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*RESEARCH ARTICLE*

# IMPROVED DESIGN AND MODELLING OF THE BACKHOE ARM OF A BACKHOE LOADER.

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## Abstract

This research presents an improved design and modelling of the backhoe arm of the backhoe loader. It entails finite element analysis (FEA) and material based optimization of the backhoe arm, aimed at improving the loading capacity of the bucket of the backhoe arm while decreasing its digging forces. The stress analysis of the boom, arm and bucket of the backhoe were done by the use of ANSYS after which the backhoe arm was modelled with the use of SOLIDWORKS, a CAD software. The results obtained showed a breaking force of the bucket as 30.34 kN and a curling force of the arm as 29.34 KN, both about 43% reduction compared to the values of 71.00 KN and 64 KN respectively, obtained by the conventional backhoe loader. The results further showed an improved bucket volume capacity leading to an increase in the number of teeth of the backhoe as opposed to the number used in conventional backhoe loaders. It was thus recommended that the number of the backhoe bucket teeth should be increased with a reduced diameter to minimise friction losses during digging while the backhoe arm should be re-designed to absorb and reduce to the bearest minimum, the vibrations produced during its working cycle.



## Introduction:-

Backhoe loader is the power driven mechanism used for digging, moving or transporting of the loose gravel, sand or soil. It is a general-purpose construction equipment which is the combination of a tractor, a loader and a backhoe (Prabhakar, 2012). The core structure of the backhoe loader is built on the tractor; the loader is fixed in the front portion were it is used to carry the loose material and the backhoe is fixed in the back portion of the backhoe loader. The backhoe loader is used for a wide variety of tasks: construction, small demolitions, and light transportation of building materials, powering building equipment and digging holes/excavating, landscaping, breaking asphalt and paving roads. Various loads are applied at the bucket tip and to the boom and digger arm. Hence, it is necessary to analyse the parts assembly to avoid failure while in working condition. From static analysis, high stress area can be found out when backhoe loader is in different load condition whereas, provision of some design changes can enhance stress minimization (C and Ikbalahemad, 2015).

The backhoe is the main tool of the backhoe loader. It's used to dig up hard, compact material, usually earth, or to lift heavy loads. The Backhoe portion has 3 segments which are the boom, stick (arm) and the bucket which is shown in Figure 1 (Prabhakar, 2012). The backhoe arm is the most important part of the backhoe loader machine. During excavation operations, the backhoe arm is opposed by various resistive forces offered by the soil and other

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materials in contact with it. It is then vital to design an arm strong enough to combat these unpredictable forces while functioning effectively even under the worst conditions.

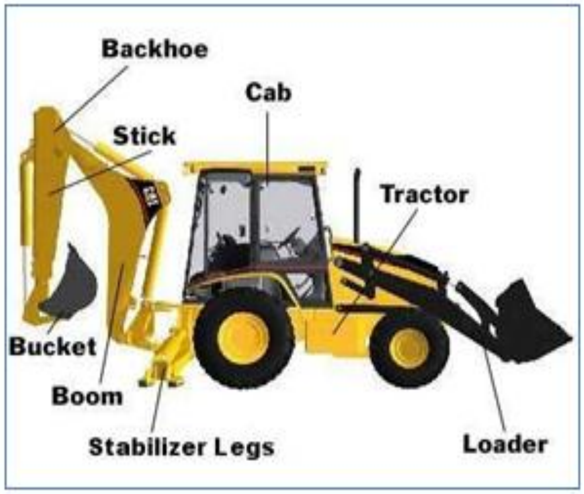



**Figure 1**:-Backhoe-loader machine (Prabhakar, 2012).

As part of the challenges associated with the backhoe-loader arms, it is important to note that during operation, as the backhoe goes through severe working conditions, the high level of stresses can damage the critical parts of the backhoe-loader such as the boom, arm, and bucket and thus, adversely affect the productivity of machine. The loader part of the loader mechanism and the backhoe are also subjected to high loads hence, the consideration of weight referred to as performance which is proportional to the power to weight ratio. Thus, it is necessary for the designers to provide not only an equipment of maximum reliability but also of minimum weight and cost, keeping design safe under all loading conditions since force analysis and strength analysis is an important step in the design of excavator parts.

Finite Element Analysis (FEA) is the most powerful technique in strength calculations of the structures working under known load and boundary conditions although, computer aided drawing model of the parts to be analysed must be prepared prior to the FEA. Also the possibility to reduce the weight of the mechanism is attainable by performing optimization task in FEA (R and Ramesh, 2016).

Literature indicates several studies focused on improving the working efficiency of the arms of the backhoe loader. Sanosh, 2017 worked on finite element analysis and optimization of boom of backhoe loader and suggested that the static analysis of the boom of a backhoe-loader will be at high performance when practiced with the finite element method (FEM). An optimum configuration was carried out to simulate and strengthen the boom under maximum loading condition and different boundary conditions. The results showed that the weight of the boom decreased by nearly 5%. (Ahmet and Eyup, 2013), in their research decided to do a static structural analysis of the backhoe-loader arms with the finite element method (FEM). According to analysis result, back and front arms of the backhoe-loader are strengthened with the use of reinforcements by nearly 20%.Juber and Manish (2012), carried out a Finite Element Analysis (FEA) of the boom of backhoe loader which is followed by the results of dynamic study of the boom of the machine and provided the platform to understand its modelling and FEA similar to the work of other researchers. The results are in the form of stress and deformation plots which can provide laudable assistance in the development of boom of backhoe loader. In their subsequent study and analysis of boom of backhoe loader with the help of FE tool, described its basic structure, stress characteristics and the engineering finite element modelling for analysing, testing and validation of backhoe loader parts under high stress zones. Li (2012), in order to study the stress and strain characteristics of working mechanism for a full hydraulic backhoe loader, a 3D entity model of the working mechanism was established by Pro/E, by importing the 3D model into ANSYS, the finite element model of organ was built by the interface of Pro/E, and three main parts of working mechanism were analysed by FEM (finite element method). A review on FEA and optimization of backhoe attachment in hydraulic excavator was done byBhaveshumar and J. (2011). The paper provided the platform to understand the Modelling, FEA and optimization of backhoe excavator attachment, thus, presenting helpful information for the development of a new excavator attachment. To reduce cost and improve strength of hydraulic mechanism, analysis and optimization of hydraulic loader was carried out by Ritenkumar (2015). Performance parameters of the mechanism were checked by carrying out the simulation of the mechanism, indicating the various methods used to make efficient simulation of complex mechanism. (R and Ramesh, 2016), worked on "Design and analysis of detachable type backhoe and loader. This paper deals with the design of backhoe components, loader components and special chassis for the tractor for a limited load of 2000N backhoe and 6000N loader. The hydraulic unit was selected to run by the tractor engine power of 50Hp whereas the original backhoe has 60Hp. The design process was carried out from determining the component dimensions that are required to withstand the load by both analytical calculations and software modelling. The components were 3D modelled using CREO PARAMETRIC modelling software and then structural analysis was carried out over the components using ANSYS.

C and Ikbalahemad (2015) developed the backhoe machine by 3D modelling using CAD software and verify the structural design by using finite element method. The study covers the detailed design, modelling and FE analysis of Backhoe Machine.Durmus (2010), worked on "Ride model and simulation of a backhoe loader". The objective of this study was to present a dynamic model of a backhoe-loader including cab dynamics in order to simulate the vibration levels transmitted to the operator. For this purpose, analytical solutions of the cab and the machine were developed by deriving the equations of motion of the system and the state space forms of the solution were implemented in the commercially available simulation software, MATLAB/Simulink. In addition to the analytical solution, a model was developed using the physical modelling toolboxes of MATLAB/SimMechanics. Cab model developed in SimMechanics is extended to simulate whole machine dynamics by inserting machine body and tire parameters. Vibration data was acquired from the machine for experimental validation of the models. Analytical and SimMechanics solution was evaluated by comparing the seat acceleration results for the same inputs. Furthermore, simulation results obtained from the models and the measurement results were found to be in agreement in both time and frequency domain.

Kilic (2009), worked on dynamic modelling of a backhoe loader with the aim to developing a dynamic model of the loader system of a backhoe-loader. Rigid bodies and joints in the loader mechanism and loader hydraulic system components were modelled and analysed in the same environment using the physical modelling toolboxes inside the commercially available simulation software, MATLAB/Simulink. System variables such as pressure, flow and displacement were measured on a physical machine and then compared with the simulation results. It was observed that the simulation results were consistent with the measurement results. It was thus shown that a prototyping time and costs can be highly reduced by implementing this model in the design process.
This research therefore presents an improvement in the design and modelling of the backhoe arm of the backhoe loader as it has to do with increasing the volume of the bucket for the backhoe arm of the backhoe loader machine. It employed a FEA software ANSYS for stress and force analysis of the bucket, the stick (arm) and the boom of the backhoe arm from materials which were selected for all the parts of the arm with the aid of modelling CAD software SOLIDWORKS to show the model of the arm. The application of the FEA software also showed the deformation of the bucket at the point under stress and the Factor of safety (FOS) of each parts.

## Materials and Methods:-

The design of any mechanical part or structure depends on a large number of criteria such as the selection of materials, the type of load and stresses caused by the load, form and size of the parts etc. The calculation for the static force analysis of the backhoe loader for the condition in which the mechanism produces the maximum breakout force has to be done. The condition for the maximum breakout force is the most critical one as it produces the highest breakout force, and thus for this condition force analysis is to be done, and will be used as a boundary condition for the static FEA. Therefore, in this paper, the professional three-dimensional modelling software Solidworks was chosen to model the backhoe arm and the model is imported into ANSYS software for further analysis.

The following assumptions are made for the purpose of this modelling:-

1. Material behaviour is linear elastic and strains are small. Therefore, linear elastic analysis will be carried out.
2. Pins and links are assumed to be rigid.
3. The loads are statically applied.
4. Material properties of structures after heat treatment are not changing.

**Materials used for the Backhoe Arm**

The materials used in the backhoe are based on “Crane Handbook” Design Data and Engineering Information used in the manufacture and application of Cranes by H.G Greiner shown in Table 1.

**Table 1.** Backhoe Component’s Material Selection.

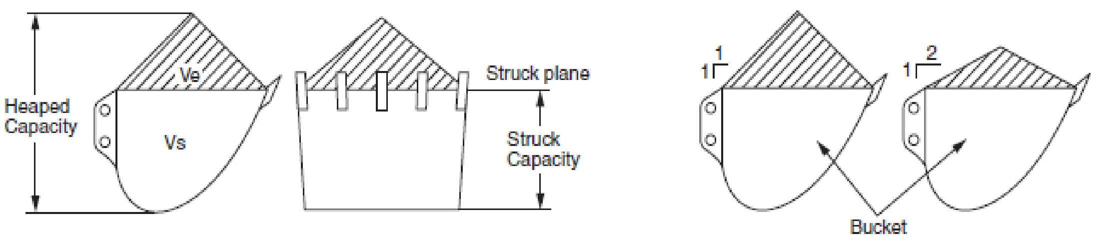
| S/N | Components | Materials Used | Material Properties |
|---|---|---|---|
| 1 | Boom | Medium Strength Alloy Steel ASTM-A514 | Ultimate Strength = 690 N/mm$^2$<br>Yield Strength = 450 N/mm$^2$<br>Poisson Ratio = 0.29<br>Density = 7900 kg |
| 2 | Stick | | |
| 3 | Bucket | | |
| 4 | Stabilizer | | |
| 5 | Support | | |
| 6 | Base/Base Pin | | |
| 7 | Hydraulic Cylinder | Mild Steel SA 36 Grade | Ultimate Strength = 450 N/mm$^2$<br>Yield strength = 250 N/mm$^2$<br>Poisson Ratio = 0.27<br>Density = 5600 kg/m$^3$ |
| 8 | Hydraulic Piston | | |

## Methods:-

**Calculations for Bucket**

**Bucket Capacity Calculations**

Bucket capacity is a measure of the maximum volume of the material that can be accommodated inside the bucket of the backhoe excavator. Bucket capacity can be either measured in struck capacity or heaped capacity as described in Figure 2.




(a) (b)

**Figure 2:-**Bucket and struck heated capacities.

The struck capacity is defined as the volume of the bucket after it has been struck at the strike plane. The strike plane passes through the top back edge of the bucket and the cutting edge as shown in Figure 2 (a). The struck capacity can be directly being measured from the 3D model of the backhoe bucket excavator.

Heated capacity is defined as the sum of the struck capacity plus the volume of excess material heaped on the bucket at a 1:1 angle of repose (according to SAE) or at a ratio of 1:2 angle of repose (according to CECE), as show in the Figure 2 (b). This is no way implies that the hoe must carry the bucket oriented in this attitude, or that all material will naturally have a 1:1 or 1:2 angle of repose.

The heaped capacity $V_h$ is given as:

$$V_h = V_s + V_e \quad (1)$$

Where, Vs is the struck capacity, and $V_e$ is the excess material capacity heaped either at 1:1 or at 1:2 angle of repose as shown in Figure 2 (b)

The Struck capacity $V_S$ as observed from Figure 2 is

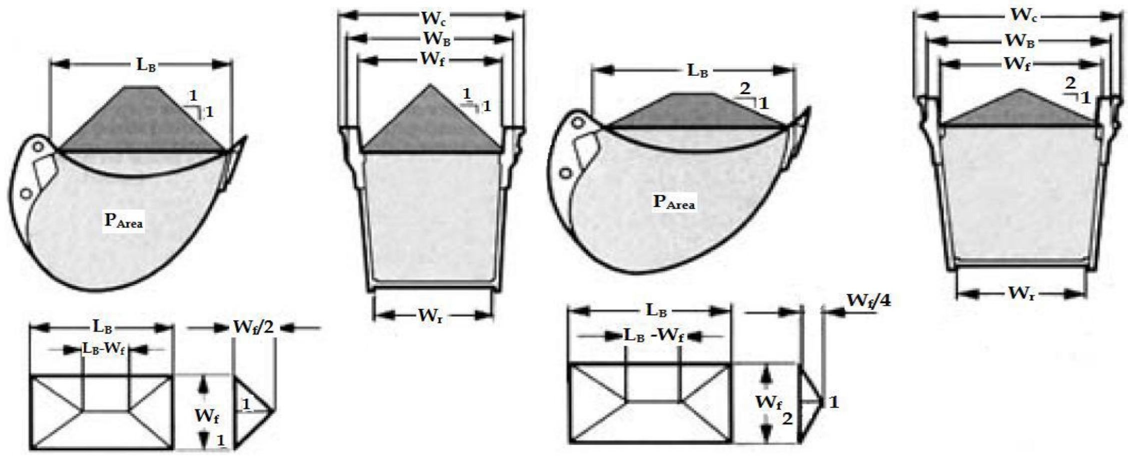
$$Vs = Pareax\frac{Wf + Wr}{2} \quad (2)$$

Excess material capacity $V_e$ for angle of repose 1:1 according to SAE J296 Standard as shown in Figure 3 (a)

$$Ve = \left(\frac{LBWf2}{4} - \frac{Wf3}{12}\right) \quad (3)$$

Excess material capacity $V_e$ for angle of repose 1:2 according to CECE section VI as shown in Figure.3 (b).

$$Ve = \left(\frac{LBWf2}{8} - \frac{Wf3}{24}\right)$$




(a) (b)

**Figure 3:-**Bucket capacity rating (A) According to SAE (B) According to CECE

The description of the terms used in Figure. 3 is as follows:

1. LB: Bucket opening, measured from cutting edge to the end of bucket base rear plate.
2. WC: Cutting width, measured over the teeth or side cutter (note that the 3D model of bucket proposed in this thesis is only for light duty construction work, so side cutters are not attached in our model).
3. WB: Bucket width measured over sides of bucket at the lower lip without teeth of the side cutters attached (so this will also not be the important parameter for the proposed 3D model of bucket as it does not contain any side cutters).
4. Wf: Inside width front, measured at cutting edge or side protectors.
5. Wr: Inside width rear, measured at narrowest part in the back of the bucket.
6. Parea: Side profile area of bucket, bounded by the inside contour and strike plane of the bucket.

**Digging Force Calculation**

Digging force is the force required to dig the terrain which is exerted at the tip of the bucket. It is are classified into bucket curling force and the arm curling force. Bucket curling force is the generated at the tip of the bucket due to bucket cylinder and arm curling force is generated at the tip of the bucket due to arm cylinder. Generally digging force is calculated at the condition at which the excavator generates maximum digging force known as maximum breakout condition of the linkages. According to SAE J1179 digging forces for maximum breakout condition is shown in Figure 4.

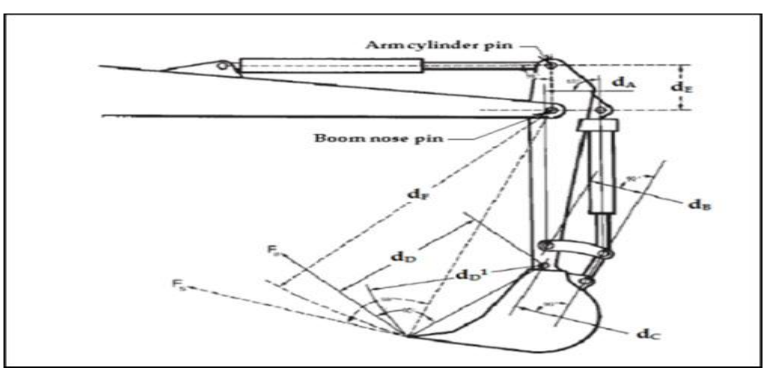



**Figure 4:-**Maximum digging force calculations **(**Khedkar, Dey, & Padasalagi, 2017)

In this $d_A$, $d_E$, $d_F$, $d_C$,$d_D$ are the distances and $F_B$ and $F_S$ are the bucket curling force and arm curling force. Bucket curling force is tangent to tip radius of bucket and it's generated by bucket cylinder. It's given by,

$$F_B = \frac{p \times \frac{\pi}{4} \times {D_B}^2 \times d_A \times d_C}{d_D \times d_B} \quad (5)$$

Here $d_B$ (mm) is the diameter of the bucket cylinder, p is the working pressure (MPa) and $F_B$ is the maximum breakout force of the bucket.
The arm curling force is given by,

$$F_S = \frac{p \times \frac{\pi}{4} \times {D_A}^2 \times d_E}{d_F} \quad (6)$$

Where, $d_F$ = bucket tip radius ($d_D$) + arm link length and$D_A$ = end diameter of the arm cylinder.
The combination of the backhoe excavator's arm crowd$F_S$ and bucket curling force$F_B$ give the machine configuration a more effective bucket penetration force per mm of bucket cutting edge than it is available with other machine types such as wheel and track loaders.

**Finite Element Analysis of Boom, Stick and Arm**
Finite element analysis (FEA) is the most powerful technique in strength calculations of structures working under loads and boundary conditions. These analyses show critical points of design early and so one improves the design before producing prototypes. A typical Finite Element Analysis (FEA) problem is composed by three essential steps: Modelling, mesh generation and solution, and post process. A parametric geometry model suitable for FEM analysis is first generated using CAD software Pro/Engineer. The subsequent mesh is generated with minimum reference to the geometry information, as only the top-level entities in the CAD model are referenced in the meshing.

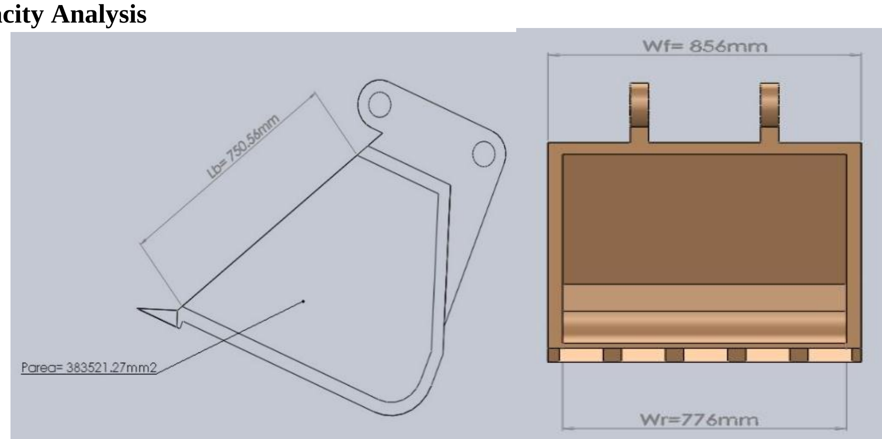
The static force analysis of the backhoe arm of the loader was carried out on the boom, arm, and bucket. These three sections of the backhoe arm are the principal sections that are affected by the loading and unloading cycles of the backhoe loader and hence, need to be optimized. They were carried out with the FEA software, ANSYS.

**Component Design**
The components of the backhoe loader were analysed using the ANSYS after calculations are made. Each of the individual components is analysed separately and then the mechanism is fully assembled. After the assembly of the arm, the final stress analysis of the backhoe arm is then carried out.
Design stress for ductile materials,

$$\sigma_{VM} \leq \frac{\sigma_y}{\text{FactorofSafety}} \quad (7)$$

$$\text{Total stress}\sigma_T = \text{Von-Misses Stress}\sigma_{VM} + \text{Normal Stress}\sigma_n \quad (8)$$

## Results and Discussion:-
**Bucket Capacity Analysis**



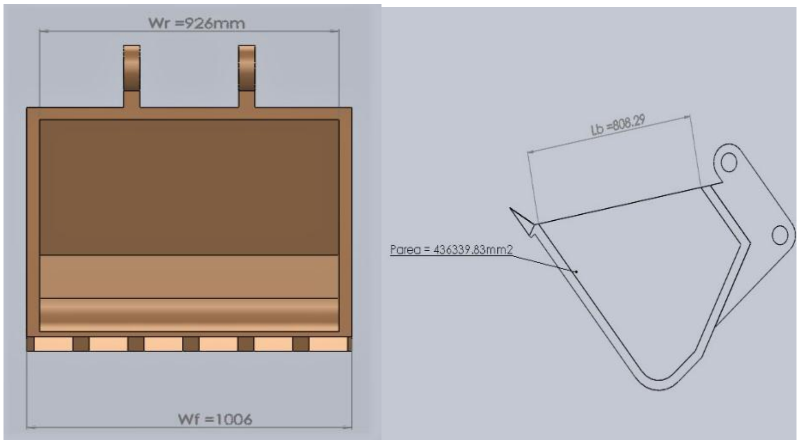
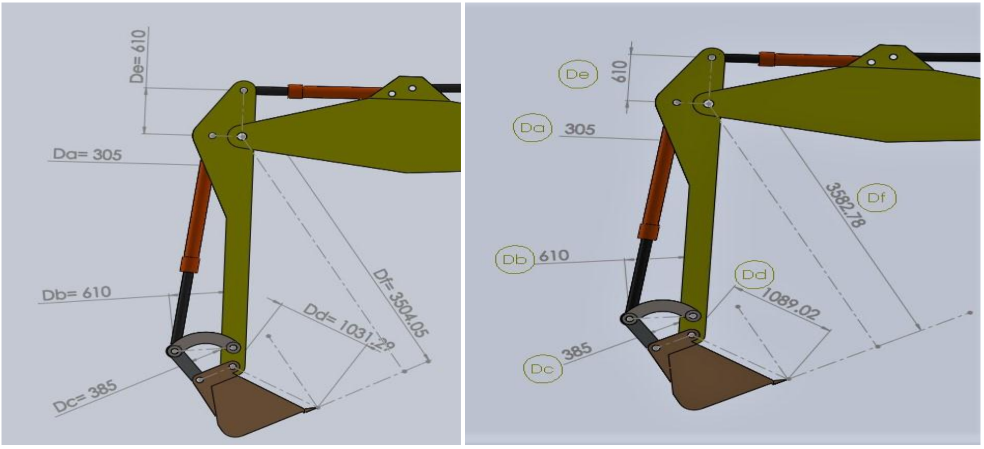



**Figure 5:-**Dimension for the bucket 1 and 2.

The analysis for the capacity of the bucket is expressed thus:
For bucket 1: $\mathrm{Vs} = 0.38352137 \text{ x } ((0.856 \text{ x } 0.776)/2) = 0.31295\mathrm{m}^3$
For bucket 2: $\mathrm{Vs} = 0.43633983 \text{ x } ((1.006 + 0.926)/2) = 0.421504\mathrm{m}^3$
The excess volume capacity of the bucket according to SAE Standard from equation (3) and (4) are;
For bucket 1: $\mathrm{Ve} = \frac{0.75056\mathrm{x}0.856^2}{4} - \frac{0.856^3}{12} = 0.07405159\mathrm{m}^3$
For bucket 2: $\mathrm{Ve} = \frac{0.80829\mathrm{x}1.006^2}{4} - \frac{1.006^3}{12} = 0.119663\mathrm{m}^3$
Therefore, the heated capacity of the bucket is given as:
For bucket 1, $\mathrm{Vh} = 0.38700159\mathrm{m}^3$
For bucket 2, $\mathrm{Vh} = 0.541167\mathrm{m}^3$

**Digging Force Calculations**




(a) From bucket 1 (b) from bucket 2

**Figure 6:-**Dimension for the arm

The following values are given from Figure 6;

From a
$\mathrm{d_A} = 305\mathrm{mm}$
$\mathrm{d_B} = 610\mathrm{mm}$
$\mathrm{d_C} = 385\mathrm{mm}$
$\mathrm{d_D} = 1031.29\mathrm{mm}$
$\mathrm{d_E} = 610\mathrm{mm}$
$\mathrm{d_F} = 3504.05\mathrm{mm}$
$\mathrm{D_A} = 102\mathrm{mm}$

From b
$\mathrm{d_A} = 305\mathrm{mm}$
$\mathrm{d_B} = 610\mathrm{mm}$
$\mathrm{d_C} = 385\mathrm{mm}$
$\mathrm{d_D} = 1089.02\mathrm{mm}$
$\mathrm{d_E} = 610\mathrm{mm}$
$\mathrm{d_F} = 38582.78\mathrm{mm}$
$\mathrm{D_A} = 102\mathrm{mm}$

$D_B = 102\text{mm}$ $D_B = 102\text{mm}$

**Bucket Curling Force Calculation**

The calculation for the bucket curl is as shown;

For bucket 1: $= \frac{21000\ x\frac{3.142}{4}x102^2x\ 305\ x\ 385}{1.03129x0.610} = 32.0343\text{KN}$

For bucket 2: $= \frac{21000\ x\frac{3.142}{4}x102^2x\ 305\ x\ 385}{1089.02x0.610} = 30.3362\text{KN}$

**Arm Curling Force Calculation**

For bucket 1: $= 21000\ x\frac{3.142}{4}x\frac{0.102x0.102x0.610}{3.50405} = 29.8762\text{kN}$

For bucket 2: $= 21000\ x\frac{3.142}{4}x\frac{0.102x0.102x0.610}{3.58278} = 029.22\text{kN}$

**Finite Element Method**

As all dimension of bucket, boom and entire arm are in proportion we use tetrahedral elements for meshing.

**Table 2.** Final Result Summary of Deformation and Stress Analysis

| S/N | PART | DEFORMATION | VON-MISSES STRESS | NORMAL STRESS | TOTAL STRESS |
|---|---|---|---|---|---|
| 1 | BUCKET | 0.62192mm | 24.304 | 9.1105 | 32.4145 |
| 2 | BOOM | 0.010594mm | 3.6088 | 3.4054 | 6.0138 |
| 3 | ARM | 0.58598mm | 35.646 | 19.386 | 55.032 |

**Bucket Analysis**

For the bucket before improvement,

**For bucket 1:**

Total deformation Equivalent stress (Von-Mises)

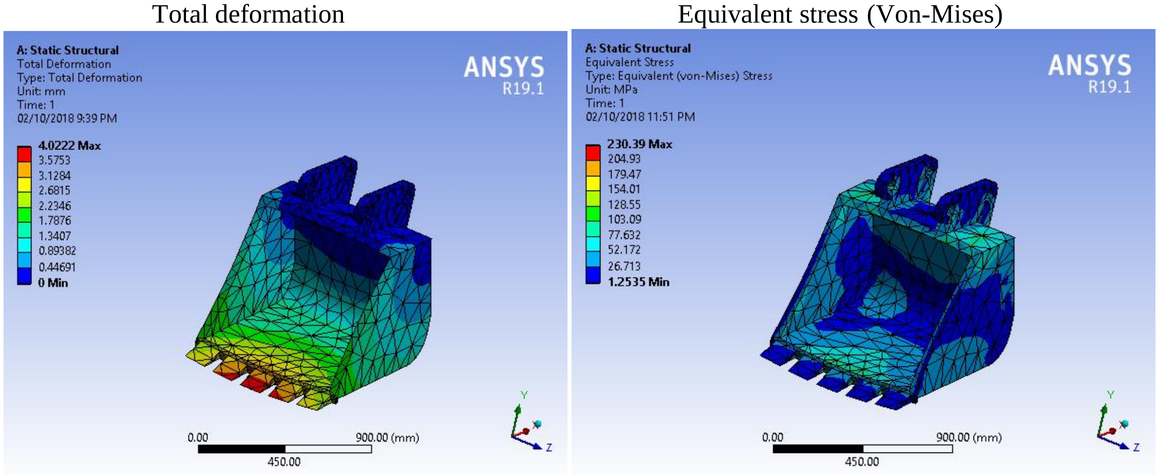



**Figure 7.**Total deformation and Equivalent stress (Von-Mises) of bucket 1.

1. Normal stress

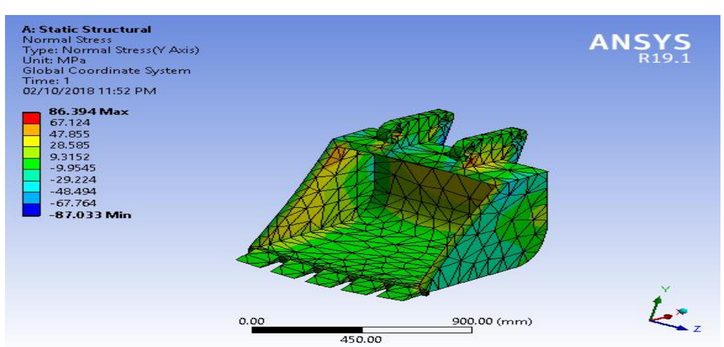



**Figure 8:-**Normal stress of bucket 1

**For bucket 2**

Total deformation Equivalent stress (Von-Mises)

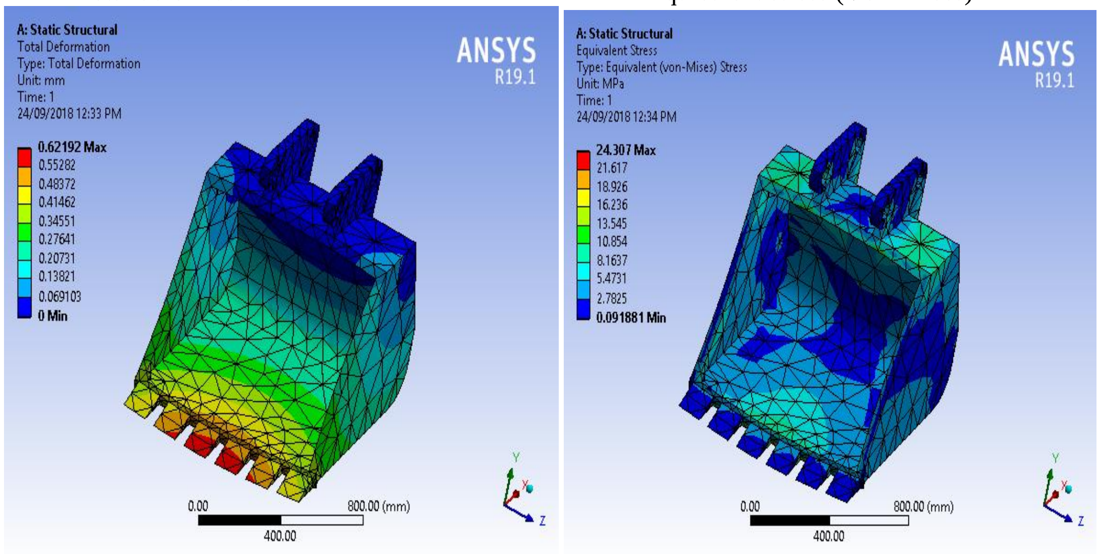



**Figure 9:-**Total deformation and Equivalent stress (Von-Mises) of bucket 2.

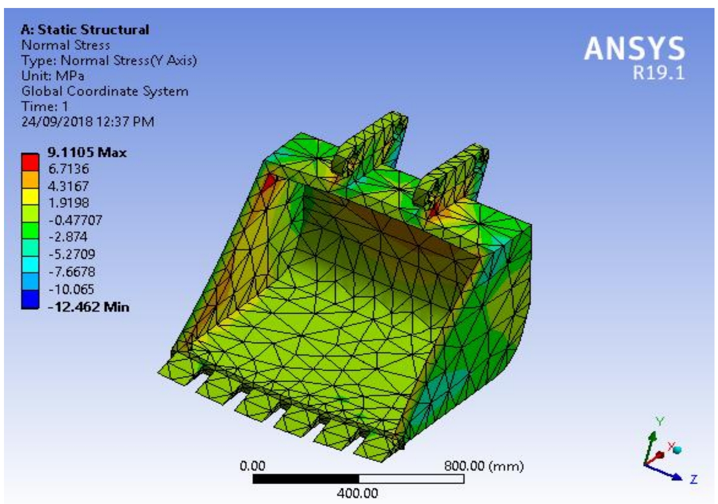



**Figure 10:-**Normal stress of bucket 2.

From the results obtained from the analysis of the bucket 1 and 2 in Figures 7, 8, 9 & 10, the displacement values are 4.022 mm and 0.62192 mm respectively, which is very small compared to the thickness of the bucket of 40 mm and hence, very acceptable.

Assuming of factor of 4 for the bucket, observation showed that the Von-Misses stress of 24.304 Mpa as well as the total stress of 32.4145 Mpa is very little compared to the yield strength of 112.5 Mpa for bucket 2, indicating a safe and acceptable design.

**Boom Analysis**

Total deformation Equivalent stress (Von-Mises)

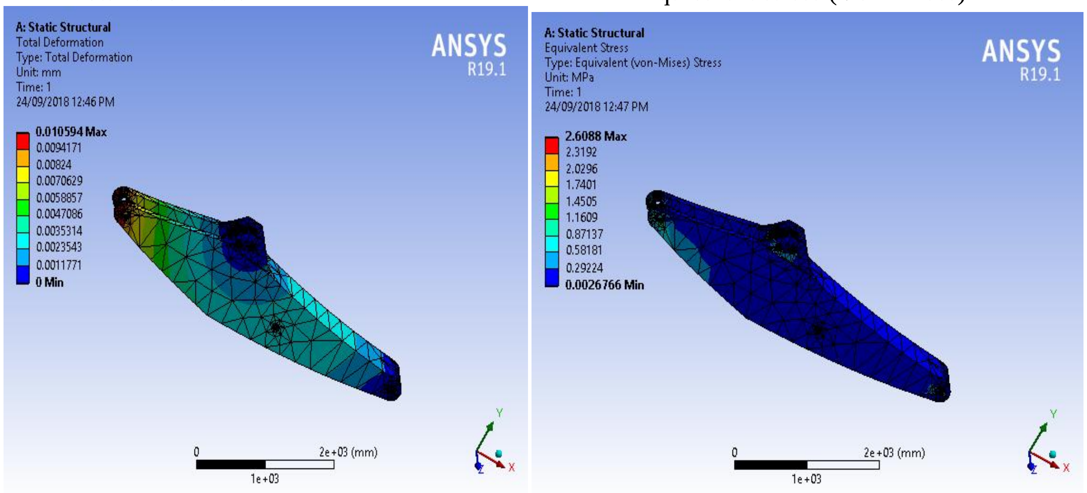



**Figure 11:-**Deformation and Equivalent stress (Von-Mises) of boom

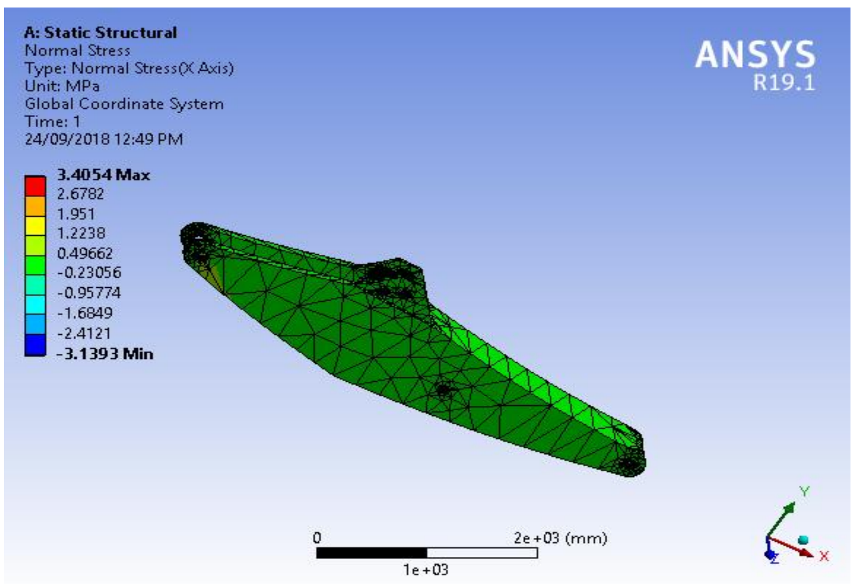



**Figure 12:-**Normal stress of boom

The results obtained from the analysis of the boom shown in Figures 11 and 12, showed a safe and acceptable design as the displacement of 0.010592 mm is very small compared to the thickness of the boom of 100 mm and with a Total stress and Von-Misses stress values of 6.0138 Mpa and 2.6088 Mpa respectively for a factor of 4 for the buckets.

**Arm Analysis**

Total deformation Equivalent stress (Von-Mises)

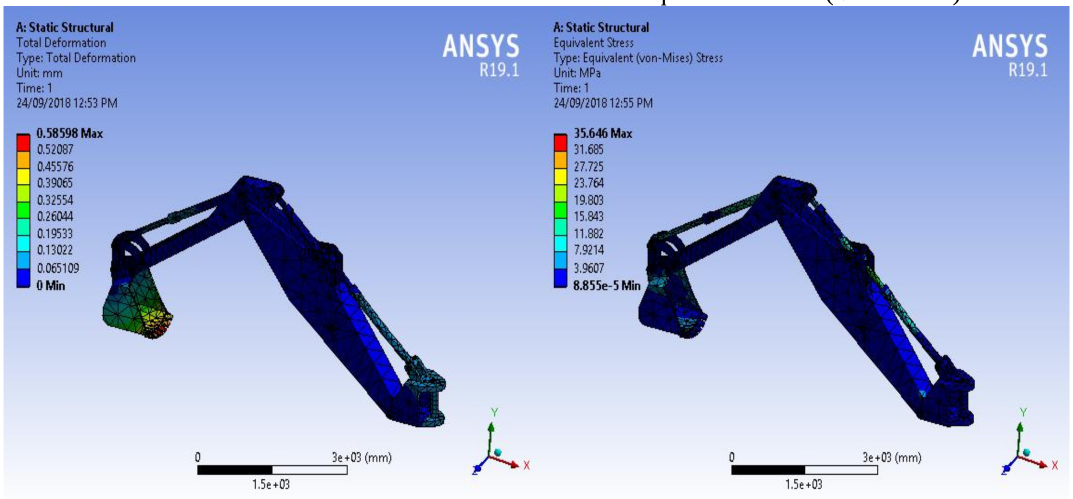



**Figure 13:-**Deformation and Equivalent stress (Von-Mises) of the arm

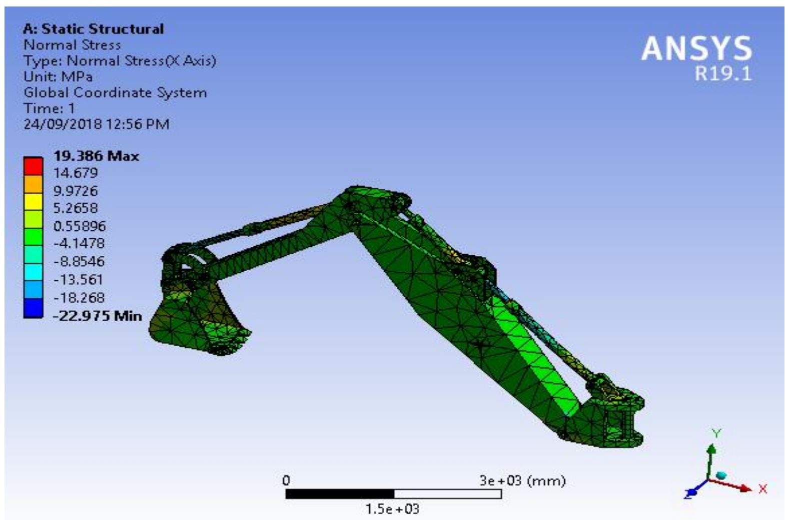



**Figure 14:-**Normal stress of the arm

From the analysis of the arm shown in Figures 13 and 14, the displacement value of 0.62192 mm is very minimal in comparison to the thickness of the arm of 70 mm. Again, for a factor of safety of 4 for the bucket, values obtained for the Von-Misses stress and the Total stress are 35.646 Mpa and 55.032 Mpa which are both very small when compared to the yield strength value of 112.5 Mpa for the arm. Hence, the design is safe and acceptable.

## Conclusion:-

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. Finally, the results obtained for breaking and curling forces (30.3362, 29.34) KN have been reduced by over 43% as opposed to 71 KN and 64 KN used by conventional backhoe loaders. It is thus recommended that the backhoe arm should be re-designed to absorb and reduce to the barest minimum vibrations produced during its working cycle. Again, the backhoe bucket teeth should be further increased and their diameters reduced to minimize the friction losses during digging.

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