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MODELLING AND ANALYSIS OF EXCAVATOR ARM

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Abstract: The Hydraulic excavator machines are heavy duty earth movers consisting of a boom, arm and bucket. It works on principle of hydraulic fluid with hydraulic cylinder and hydraulic motors. The Hydraulic excavator arm operation require coordinated movement of boom, arm and bucket. The important criteria for the design to be safe is that, the digging forces developed by actuators must be greater than that of the resistive forces offered by the surface to be excavated. The main objective of this paper is to perform design and analysis Excavator Arm for the calculated Force. The CATIA software is used for making the 3D model of the excavator arm linkage. By using ANSYS workbench software analysis of the excavator arm is done at existing digging force and lifting force. Excavator bucket is very crucial element of hydraulic excavator. The whole loads of excavated materials have been carried out by this element. As the present mechanism used in excavator arm is subjected to deformation and bending stresses during lifting and digging operation respectively, because of which failure occurs frequently at the bucket end of the arm.

Keywords: **Excavator arm, Digging force, CATIA, ANSYS 16.0.**

1. INTRODUCTION

Earth moving excavation represents a huge potential and a favourable approach for many earthmoving operations including construction, mining, agricultural, forestry, military applications and especially for cleaning up hazardous areas. Rapidly growing rate of industry of earth moving machines is assured through the high-performance construction machineries with complex mechanism and automation of construction activity [1].

An excavator is an engineering vehicle consisting of a backhoe with cabin for the operator and engine is used for power generation. Hydraulic system is used for operation of the machine while digging or moving the material. Excavators are used primarily to excavate below the natural surface of the ground on which the machine rests and load it into trucks or tractor pulled wagons or onto convey or belts [1].

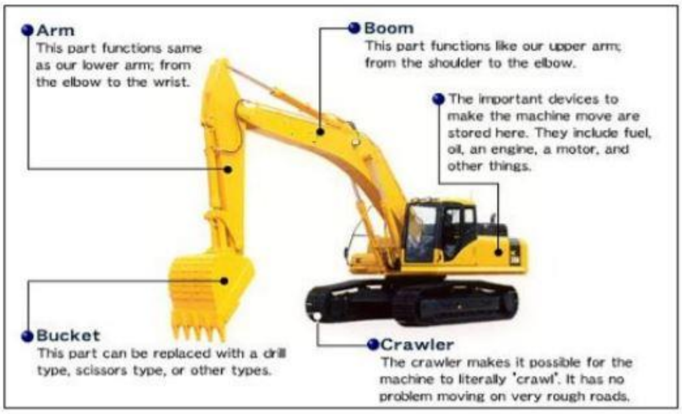

Figure 1 Different parts of hydraulic excavator arm

There are many different operating attachments available. With the options in types, attachments, and sizes of machines, there are differences in appropriate applications and therefore variations in economic advantages. The main components of the hydraulic excavator back hoe are Boom, Arm and Bucket. Excavators are used in many roles as follows:

- Digging of trenches, holes, foundations etc.
- Demolition

• General grading/landscaping

• Heavy lift, e.g. lifting and placing of pipes

• River dredging etc.

As per the varying size of the machine they are called as "mini excavators" or "compact excavators". Often the bucket can be replaced with other tools like a breaker or a grapple. Hydraulic excavators are classified by the digging motion of the hydraulically controlled boom and stick to which the bucket is attached.

2.LITERATURE REVIEW

Rosen Mitrev and et.al in their paper they have studied the dynamic stability of a hydraulic excavator during performing lifting operations. The developed dynamic model with six degrees of freedom considers the base body elastic connection with the terrain, the front digging manipulator links, and the presence of the freely suspended payload swinging. They have found that the excavator overturning stability while following a vertical straight-line trajectory decreases during the motion from the higher to the lower part of the trajectory. If the stability coefficient is close to 1, the payload swinging can cause the separation of a support from the terrain; nevertheless, they have found that the excavator overturning stability can be restored [2].

Bilal Pirmahamad Shaikh and et.al in their paper they taken the maximum digging force condition as the boundary condition and loading condition to carry out static finite element analysis for different excavator bucket tooth. They have found that the Stresses below yield strength obey Hook's law, so deformation in elastic limits. From results it can be seen that stresses are still below safe stress/ allowable stress value so more material can be removed. They have found in the results that the tiger and twin tiger teeth stresses are above safe stresses [3].

B. Govinda Reddy and et.al in their paper they have done the analytical and Ansys results percentage error. The stress at the Tip of teeth of an Excavator bucket is calculated 86.39 MPA and stress due to shearing of rivet is calculated 187.67 MPA by analytically. The stress at the tip of the teeth is calculated 112.98 MPA and stress due to shearing of rivet 157.47 is calculated. Percentage error between analytical result and Ansys result are 13.69 % and 6.72 %. From the above results they have suggested that the bucket used for the excavation purpose should be properly checked for its application on the basis of the soil strata [4].

Bhaveshkumar P. Patel and et.al in their paper they have developed a dynamic model of the backhoe in digging mode using L-E approach. The proposed dynamic model can be used as the basis for automating the digging operation of the backhoe. This can be accomplished by designing the controller so that the entire system can be operated in autonomous mode [5].

Takashi YAMAGUCHI and et.al in their paper they reports on the results of performing experimental measurements of the motion of a hydraulic excavator operated by a human operator and analysing the data obtained by the measurements in order to achieve autonomous control of excavating and loading work by hydraulic excavators based on the skill of experienced operators. the ground materials that are the object of the excavation and loading work by a hydraulic excavator have non-uniform properties, so it is difficult to know the properties in the entire work range before performing the work [6].

Sharanagouda A Biradar and et.al in their paper they have calculated the forces acting on the excavator bucket teeth according to the standard SAE J1179 as 60 KN and also the bucket capacity is calculated according to the standard SAE J296 as 0.75 m3. The stress at the tip of teeth of an excavator bucket is calculated. As per the analysis results, they have suggested that the bucket used for the excavation purpose should be properly checked for its application on the basis of the soil strata [7].

Dharmesh h. Prajapati and et.al in their paper they have concluded that the capacity of bucket has been increased up to 300 kg from 150 kg. they have modified design and increased capacity also by adding two more teeth to full feel the functional requirements. They have checked the design of excavator bucket under different loads. They have increased the volumetric capacity as well as reduce the total deformation of modified bucket [8].

Altaf S. Shaikh and et.al in their paper, the forces on the excavator are calculated and the forces flowing to excavator arm are determined. The analysed part shows there is a scope for optimization. The optimizations of the excavator part are carried out by different iterations and finally the optimized results are obtained. Excavator arm is fabricated and experimentally tested. The FEM results and experimental results are made a comparable study and the validation shows close variance. From comparison of weight of existing model and optimized model it is seen that Overall weight reduction of 5% approximately has been achieved [9].

Sachin B. Bende and et.al in their paper they have modified the Design of the excavator arm and analysis of the design. From the analysis results, they have proved that the design is safe for the calculated digging force. During designing of excavator arm, they have taken the important factors into account they are productivity and fuel consumption. Since, dislocation of the pin at bucket end and the cracking at the adapter end is eliminated by reducing the digging force. But reduction in digging force directly affects the productivity. So, the bucket capacity is increased to compensate for the loss in production due to the reduction in digging force. Also, fuel consumption is less due to the reduction in digging force. Finally, the results of the proposed model are compared with the existing model [10].

A. V. Pradeep and et.al in their paper they have designed excavator bucket and analysis is done for three materials, i.e. steel, wrought iron and cast iron. They have found out the von-mises stresses, deformation and the strain energy for all the three materials. They have made the comparisons among them. Steel and wrought iron has lesser stresses developed when compared to cast iron. Form the results they have concluded that the steel can be replaced with wrought iron [11].

P. Govinda Raju and et.al in their paper, the static structural analysis of the arm and bucket is done and the maximum shear stress and deformation developed in the model is shown. From their study the total weight of the arm is reduced by 50%. The capacity of the bucket is also increased [12].

G. Ramesh and et.al in their paper topology optimization approach is presented to create an innovative design of an excavator Lower Arm. Final comparison in terms of weight and component performance illustrates that structural optimization techniques are effective to produce higher quality products at a lower cost. The Lower Arm has further undergone weight reduction using the material selection through the usage of ALTAIR RADIOSS SOFTWARE. 9.28% of weight is reduced from the base model and it is stiffer [13].

Roshan V. Marode and et.al in their paper, the backhoe-loader bucket have been analysed with the maximum loads and boundary conditions using FEM. Analyses have been carried out for the maximum hydraulic cylinder forces. Symmetrical boundary conditions have been examined along with the fatigue life. The theoretical life cycle of any component in ANSYS is considered as 106 and in the present study the estimated life cycle is 0 for very small region. This can further be improved by changing the shape of the feature. It is also observed that the life of the component reduces considerably as it undergoes fatigue loading [14].

Niteen S. Patil and et.al in their paper all the iterations have been carried out the final iteration shows the better results. Therefore, they have been concluded that it would be a better replacement for the conventional model. After the optimization the total weight reduction of approximately 120 kg is achieved in turn it would increase to the performance of the boom and hence the cost reduction. As the yield strength of the material is 1000MPa, the stresses are within limit and hence the design is safe (3.45 factor of safety) [15].

R M Dhawale and et.al in their paper the mini hydraulic backhoe excavator attachment is developed to perform excavation task for light duty construction work. Based on static force analysis finite element analysis is carried out for individual parts as well as the whole assembly of the backhoe excavator with and without consideration of welding. It is clearly depicted that the stresses produced in the parts of the backhoe excavator attachment are within the safe limit of the material stresses for the case of with and without consideration of welding [16].

Janmit Raj and et.al in their paper the FEA of excavator boom was done in various operating states, simulating actual working conditions in software. The studies shown that mostly higher stress concentration occur at bottom plate of the boom near boom cylinder connecting seat. The forces at each hinge point were calculated mathematically [17].

Sujit Lomate and et.al this paper basically focused on an Analysis and Optimization of Excavator Bucket. The results were supported with an experimental validation for verifying the actual distortion and FEA results. Following are concluding remarks based on the analysis performed on bucket model & Bucket validation at ARAI. Model of Bucket is analysed under 4 different loading conditions to find out the bucket distortion, and bucket distortion is compared with regular bucket. It is observed that the stresses in 1.8 cum design when analysed for 1/3 offset and for full offset are lesser than 1.9 cum Current production bucket [18].

Chinta Ranjeet Kumar and et.al in their paper the main changes in the model are done by adding rectangular ribs, round ribs and half sphere ribs to the inner surface of the bucket and also EN19 Steel material was replaced with AISI1059 Carbon Steel for better results. Static and buckling analysis on the excavator bucket is done. By observing the analysis results, the stress values for half sphere ribs are less than other three models. When, they compare the results for materials, the stress value is less for AISI 1059 Carbon steel and also its weight is less compared with EN19 Steel [19].

Swapnil S. Nishane and et.al in their paper By modelling and analysis of backhoe excavator bucket they have been observed that, the values of von-mises or equivalent stresses for existing and optimized bucket become less difference, but the area of stress in optimized backhoe excavator bucket is reduced as compared to existing one. Also, the value of deformation and stress intensity optimized HORDOX-400 excavator bucket becomes 2.138mm & 201MPa respectively, are less as compared to other materials. The life of existing bucket material is of 22760 min cycles. but by analysing and comparing with different materials, they have been found that the life of optimized HORDOX-400 excavator bucket 66102 min. which is better than existing & optimized – 500 material [20].

Khedkar Y and et.al in their paper Analytical soil-tool interaction models are utilized to calculate resistive forces exerted during digging operations. The digging force is higher than the resistive force so the bucket design is proficient for digging. From the graphs, it's clear that resistive force is increasing as the tool depth below the soil, bucket width and rack angle so it's necessary to select optimum value of bucket width and rack angle while designing bucket. With the static force analysis, we come to know about forces acting at joints of the bucket for each angle of lift and digging [21].

Y Madhu Maheswara Reddy and et.al in their paper by modelling and analysis of backhoe excavator bucket tooth it has been observed that, the values of von-misses or equivalent stresses for existing and optimized bucket become less difference, but the area of stress in optimized backhoe excavator bucket tooth is reduced as compared to existing one [22].

R. Jaison and et.al in their paper a detachable backhoe and loader components are designed to be fitted on a agricultural tractor to lift a load of 2000N and 6000N respectively. This attachment can be removed once its work is completed and the tractor can be used for other purposes like ploughing, carrying loads etc. This backhoe is preferred for trenching and digging in the fields where the trenching process will be carried out often and to carry waste from fields through the loader [23].

Dhanpal N in his paper Analytical soil-tool interaction models are utilized to calculate resistive forces exerted during digging operations. The digging force is higher than the resistive force so the bucket design is proficient for digging. From the graphs, it's clear that resistive force is increasing as the tool depth below the soil, bucket width and rack angle so it's necessary to select optimum value of bucket width and rack angle while designing bucket. With the static force analysis, we come to know about forces acting at joints of the bucket for each angle of lift and dig [24].

P Mahesh Babu and et.al in their paper the digger arm is developed to perform excavation task for light duty construction work. Based on static force and dynamic force loads, finite element analysis is carried out for digger arm. It is clearly depicted that the stresses produced in the component of the digger arm are within the safe limit of the material stresses for the case of static and dynamic load conditions. It is also clearly depicted that the fatigue life cycle of the digger arm is more by 42.6% for modified digger arm compared to original digger arm. Based on results they have conclude that optimization can help to reduce the initial cost of the digger arm as well as to improve the functionality and life cycle as the digger arm operates in worst working conditions. The optimization also helps to avoid frequent failure of digger arm which may cause the entire system become idle and lead to a commercial loss to the owner [25].

Rahul Mishra and et.al in their paper the capacity of bucket has been calculated according to SAEJ296. The bucket specification is the most superior when compared to all other standard model. The breakout force is calculated by SAEJ1179. The SAE provide the breakout and digging force. For max. breakout force condition but for autonomous application it is important to understand. Which are improved bucket geometry for more efficient digging and loading of material. And heavy-duty robust construction for increased strength and durability [26].

J Subba Raju and et.al in their paper Working range in one of the important characteristics of backhoe mechanism. To estimate the working range, a forward kinematical modal and its computer algorithm was developed. Working range computed from computer algorithm was validated with virtual and physical prototype of BEML designed excavator. Results were consistent and proved to be right. This paper emphasizes the significance of structural parameters of backhoe, sequence of design and design validation procedures. This work lays foundation for analyzing the backhoe from stability and digging forces point of view Also, developing a customised tool in MSC Adams, which adapts the concept of mathematical modelling and its computer algorithm, will reduce the design efforts [27].

Zhigui Ren and et.al in their paper, the accurate calculation of the theoretical digging force shows many applications, not only in the optimal design of the excavator and the evaluation of the excavator’s digging performance, but also in trajectory planning and control automation. In TDFCM model the normal resistance and resistance moment are simplified and ignored. Based on the resistance characteristics, the LDF model is established in this paper, simultaneously taking the tangential force, normal force, and the bending moment into consideration. Taking the digging resistance by testing for a 35t hydraulic excavator with backhoe attachment as the standard, this research compares the calculation results of the TDFCM model with those of the LDF model proposed in their paper [28].

Suraj R. Jiddewar and et.al in their paper, in order to eliminate the problem of dislocation due to the play in pin at the bucket end of the excavator arm, the analysis of the problem occurring is done to find out the reasons behind the failure of excavator arm. While performing various operations, excavator arm is subjected to various stresses because of the force acting on it. Due to this all the parts of excavator arm gets deformed and when deformation exceeds a certain limit failure takes place. During working a large amount of vibration is always occurs in an excavator arm assembly and the frequency of this vibration is always changing at different moments during operation [29].

Prof. C. K. Motka and et.al in their paper Stresses are well within the allowable stress for the entire area. There is a stress concentration at the point where the max force is applied, which can be neglected. The total deformation at the bucket teeth point is 5.5292 mm which is negligible. By increasing pin diameter at critical loading points strength of the assembly is increased. By changing the material of the components of backhoe and by trial and error method in ANSYS results are obtained [30].

Amol B. Bhosale and et.al in their paper the work is carried out on the boom component of the hydraulic excavator. The linear static analysis is done to find out the linear static characteristics of the boom component. From those static characteristics, the structural weight optimization is carried out. Then boundary conditions are applied on existing model and on 4 cases considered for weight optimization using varying thickness of boom plates. The maximum value of stress is considered in each case and weight optimization is done with factor of safety as 1.5. Structural weight optimization gave the total weight reduction of 715 kg (36.4%). Comparison in results of the Von Mises stresses obtained by numerically and analytically is very less and total variation in result is of only 1.12% which shows that the result of structural weight optimization performed numerically is accurate and acceptable [31].

Ahmet Erklig and et.al The backhoe-loader back and front arm have been analysed with the maximum loads and boundary conditions using FEM. Analyses have been carried out for the maximum hydraulic cylinder forces. Symmetrical and unsymmetrical boundary conditions have been examined. With respect to analyses results, the backhoe-loader arms need an improvement to increase its strength. Two different improvements have been performed for arms. After improvements, safety factor is increased to 1.98 from 1.59 for back arm. Strength of the back arm has been increased by 24.5%. For front arm, safety factor has been increased to 2.18 from 1.94 at the symmetrical loading while loader cylinder is active [32].

Bhaveshkumar P. Patel and et.al in their paper The FEA and optimization is versatile tool for designing the backhoe attachment in hydraulic excavator. By conducting FEA it is very easy to identify weak components through strength analysis of excavator attachment and corrections are possible in early stage of design. Topology optimization may give better results by changing the initial topology. Genetic algorithm (GA) and neural network is a powerful tool for optimization. Better lighter and cheaper designs can be obtained by using Finite Element Method and optimization techniques [33].

Y Madhu Maheswara Reddy and et.al in their paper by modelling and analysis of backhoe excavator bucket tooth it has been observed that, although, the values of von-misses or equivalent stresses for existing and optimized bucket become less difference. the value of deformation and stress intensity optimized by using material HSS, therefore the values of the HSS is become 5.9645x10-6 M and 23.58MPain static structural analysis, 6.36x10-6 M and48.0 MPa in couple field analysis respectively, are less as compared to other materials. Failure of excavator bucket tooth is due to abrasive wear and impact loading [34].

Qingying Qiu and et.al in their paper A new design parameter setting method based on the combination of the lengths and angles between hinge joints is successfully applied to the description of the hydraulic excavator working device, which can improve the effectiveness of sample points. It is impractical for all of the design variables to be merely expressed by length in conventional methods because so many conflicts would occur after sampling. Compared with the original methods, this method takes full advantage of basic trigonometric functions, ensuring that all sample points are valid. Multiple surrogate models are then adapted to comparatively study the optimal design of the hydraulic excavator working device [35].

Sumar Hadi and et.al in their paper design and analyse a Trapezoidal bucket excavator by using ANSYS R15.0. Maximum strain that display in a fixed position of top the bucket. Maximum deformation will occur at the end of the tooth to the entire body. They recommended that buckets used for excavation purposes should be checked properly for their application based on soil. Also geometry is one of the parameters and effects of deformation during the lifetime a bucket [36].

Vishwajeet A. Patil and et.al in their paper the excavator bucket is developed to perform excavation task for light duty construction work. The bucket and arm digging forces are found out by calculations. By using different material properties and based on static force loads, finite element analysis is carried out for excavator bucket. Using stress values, the fatigue life is carried out which gives the cycle time life converted to hrs. Also, but using online e-fatigue calculator results are validated. By using the result, the stress points are carried out and the optimized bucket model is created [37].

Kishore Krishna M and et.al in their paper the offset boom attachment is developed to perform excavation task for light duty construction work like trenches and pipe laying work. Based on static force analysis finite element analysis is carried out for individual parts. The analysis results indicate that the stresses produced in the parts of the attachment are very less equal to limiting (safe) stress of the parts material. The total deformation is also found to be negligible when compared to thickness of the attachment part. In future, there is a scope to perform the structural optimization of the boom attachment for weight reduction. Optimization can help to reduce the initial cost of the attachment as well as to improve the functionality in context of controlling of the excavation operation. Using a swing set cylinder with trunnion mounting can be used [38].

Anthony Kpegele Le-ol and et.al in their paper the backhoe arm of the backhoe loader has been analysed using FEM and the results obtained showed that the volume capacity of the bucket can be improved while decreasing the breaking force of the bucket and curling force of the arm. It has been further shown that this increase in the bucket capacity leads to an increase in the number of teeth of the backhoe as opposed to the number used in conventional backhoe loaders. It proves that the breaking force used by conventional backhoe loaders by the backhoe arms can be reduced greatly without sacrificing the functionality of the equipment and this leads to power and cost conservation. This implies that a decrease in the breaking force does not lead to increased deformation of the bucket [39].

Fahim Mahmud Khan and et.al in their paper the modification of the arm is possible from different perspectives as seen in this paper. The thickness of different components of the arm can play a vital role in the modification which has been discussed in details. The results found from the displacement and von-misses stress using different materials have been used to decide whether the designs are safe or not [40].

Manisha P. Tupkar and et.al in their paper the stress at the Tip of teeth of an Excavator bucket is calculated 96. 39 MPA and stress due to shearing of rivet is calculated 157.67 MPA by analytically. The stress at the tip of the teeth is calculated 112.98 MPA and stress due to shearing of rivet 167.42 is calculated. Percentage error between analytical result and Ansys result are 14.69 % and 5.82 %. They suggested that the bucket used for the excavation purpose should be properly checked for its application on the basis of the soil strata [41].

Alexander Gurko and et.al in their paper they investigate a new controller to do digging trajectory tracking for a robotic excavator. The controller requires two circuits: the first circuit calculates the main control using the CTC, and the aim of the second one is to provide an additional control to compensate effect of uncertain factors on the basis of differential games with quadratic cost. The mathematical tool of R-functions as the alternative of the linear matrix inequality approach to constructing information sets of the excavator arm state is used. The practical value of the proposed controller is in providing an upper bound on a given performance index at any uncertainties from the given bounded set, as well as in requiring a relatively low computational capability compared to other reviewed methods [42].

Hemanth Kumar BL and et.al in their paper The Deformation and Stress plot for the Backhoe Boom are obtained from the analysis results. The stresses are less than the fatigue strength of the material. The design of Backhoe Boom assembly is safe [43].

K.Guna Sekhar and et.al in their paper The stress at the Tip of teeth of an Excavator bucket is calculated 96.39 MPA and stress due to shearing of rivet is calculated 157.67 MPA by analytically. The stress at the tip of the teeth is calculated 112.98 MPA and stress due to shearing of rivet 167.42 is calculated. Percentage error between analytical result and Ansys result are 14.69 % and 5.82 %. As per the above analysis, they suggested that the bucket used for the excavation purpose should be properly checked for its application on the basis of the soil strata. And considering the failure of the tooth and rivet due the impact loading [44].

Bikash Rai and et.al in their paper they show that by determining various reaction forces a rotary joint can be designed for the excavator arm, which facilitate the rotation of the arm and increase the productivity. This is very important to analyse all the forces during designing process, selection of material, power of motor. The excavation could be carried out in different position of the bucket [45].

Bhaveshkumar Patel and et.al in their paper the capacity of the bucket has been calculated according to the standard SAEJ296 and comes out to be 0.028m$^{3.}$ this bucket specification is the most superior when compared to all the other standard mini hydraulic excavator madels. The breakout force calculation is done by standard SAE J1179 and comes to be 7626 N. The static force analysis performed by considering the maximum breakout force configuration and can be used as the boundary conditions [46].

Adrien Michel and et.al in their paper they looked at the kinematic manipulability of a mechanical excavator used for construction purposes. The movement of the excavator arms were modelled as a 3DOF manipulator and the corresponding DH and screw parameters were obtained. A simplified model of the excavator was simulated using MATLAB and the kinematic manipulability ellipses were determined. A parametric study was conducted to study the effects of joint angles and length ratios between arms and concluded that joint 2 had the largest effect on the manipulability [47].

S. Zarotti and et.al in their paper A method to outline energy use characteristic of hydraulic excavators is described an applied to a standard trench digging cycle. comparing energy use characteristics of the same machine doing the standard cycle without actually moving earth (simulated digging) and moving earth (actual digging), external load influences are defined, and compatibility with fuel consumption evidences is shown [48].

J. L.R. Manoj Kumara and et.al in their paper the design of a barge which is capable of operating along narrow canals. The barge was designed to withstand the dead weight of barge and the machine engaged for dredging the canal. The draught of the barge was 557 mm which is an acceptable amount of draught for the canals to be dredging by SLLRDC. Small excavators which have similar specifications as our Doosan 35Z excavator are being produced by all of the major construction equipment manufacturing companies [49].

Masayuki Kagoshima and et.al in their paper The SK80H, developed as an 8-tonne class hydraulic excavator, has been introduced as an example of volume production machinery designed as energy-saving construction equipment. The hybrid system has enabled the downsizing of the engine, improved the fuel efficiency by 40% and achieved a significant noise reduction. We will continue to strive to further improve our product on the basis of field information, such as actual operational data, and also to reduce the cost and increase productivity, while maintaining the above-described fuel efficiency performance [50].

Nimisha Raj and et.al in their paper this manual earth digger machine reduces cumbersome digging operation and improves operator level comfort. It will give a new dimension of comfort add to the operator to work in garbage handling, constructional work levelling pile of soil hose gardening. In future we work on paddle hydraulic arm for increasing and decreasing, digging force and man efforts [51].

3. PROBLEM STATEMENT AND OBJECTIVE

3.1 Problem statement

Backhoe Loader is versatile machine and able to operate in different conditions. It is used in different excavation operation like trench digging, laying pipes, construction etc. nowadays buildings are built very closely, for the heavy-duty excavator it is difficult for them to operate in this condition. We have developed with a minimum dimension of the parts for the construction.

3.2 Objective

The objective of the paper is to Generate the required dimensions of the parts and analyses the excavator arm. The Analysis is done using ANSYS software by loading the max force to the excavator arm by static analysis and modal analysis to find the natural frequencies of the excavator arm.

4. METHODOLOGY

The Generation of the 3D model consists of mathematical and computer modelling. We have gone through the mathematical calculations and obtained the geometrical parameters for the 3D model of an excavator parts. Once the design calculations are carried out the final dimensions are fixed and Modelling of the excavator arm is carried out using modelling software CATIA.
After the completion of parts modelling parts are subjected to Structural analysis and modal analysis using Finite Element packages like ANSYS.

5. DESIGN PROCEDURE FOR EXCAVATOR ARM

5.1 Base Calculations for stick and boom:

The ultimate strength of medium strength alloy steel (ASTM – A 514) = 690 N/mm^2

Factor of safety = 5

According to Roymeck, UK standards "FOS=5", should also be used with better-known materials that are to be used in uncertain environments or subjected to uncertain stresses". As the backhoe experiences uncertain stresses by the angle of digging suppose if the bucket angle is around 45^0, the digging force experienced by the bucket will be less similarly if the bucket angle moves away from 45^0, the force required will be more it depends on the skill of the operator.

Table1 Properties of medium strength alloy steel

S.no	Property name	Value
1	Ultimate strength	690Mpa
2	Yield strength	450Mpa
3	Poisons ratio	0.29
4	Density	7.85g/cm^3
5	Tensile strength	850Mpa
6	Shear modulus	80Mpa
7	Bulk modulus	140Mpa

Therefore, allowable yield strength = 690/5 = 138 N/mm^2.

5.1.1 Breakout force calculation

In general terms it is the amount of force that the tip of the bucket teeth or leading edge of the cutting on the bucket can exert against the material trying to load. Maximum load is taken as 2500N.

Maximum lifting capacity = Breakout force/ (Factor of safety-1)

Breakout force = 10000N.

Assumptions made in calculation

- The boom cross section is idealized to channel section,175*133, plate thickness=6mm.
- The stick cross section is idealized to box cross section,142*117, plate thickness=6mm
- The maximum bending stress does not exceed the allowable yield strength 138N/mm^2.

Thus, in bending equation instead of the bending stress the allowable yield strength value is substituted.

5.1.2 Finding the stick length using bending equation

Where M = bending moment in N-mm^2, I = moment of inertia in mm^4, σ = bending stress in N/mm^2 and y = distance from neutral axis to outer surface in mm.

In the bending equation instead of the bending stress the allowable yield strength value is substituted as per the assumption made as the maximum bending stress will be allowed to reach the value of 230 N/mm^2.

$$\frac{M}{I} = \frac{\sigma}{y}$$

$$\frac{M}{0.869X10^2} = \frac{138}{71}$$

M = 16890422.5 N-mm.

Moment = force x distance.

Were, force is taken as the breaking force as calculated which is 10000N. Therefore, the link length is:

stick length = Moment / Breaking force

= 16890422.5 / 10000

= 1700mm

5.1.3 finding the boom length using bending moment equation

Where M= bending moment in N-mm, I = moment of inertia in mm^4, σ = bending stress in N/mm^2 and y = distance between the

$$\frac{M}{I} = \frac{\sigma}{y}$$

neutral axis to the outer surface in mm.

In the bending equation instead of the bending stress the allowable yield strength value is substituted as per the assumption made as the maximum bending stress will be allowed to reach the value of 138 N/mm^2.

$$\frac{M}{1.356X110^7} = \frac{230}{87.5}$$

M = 21386057 N-mm

Moment = force x distance

Were, the force is taken as the bucket curl or breaking force as calculated which is 10000N. Therefore, the link length is:

Boom length = Moment / Breaking force

= 21386057 / 10000

= 2140mm

5.1.4 Finding the pin diameter using bending equation

Where M = bending moment in N-mm^2, I = moment of inertia in mm^4, σ = bending stress in N/mm^2 and y = distance between the

$$\frac{M}{I} = \frac{\sigma}{y}$$

neutral axis to the outer surface in mm.

F.O.S = 2.5

σ = 690/2.5

= 276 N/mm^2

$$d^3 = \frac{1000*90*64}{|\pi*276*2}$$

by solving cubic equation, the diameter of the pin.

d = 32 mm.

5.2 Bucket capacity calculation

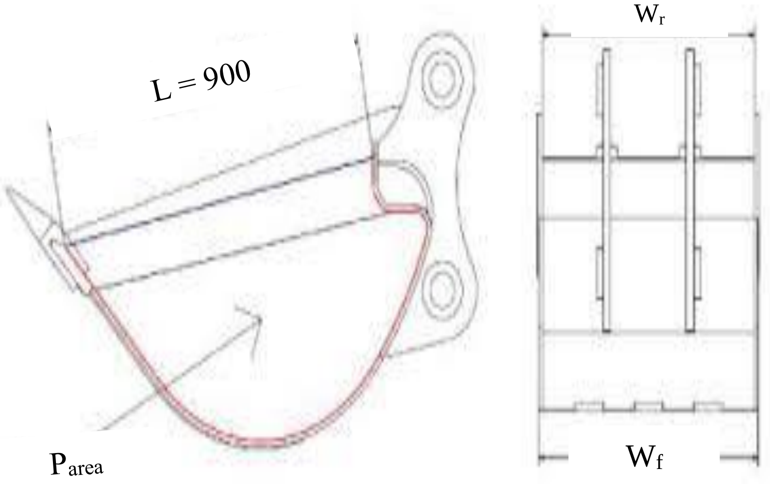

Figure 2 Bucket parameters [52]

Bucket capacity is a measure of the maximum volume of the material that can be accommodated inside the bucket of the backhoe excavator.

The bucket capacity can be calculated by:

$V_{BC} = V_{DC} + V_{EC}$

Were, VBC = Bucket capacity, VDC = Dump capacity (struck capacity) and VEC = Excess capacity, W_f =1000, W_r =952.

The dump capacity (VDC) can be calculated by:

$$V_{DC} = \frac{w_f + w_r}{2} \times p_{area}$$

$$V_{DC} = \frac{1000+952}{2} \times 435080$$

$$V_{DC} = 424638080\text{mm}^3$$

$$V_{DC} = 0.424638080 \text{ m}^3$$

Were, p_{area}= area of inner surface of bucket, W_r = inside width of the bucket and W_f = Outer width of bucket. Excess material capacity V_{EC} for angle of repose 1:1 according to SAE J296.

$$V_{EC} = \frac{L * W_F^2}{4} - \frac{w_r^3}{12}$$

$$V_{EC} = \frac{900*1000^2}{4} - \frac{952^3}{12}$$

$$V_{EC} = 153099883 \text{ mm}^3$$

$$V_{EC} = 0.153099 \text{ m}^3.$$

$$V_{BC} = V_{DC} + V_{EC}$$

$$V_{BC} = 0.424638080+0.153099$$

$$V_{BC} = 0.5777 \text{ m}^3$$

5.3 Calculating digging force and curling force

Bucket digging force is calculated by finding the curling force of the bucket (FB) and the mass force of the bucket (FS) as shown below.

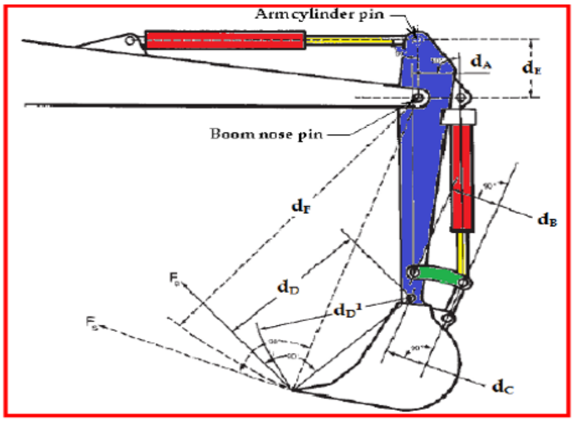

Figure 3 Parameters used in excavator arm calculations [52].

Digging force is nothing but force required to dig the terrain. These forces are exerted at the tip of the bucket. Digging forces are classified into bucket curling force and arm curling force. Bucket curling force is the force generated at the tip of the bucket due to bucket cylinder and arm crowd force is the force generated at the tip of the bucket due to arm cylinder.

5.3.1 The calculating the curling force

$$F_B = \frac{P * \frac{\pi}{4} * D_B^2}{d_D} * \frac{d_A * d_C}{d_B}$$

$$F_B = \frac{16 * \frac{\pi}{4} * 32^2}{1126} * \frac{627 * 270}{360}$$

$$F_B = 5374\text{N}$$

The curling force of the bucket is 5374N.

Where d_A=627mm, de=430mm, d_B=360mm, d_C=270, d_D=1126, $d_D{}^1$=872, d_F=2450mm, D_A=D_B=32, P=16N/mm^2 (working pressure).

5.3.2 The calculating the digging force

$$F_S = \frac{D_A^2 * d_E}{d_F} * P * \frac{\pi}{4}$$

$$F_S = \frac{32^2 * 430}{2450} * 16 * \frac{\pi}{4}$$

F_S= 2258N

5.4 MODELLING OF EXCAVATOR ARM

The following procedure is followed for modelling EXCAVTOR ARM in CATIS V5. Basic-2D sketch of base for support is created with the dimensions then develop this sketch into 3D by using pad command. Then create the 2D sketch of the boom then go to generative shape design and give volume extrude. Then create the arm D sketch and exit the workbench give the pad and shell command is used. Then create the 2D sketch of the bucket side next pad and shell command. Then assemble all the parts together as shown in figure.

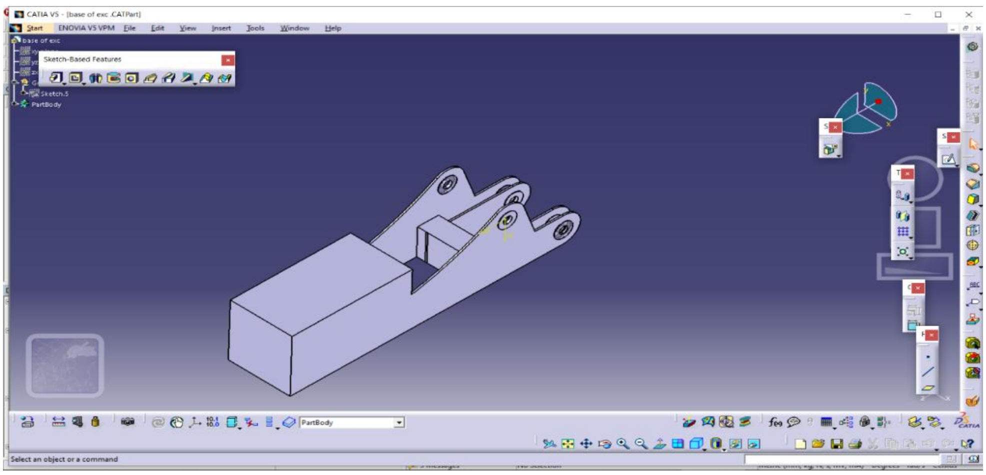

Figure 4 Modelling of base

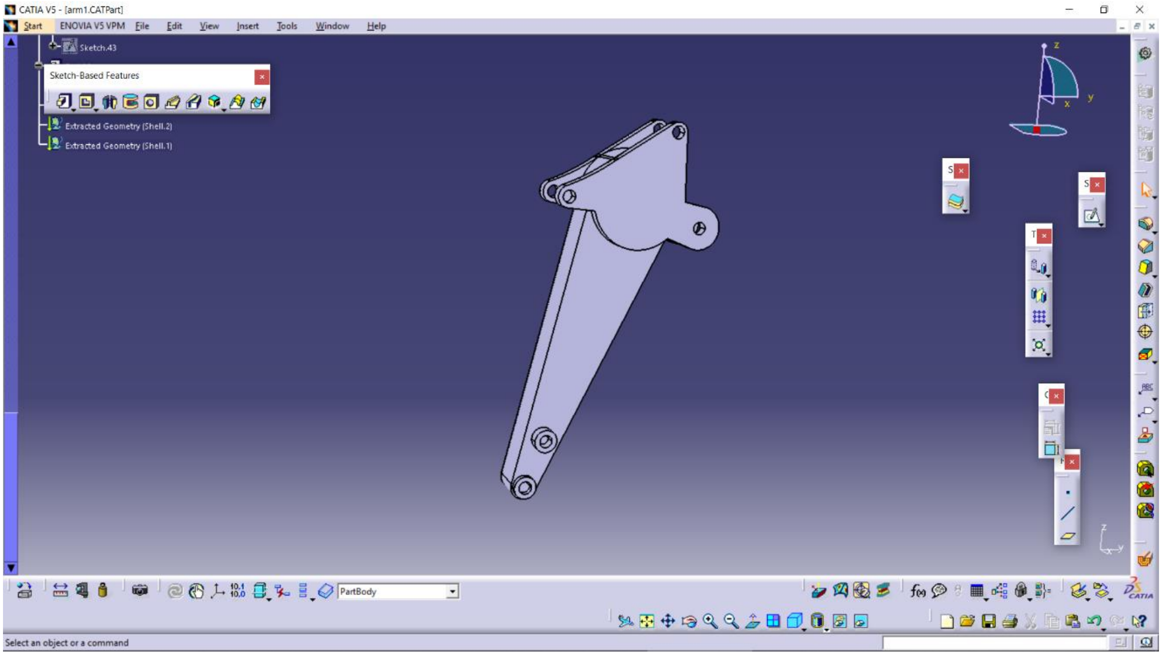

Figure 5 Modelling of arm

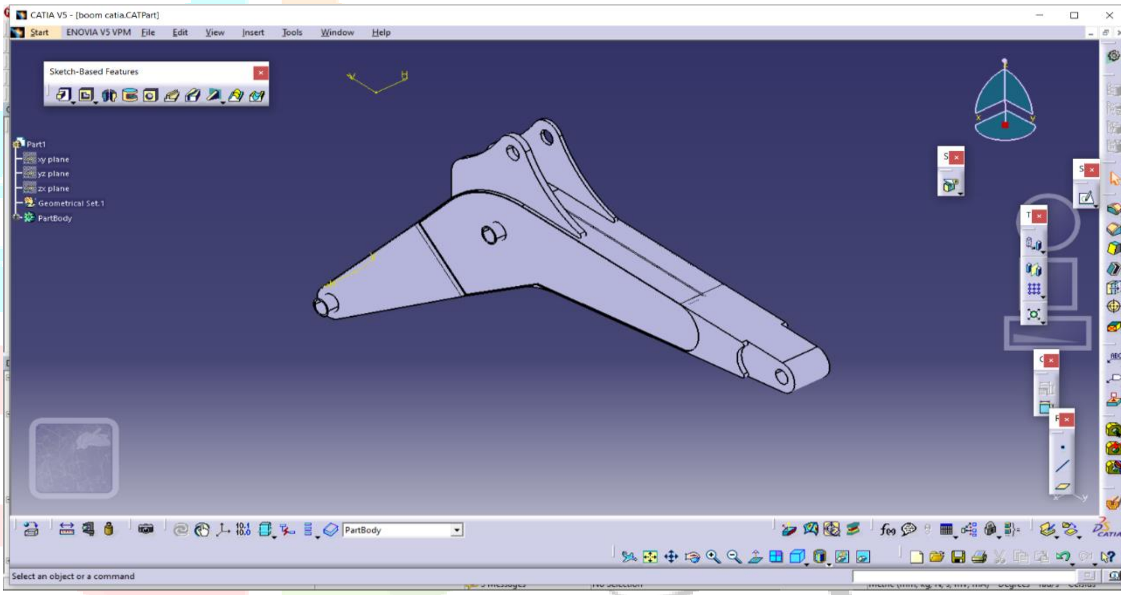

Figure 6 Modelling of boom

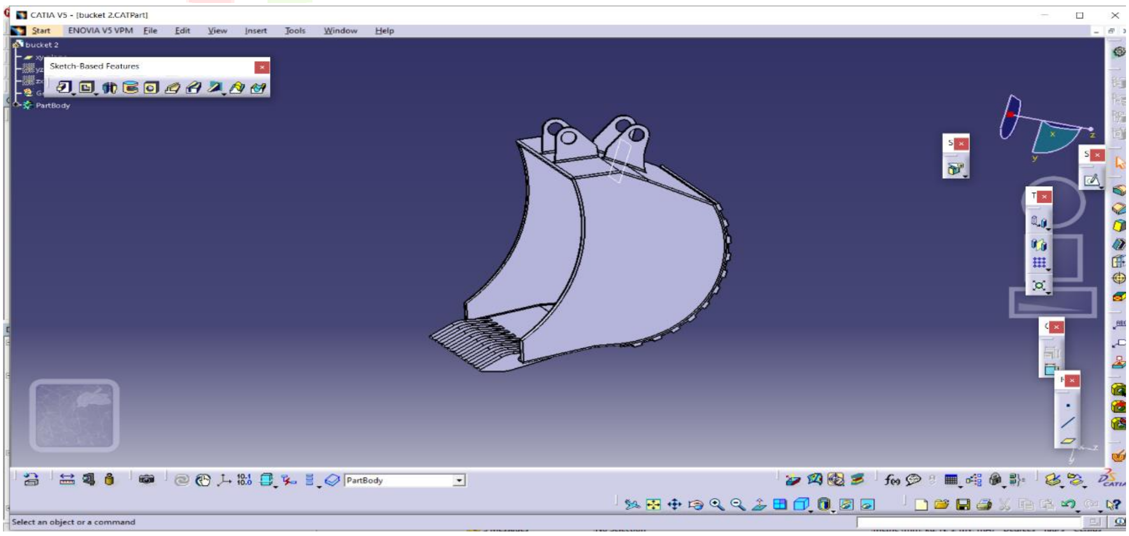

Figure 7 Modelling of bucket

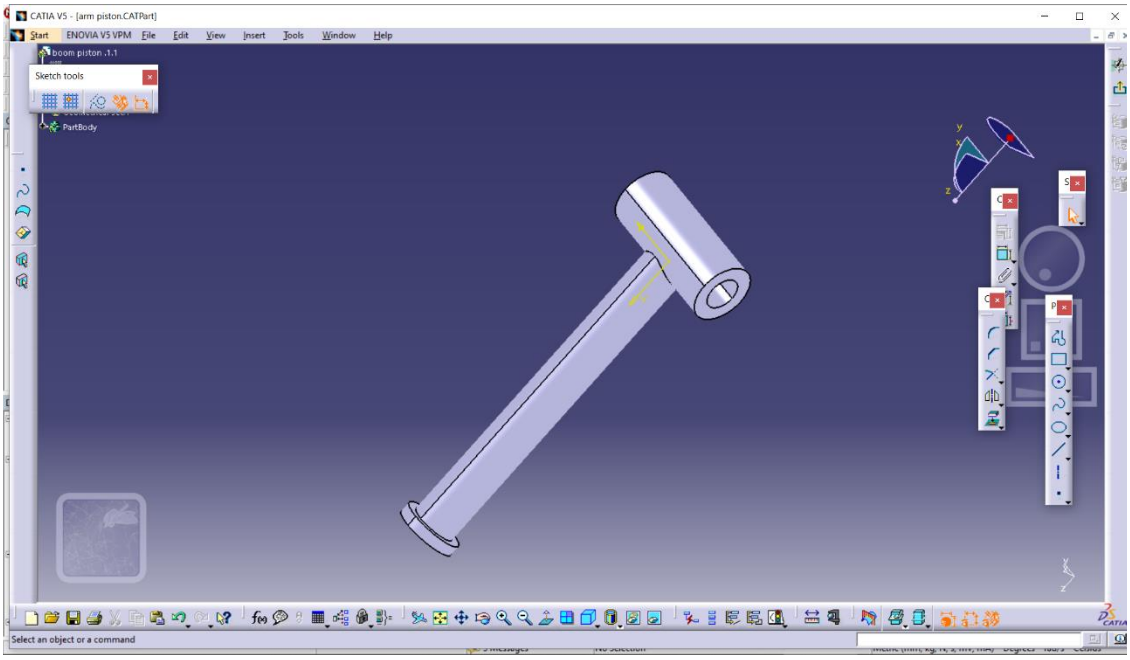

Figure 8 Modelling of piston

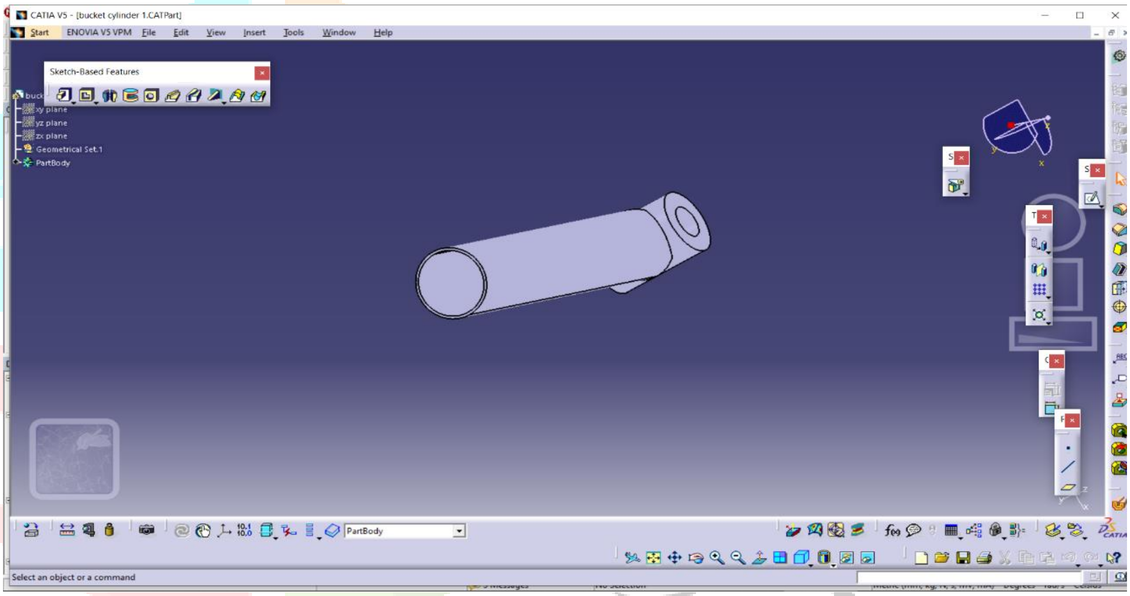

Figure 9 Modelling of cylinder

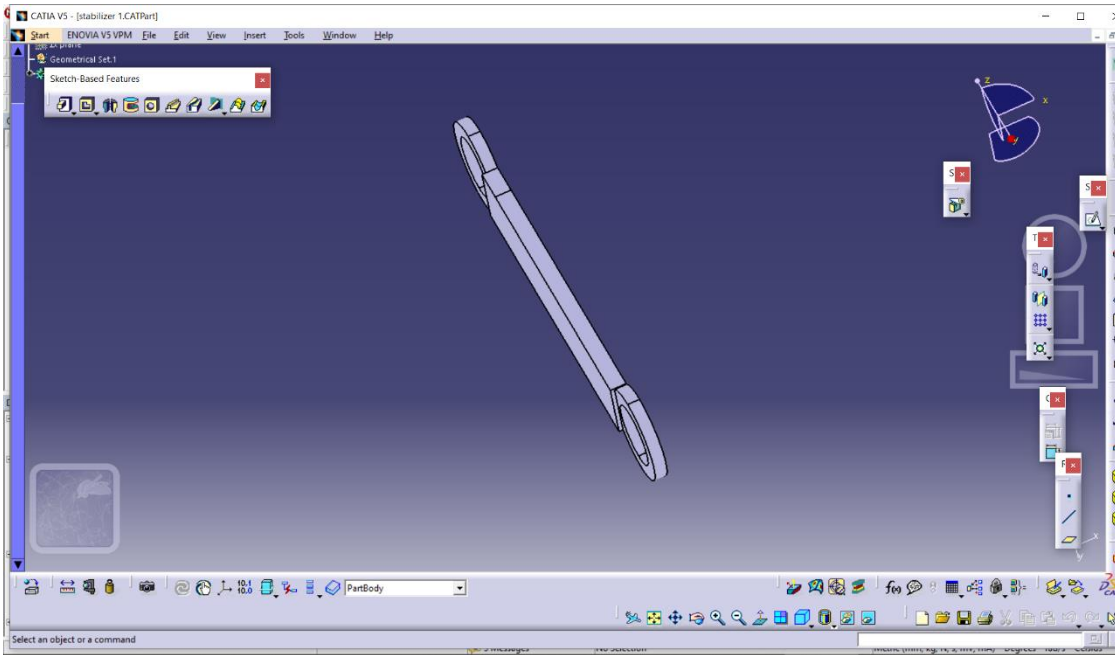

Figure 10 Modelling of stabilizer

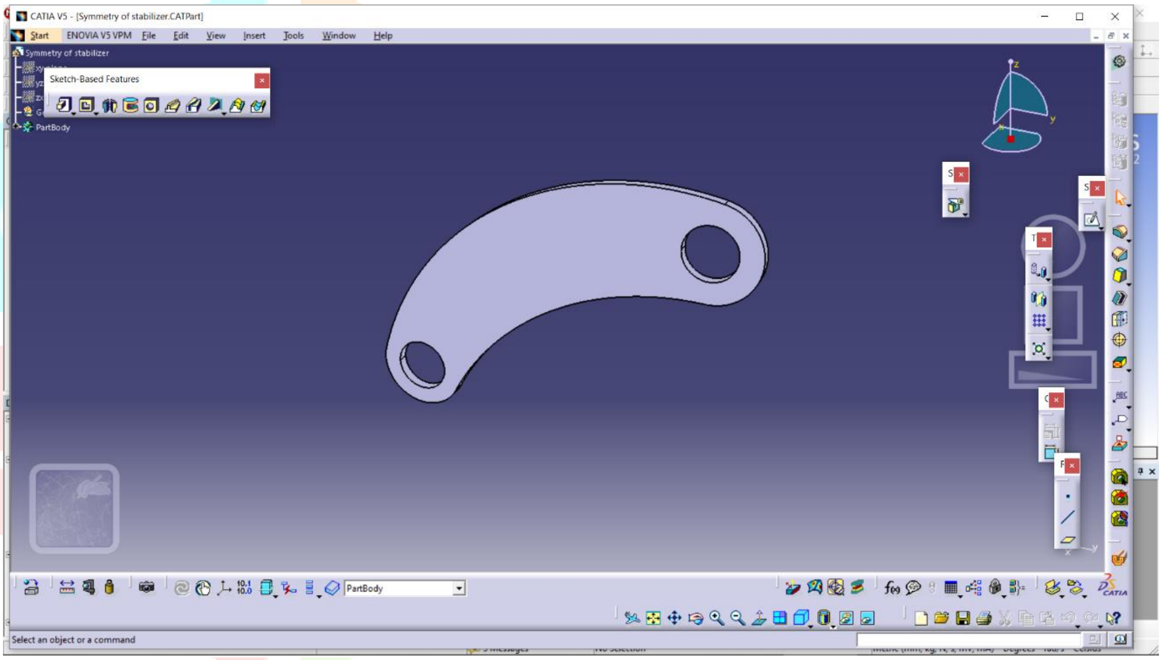

Figure 11 Modelling of stabilizer

6.ANALYSIS OF EXCAVATOR ARM

6.1 STATIC STRUCTUTAL ANALYSIS OF ARM

Finite Element Analysis is a powerful tool to analyse the designed engineering parts for its strength. The designed parts must robust in design and sustain against all kind of loading conditions as well as it should be work satisfactorily during its performance throughout its design life. Normally backhoe excavators are working in severe working environment with cyclic operation. During the design stage it is very important to check the strength of the various parts of the backhoe excavator for maximum breakout force condition. This can be achieved by performing the Finite Element Analysis of all the parts of the backhoe excavator attachment.

For analysis the software needs all the three dimensions defined. It cannot make calculations unless the geometry is defined completely. Thus, CAD model of the Excavator arm is converted to STEP file for importing geometry of excavator arm model into the design modeler of ANSYS 16.0. After completion of importing geometry as excavator arm, then after apply the materials.

Ansys meshing technology provide physical preferences that help to automate the meshing process. For an initial design, a mesh can often be generated in batch with an initial solution run to locate regions of interest. Further refinement can then be made to the mesh to improve the accuracy of the solution. Default Meshing is done on the model, the generated mesh is shown in the figure as follows.

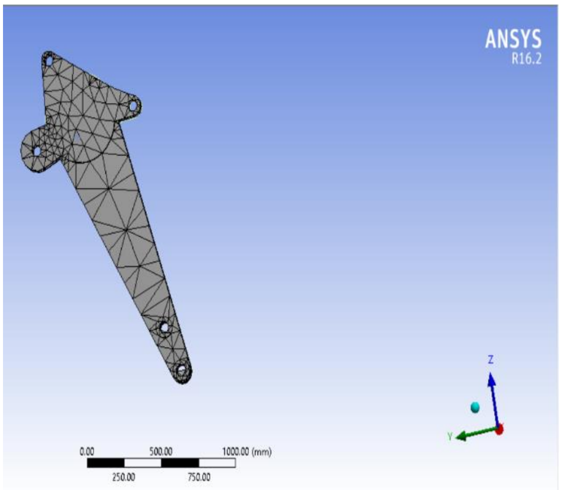

Figure 12 Generation of mesh of arm

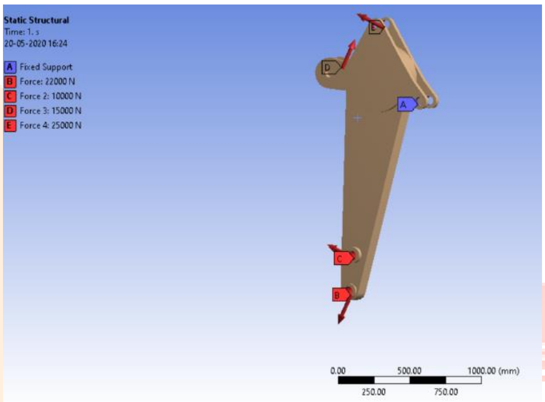

Figure 13 Boundary conditions of arm

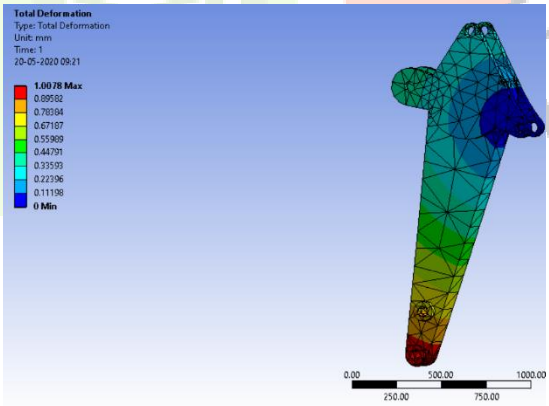

Figure 14 Total deformation of arm

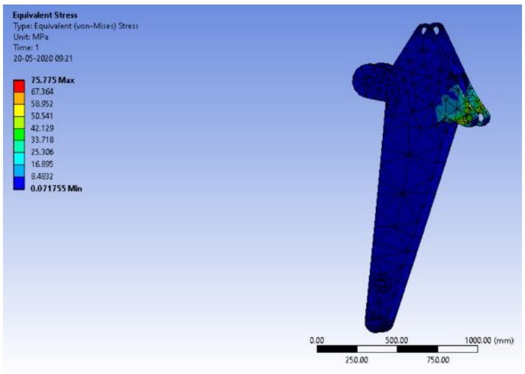

Figure 15 von-misses stress of arm

The above figures show the results of the static structural analysis of arm. The deformation developed is less than the thickness of the plate of the arm. The thickness of the plate is 6mm whereas the deformation is 1.007mm and the maximum von-misses stress is developed at the cylinder mounting. The developed stress due to the loads is 75 N/mm^2 is less than the yield strength of the material. The yield strength of the material is 450Mpa.

6.2 Static structural analysis of boom.

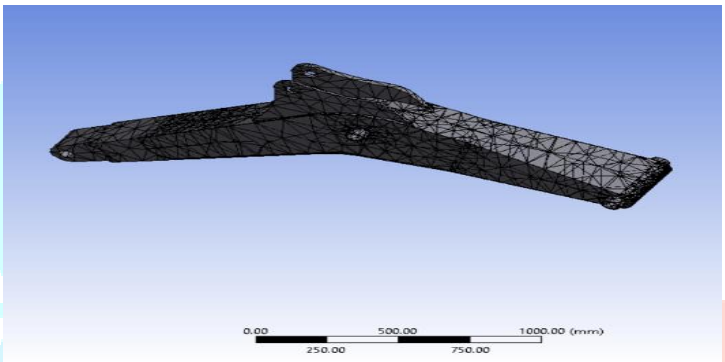

Figure 16 Meshing of boom

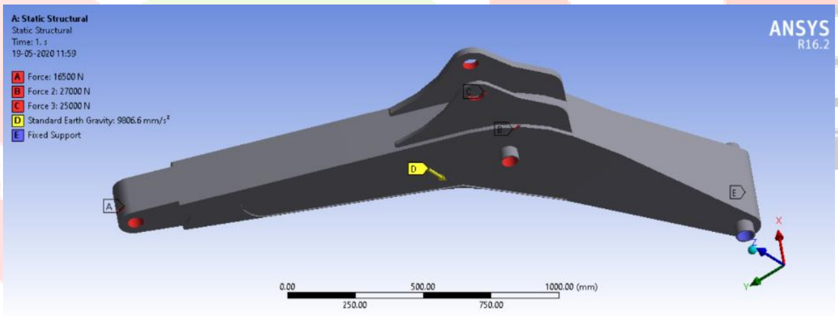

Figure 17 Boundary conditions of boom

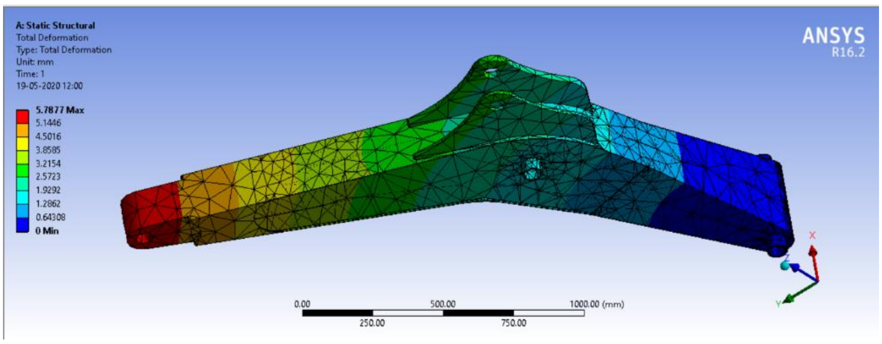

Figure 18 Total deformation of boom

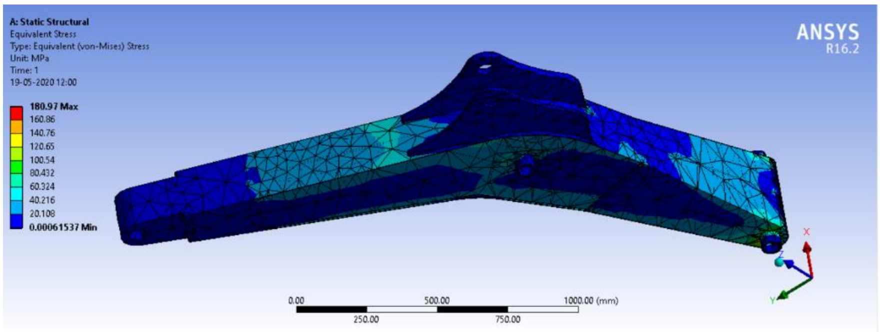

Figure 19 von-misses stresses of boom

The above figures show the results of the static structural analysis of boom. The deformation developed is less than the thickness of the plate of the arm. The thickness of the plate is 6mm whereas the deformation is 5.78mm and the maximum von-misses stress is developed at the base connecting. The developed stress due to the loads is 180.97 N/mm^2 is less than the yield strength of the material. The yield strength of the material is 450 Mpa.

6.3 static structural analysis of bucket

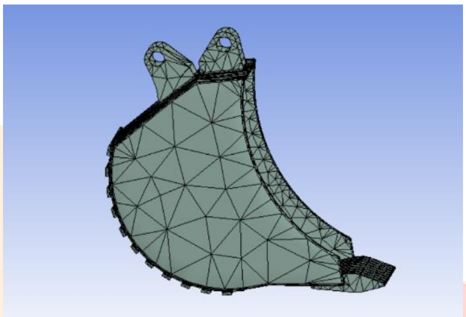

Figure 20 Generation of mesh of bucket

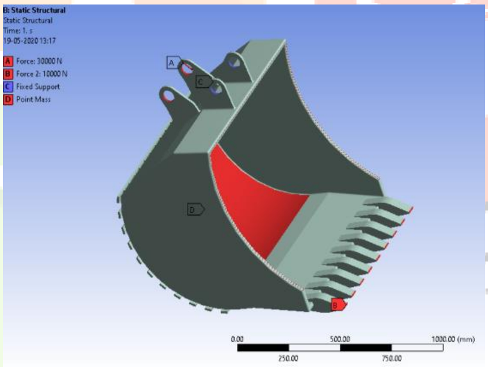

Figure 21 Boundary conditions of bucket

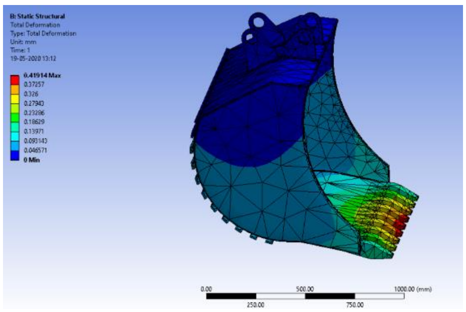

Figure 22 Total deformation of bucket

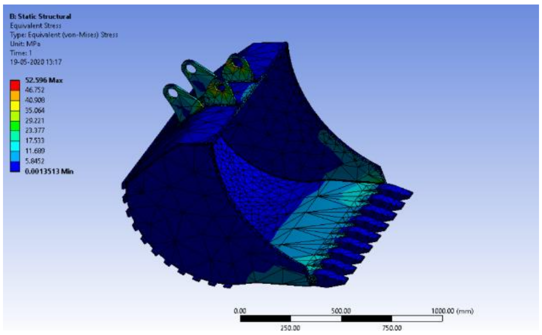

Figure 23 Von-misses stresses of bucket

The above figures show the results of the static structural analysis of bucket. The applied load on the bucket teeth tip is 5000N. The deformation developed is less than the thickness of the teeth of the bucket, whereas the deformation is 0.0041mm and the maximum von-misses stress is developed at the cylinder mounting. The developed stress due to the loads is 52.71N/mm^2 is less than the yield strength of the material. The allowable yield strength of the material is 450 Mpa.

6.3.1 Static structural analysis of bucket

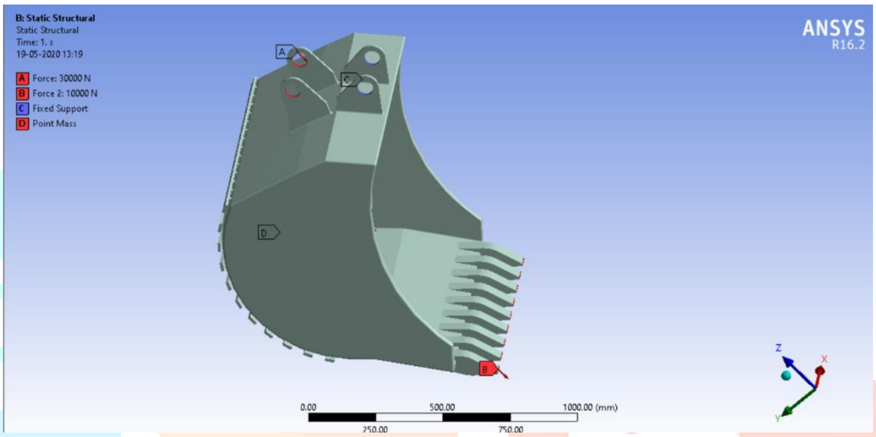

Figure 24 Boundary conditions of excavator

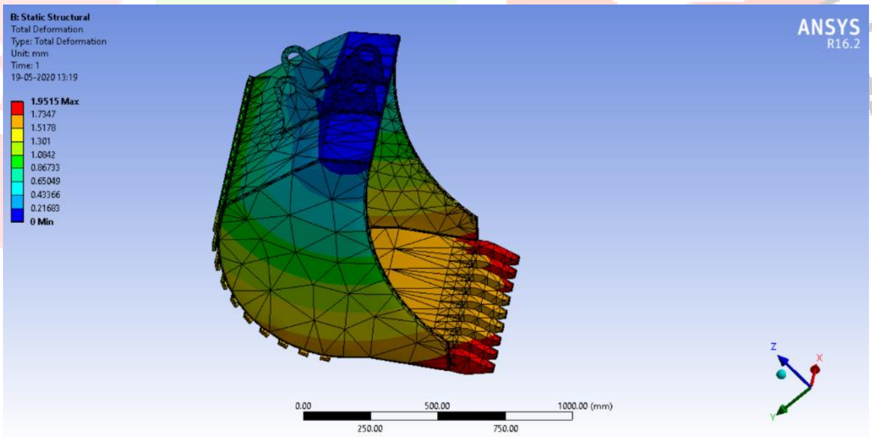

Figure 25 Total deformation of bucket

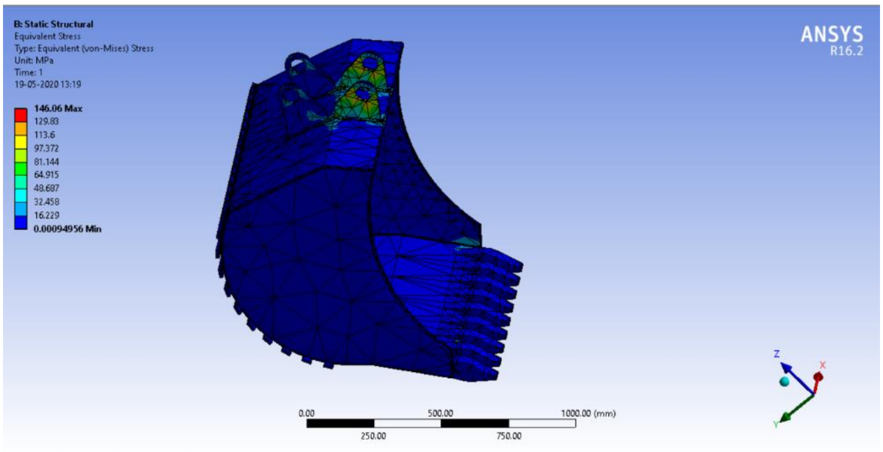

Figure 26 Von-misses stresses

The above figures show the results of the static structural analysis of bucket. The applied load is tangential to the bucket teeth tip. The deformation developed is less than the thickness of the teeth of the bucket, whereas the deformation is 1.951mm and the maximum von-misses stress is developed at the cylinder mounting. The developed stress due to the loads is 146.06N/mm^2 is less than the yield strength of the material. The yield strength of the material is 450 Mpa.

6.4 MODAL ANALYSIS

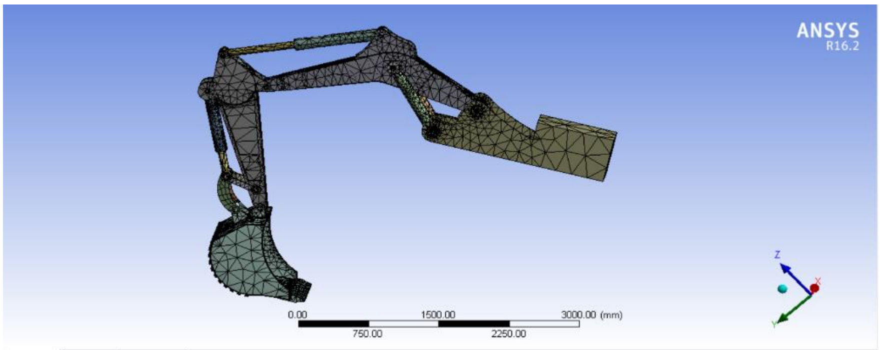

Figure 27 Meshing of Excavator arm

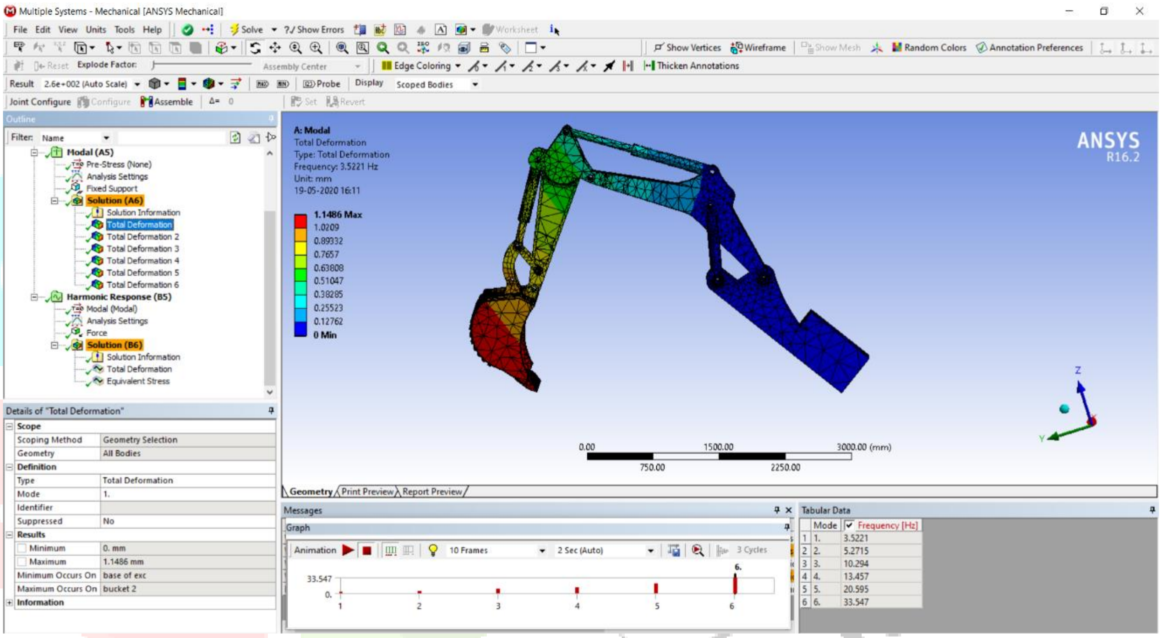

Figure 28 Total deformation due to minimum frequency

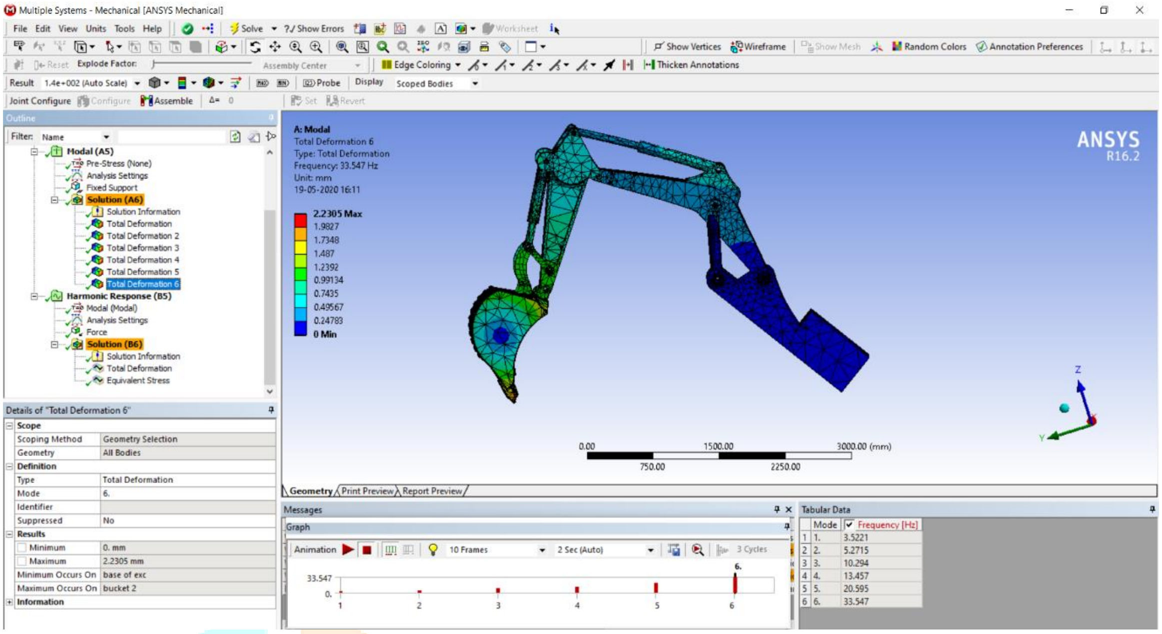

Figure 29 Modal Analysis

Modal analysis is used to determine the vibration characteristics of the design mechanism of the machine components, namely the structure's inherent frequency and vibration mode which are important parameters of the structural design. In modal analysis the frequencies are developed in the structure for the six different modes. The least frequency developed in the structure is 3.52Hz.

7. RESULTS AND DISSCUSIONS

7.1 Structural analysis

Table 1 structural analysis

S.NO	Structural analysis		
1	Arm	Total deformation	1.007 mm
2	Arm	Directional deformation	0.1255mm
3	Arm	Equivalent elastic strain	0.000419
4	Arm	Equivalent stress	75.775 Mpa
5	Boom	Total deformation	5.7877 mm
6	Boom	Directional deformation	0.514596 mm
7	Boom	Equivalent elastic strain	0.00089
8	Boom	Equivalent stress	180.97Mpa
	Bucket	Total deformation	1.9515 mm
10	Bucket	Directional deformation	0.111434 mm
11	Bucket	Equivalent elastic strain	0.00076303
12	Bucket	Equivalent stress	146.06 Mpa

In structural analysis the total deformation developed due to force acting on the excavator parts is less than the thickness of the sheet material used for the manufacturing. The level of stress developed is below the yield strength of the material used i.e., 450Mpa.

7.2 Modal Analysis

Table 2 Modal analysis

S .no	Mode	Frequency [Hz]	Deformation
1	1	3.52	1.1486
2	2	5.2715	1.1538
3	3	10.94	1.3674
4	4	13.457	1.8007
5	5	20.595	2.75
6	6	33.547	2.2305

Modal analysis is performed to find the fundamental frequencies (Modes) and their associated behaviour (Mode shapes). In modal analysis the initial minimum frequency developed is 3.52 Hz.

8. CONCLUSION

We designed an Excavator bucket by using CATIA V5 software and analysis is done by ANSYS 16.0 software. From the analysis results, it is proved that the design is safe for the calculated digging force. During designing of excavator arm, the important factors taken into account are factor of safety, breakout force and maximum lifting capacity. The maximum stress values were found at the cylinder mountings. The material used is the medium strength alloy steel. Yield strength of the material is 450Mpa. From the analysis results it is observed that the stress developed due to applied force is less than the yield strength of the material for boom, arm and bucket. The deformation values are also less than the plate thickness i.e.,6mm. from the modal analysis, the deformations developed due to the natural frequencies levels is also less.

9. FUTURE SCOPE OF THE PROJECT

Because of the surge of medium density housing construction, together with changing regulations have caused builders to rethink construction methods, including the use of construction equipment. In some cases where homes are built only meters apart, the mobility of the excavating machines in such compact areas along with the movement of materials, and the size of the excavating equipment play a crucial role in excavation.

This indicates the urgency in the development of such compact excavation equipment or a compact backhoe excavator attachment which has higher digging depth, dumping height, and digging reach with minimum dimensions so that the machine can be easily accommodate in the workspace. Thus, developed compact backhoe attachment parts have to be equally better in strength as the parts of the heavy-duty backhoe loaders are. Design of such a backhoe excavator attachment is carried out by developing a 3D model.

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