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Development of Backhoe Machine By 3-D Modelling using CAD Software and Verify the Structural Design By using Finite Element Method

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Abstract

Present study covers the detailed design, modelling and FE analysis of Backhoe Machine. . Backhoe Loader is the the rear part of the excavator machine. The backhoe loader is used for a wide variety of tasks: construction, small demolitions, 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. So it is necessary to analyze the parts assembly to avoid failure while it is in working condition. From static analysis, high stress area can be found out when Backhoe Loader is in different load condition. Also by providing some design changes, stress can be minimized.
Keywords: Backhoe Loader; Stress Analysis; FEM

I. INTRODUCTION

Today in the machine age when the use of machines is increasing for the earth moving works, considerable attention has been focused on designing of these earth moving equipments. Achievement of an ambitious and 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. Bulldozers, scrapers, motor graders, excavators and other machines are widely used for most arduous earth moving work in engineering construction. Thus it is very much 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 by careful stress analysis of the machines.

II. STATIC FORCE ANALYSIS

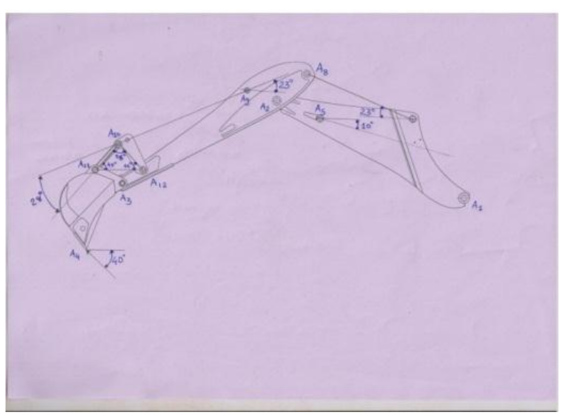


Fig. 1: Free Body Diagram of Backhoe Assembly

A. *Bucket Static Force Analysis:*

The reaction force on the bucket teeth at point A_4 due to breakout force 60100 N acts at the angle 40° for configuration of the breakout force condition.Reaction forces on point A_4 is resolved in horizontal (X) and vertical (Y) direction.

$F_{4H} = F_B * \text{Cos}(\alpha_1)$
$= 46039.27$ N

Where α_1 is the angle between the breakout force of bucket and the ground level as horizontal reference surface of 40º

$F_{4V}=F_B * \sin(\alpha_1)$

$=38631.53$ N

Now considering the bucket in equilibrium $\Sigma M = 0$, taking moment about the bucket hinge point A_3 leads to

$F_4 \cdot l_4 - F_{gb} \cdot l_{gb} = F_{11} \cdot l_{11}$

:. $F_{11} = 182032.46$ N

$F_{11H}=F_{11} \cdot \cos \beta 11$

$=121803.49$ N

$F_{11V}=F_{11} \cdot \sin \beta 11$

$=135276.48$ N

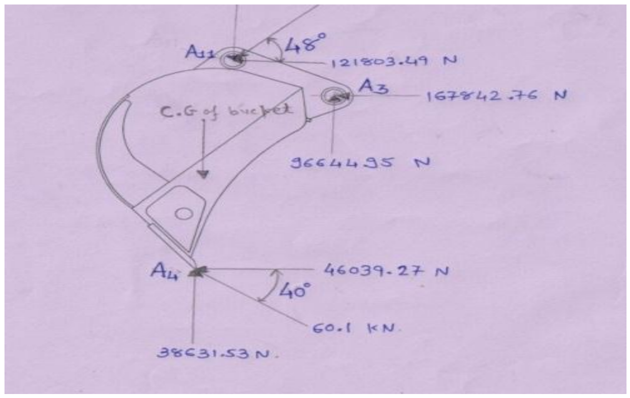


Fig. 2: Free Body Diagram of Bucket

Table - 1
Forces on Bucket

Joint of the bucket	*Forces (N) Horizontal (X) component*	*Forces (N) Vertical (Y) component*
A_4	-46039.27	38634.53
A_{11}	-121803.49	-135276.48
A_3	167842.76	96644.95

The negative sign shows the force acting in the leftward direction for horizontal component of the force and downward direction for vertical component of the force.

B. Arm Static Force Analysis:

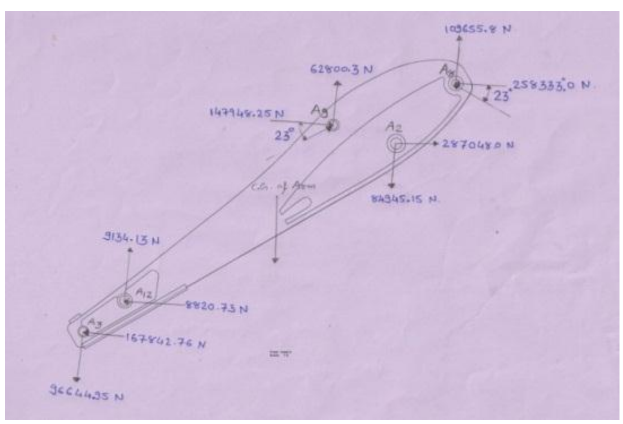


Fig. 3: Free Body Diagram Of Arm

The Force (F_{12}) is the force acting on the intermediate link ($A_{10}A_{12}$) from the idler link ($A_{11}A_{10}$) at an angle β_{10} of 86º

$F_{12}=F_{11} \cdot \cos \beta_{10}$

$=12697.94$ N

$F_{12H}=F_{12} \cdot \cos \beta_{12}$

$=8820.73$ N

$F_{12V}=F_{12} \cdot \cos \beta_{12}$

$=9134.13$ N

The force F_9 is acting on arm through the bucket cylinder, at an angle β_{10a} of 28º $F_9=F_{11} \cdot \cos \beta_{10a}$

$=160725.12$ N

The force F_9 can be resolved in horizontal (X) and the vertical (Y) directions. Here, β_9 is the angle made by force on arm through bucket cylinder with horizontal reference of 23º

$F_{9H}=F_9 \cdot \cos \beta_9$

$=147948.25$ N

$F_{9V}=F_9 \cdot \cos \beta_9$

$=62800.3$ N

Considering the arm in equilibrium ΣM = 0 and taking moment about the arm to boom hinge point (A2) leads to;

$F_8 \cdot l_8= -(F_{3V} \cdot l_{3H})- (F_{ga} \cdot l_{ga})+ (F_{3H} \cdot l_{3V})+(F_{12} \cdot l_{12})+(F_9 \cdot l_9)$

$F_8 = 280642.76$ N

$F_{8H}= F_8 * \text{Cos } \beta_8$

$= 258333.027$ N

$F_{8V}= F_8 * \text{Sin } \beta_8$

$= 109655.8$ N

Table - 2
Point Forces on Arm

Joint of the bucket	*Forces (N) Horizontal (X) component*	*Forces (N) Vertical (Y) component*
A_3	*-167842.76*	*-96644.95*
A_{12}	*-8820.73*	*9134.13*
A_9	*147948.25*	*62800.3*
A_8	*-258333.0*	*109655.8*
A_2	*-287048.0*	*84945.15*

C. Boom Static Force Analysis:

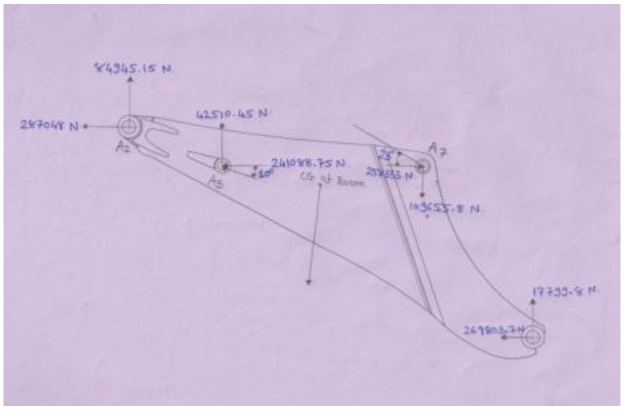


Fig. 4: Free Body Diagram of Boom

force F_7 is the force acts by arm at point A_7 through arm cylinder which is same as the force F_8 but direction is opposite. The force F_7 can be resolved in horizontal (X) and the vertical (Y) directions Here, β_7 is the angle made by force on boom through arm cylinder with horizontal reference at point A_7 of 23º

$F_{7H}=258333.027$ N

$F_{7V}=109655.8$ N

Considering the boom in equilibrium ΣM = 0 and taking moment about the arm to boom hinge point (A_1) leads to;

$(F_5 * l_5)= - (F_{2H} * l_{2V}) - (F_{gbo} * l_{gbo}) + (F_{2V} * l_{2H}) + (F_7 * l_7)$

$F_5 = 244807.94$ N

$F_{5H}=F_5 \cdot \cos \beta_5$

$=241088.75$ N

$F_{5V}=F_5 \cdot \cos \beta_5$

$=42510.45$ N

The forces on each of the joints of the boom are shown in table 7.3.

Table – 3
Point Forces on Boom

Joint of the bucket	*Forces (N) Horizontal (X) component*	*Forces (N) Vertical (Y) component*
A_2	*-287048.0*	*84945.15*
A_7	*258333.02*	*-109655.8*

A_5	-241088.75	42510.45
A_1	269803.7	17799.8

III. DESIGN CALCULATIONS

1) Old Material:

Pin: Carbon steel EN 9 grade

2) New Material:

Tooth and tooth plate of bucket: Hardox 400
Pin: Carbon steel EN 9 grade
Remaining parts: SAILMA 450 HI

A. Design Calculations of Pins Bucket Pin Design:

The bucket pin is checked for bearing, shear and bending failure.

- Diameter of bucket pin, db=60 mm
- Total length of the bucket pin, lpb=300 mm
- Effective length of the bucket pin, lpbe=190 mm

Let us first check the bucket pin for bearing failure. For the calculation, the effective length of the bucket pin (the portion of bucket pin which is in contact with the arm) is considered.

- The resultant force acting on the bucket pin is,

$Fpb=F3=\sqrt{F3H\ 2+\ F3V\ 2}$ F3= 186656 N

- Also, the bearing force acting on the bucket pin is,

  $Fpb=d_b \cdot l_{pbe} \cdot p_b$

  P_b= 186656 / (60*190)

  =16.37 Mpa

Here, the bearing pressure induced in the bucket pint is very less compared to the allowable bearing pressure of bucket pin material carbon steel EN 9 as 130 MPa. So, the bucket pin is safe in bearing

- Considering bucket pin is in double shear and the shear area of the bucket pin is, $A_{pb}=2\cdot\pi/4\cdot(db)^2$

Now, shear force acting on the bucket pin is,

$F_{pbfs}=2\cdot(\pi/4)\cdot(d_b)^2\cdot\tau_{shear}$ $\tau_{shear}=(2*186656)/(\pi*60^2)$

= 33.009 Mpa

Here, the shear stresses developed in the bucket pin is very less compared to allowable shear stress of bucket pin material steel EN 9 as 160 MPa. So, the bucket pin is safe in shear.

Consider bucket pin as simply supported beam with uniformly distributed loading condition as shown in Fig. 5.1 and now let us check bucket pin for bending failure.

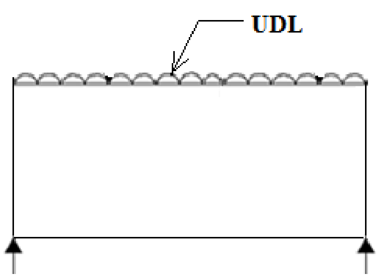


Fig. 5: Pin with UDL Condition

$\sigma_{max}=(M_b * Y_{max})/I_b$

$=(4*F_b*L)/(\pi*d^3)$

= 209.057 Mpa

The maximum bending stress developed in the bucket pin is within the stress limit of the bucket pin material, therefore the design of bucket pin is safe in bending.

By using the above equations for all pins, obtained results are plotted in below table.

Table - 4
Pin Calculations

Pin	*Axial force N*	*Old pin dia. Mm*	*New pin dia. Mm*	*Total length mm*	*Effective length mm*	*Maximum Bearing pressure, Mpa*	*Maximum shear stress Mpa*	*Maximum bending stress Mpa*
A1	*270390*	*70*	*70*	*449*	*265*	*14.57*	*35.13*	*265.99*
A7	*280642*	*55*	*55*	*318*	*120*	*42.52*	*59.06*	*257.7*
A10	*182032*	*45*	*50*	*252*	*102*	*35.69*	*46.35*	*184.13*
A11	*182032*	*55*	*60*	*326*	*190*	*15.96*	*32.19*	*203.87*
A12	*12697*	*40*	*40*	*316*	*200*	*1.58*	*5.05*	*50.52*
A3	*186656*	*55*	*60*	*300*	*190*	*16.37*	*33.09*	*209.57*
A8	*280642*	*60*	*65*	*296*	*176*	*24.53*	*42.28*	*229.06*
A5	*244807*	*60*	*60*	*306*	*170*	*24.00*	*43.29*	*245.32*
A2	*299353*	*65*	*65*	*262*	*177*	*26.01*	*45.10*	*245.66*
A9	*160725*	*50*	*55*	*306*	*170*	*17.18*	*33.82*	*209.10*

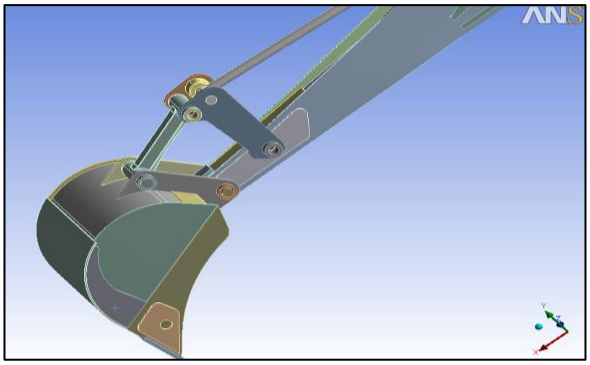
Above table shows the calculation for all pin which shows that all pins are more safe. By comparing the above results with the materials properties it is found that maximum bearing pressure, maximum shear stress and maximum bending stress is within the limit of allowable stresses.

B. Analysis of Backhoe Assembly:

1) *Scope of Analysis*

The Analysis report presented herewith deals with the Finite Element Analysis of the backhoe. It presents the structural analysis of the backhoe assembly.

IV. GEOMETRY



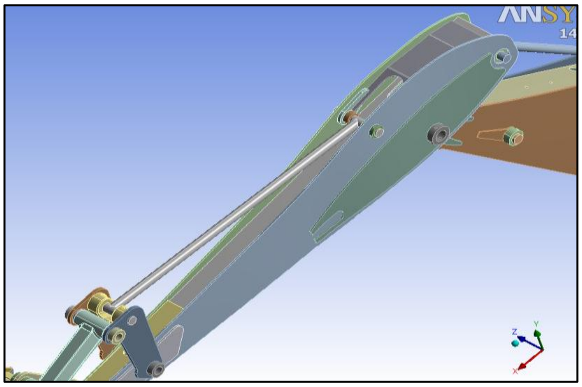


Fig. 6: Views of Backhoe Assembly

A. Load Case:

Structural Analysis has been performed by applying the breakout force at the teeth of bucket.

1) *FEA Details – Mesh Details:*

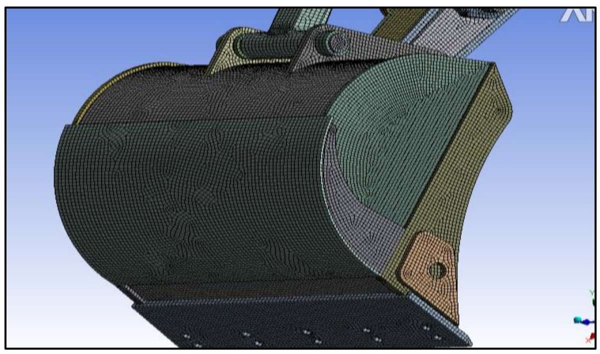
Table – 5
FE Model Summary

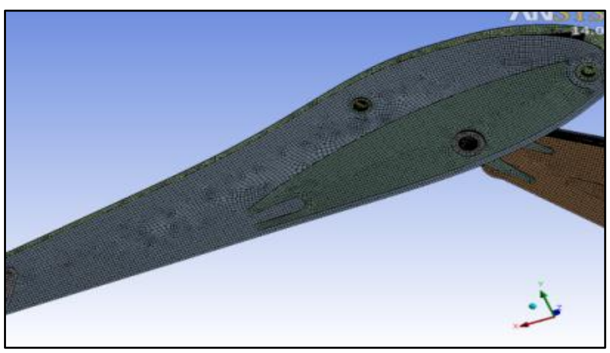
Description	*Quantity*
Total Nodes	*1136758*
Total Elements	*269741*

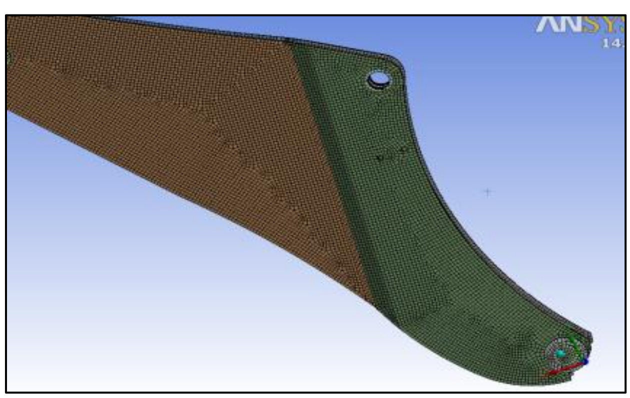
Table – 6
Elements Type Summary

Generic Element Type Name	*Ansys Name*	*Description*
10 Node Quadratic Tetrahedron	*Solid187*	*10 Node Tetrahedral Structural Solid*
20 Node Quadratic Hexahedron	*Solid186*	*20 Node Structural Solid*
20 Node Quadratic Wedge	*Solid186*	*20 Node Structural Solid*
20 Node Quadratic Pyramid	*Solid186*	*20 Node Structural Solid*

V. MESHING OF ASSEMBLY







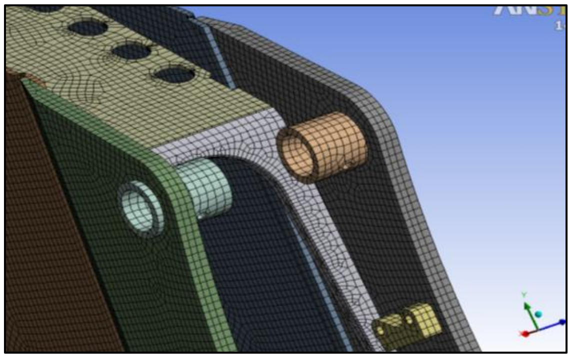


Fig. 7: Meshing of Assembly

- Constraints

Following Constraints are applied in order to simulate the actual Joint for FEA:

A. Bonded Contact Between Each Part To Simulate Welded Joint:

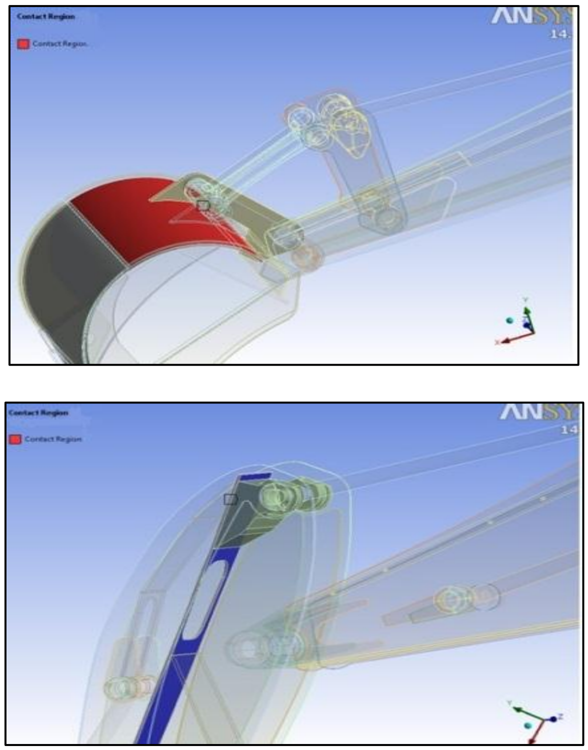
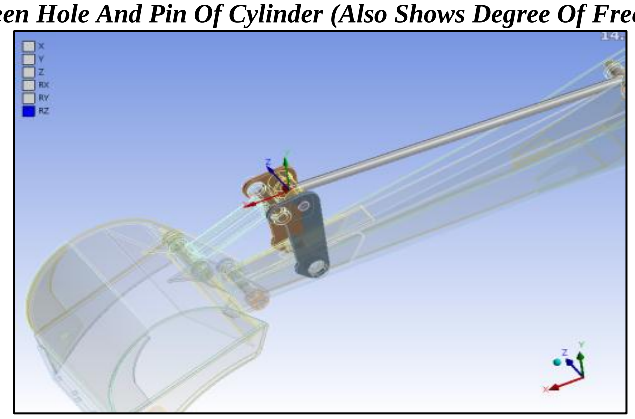


Fig. 8: Applied Constraints 1

B. Rigid Link Connection Between Hole And Pin Of Cylinder (Also Shows Degree Of Freedom Of Part):



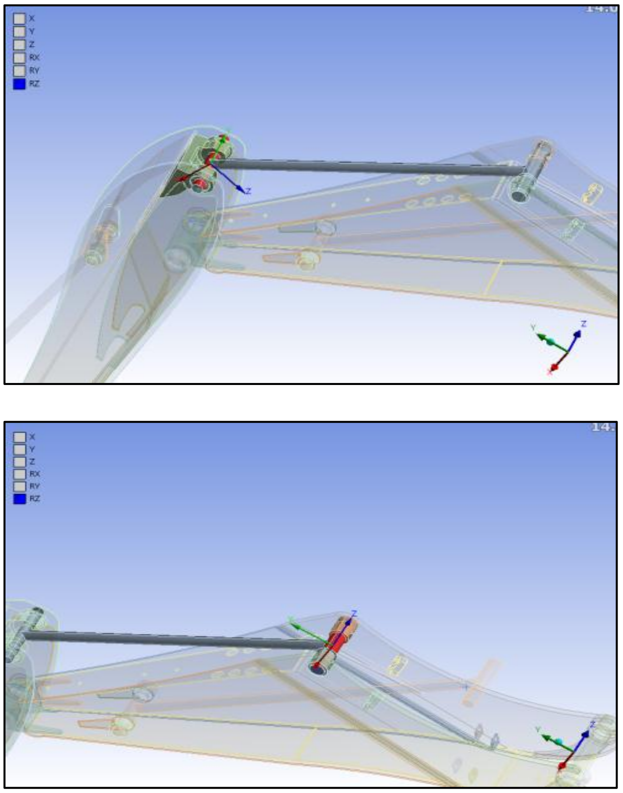


Fig. 9: Applied Constraints 2

Here each pin constrained to rotate about Z axis only. Also cylinders are considered as a rigid link.so rotation about X and Y are zero.so each elements are allowed to rotate in XY plane with respect to Z axis.

1) *Loads:*

Load Case : Structural analysis by applying breakout force at the teeth location

2) *Fix support Self Weight:*

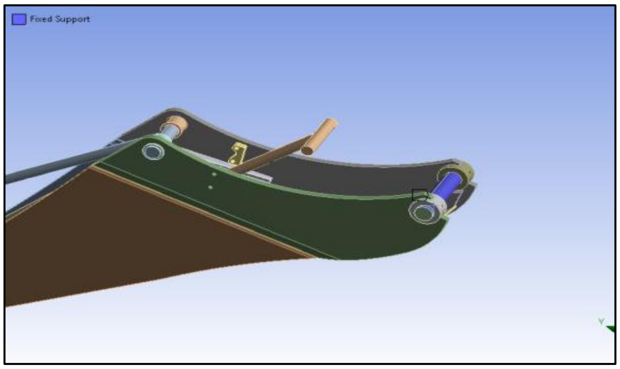


Fig. 10: Fixed Support

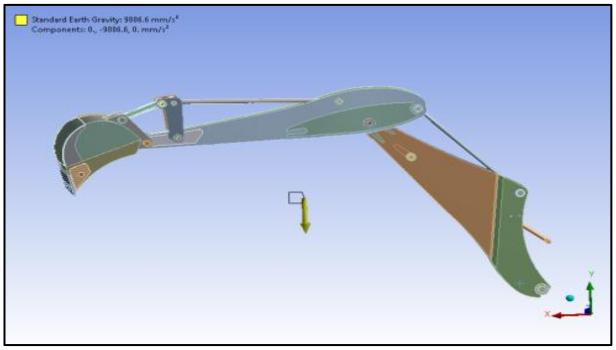


Fig. 11: Self-Weight

3) *Breakout Force:*

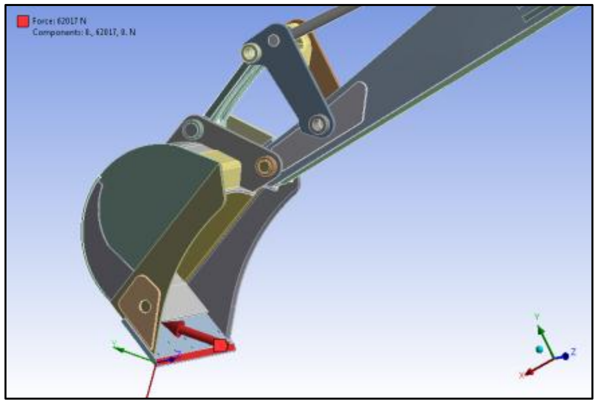


Fig. 12: Applied Breakout Force

C. Results:

Load Case - Total Deformation

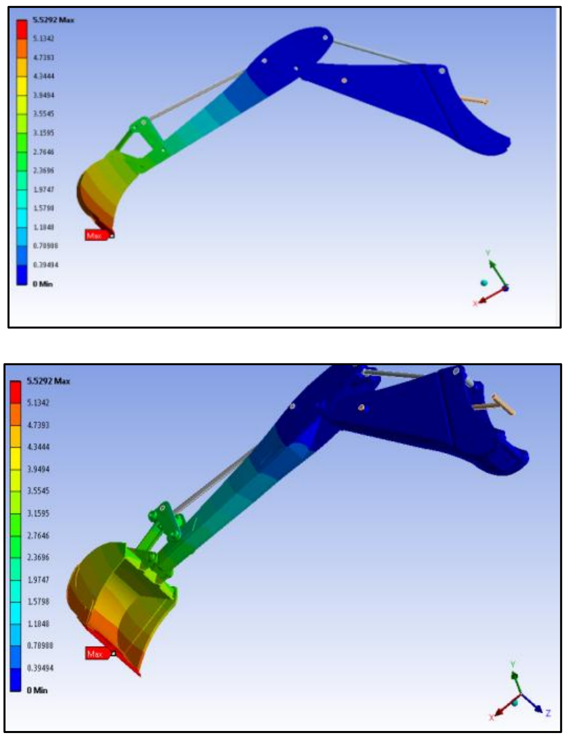
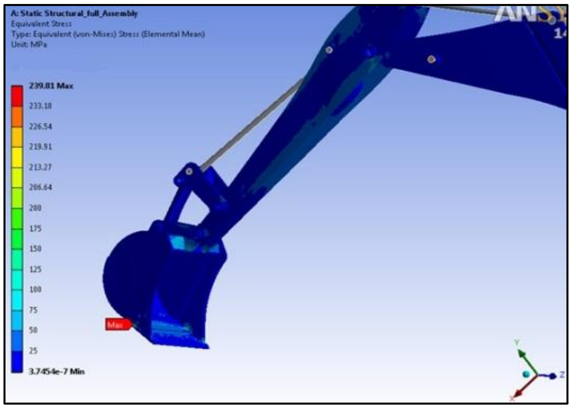


Fig. 13: Total Deformation

D. Equivalent Stress:



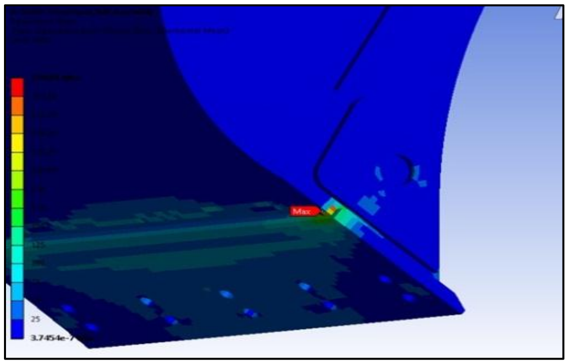


Fig. 14: Equivalent Stress (Von Mises Stress)

VI. RESULTS

1) 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.
2) The total deformation at the bucket teeth point is 5.5292 mm which is negligible.
3) By increasing pin diameter at critical loading points strength of the assembly is increased.
4) Variation in pin diameter is shown in table
5) By changing the material of the components of backhoe and by trial and error method in ANSYS following results are obtained

Table - 7

Component	*Modifications*		*Total weight (Kg)*		
	Thickness before modification mm	*Thickness after modification mm*	*Weight before optimization*	*Weight of optimized model*	*% Reduction*
Bucket	*No change in bucket thickness so as to strengthen it more.*		*240*	*240*	*0%*
Dipper	*12 mm side plates.*	*10 mm*	*337*	*285*	*52 kg 15%*
	8 mm top and bottom covers.	*7 mm*			
Boom	*25 mm side plates at pivot joint*	*24 mm*	*380*	*340*	*40 kg 11 %*
	12 mm side plate	*10 mm*			
	8 mm top and bottom covers	*7 mm*			

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