what is required for resolution of displacements. With 828 nodes and 100 elements applied, the mesh model of the single horizontal bed plate then appears, as seen in Figure 4.29.

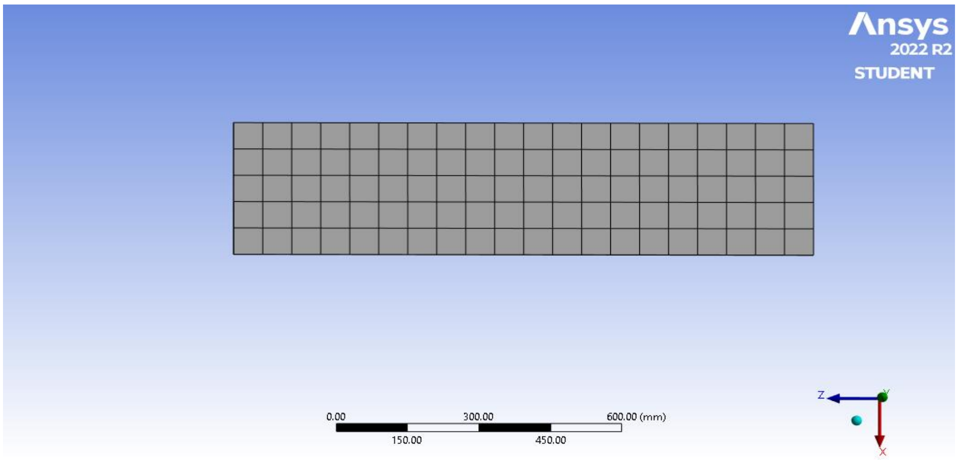


Figure 4. 29.Meshed model of Single horizontal bed plate (at: L =1220mm)

ii. Apply boundary & loading condition

Force constraint: When observing the single horizontal bed plate suspended by the two auxiliary cylinders, both sides of the single horizontal bed plate are connected with a pin, assembled using links of 240 mm pins. The characteristic single horizontal bed plate transfers compressive or tensile stress to vertical columns. The load is applied at the center of the single horizontal bed plate, and it acts as the static motion. So, static stress and total deformation analysis on a single horizontal bed plate with an acting compressive load of 81kN (simple supporting beam) force is applied to the center of a single horizontal bed plate when it is fixed at one end and displaced at the other. The equivalent (Von Misses) stress is a value used to determine if a given material will yield or fracture, as shown in boundary and loading condition Figure 4.30.

Displacement constraint: The single horizontal bed plate is held in place by pins at both ends when we study the real behavior of a single column on the ECMHPM. As a result, the fixed support at one end of a single column only affects the x and Z-axis components of displacement. The y-component's other ends are unattached.

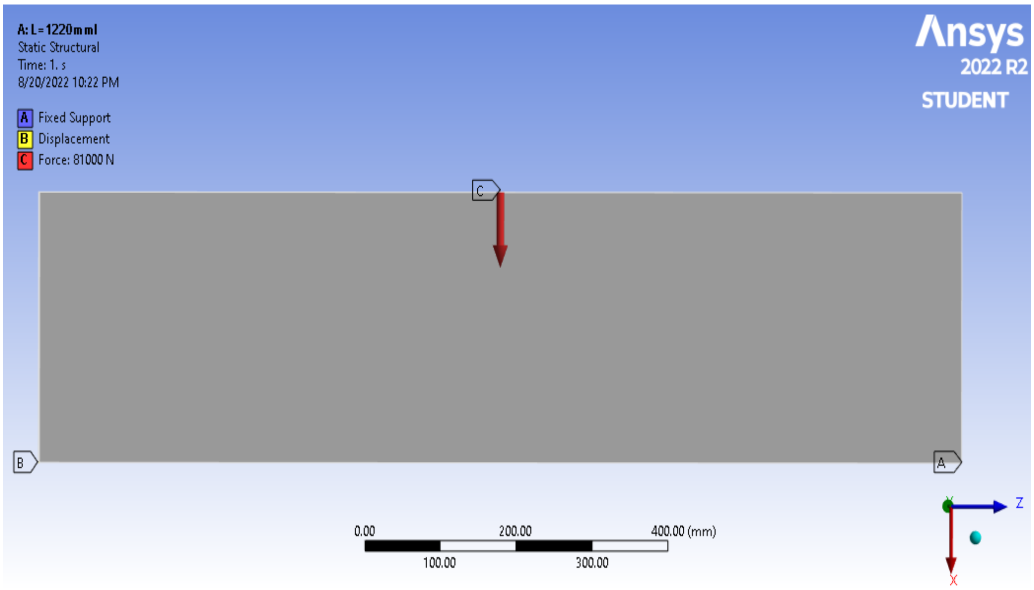


Figure 4. 30.Single horizontal bed plate structure view: at L=1220mm.

iii. Model is analysed using ANSYS software for stresses and deflections

The analysis result of the newly designed model-3 for the Maximum von- Mises stress values is 284MPa and the maximum value of total deformation is given as 0.41251mm. The maximum stress started at the edge side of the single horizontal bed plate and the maximum deformation occurs at the position of the main cylinder; it shows that the red color is for the critical region beside the minimum value of stress and deformation is 0.39727MPa and 0mm, respectively. It indicates the blue color in the safe region. The following is a clear comparison between the two: theoretical analysis and computer-aided analysis are given in Table 4.7.

Table 4.7.Theoretical and Computer aided analysis single horizontal bed plate; at L=1220mm.

Analysis	Theoretical Results	Computer aided analysis
Maximum von- Mises stress	272MPa	284Mpa
Total deflection	0.03546578mm	0.41251mm

Validation: As can be seen from Table 4..7 demonstrates that the theoretical stress of 270MPa and the maximum stress of 284MPa obtained with computer analysis are almost equal and less than the yield strength of the single horizontal bed plate of 340MPa because its material is structural steel. The maximum displacement of 0.425589mm, these results show that the design of the single horizontal bed plate at L = 1220mm is safe and the machine will not fail under operation since the factor of safety (1.7) is greater than 1.

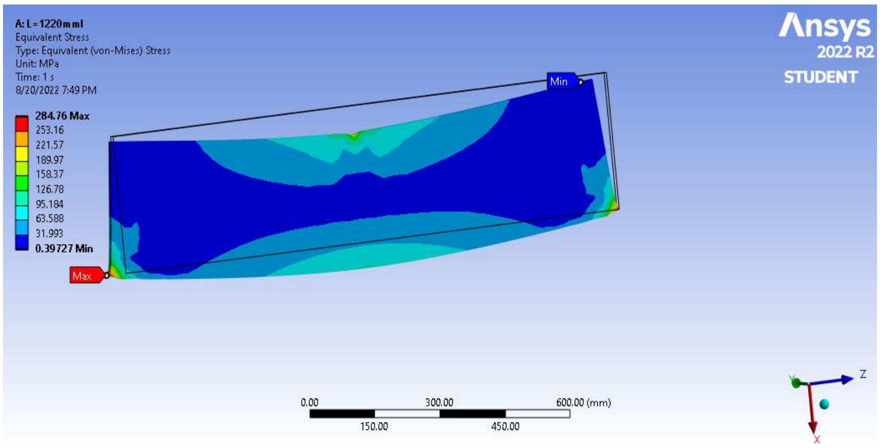


Figure 4. 31.Stress analysis of horizontal bed plate acting Main cylinder: when Pin at one end and roller at the other :(at L=1220mm).

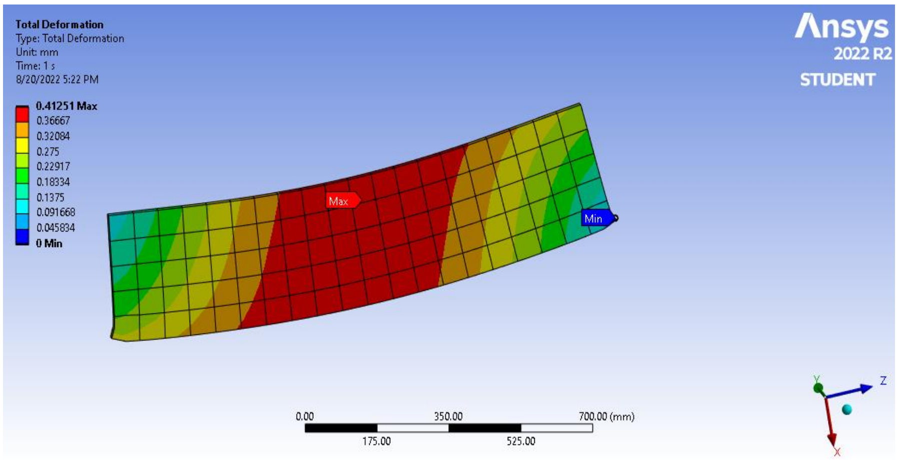


Figure 4. 32.Total deformation analysis of horizontal bed acting Main cylinder: when pin at one end and roller at the other: (at L=1220mm).

b) Single Horizontal bed (at L= 1220 mm, C = 100mm)

Model-3 is created using space claim plat form and & export to ANSYS design modular:(at: L=1220mm, C = 100mm).The geometry of a single horizontal bed plate at L = 1220mm and C = 100mm can be created using the space climate platform and offset using design modular, then produced as a file compatible with ANSYS Workbench. Make your selections for the geometry's attributes before attaching and importing it from Design Modeler. Only the design modeler's application offers options like solid bodies, surface bodies, line bodies, and parameters, attributes, named selections, material characteristics, analysis type 2D or 3D, importing coordinate systems, and importing work points. Figure 4.33 clearly shows the finished model, which is based on the space climate platform and represents a single horizontal bed plate.

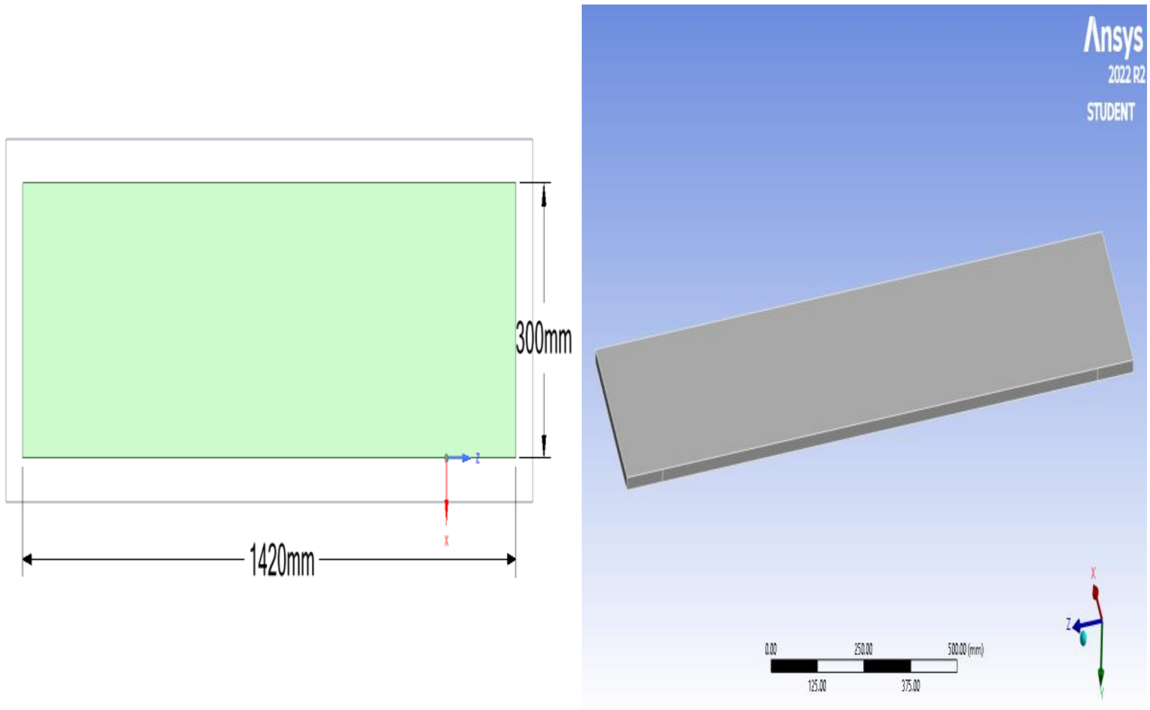


Figure 4. 33.3D model of single horizontal bed plate (at: L=1220mm, C = 100mm)

i. Selection of element and define meshing:

To enable the smooth distribution of contact stresses, provide a sufficient mesh density on the contact surfaces. Additionally, make sure that the mesh density is sufficient to resolve 53 stresses; regions where stresses or strains are important call for a finer mesh than what is required for resolution of displacements. With a mesh element size of 10 mm, 41481 nodes, and 7320 elements, the mesh model of the single horizontal bed plate then appears as shown in Figure 4.34.

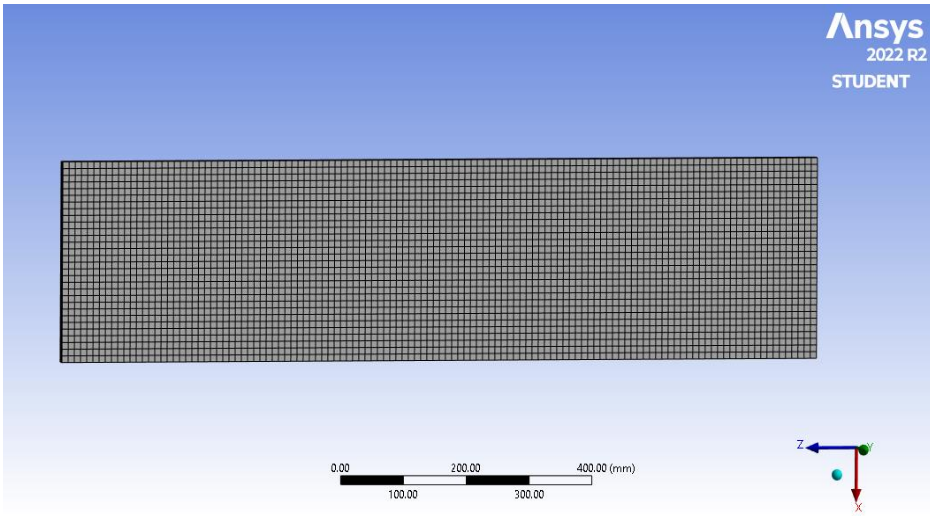


Figure 4. 34.Meshed model of Single horizontal bed plate (at: L=1220mm, C = 100mm)

ii. Boundary & loading conditions

Force constraint: You can see the single horizontal bed plate, which is supported by two Auxiliary cylinders, at L = 1220mm, C = 100mm. A single horizontal bed plate that supports the engine holds the engine block on both sides of the auxiliary cylinders. A single horizontal bed plate transmits compression or tensile stress to vertical columns. The load is imparted to one horizontal bed plate's corner, and it creates a static motion. Consequently, static stress and total deformation analysis on a single horizontal bed plate with an acting compressive load of 10kN is performed (overhanging; simple supporting beam). The amount of force applied to a single horizontal bed plate's corner edge when it is fixed at the right end and displaced at the left end as shown in Figure 4.35.

Displacement constraint: When we study the single horizontal bed plate's real behavior on the ECMHPM. We see that both of its ends are supported by auxiliary cylinders. As a result, the permanent support at one end of a single horizontal bed plate also serves as the x and Z-axis components for displacement. The y-component's other ends are unattached.

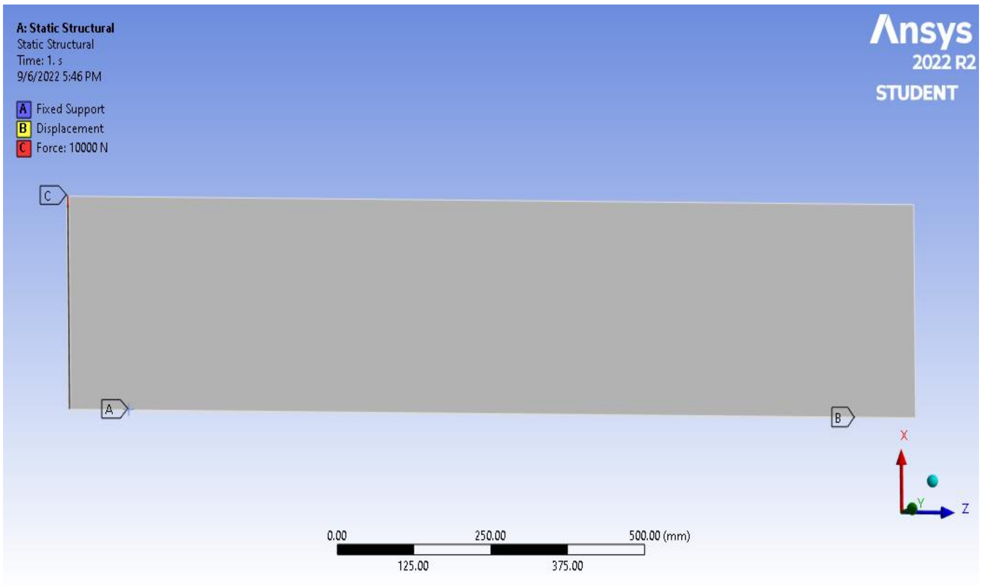


Figure 4. 35.Single horizontal bed plate with boundary condition (at: L = 1220mm & C= 100mm)

iii. model is analyzed using ANSYS software for stresses and deflections

The newly designed (Model-3) analysis result for the maximum value of stress values is 5.5262Mpa, and the maximum value of total deformation is 7.9588 mm. It demonstrates that the blue color at the safe region, as well as the minimum values of stress and deformation, are 0.008029MPa and 0 mm, respectively. Indicates that the red color is for the critical region. The following is a clear comparison between the two: theoretical analysis and computer-aided analysis are given in Table 4.8.

Table 4.8.Theoretical and Computer aided analysis single horizontal bed plate (at: L=1220mm, C = 100mm)

Analysis	Theoretical Results	Computer aided analysis
Maximum von- Mises stress	20.8MPa	18.788Mpa
Total deflection	0.0259mm	0.016839mm

Validation: Table 4.8., which shows that the theoretical stress of 20.8MPa and the maximum stress of 18.788MPa derived from computer analysis are nearly identical and lower than the 340MPa yield strength of the horizontal bed plate since it is made of structural steel, is a good example of this. 0.0259mm is the largest displacement possible. These findings demonstrate the safety of the single horizontal bed plate design at L = 1220 mm and C = 100 mm, and the fact that the machine will not malfunction while in use due to the higher than one safety factor.

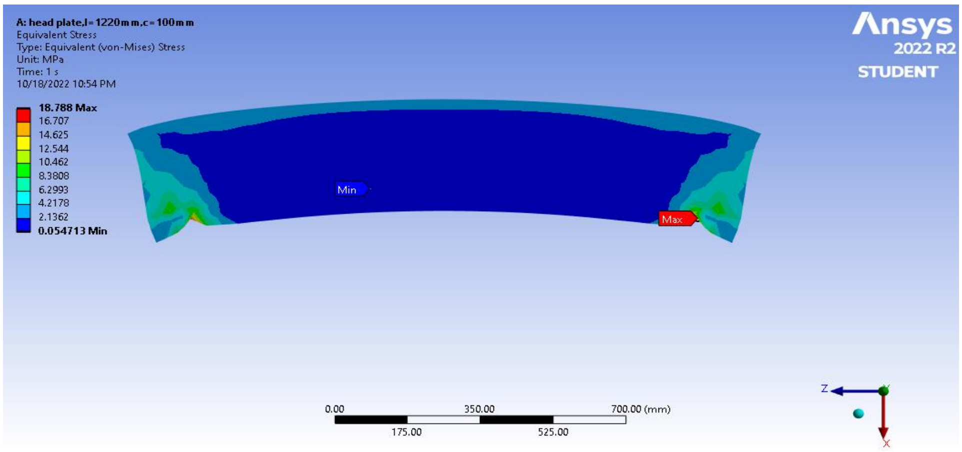


Figure 4. 36.Stress Analysis of Horizontal Bed Acting twin Concentrated Forces (at: L=1220mm, C = 100mm)

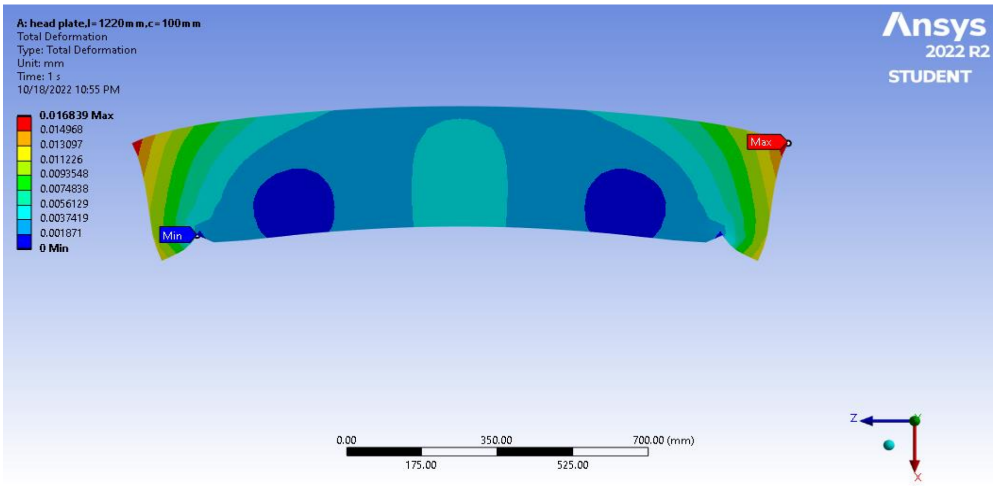


Figure 4. 37.Total deflection on horizontal bed acting with twin concentrated forces (at: L=1220mm, C = 100mm)

c) Single Horizontal bed (at: L=1420mm)

Model is created using space claim plat (at: L=1420mm) .The geometry of a single horizontal bed plate at L = 1420 mm can be created using the space climate platform and produced as a file compatible with ANSYS Workbench. Make your selections for the geometry's attributes before attaching and importing it from Design Modeler. Only the design modeler's application offers options like solid bodies, surface bodies, line bodies, and parameters, attributes, named selections, material characteristics, and analysis type 2D or 3D, importing coordinate systems, and importing work points. This is based on the space climate platform and represents a single horizontal bed plate as shown in Figure 4.38.

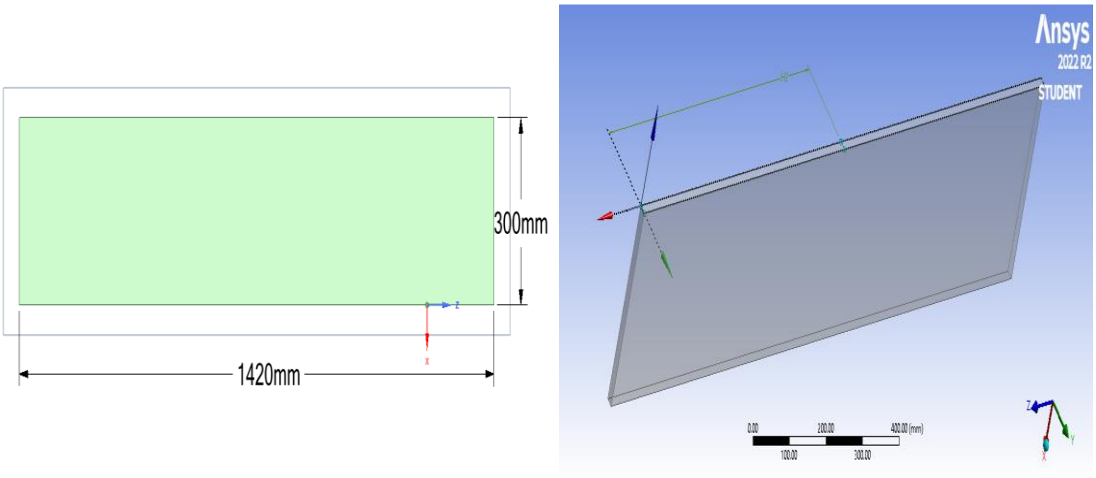


Figure 4. 38.3D model of single horizontal bed plate (at: L=1420mm)

i. Selection of element and define meshing:

To enable the smooth distribution of contact stresses, provide a sufficient mesh density on the contact surfaces. Additionally, make sure that the mesh density is sufficient to resolve 53 stresses; regions where stresses or strains are important call for a finer mesh than what is required for resolution of displacements. With a mesh element size of 10 mm, 41481 nodes, and 7320 elements, the mesh model of the single horizontal bed plate then appears as shown in Figure 4.39.

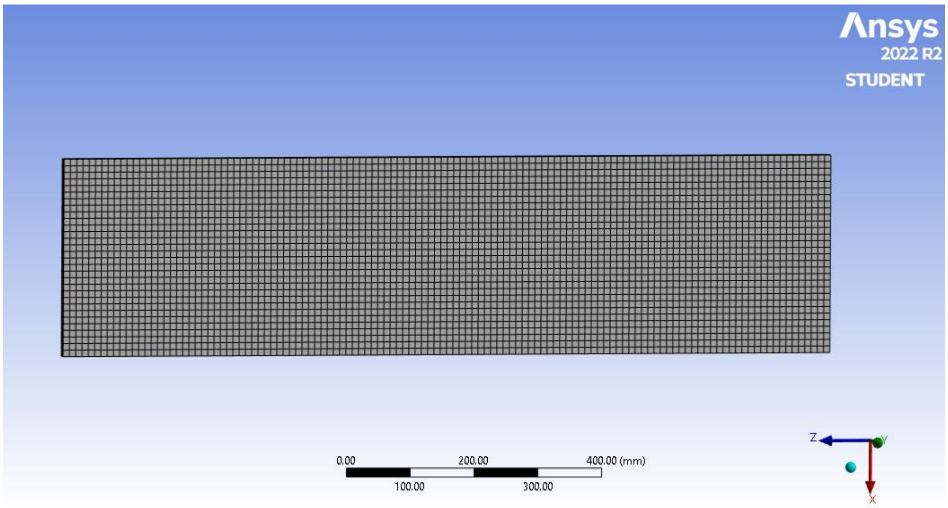


Figure 4. 39.Meshed model of Single horizontal bed plate (at: L=1420mm)

ii. Boundary & loading conditions

Force constraint: When observing the Single horizontal bed plate at L=1420mm suspended by the two auxiliary cylinders. Both side of auxiliary cylinders acting on the Single horizontal bed plate which is lifting heavy wt from ground. Single horizontal bed plate; transfer compressive or tensile stress to vertical columns. the loads is applied at the center of the Single horizontal bed plate and it act as the statics motion.so, statics Analysis of stress and total deformation in a Single horizontal bed plate; acting compressive load of 10kN(simply supporting beam). The force Application at center of Single horizontal bed plate when fixed at right end and displace at the left side.so, Equivalent (Von Misses) stress is a value used to determine if a given material will yield or fracture as shown in Figure 4.40.

Displacement constraint: When we considering the actual behavior of Single horizontal bed plate on the ECMHPM. Both end of Single horizontal bed plate is hold auxiliary cylinders. So, one end of single column is fixed support and in displacement alone x and z-axis components. The other ends of the y- component are free.

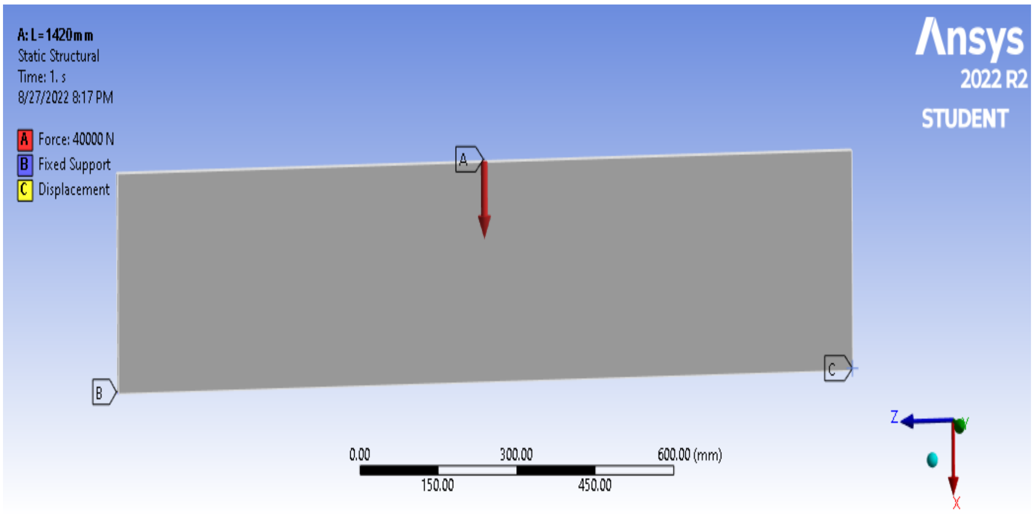


Figure 4. 40.Single horizontal bed plate with boundary condition (at L=1420mm)

iii. model is analysed using ANSYS software for stresses and deflections

The analysis result of the newly designed (model-3) for the maximum value of stress values is 341.02Mpa and the maximum value of total deformation is given as 0.28397mm. It demonstrates that the blue color at the safe region, as well as the minimum values of stress and deformation, are 0.011262Mpa and 0mm, respectively. Indicates that the red color is for the critical region. The following is a clear comparison between the two: theoretical analysis and computer aid analysis are given in Table 4.9.

Table 4.9. Theoretical and Computer aided analysis single horizontal bed plate (at L=1420mm)

Analysis	Theoretical Results	Computer aid analysis
Maximum von- Mises stress	312.5MPa	341.58MPa
Total deflection	0.04142mm	0.28397mm

Validation: Table 4.9., which shows that the theoretical stress of 312.5MPa and the maximum stress of 341.02MPa achieved from computer analysis are nearly equal and less than the structural steel's yield strength of 355MPa shows that. 0.28397 mm is the largest displacement possible. These findings demonstrate the safety of the single horizontal bed plate's design at L = 1420 mm. The machine won't malfunction while it's in use because the safety factor (1.13) is higher than 1.

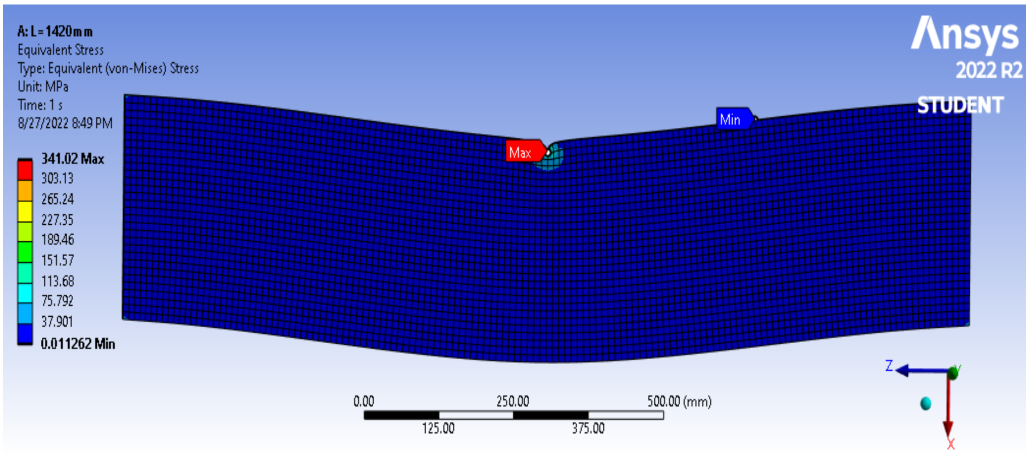


Figure 4. 41.Stress analysis of horizontal bed acting with twin concentrated forces

(at: L=1420mm)

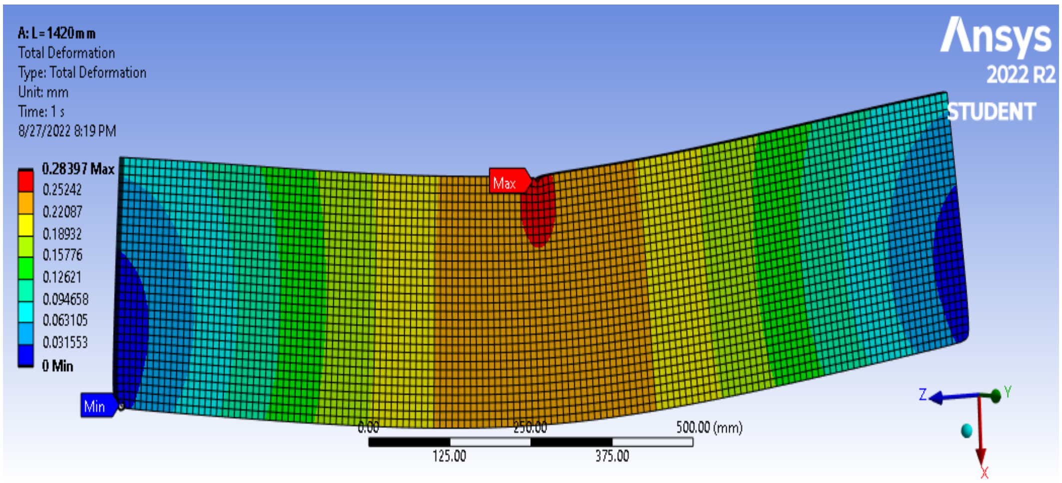


Figure 4. 42.Total deflection on horizontal bed acting twin concentrated forces

(at: L=1420mm)

4.1.3 Fabrication of newly designed machine

Measurement is one of the processes used in the construction of this electrically controlled multipurpose hydraulic press machine. , marking out, cutting, drilling, welding, fastening, grinding, and painting. The following procedure is used to fabricate this multipurpose hydraulic press machine:

i. Safety equipment is used.
ii. I-channel sections with dimensions 240x100x10mm are select. Cutting a 2200mm length with oxygen gas, drilling six 30mm bolt holes, and welding a 220mm x 300mm (16mm) foot plate on the bottom.
iii. Select a 6000x1000x12mm plate and cut off two 1300 x 500 x 12mm (02) head plates using oxygen gas, then drill six 30mm bolt holes.
iv. Cutting the thickness of the 16mm plate cylinder flange to 220mm x 300mm, shaping it using a lathe machine, and drilling six 30mm bolt holes into it.
v. Keep90^0 is a supporter of two head plats. Weld each of them.
vi. Horizontal bed Plates are 6000x1000x16mm (02 pieces) in steel and are connected with 60-mm-diameter, 220mm-long (04 piece) pins. It weighs approximately 100 kg and has a dimension of 1620mm x 300mm.

4.1.4 Assembling procedure of newly designed machine

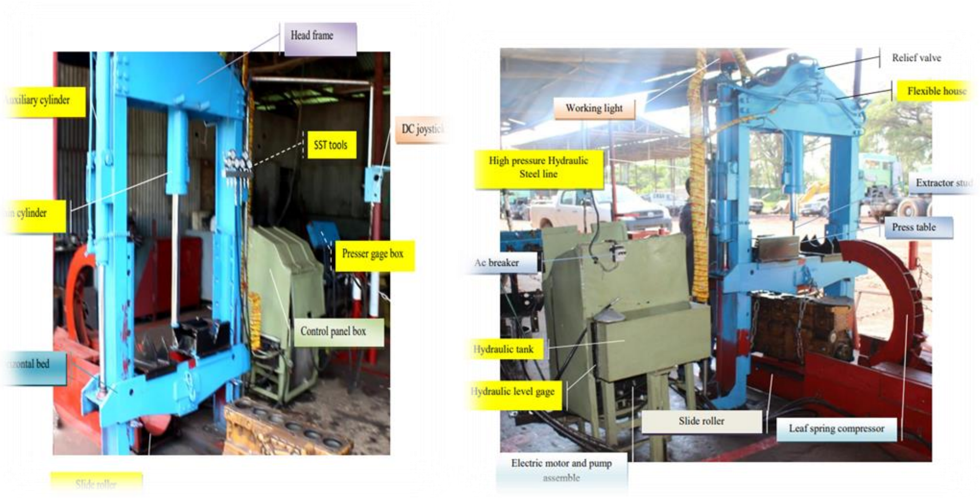
i. Inserting a horizontal bed assemble through two I-channels.
ii. A Tightening head frame assembles with the head of an I-channel using mm32 bolts.
iii. Tightening the main cylinder flange with the head supporter using mm32 bolts.

iv. Tightening the foot plate at the bottom side of the I-column and the ground plate (keep 90^0 positions).

v. Two auxiliary cylinders are tightening on auxiliary support on the I-column and piston rim with the horizontal bed.

vi. Assemble the hydraulic (pipe lines, tank), electrical motor assemble, control panel unit, main switch, joystick.

vii. The Slide Roller (130cm x 45cm) is located at the bottom of the horizontal bed/table. This is a slide back and forth. It slides on a (320cm x 55) length of angle plate fixed with cement concrete on the ground. It is driven by a hydraulic motor at the back of the roller. It shows in actual views of Figure 4.43.



a) Front view b) Rear side view

Figure 4. 43.Assembled of ECMHPM

4.1.5. Testing of hydraulic press machine under different loads

The hydropower system is one component of ECMHPM. The main and auxiliaries cylinders, the relief valve, the dual acting solenoids, the filter electric motor, and the joystick unit are the key parts of this system. The hydraulic oil starts to flow through the open solenoid controlled by the joystick unit when the electrical motor unit turns on, and it then starts to flow outward or inward the piston assembly. However, our machine is in danger because the hydraulic motor's maximum pumping capability is 3050 psi (20.9mpa). To reduce the pressure to 3000 psi, a pressure relief valve was installed through the hydraulic system (20.6mpa). As more unpaid electrical and oil bills are displayed figure

4.44.

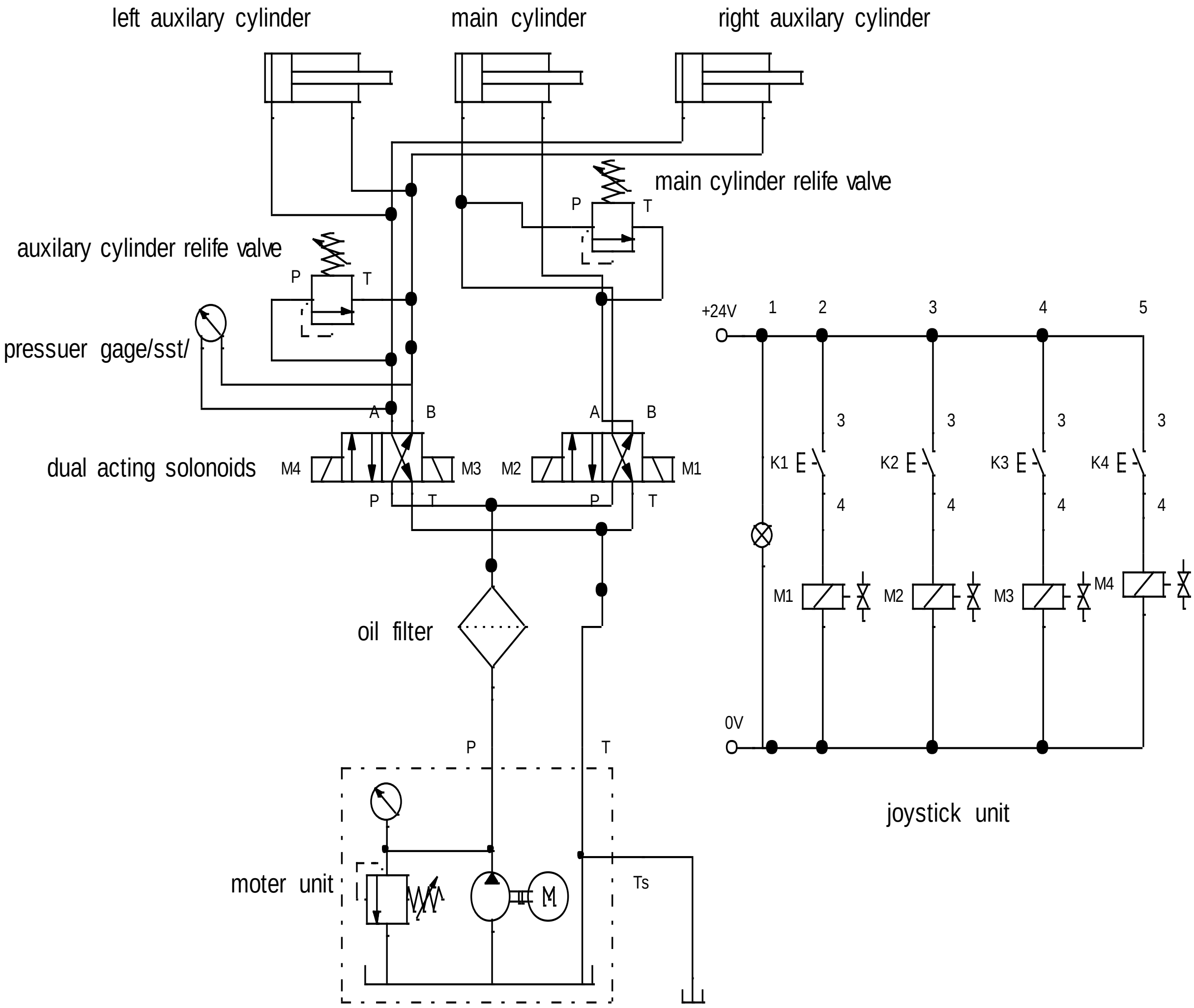


Figure 4. 44.Arrangement of hydroelectric system on ECMHPM

4.1.5.1. Internal and external leakage test

As the names imply, internal and exterior leaks are holes that develop within the hydraulic system. Some of these leaks are purposeful and take the form of tiny openings or confined passageways that let hydraulic fluid flow from one region of high pressure to another in order to cool or clean particular parts. Despite the fact that both testing methods exert their greatest pressure at the cylinder's inlet, the pressure tester (SST) is situated between the hydraulic pump and the power cylinder (attention: never apply pressure until you fix the cylinder). Less is seen below.

Internal and external leakage test: when piston downward stroke

For the leakage test to be effective, make sure that the hydraulic cylinder resistance at the rim side is such that the maximum pressure gage reads 20.6MPa (3050 psi) but is free of resistance so that the pressure reads a minimum pressure of roughly 2.5 PSI. These are done after the air has been released from the system; there is no fluctuation at this time. As a result, the system runs without interruption or pressure decreases. Shown more Figure 4.45.

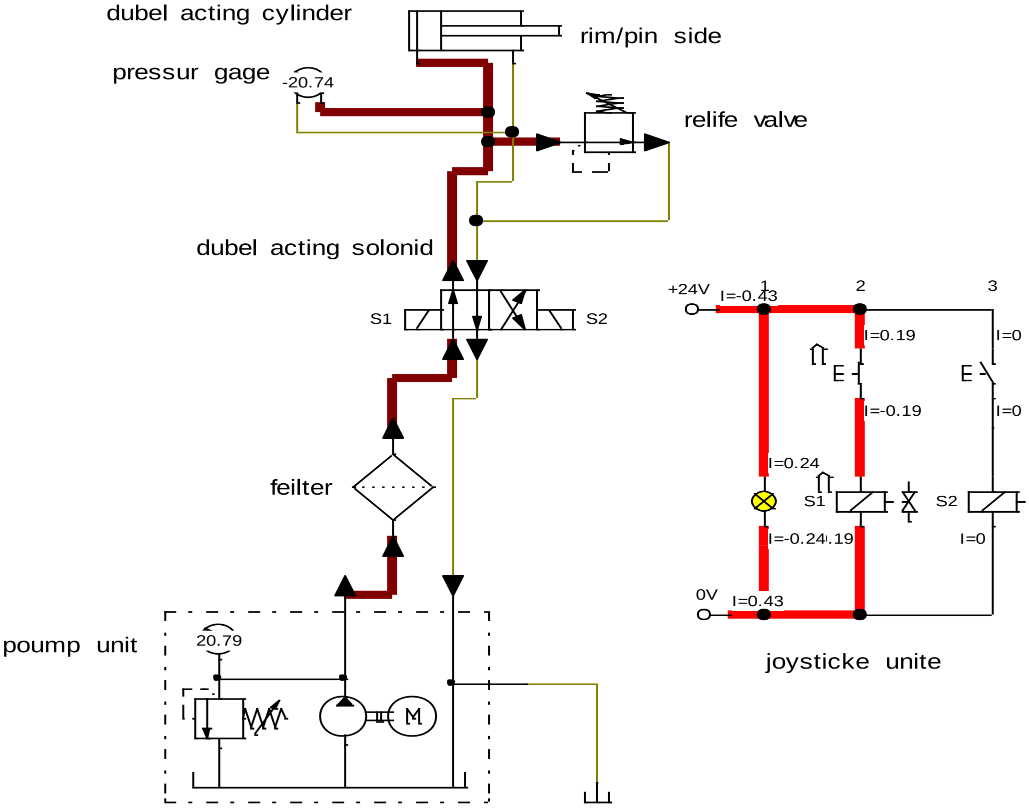


Figure 4. 45.Internal and external leakage tests were performed with load and free applied

Testing: It is seen in Figure 4.45. Hydraulic oil is transferred from the reservoir to the hydraulic cylinder through the electrical solenoids. When the electrical solenoid (s1) starts on, the oil pressure goes to the main cylinder, and on the piston's downward stroke, when it is loaded or free at the front of the rim side, check for both internal and external leakage. The pressure gage's state indicates that if it is dropping from its maximum point, there will be a pressure decrease in the interior or exterior cylinder, conduits, and hydraulic lines. However, our testing showed that there was no pressure reduction when running the test.

Internal and external leakage test: when piston upward stroke

For the leakage test to be effective, make sure that during the hydraulic piston upward (inward)stroke (Figure 4.46), the resistance at the rim side is such that the maximum pressure gage reads 20.6MPa (3050 psi) when it is free of resistance so that the pressure reads a minimum pressure of roughly 2.5 PSI. Because of area variation, force variation will be expected. After the air has been released from the system; there is no fluctuation at this time. As a result, the system runs without interruption or pressure decreases. Shown more Figure 4.46.

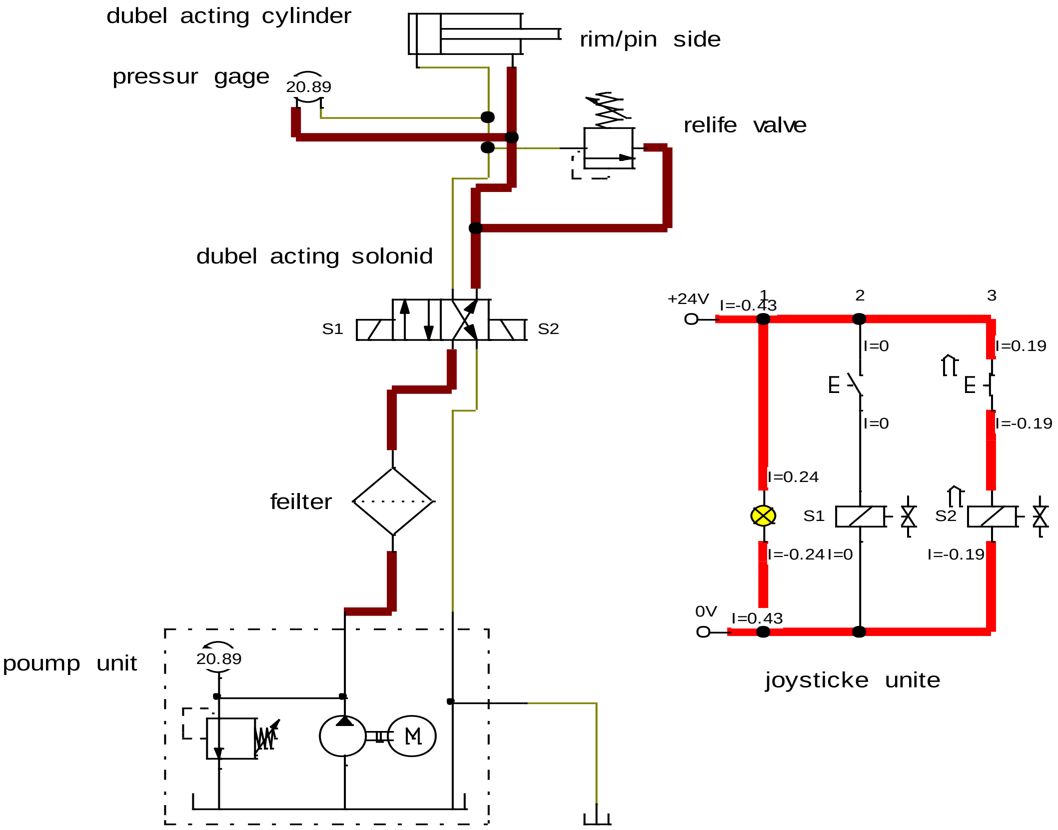


Figure 4. 46.Internal and external leakage tests under load

Testing : As seen in Figure 4.46, when the electrical motor and s2(solenoid) start on, hydraulic oil is delivered from the reservoir to the hydraulic cylinder (rim side) through the hydraulic presser gage, which is fixed or free at the front of the rim side to check the internal leakage. The condition of the presser gage indicates that if it is decreasing from its maximum point, there will be a pressure drop between the piston, cylinder housing, and others. But our testing revealed that there was no pressure drop during the testing.

4.1.5.2. Specific Load test

a. Application of 54kN on torque rod bushing;

When the piston starts to move downward, the head frame and columns assemble in an upward direction. Actual performance is considered when applied pressures of 1000 psi

(6.9MPa) are observed from SST tools during torque rod bushing assembly and disassembly without internal and external leakage of the main cylinder. Fatigue, leakage, cracking, and banding are not observed on all machines, including electrical and hydraulic systems with effective performance. Obviously, the actual view shows Figure 4.47.

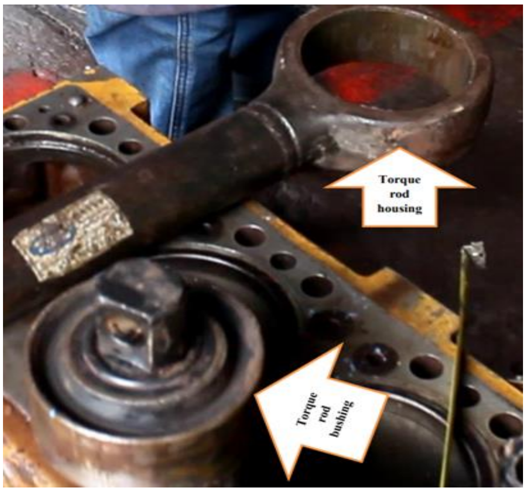


Figure 4. 47.Assembled & disassembled torque rod bushing

b. Application of 11kN engine liners

When the piston starts to move upward, the head frame and columns assemble in a starched downward direction. Actually, performance is considered when applied 250 psi (17kpa) pressures are observed from SST tools during extracting engine liners. However, the cylinder head bolt's maximum tightening pressure is 11kPa and direct intact on wet type engine liners. During performance, fatigue, leakage, cracking, and banding are not observed over all machines, including electrical and hydraulic systems, with effective performance. Obviously, it shows figure 4.48.

Figure 4. 48.Extracting engine liners

4.1.6 Discussion Cost summery

Let's first keep in mind that everything is dissolved material when talking about the machine's cost. As a result, it is preferable to calculate the cost of a machine using the stock balance sheet of the Wonji/Shoa sugar factory's purchase requests. Additionally, Wonji/Shoa sugar mill uses payment papers from daily laborers to focus on labor costs and machinery costs according to market rental providers and accordingly. As a result, the overhead expense will be 3,589EB, and our machine's maximum profit potential is 19,398.70EB. The selling price is actually 200,000EB. But the market value at the time determines this worth.

Figure 4. 49.Cost summery

No	Cost variables	Summery
4.1.6.1	Raw material cost	186,678EB
4.1.6.2	Material wastage 2.5%	131.34 EB
4.1.6.3	Machine cost	1,405EB
4.1.6.4	Labor cost	2,184EB
4.1.6.5	Overhead cost 5%(C+D)	3,589 EB
4.1.6.6	Profit 10%(A+B+C+D+E)	19,398.7EB
4.1.6.7	Sells tax 15%	129,324EB
	Selling price	200,000EB

4.1.7. Comparison b/n existing and newly designed machine

Based on a review of published information on the design and analysis of existing electrical and manual operating hydraulic press machines, the following is a basic comparison between the newly designed machines (ECMHPM).

Figure 4. 50.Comparison b/n newly design and existing electrical and manual Hydraulic press machine

S/N	Newly Designed Machine (ECMHPM)	Existing Electrical And Manual Hydraulic Press Machine
4.1.7.1	Easily detachable	Not detachable.
4.1.7.2	Easily transportable place to place and less man power required	Used for lifting heavy machineries
4.1.7.3	Movable joystick can be used according to operational situation	Most of fixed

4.1.7.4	Easy lift up and lower Horizontal table (bed). It helps to elevate heavy load from ground to proper position using joystick	Effort needed (cable)
4.1.7.5	Used for both pushing and pulling action	Only used for pushing action

4.2. Discussion

This chapter focuses on the results of the Theoretical and Computer aided analysis each models. The Computational analysis presented for applied different loads on; Model one, Model two and Model three. So the Theoretical and Simulations with the help of detected performance of fabrication machine either safe or not. According to R.S. KHURMI J.K. GUPTA, 2005) study of Stress Strain diagrams are used to study the behavior of a material from the point it is loaded until it breaks. Detail Show below figures

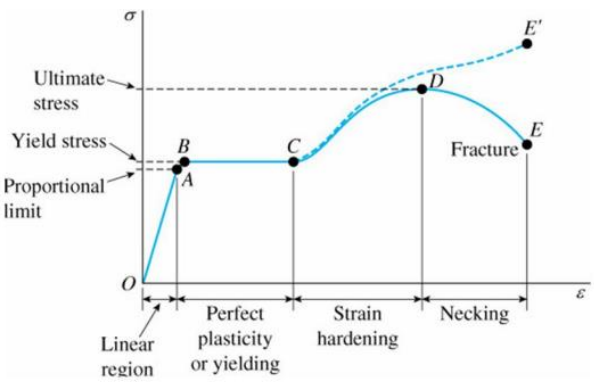


Figure 4. 51.Stress-strain Diagram of structural steel

When we see from the diagram that from point O to A is a straight line, which represents that the stress is proportional to strain. Beyond point A, the curve slightly deviates from the straight line. It is thus obvious, that Hooke's law holds good up to point A and it is known as proportional limit. It is defined as that stress at which the stress-strain curve begins to deviate from the straight line. It may be noted that even if the load is increased beyond point A up to the point B, the material will regain its shape and size when the load is removed. This means that the material has elastic properties up to the point B. This point is known as elastic limit. It is defined as the stress developed in the material without any permanent set. Since the above two limits are very close to each other, therefore, for all practical purposes these are taken to be equal.

If the material is stressed beyond point B, the plastic stage will reach i.e. on the removal of the load; the material will not be able to recover its original size and shape. A little consideration will show that beyond point B, the strain increases at a faster rate with any increase in the stress until the point C is reached. At this point, the material yields before the load and there is an appreciable strain without any increase in stress. In case of mild steel, it will be seen that a small load drops to D, immediately after yielding commences. Hence there are two yield points C and D. The points C and D are called the upper and lower yield points respectively. The stress corresponding to yield point is known as yield point stress. At pt. D, the specimen regains some strength and higher values of stresses are required for higher strains, than those between A and D. The stress (or load) goes on increasing till the point E is reached. The gradual increase in the strain (or length) of the specimen is followed with the uniform reduction of its cross-sectional area. The work done, during stretching the specimen, is transformed largely into heat and the specimen becomes hot. At E, the stress, which attains its maximum value is known as ultimate stress. It is defined as the largest stress obtained by dividing the largest value of the load reached in a test to the original cross-sectional area of the test piece. After the specimen has reached the ultimate stress, a neck is formed, which decreases the cross-sectional area of the specimen, A little consideration will show that the stress (or load) necessary to break away the specimen, is less than the maximum stress. The stress is, therefore, reduced until the specimen breaks away at point F. The stress corresponding to point F is known as breaking stress.

Therefore, according to the above disruption considered that the following result and discussion more brief below. Each result discussed on Theoretical Results versus Computer aided analysis comparison of single Head plate (model-1), single column (model-2), and single horizontal bed: (model-3) are the following Discussion done.

4.2.1 Discussion of Head plate (model-1)

Table 4. 10. Theoretical versus Computer aided analysis comparison of single Head plate

Analysis	Theoretical Results	Computer aided analysis
Maximum von- Mises stress	152Mpa	152.28MPa
Total deflection	0.32530mm	0.35276mm
Bending stress	72.9Mpa	73MPa
total strain	0.00025	
Safety factor	1.9	

Maximum von-Mises stress analysis of the head frame assembles in a starched upward direction as soon as the piston begins to move downward (compression action). Based on 152MPa of Theoretical Results and 152.28MPa of computer-aided analysis. The yield strength of 320Mpa can be withstood by the newly created single head plate. The material can withstand the application of the imposed load because the stress value is below the yield value (proportional limit). The safety aspect is critical in determining whether or not the manufacturing equipment is secure. The value of the safety factor is directly related to yield stress and inversely proportional to allowed stress. So, according to the Theoretical Results at 81kN, the maximum stress induced in the single head plate is 152MPa, with a total deflection of 0.32530mm and a total strain of 0.00025. The maximum stress induced, which is less than the yield stress of the structural steel (320Mpa), means the design is safe. On the other hand, the maximum load is applied. The bending stress value is represented Theoretical Results as 72.9MPa, and computer-aided analyses provide a value of 73MPa (figure A.1). The comparison revealed that the error was nearly within the allowable range but that the greatest bending stress value was situated on the compression side of the head plate (position of application load) and the Theoretical Results calculations of the stresses on the single head plate show that the bending moment is 26325kNmm. The bending stress on the head plate is 72.9MPa less than the stress value (Akshay Vaishnav et al. 2016). Therefore, according to the factor of safety, 1.9 is the ratio of yield stress to the allowable stress. The result of the factor of safety shows that the head plate will not fail and the design is safe.

4.2.2 Discussion of column (model-2)

Table 4. 11. Theoretical versus Computer aided analysis comparison of single column

Analysis	Theoretical Results	Computer aided analysis
von- Mises stress	19MPa	22MPa
Buckling	182kN	188kN
Total deflection	0.21214mm	0.21205mm
Strain	0.000096	
Safety factor	16	

The vertical column stretched upward when the main cylinder applied downward direction the Theoretical Results of 182KN and computer assistance results of 188KN (buckling). When the values are compared, they are nearly identical. Both these show that both analyses are sound; however, buckling occurs on vertical columns because the critical load value is greater than the applied load ($F_{cri} > F_{app}$) when the vertical columns are tending to pull upward (tensile action). Both the computer-aided analysis and the theoretical analysis stress value of 19MPa are almost the same value at much below yield strength (320MPa). The material was indeed able to resist the imposed load because of the safety factors. Even though the greatest stress begins at the bottom of the I-column. The same values were taken into account in all theoretical and computer-aided assessments. So, according to Theoretical Results, the maximum stress induced in the columns is 19MPa at 81kN, the total deflection is 0.21214mm, and the strain value is 0.000096. However, the safety factor is inversely correlated with allowed stress and directly proportionate to the yield stress of structural steel. Since factor safety has a value greater than 1, it is decided that the columns assembled are not easily exposed to buckling.

4.2.3 Discussion of Horizontal bed (model-3)

a. **Horizontal bed: (at L=1220mm)**

Table 4. 12. Theoretical versus Computer aided analysis comparison of single horizontal bed (at: L=1220mm)

Analysis	Theoretical Results	Computer aided analysis
Bending stress	102.9375MPa	102.9375MPa
Maximum von- Mises stress	272MPa	284MPa
Total deflection	0.03546578mm	0.41251mm
total strain	0.000029	
Safety factor	1.176	

Based on 272MPa of Theoretical Results and 284MPa of computer-aided analysis. A yield stress of 340 N/mm^2 can be withstood by the newly created horizontal bed plate. The material can withstand the application of the imposed load because the stress value is below the yield value. However, the safety aspect is crucial to determining if the manufacturing equipment is secure or not. The value of the safety factor is directly related to yield stress and inversely proportional to allowed stress. So, according to Theoretical

Results, the maximum stress induced in the single horizontal bed plate is 272MPa at 162kN and the total deflection is 0.03546578mm and total strain 0.000029. The maximum stress induced, which is less than the yield stress of the structural steel (340Mpa), the factor of safety is 1.176. Finally, the design is safe for a maximum load applied at a length of 1220mm. As a result, the horizontal bed can support a load that is put on it in order to induce bending stress. a single horizontal bed plate's compression side (position of application load)above 150 mm and tensile at the bottom side of 150 mm. Consequently, the Theoretical Results show that the bending stress levels (102MPa) are lower than the allowable stress values (272MPa). So, According to (Akshay Vaishnav et al. 2016), they show that the horizontal bed plate is unquestionably secure.

b. Horizontal bed (at L=1220mm, C=100mm).

Table 4. 13.Theoretical versus Computer aided analysis comparison of single horizontal bed (at L=1220mm, C=100mm)

Analysis	Theoretical Results	Computer aided analysis
Bending stress	4.1666MPa	5.5262MPa
Maximum von- Mises stress	20.8MPa	18.788MPa
Total deflection	0.0259mm	0.016839mm
total strain	0.000018	
Safety factor	16	

The maximum bending stress determined by computer-aided analysis is 5.5262MPa, while the bending stress value in terms of numbers is 4.1666MPa. The comparison revealed that although the application load, where the maximum bending stress value was found, was at the center of the horizontal bed plat, the inaccuracy was practically within the permissible range. According to the Theoretical Results of the 20.8MPa and 18.788MPa computer-aided analyses on machine structures within their acceptable values and the strain value of 0.000018, the bending moment is 1000kNm. At L = 1220mm and C = 100mm, the bending stress is much less than the amount of allowed stress on the horizontal bed plate. So, according to (Akshay Vaishnav et 2016), the horizontal bed plat is safe from fracture. Additionally, the horizontal bed plate yield strength is 345MPa since structural steel was used as the material. Due to the safety factor, the yield stress to permissible stress ratio is 16. So the horizontal bed frame is safe under the given length.

c. Horizontal bed (at L=1420mm)

Table 4. 14.Theoretical versus Numerical comparison of single horizontal bed (at L=1420mm)

Analysis	Theoretical Results	Computer aided analysis
Bending stress	7.3958Mpa	7.3958MPa
Maximum von- Mises stress	312.5Mpa	341.58MPa
Total deflection	0.04142mm	0.28397mm
total strain	0.000003	
Safety factor	1.1	

The compression side of a single horizontal bed plate (position of application load) is where the greatest bending stress values are located, and the lower side of the horizontal bed plate clearly displays the minimum value. Maximum Stress Evaluation The comparison of the Theoretical Results findings for the bending stress value of 7.3958MPa and the computer-aided analysis of 7.3958MPa The horizontal bed's compression side is where the highest bending stress values are positioned (where; above neutral axis y=150mm and the application load is placed), and the lower side of the horizontal bed plate is plainly where the minimum value is indicated (where; below neutral axis y=150mm and tensile position). From analyses, it is conceded that the value of bending stress is less than the maximum von-Mies stresses, which indicates that the horizontal bed plate is safe. In addition, it shows that the safety factor value is inversely related to allowed stress and directly proportional to yield stress (345MPa) and the value of total deflection is 0.04142mm and the strain value is 0.000003. Also, the safety factor is 1.1. It was finally concluded that the horizontal bed plat frame could be at a given length.

CHAPTER FIVE

CONCLUSION AND RECOMMENDATION

5.1 Conclusion

A single head plate is one of the main parts of the head frame assembly. These models are only taken into account for the head frame assembly single head plate, but the head frame assembly takes into account that the piston starts moving downward (compression action) and that the head frame assembly starts moving upward (tensile action). Both circumstances are well understood. But a quicker outcome is anticipated with a single head plate. From both theoretical and computer-aided analyses of the newly designed single-head plate, less value was obtained from yield values. We can therefore conclude that the head frame assembly is constructed of materials that can withstand the main cylinder maximum load in both tensile and compression action. Then factors of safety have values that are greater than one. So we can conclude that the head frame assembled without fracture. On the newly fabricated electrical control multi-purpose hydraulic press machine, there are two vertical columns. The main cylinder's tensile force and compression pushed downward, causing this column to be in downward and upward static motion. Both actions are obviously carried out on the actual machine, but maximal force is supplied to the vertical column at the calculated time because ductile material has the same tensile and compression strengths (R.S. Khurmi & J.K. Gupt, 2005). When we observe the stress value and safety factor acting on the main cylinder, down and upward motions have entirely rendered the vertical motion. Finally, from the analysis, it is evident that while both vertical column assemblies are safe during operational conditions, the horizontal bed plate (at L = 1220mm) is only suitable for a single horizontal bed plate in a horizontal bed assembly. The horizontal bed assembly considers that the piston starts to move downward (compression action). The horizontal bed assembly starts to move in a downward direction. But if a single horizontal bed plate is used, a shorter result will be expected. Less-than-yield values are derived from Theoretical Results and computer-aided analyses of the newly designed single horizontal bed plate. So, we can conclude that horizontal beds (L = 1220 mm) are made of material able to resist the maximum load of the main cylinder for compression action. Factor in that safety is more valuable than limits. So, the horizontal bed is safe from fracture. According to workpics, there were no signs of fracture on the horizontal bed assembly during actual performance. Horizontal bed (L = 1220mm, 100mm) One of the main objects of the horizontal bed assembly is to hold down the engine block during the extraction of the engine liners. But the horizontal bed assembled was suspended by auxiliary cylinders. The auxiliary cylinder forces are greater than the main cylinder forces, unless the engine block starts to lift up from the head of the slider roller. It is difficult to remove the liner from the engine block. Mathematically express the action and reaction forces implied (F auxiliary cylinders > F main cylinder). According to the analysis, a single horizontal bed plate that is safe according to Theoretical and computer-aided analysis means that the horizontal bed assembly is also safe from fracture. Horizontal bed (L=1420mm) One of the main objects of the horizontal bed assembly is to lift heavy workpieces from the ground to a specific position. The horizontal bed was also suspended by the two auxiliary cylinders. According to the analysis, there is a single horizontal bed plate whose stress value is less than its yield value, so during operation the horizontal bed assembly is safe from fracture. Finally, consider cost affordability. A cost study clearly demonstrates that it is more affordable than purchasing outside due to the material's ability to dissolve, the need for less labor and electricity, the absence of pollution, and the addition of an additional application. Finally, an electrically controlled multipurpose hydraulic press machine must be built based on the design and solutions to some problems.

5.2 Recommendation

- According to the analysis, the overall strictures of the electrically controlled multipurpose hydraulic press machine make it preferable to fabricate from materials with homogeneous chemical and mechanical properties, based on the cylinder output. Unless stress distribution is very poor and causes fracture,
- The auxiliary cylinders' total applied load is 20kN in the downward direction on both sides; the aim is only to hold the engine block by using the horizontal bed assembly and to avoid extra movement. So, never apply excessive force to the cylinder block. Unless the damage occurs,
- One of the characteristics of this machine is that it is easily detached because it minimizes time, human energy, and cost. If you require transportation from one location to another, never use a fork lift or other heavy lifting machinery.
- This new technology is very important to our organization. According to this view of conservation, it should be kept and distributed for all under the Sugar Corporation factory to share experience on the fabrication of machines from dissolved materials.

- Finally, ASTU should consider the technology transfer and performance of a machine. Series applies to superior researchers for further study to improve machines. It helps to give training for our organization or others, it helps create a new job for microenterprises ‘’how to manufacture it?’’ and minimize extra foreign currency in our country, and so on.

5.3 Future research work

- future researchers could focus on the effect on:
  - ✓ In this study fatigue bolts between I-columns and plates has got less focus;
  - ✓ Extracting of assembly, fatigue between bolts main cylinder and Extracting bolt throughout operation condition.
  - ✓ Fatigue will be expected due to redundant operation.
  - ✓ Further studies on assemble and disassemble of leaf spring can be conducted.

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APPENDICES

Appendix A. Maximum and minimum bending stress

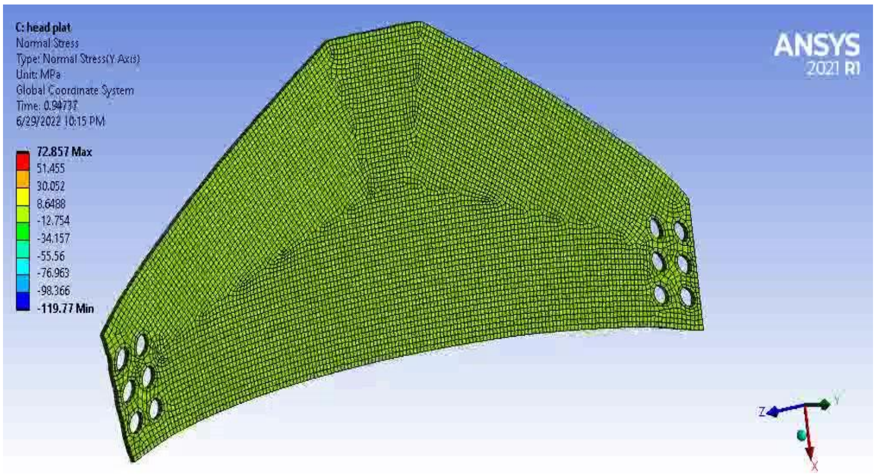


Figure A. 1. Maximum and minimum bending stress of single head plate

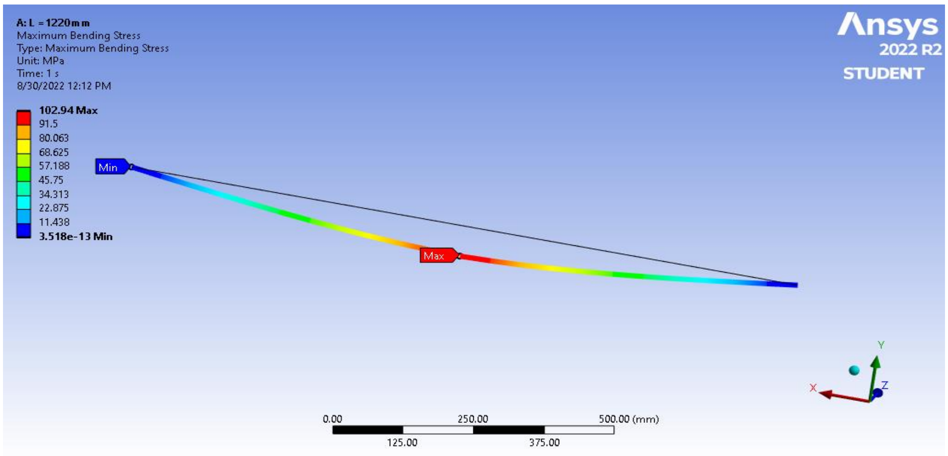


Figure A. 2.Maximum and minimum bending stress of single horizontal bed plate (at L= 1220mm)

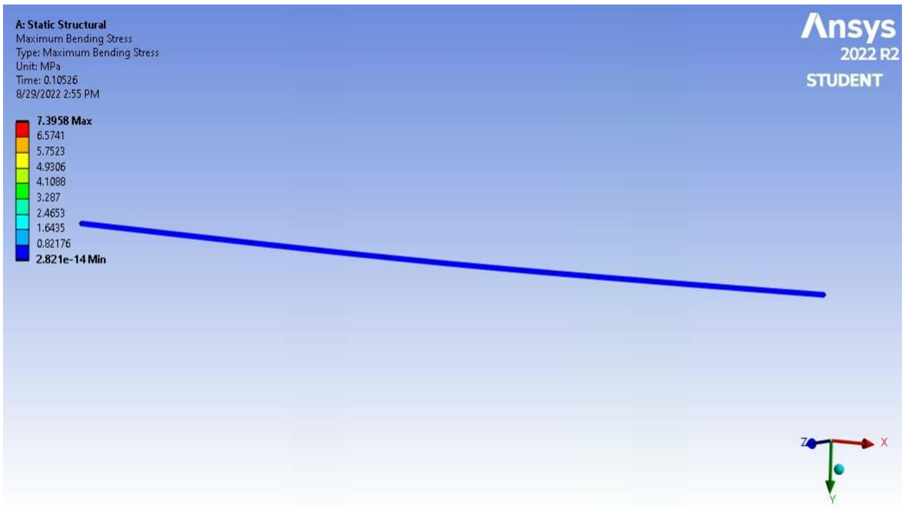


Figure A. 3. Maximum and minimum bending stress of single horizontal bed plate (at L= 1420mm)

Appendix B. Production and operational cost of ECMHPM

Tabel B. 1.Material for Fabrication of ECMHPM

No	Type Material	Standard Size	Unit Price	Used Material	Total Price
1	Plate	6000x1000x12mm	14,850	1300x1000x12mm (02)	6,435
2	Plate	2000x1000x16mm	13,200	1000x1000x16mm (02)	13,200
3	I-profile	6000x240x100x10mm	40,000	2200x240x100x10mm (02)	29,333.30
4	main cylinder	-	50,00	1	50,000
5	Auxiliary cylinder	-	40,000	2	80,000
6	Bolts	M30x49mm	90	3780	3780
7	Cutting disc	178*22*3.2mm	66	2	132
8	Grinding disc	180*22*7mm	78	4	312
9	Calcium carbide	Bottle	17000	10kg	1700
10	Oxygen	Liter/bottle	150	1	150
11	electrodeΦ4.00 mm	7018	158.60	½	70.30
12	electrodeΦ3.25	7018	130	½	65

	mm				
13	Hydraulic oil	15Litt. (10w)	80	15	1,200
14	painting	-	150	2	300
Subtotal					186,678 EB

Tabel B. 2.Machine And Labor Costs

no	Type of machine	Machine Cost/hr.	Working hour	Cost	Labor cost/hrs.	Working hour	Cost	Total price
1	Grinding machine	15	4	60	41	8	328	31.25
2	Lath machine	15	3	45	150	8	1200	490
3	Drilling machine	200	4	800	41	8	328	964
4	welding machine	150	2	300	41	4	164	381
5	Painting machine	100	2	200	41	4	164	282
Sub total				1405 EB			2184EB	2148.25 EB

Tabel B. 3.Cost summery

No	Cost variables	Summery
A	Raw material cost	186,678EB
B	Material wastage 2.5%	131.34 EB
C	Machine cost	1,405EB
D	Labor cost	2184EB
E	Overhead cost 5%(C+D)	3589 EB
F	Profit 10%(A+B+C+D+E)	19398.7EB
G	Sells tax 15%	129,324EB
	Selling price	200,000EB

Tabel B. 4.specification of standard component

Type of material	Qut.	Base thickens	width	height	Diameter	
					Int.	Ex.
Bolts/32M/	36	-	-	29mm	29.5mm	30mm
Center pins /steles steel/	04	-	-	300mm	39.5mm	40mm
I-channel	02	10mm	100mm	2200mm	39.5mm	-
plate Head plate/12mm/	02	12mm	1300mm	500mm	29.5mm	-
Horizontal bed	02	16mm	1620mm	300mm	350mm	-
main piston	01	-	-	800mm	100mm	150mm
auxiliary pistons	02	-	-	1100mm	50mm	90mm
Hydraulics steel pipe	04	15mm	-	1700mm	0.22mm	15mm
Hydraulics rubber hose	10	24mm	-	900mm	0.35	24mm
Side steel pipe	02	15mm	-	940mm	0.22mm	15mm
Type oil (15 litter)		10w				
Maximum out put pressure of hydraulic pump		3050psi				
Direct current control voltage		24v solenoids				
Type of press		H-type/vertical operation/				
Transformer power		Input voltage 220/380v output voltage 25v&27v				

Appendix C. observation of ECMHPM

Figure C. 1.Full Front View of ECMHPM

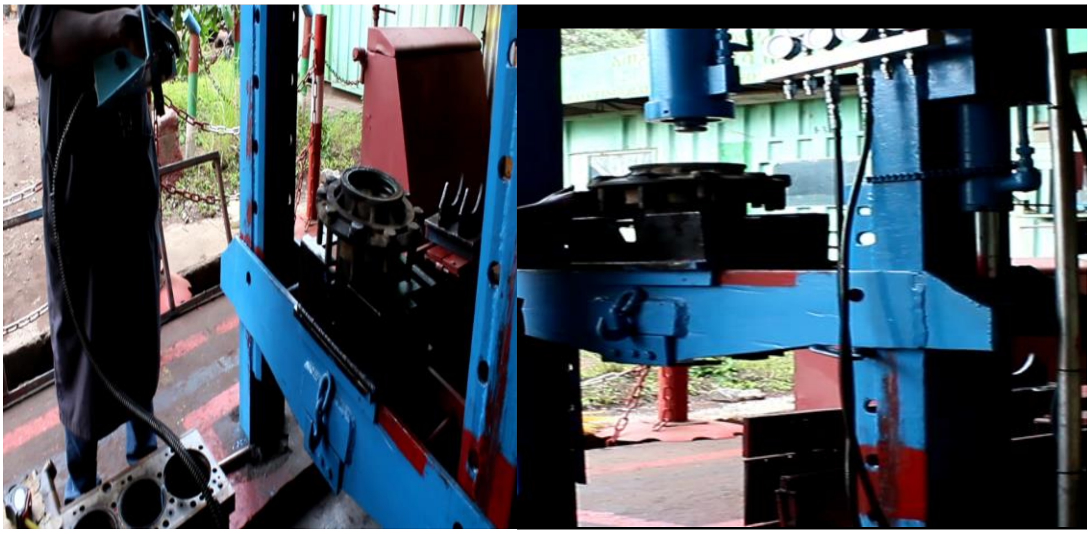
Figure C. 2. rear view of ECMHP

Appendix D. Additional application of ECMHPM

a) (b)

Figure D. 1.show position of joystick: A. supporter B. on hand

Appendix E. Lift up heavy workpice from ground to proper position



(a) (b)

Figure E. 1.Lifts up heavy workpice from ground to proper position; A) at the bottom level B) keeping proper position

Appendix F. Assembling and disassembling of torque rod bushing

a) Assemble and dissemble process of torque rod b) Disassembling of torque rod bushing

Figure F. 1.assembles and disassemble torque rod bushing

Appendix G. Assembling and disassembling liner from engine block

a) Removing liner from engine block

(b) Dismantling from extractor

(C) Extractor Assembling

Figure G. 1.Assembling and disassembling liner from engine block

Sugar Corporation

Business Name

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WONJI/SHOA SUGAR FACTORY

Document No.
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ቁጥር 25/2106/1.1.5

ቀን 09/07/2012 ዓ.ም

<u>ለሚመለከተው ሁሉ</u>

ጉዳዩ ፡- <u>ለአቶ ደመላሽ ጅሩ እየሰሩ ለሚገኘው ሥራ እውቅና ስለመስጠት</u>

ጥር 14 ቀን 2012 ዓ.ም እየሰሩ ከሚገኘው የማሻሻያ ሥራ /Modification/ ጋር ተያይዞ እውቅና እንዲሰጣቸው በደብዳቤ ጠይቀዋል፡፡

በመሆኑም ግለሰቡ ለወንጂ/ሸዋ ስኳር ፋብሪካ ለጥገና ስራ ከፍተኛ አገልግሎት ይሰጣል ተብሎ የታመነበትን እና እንደ ተቋም የተጀመረውን ተቋማዊ የሪፎርም ስራዎችን ወደተሻለ ደረጃ ለማድረስ የበኩሉን አስተዋፅኦ ያበረክታል ተብሎ የታመነበትን ከፍተኛ ስራ ከወዳደዱቁ እና አያሥፈልም ተብለው ተለይተው ከተቀመጡ ቁሳቁሶች በመጠቀም በተሻለ እና እጅግ ዘመናዊ በሆነ መልኩ ለአጠቃቀም ቀላል ፤ የሰው ጉለበትን እና የስራ ሰዓት ብክነትን በመቀነስ በተቀላጠፈ መልኩ ስራን ለመስራት የሚያስችል "ሀይድርሊክ ፕሬስ ማሽን" እየሰሩ መሆኑን በመገንዘብ በቀጣይም ሙሉ ለሙሉ ወደስራ በመግባት ለምርትና ምረታማነት እድገት ከፍተኛ ውጤት ማስመዝገብ የሚያስችል እና ለሌሎች ማንፋክቸሪንግ ኢንዱስትሪ ተሸጋሪ ተሞክሮ እንደሚሆን በማመን ለጅምር ስራው እውቅና የተሰጠው መሆኑን እናሳውቃለን ፡፡

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  <u>ወንጂ</u>

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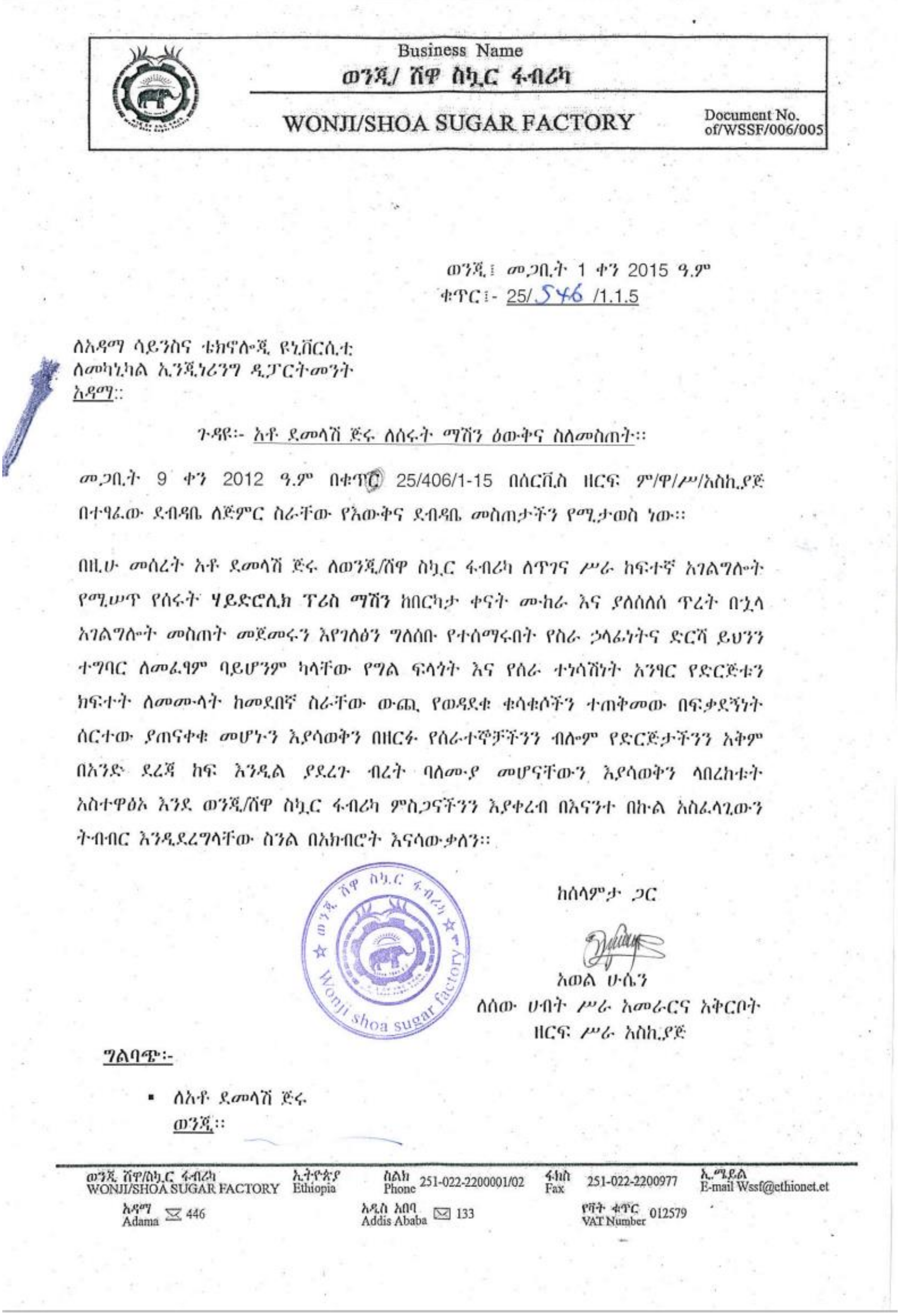

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WONJI/SHOA SUGAR FACTORY
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ወንጂ፤ መጋቢት 1 ቀን 2015 ዓ.ም
ቁጥር፤- 25/546 /1.1.5

ለአዳማ ሳይንስና ቴክኖሎጂ ዩኒቨርሲቲ
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አዳማ::

ጉዳዩ፡- አቶ ደመላሽ ጅሩ ለሰሩት ማሽን ዕውቅና ስለመስጠት፡፡

መጋቢት 9 ቀን 2012 ዓ.ም በቁጥር 25/406/1-15 በሰርቪስ ዘርፍ ም/ዋ/ሥ/አስኪያጅ በተፃፈው ደብዳቤ ለጅምር ስራቸው የእውቅና ደብዳቤ መስጠታችን የሚታወስ ነው፡፡

በዚሁ መሰረት አቶ ደመላሽ ጅሩ ለወንጂ/ሸዋ ስኳር ፋብሪካ ለጥገና ሥራ ከፍተኛ አገልግሎት የሚሠጥ የሰሩት ሃይድሮሊክ ፕሪስ ማሽን ከበርካታ ቀናት ሙከራ እና ያለሰለሰ ጥረት በኂላ አገልግሎት መስጠት መጀመሩን እየገለፅን ግለሰቡ የተሰማሩበት የስራ ኃላፊነትና ድርሻ ይህንን ተግባር ለመፈፀም ባይሆንም ካላቸው የግል ፍላጎት እና የሰራ ተነሳሽነት አንፃር የድርጅቱን ክፍተት ለመሙላት ከመደበኛ ስራቸው ውጪ የወዳደቁ ቁሳቁሶችን ተጠቅመው በፍቃደኝነት ሰርተው ያጠናቀቁ መሆኑን እያሳወቅን በዘርፉ የሰራተኞቻችንን ብሎም የድርጅታችንን አቅም በአንድ ደረጃ ከፍ እንዲል ያደረጉ ብረት ባለሙያ መሆናቸውን እያሳወቅን ላበረከቱት አስተዋፅኦ እንደ ወንጂ/ሸዋ ስኳር ፋብሪካ ምስጋናችንን እያቀረብ በእናንተ በኩል አስፈላጊውን ትብብር እንዲደረግላቸው ስንል በአክብሮት እናሳውቃለን፡፡

ከሰላምታ ጋር

አወል ሁሴን
ለሰው ሀብት ሥራ አመራርና አቅርቦት
ዘርፍ ሥራ አስኪያጅ

ግልባጭ፡-

- ለአቶ ደመላሽ ጅሩ
  ወንጂ::

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Figure G. 2.Recognition letter of wonji/shoa sugar factory