Analysis of Bucket Teeth of Backhoe Excavator Loader and its Weight Optimization

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Abstract— **An excavator is heavy construction equipment, which is commonly used in construction work, mining work etc. According to off highway research about Equipment analysis in India , backhoe loader market continuously growing. One of the parameters affecting the productivity of machine is bucket tooth. A poor design of teeth results in poor excavation and ultimately reduces its efficiency. Teeth are the parts of excavator, which always offer negligible attention by excavator manufacturing industry. For maximum digging force condition the evaluated digging force can be used as boundary condition and loading condition to carry out static finite element analysis of excavator bucket tooth. This paper is mainly concentrate on backhoe excavator loader on wheel. For general backhoe loader digging forces find out by SEA J1179 standard. According SEA J1179 arm cylinder crowd force and bucket cylinder curl force calculated and maximum bucket cylinder curl force is used for critical analysis of excavator bucket tooth at maximum breakout force condition. General purpose excavator bucket tooth like standard bolt on tooth and heavy duty fanggs digging tooth, long tooth, tiger tooth ,twin tiger tooth and abrasion tooth are analyzed for their smallest series dimension and by observing their von mises stresses , total deformation patter necessary optimization are suggested and checked.**

Keywords— Analysis bucket teeth, backhoe loader

I. INTRODUCTION

An excavator is heavy construction equipment, which is commonly used for construction work and in mining industry for digging holes, trenches, foundation and other things. Design of backhoe link mechanism is tough task by considering digging force developed through actuator during digging operation must be greater than resistive force offered by ground. The teeth are main contacting part of it, which is first come in contact with ground, while doing excavation, so total force pass through teeth, therefore more damage occurs to teeth.

The backhoe hydraulic excavator or backhoe loader is by far the most popular construction machine in India. According to 'Equipment analysis : India backhoe loader'[7] report backhoe loader accounting for around 45 per cent of the mobile construction machinery market. This market still expected rise as Indian government focuses on developing the country's infrastructure in future.

Society of Automotive Engineer (SAE) is the Engineering Society For Advancing Mobility Land Sea Air and Spaces, Warrendale. define a Surface Vehicle Standard SAEJ1179 for excavator backhoe link mechanism with the help of this we can find out excavation forces. Bhaveshkumar P. Patel [3],consider backhoe kinematics, differential motion, static and dynamic model of backhoe mechanism to develop development of generalized breakout and digging force model. Examination of an optimized replaceable cutting tooth for rock cutting process is done by Zoltan Virag [5]. Also Jan Maciejawski and Andrej Jarzebowski[4], check the influence of teeth on the earth working process experimentally for change in geometry and materal.

II. PROBLEM DEFINITION

There is no design procedure present to design excavator bucket tooth. So better teeth design in the excavation process has been always a challenging task for the engineers. A poorly designed teeth always results in poor excavation of the ground, higher wear of the tool, wastage of the time, and power, and thus reducing the overall productivity of the excavation operation. Manufactures of excavators only consider no. of teeth required for their bucket and weight of teeth. Also tooth manufactures provides unnecessary extra material for wear compensation and lack of design procedure ,they only follow their previous experience results. This paper mainly concentrate on backhoe excavator loader bucket teeth analysis and their optimization process.

III. BACKHOE DIGGING FORCES ACCORDING TO STANDARD SAE J1179

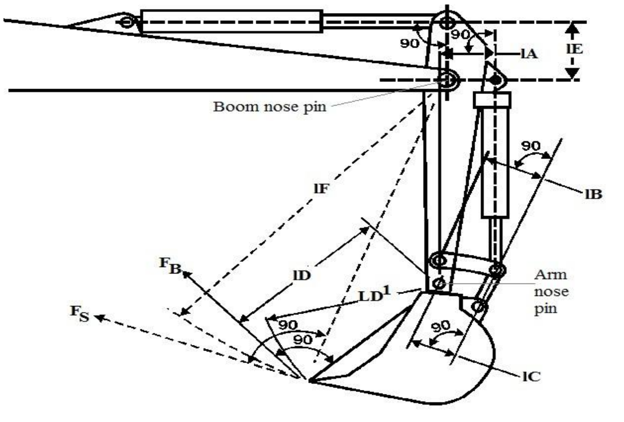


Fig. 1 Determination of digging forces by following the standard SAE J1179

Bucket penetration into a material is achieved by the bucket curling force (F_B) and arm crowd force (F_S). The rating of these digging forces is set by SAE J1179 standard "Surface Vehicle Standards - Hydraulic Excavator and Backhoe Digging Forces" (SAE International, 1990). These rated digging forces are the forces that can be exerted at the outermost cutting point (that is the tip of the bucket teeth). These forces can be calculated by applying hydraulic pressure to the cylinders providing the digging force.

Fig.1 shows the measurement of bucket curling force F_B, arm crowd force F_S, the other terms in the figure lA, lB, lC, lD, lD^1, lE, and lF shows the distances of different part.

According to SAE J1179: Maximum radial tooth force due to bucket cylinder (bucket curling force) F_B is the digging force generated by the bucket cylinder and tangent to the arc of radius lD^1. The bucket shall be positioned to obtain maximum output moment from the bucket cylinder and connecting linkages. F_B becomes maximum when distance lA reaches maximum, because rest of the distances in (1) are constant.

$$F_B = \frac{Bucket\ cylinder\ force}{l_D}\left(\frac{l_A \times l_C}{l_B}\right) \quad (1)$$

Where, *Bucket cylinder force = (Working pressure) × (End area of bucket cylinder)*

If the end diameter of the bucket cylinder = DB (mm) and the working pressure is P (MPa) and other distances are in mm then (1) can be written as:

$$F_B = \frac{P \times (\pi/4) D_B{}^2}{l_D}\left(\frac{l_A \times l_C}{l_B}\right) \quad (2)$$

Equation (2) determines the value of the bucket curl or breakout force in 'N'. Now let us determine the maximum radial tooth force due to arm cylinder F_S. Maximum tooth force due to arm cylinder is the digging force generated by the arm cylinder and tangent to arc of radius dF. The arm shall be positioned to obtain the maximum output moment from the arm cylinder and the bucket positioned as described in the case of maximum bucket curl force (Max. bucket tangential force). While calculating maximum force F_S occurs, when the axis in the arm cylinder working direction is at a right angle to the line connecting the arm cylinder pin and the boom nose pin as shown in Fig. 1.

$$F_S = \frac{Arm\ cylinder\ force \times l_E}{l_F} \quad (3)$$

$$F_S = \frac{P \times (\pi/4) D_A{}^2 \times l_E}{l_F} \quad (4)$$

Where, *lF = bucket tip radius (lD) + arm link length and DA = end diameter of the arm cylinder.*

A. Backhoe loader excavator

Fig. 2 Backhoe loader excavator

lA = 478 mm,

lB = 409 mm,

lC = 336 mm,

lD = 1017 mm,

lE = 530 mm, and

lF = 2362 mm.

Working pressure P = 210 bar or 21 MPa,

DA = DB = 90 mm

Bucket curl or breakout force F_B = 51584 N and

Arm crowd force or digging force F_S = 29977 N

IV. BUCKET TEETH OF EXCAVATOR LOADER

Considering different types of teeth and their use from [8] and [6] here following types of teeth selected for analysis

Type	Description	Image
Std. bolt on teeth	For general working condition	
Fanggs digg teeth	For general-purpose to heavy-duty digging conditions. Provide a clean start and close crowd. Flat bottom for improved bucket control.	
Long teeth	Use in high abrasion application Added wear material in tip for longer wear life.	
Tiger teeth	Long, aggressive teeth designed for breaking extremely tight soil. Used primarily as center teeth	
Twin tiger teeth	For densely compacted material. Use primarily as a corner tooth but can be used in center.	
Abrasion teeth	Use where extended tip life is needed in highly-abrasive conditions. Heavy-duty teeth have more wear material than the standard teeth.	

J.E. Farnandez et al. [2] present paper on material selection to excavator teeth in mining industry from that it is concluded that alloys on basis of Chromium , Niobium, Vanadium and boron have low wear ratios. Here high wear resistance Hardox alloy steel [9][10] is used.

Table 1 Material properties

Properties	Alloy steel Hardox 400	Alloy steel Hardox 500
Density (kg/ m3)	7850	8050
Modulus of elasticity (MPa)	2.1*105	2.1*105
Poisson's ratio	0.29	0.29
Yield strength (MPa)	1000	1300
Ultimate tensile strength (MPa)	1250	1550
Impact toughness (J)	30	30
Brinell hardness	370-500	470-540

Standard bolt on teeth

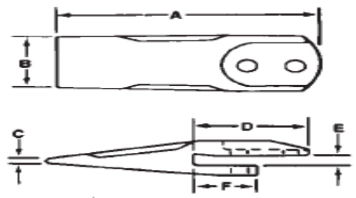


Fig. 3 Standard bolt on tooth

A	B	C	D	E	F	G	Hole spacing
324	51	5	67	24	184	23°	101

Caterpillar - specifications

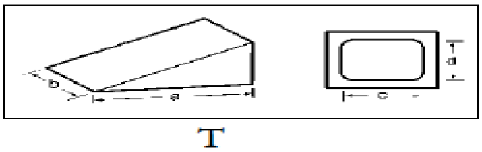


Fig. 4. Caterpillar Tooth

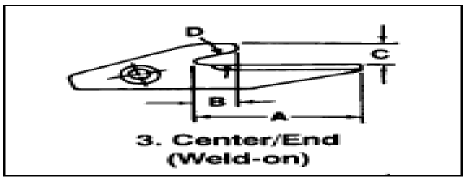


Fig. 5 .Caterpillar Adaptor

❖ Series 20

Style	Description	a (mm)	b (mm)	c (mm)	d (mm)
T	Fanggs dig	149	44	36	47
T	Long	144	51	36	47
T	Tiger	165	point	36	47
T	Twin tiger	165	70	36	47
T	Abrasion	144	55	36	47
3	Adaptor	100	25	25	23°

These are lowest series dimension selected for analysis , which involves lowest volume of material.

V. STATIC FORCE ANALYSIS

Bhaveshkumar P. Patel [4],consider backhoe kinematics, differential motion, static and dynamic model of backhoe mechanism to develop development of generalized breakout and digging force model. There is no need to analyze every position and orientation (collectively known as the configuration) of the mechanism from the available breakout and digging forces, in static analysis one configuration of the mechanism has to be decided first for which the analysis is to be carried out. From all the configurations, the maximum breakout force condition is the most critical one as it produces the highest breakout force, and thus for this condition the force analysis is done, and will be used as a boundary condition for static FEA.

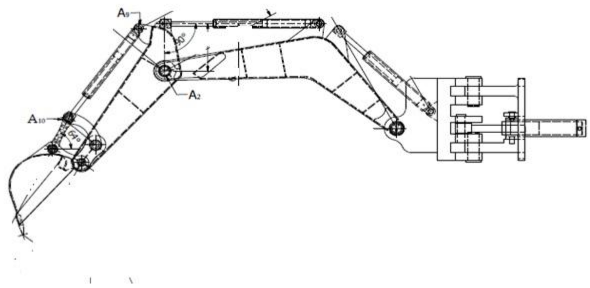


Fig.6 Maximum breakout force configuration[5]

For this loader, Bucket curl or breakout force F_B = 51584 N and Arm crowd force or digging force F_S = 29977 N , F_B is greater than F_s therefore it is selected for static analysis of tooth. Consider bucket have 3 no. of tooth, therefore total force applied on each tooth end is 51584/3= 17195 N at an angle 32° with plane which hold tooth as shown in fig. 7. For critical analysis of teeth force applied to tooth cutting face and for standard bolt on teeth plate is stationary and for other types of teeth adaptor faces which are going to weld with bucket are fixed stationary as shown in fig.8.

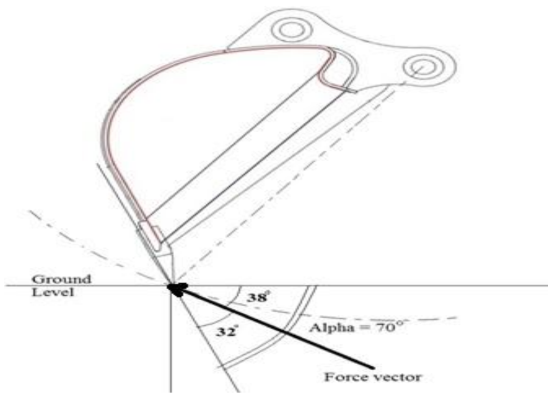


Fig.7 Free body diagram of bucket

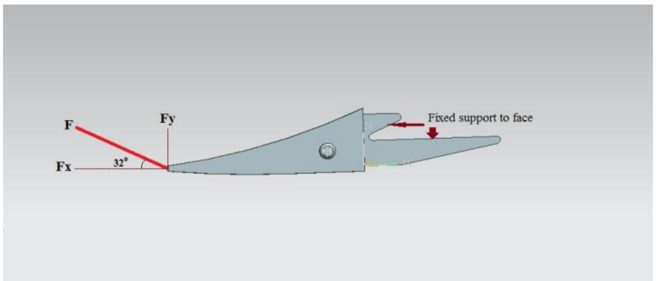
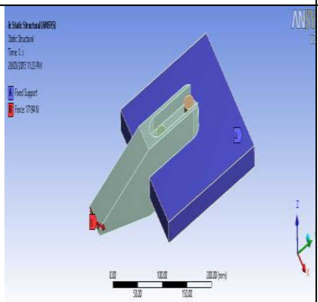
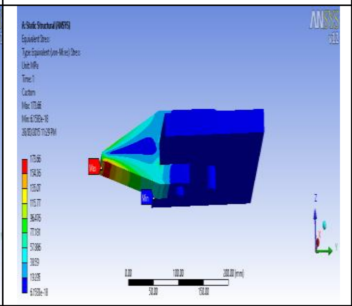
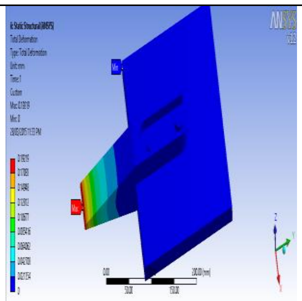
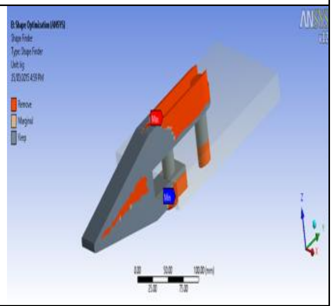
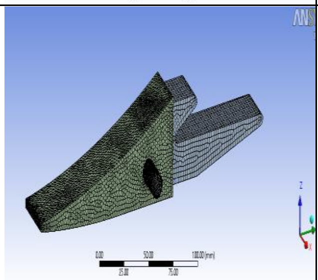
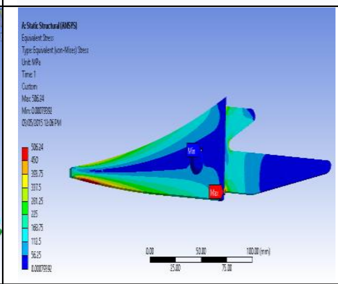
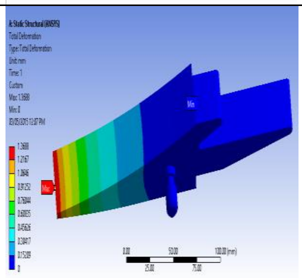
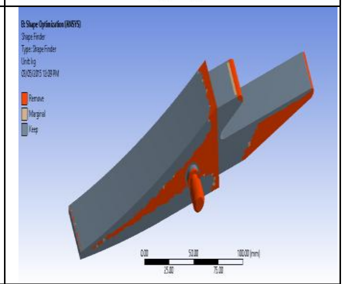
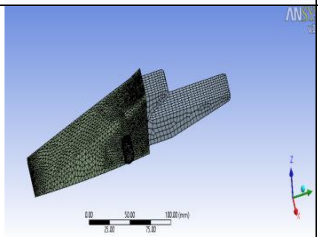
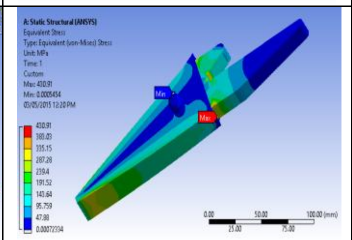
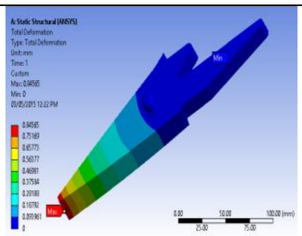
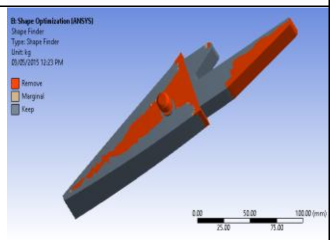
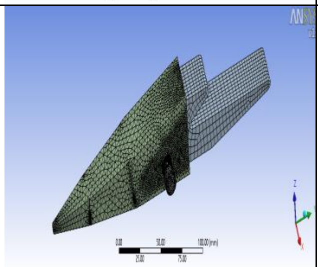
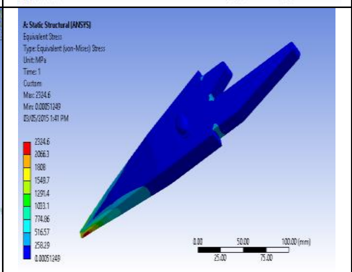
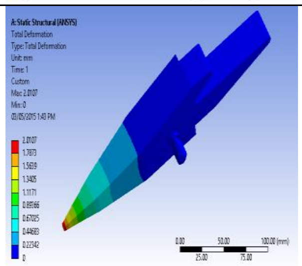
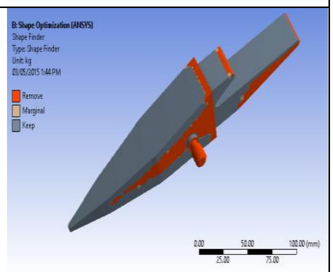
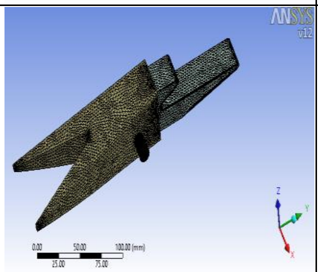
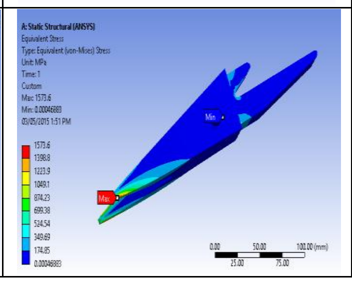
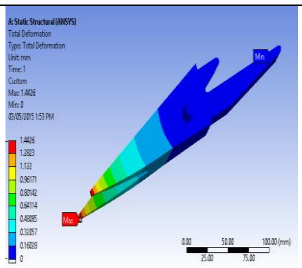
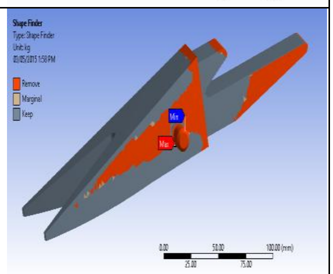
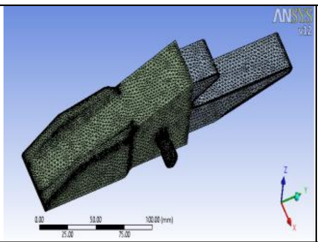
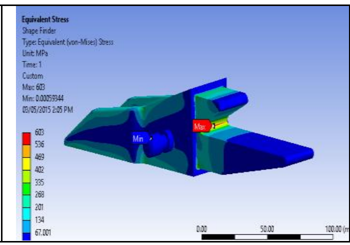
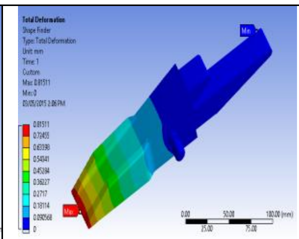
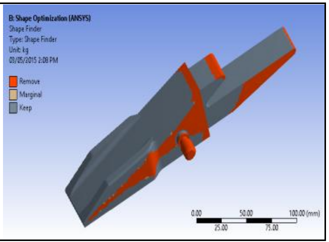


Fig. 8 Boundary condition for fanggs dig tooth

Table 2 Analysis of teeth in ANSYS workbench 12

	Boundary condition/ Meshed	Equivalent von-Mises stresses	Total deformation	Shape optimization
A				
B				
C				
D				
E				

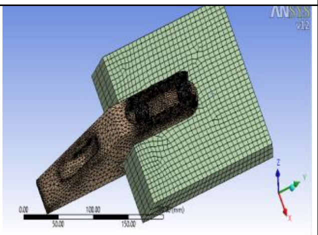
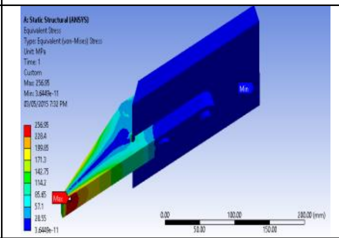
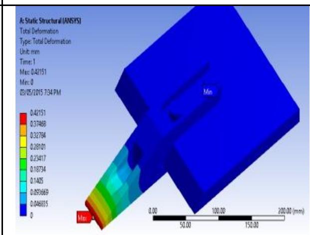
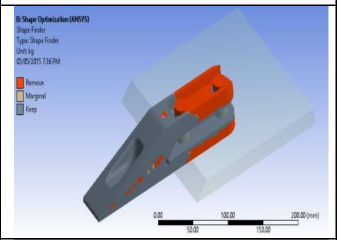
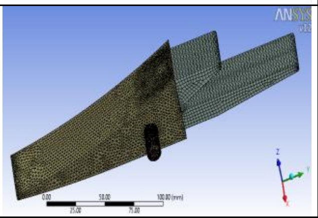
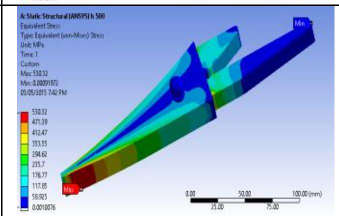
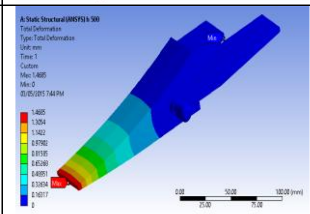
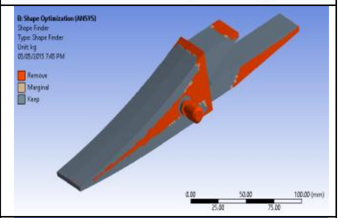
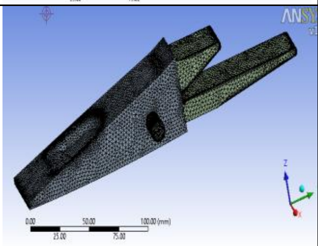
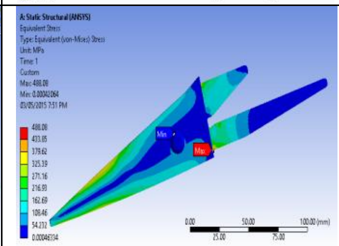
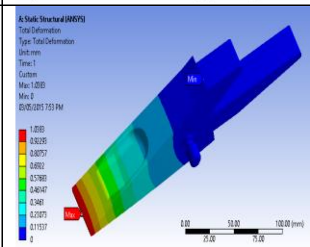
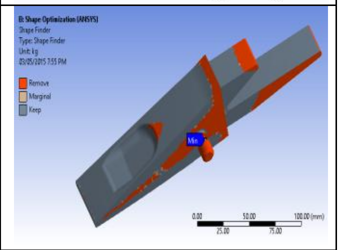
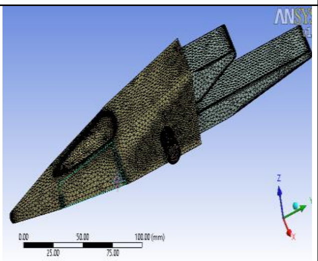
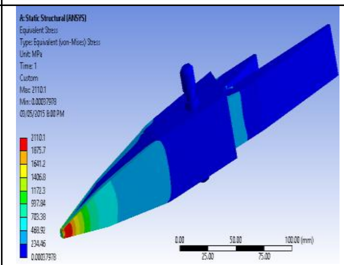
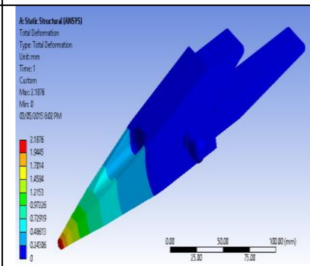
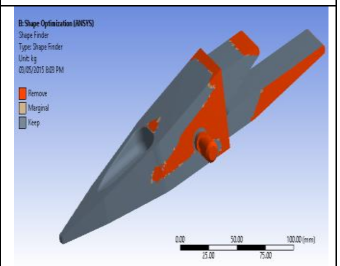
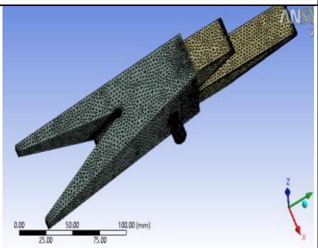
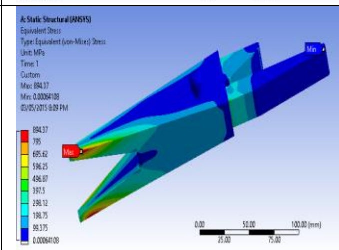
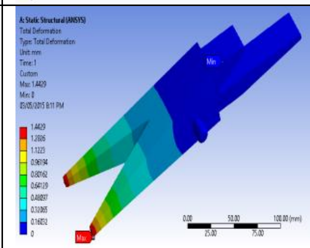
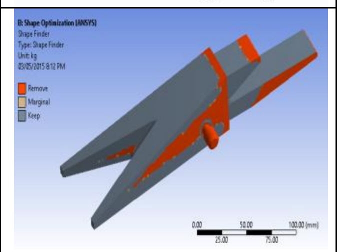
F				

A-General purpose bolt on tooth, B- Fanggs dig tooth, C- Long tooth, D- Tiger tooth, E- Twin tiger tooth, F- Abrasion tooth

After analysis , clearly seen that at different type of tooth stress concentration at different location. After understanding stress distribution ,deformation and result shown by shape optimization but result shown by shape optimization tool of ansys we can directly implement because it affects in bucket filling process.

VI. OPTIMIZATION

Table 3 Analysis of optimized teeth in ANSYS workbench 12

	Meshed	Equivalent Von-Mises stresses	Total deformation	Shape optimization
G				
H				
I				
J				
K				

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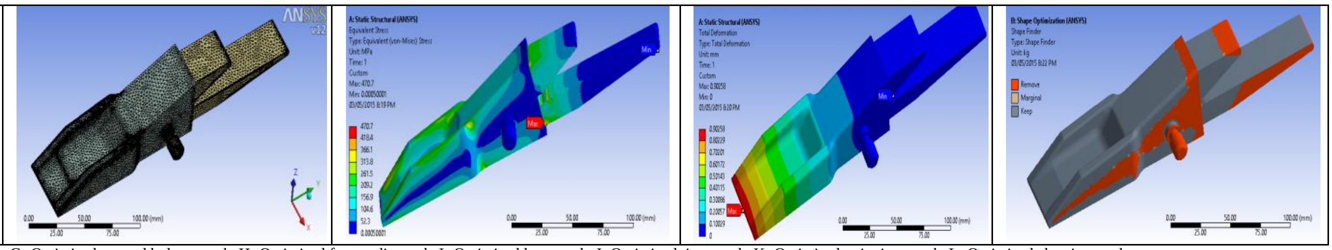

G- Optimized general bolt on tooth, H- Optimized fanggs dig tooth, I- Optimized long tooth, J- Optimized tiger tooth, K- Optimized twin tiger tooth, L- Optimized abrasion tooth

VII. Results and Discussion Table and Graph

Alloy steel (Hardox 400) for standard bolt on teeth

Safe stress = (Maximum Yield strength)/(Factor of safety)

= 1000/2

= 500 Mpa

If σVM< σ , so the design of tooth is safe.

Alloy steel (Hardox 500) for fanggs sig, long, tiger, twin tiger, abrasion

Safe stress = (Maximum Yield strength)/(Factor of safety)

= 1300/2

= 650 Mpa

If σVM< σ , so the design of tooth is safe.

Table 4 Result comparison between general and optimized /modified teeth

Type of tooth	Tooth Volume (mm^3)	Maximum total deformation (mm)	Maximum Equivalent von-mises stress (MPa)	Safe/ Unsafe
Std. Bolted	$2.5475*10^5$	0.19219	173.66	Safe
Modified Std. Bolted	$2.519*10^5$	0.42151	256.95	Safe
Fanggs dig	$1.769*10^5$	1.3688	506.24	Safe
Modified Fanggs dig	$1.715*10^5$	1.4685	530.32	Safe
Long	$2.026*10^5$	0.84565	430.91	Safe
Modified Long	$1.9047*10^5$	1.0383	488.08	Safe
Tiger	$1.9202*10^5$	2.0107	2324.6	Unsafe
Modified Tiger	$1.814*10^5$	2.1876	2110.1	Unsafe
Twin tiger	$2.299*10^5$	1.4426	1573.6	Unsafe
Modified Twin tiger	$2.2726*10^5$	1.4429	894.37	Unsafe
Abbrasion	$2.062*10^5$	0.81511	603	Safe
Modified Abbrasion	$1.9447*10^5$	0.90258	470.7	Safe

VIII. Conclusion

- Stresses below yield strength obey Hook's law, so deformation in elastic limits.
- Forces and boundary condition validated according to paper Zoltan Virag.[8]
- Sharp corner in load carrying area give rise stresses nearly 3-4 times as compare to surrounding, so avoid it.
- From results it can be seen that stresses are till below safe stress/ allowable stress value so more material can be remove. In case of tiger and twin tiger teeth stresses are above safe stresses but this type of teeth are used for excavation of densely compacted soil/material normally in such cases teeth are not subjected to such high force so modificaton can be done.
- In this paper , analysis is done for maximum force which can be generated by system/ backhoe link mechanism but in working condition system generate only required amount of force if excavation process required more force than system generate then system stop working.

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