Computer Aided Structural Analysis of Axle Tilting Effect on Tractor Front Axle Support

Aduloju, Sunday Christopher*,
National Engineering Design Development Institute, Nnewi, Anambra State, Nigeria
Email: chrisaduloju@yahoo.com
Mgbemena, Chinedum Ogonna,
Department of Mechanical Engineering, Federal University of Petroleum Resources, Effurun, Delta State, Nigeria
Email: mgbemena.ogonna@fupre.edu.ng
Ebhota, Williams Saturday,
National Engineering Design Development Institute, Nnewi, Anambra State, Nigeria
Email:willymoon2001@yahoo.com
Bolarinwa, Gabriel Oladeji
National Engineering Design Development Institute, Nnewi, Anambra State, Nigeria
Email:dejiino@yahoo.com

Abstract

Studies were carried out to determine the worst load case scenario on Ursus 3512 60hp Tractor Front Axle Support using MATLAB to analyze 36 load cases. The worst load case scenario was found to be the dynamic load case of 3g with wheel force of 9810N and axle angle of 12^0.The load on the Pivot hole of the tractor support was found to be 56057N. The component was modelled using Pro-Engineer (now CREO Elements). The structural analysis of component was done with the worst load case using Pro-Mechanica. The working or design stress was generated from the analysis within best convergence limit. The estimated maximum limiting yield stress estimated as von Mises stress of the front axle support is 88 N/mm^2. The factor of safety for the component design was calculated to be 2.84 which were lower than the recommended value of 5.5 to 6.5. The Minimum allowable stress and Maximum allowable stress for the Component design would be 484N/mm^2 and 572N/mm^2.

Keywords: Tractor; structural analysis; Matlab; Pro-Engineer; Pro-Mechanica; Factor of Safety; Front Axle support.

1. Introduction

The tractor is an off road vehicle employed for use in farming, haulage, transportation, heavy earth moving etc. The tractor has a front axle which supports the whole body at the front which is known as 'dead axle'. The front axle and its supports encounter the worst load conditions during operations.

Several studies have been carried out by researchers on Agricultural Tractors. Tests, analysis and design modification have been carried out by field tests and more recently by Computer Aided Engineering Approach.

Späth [1] investigated the loads, which acts on a tractor chassis. For this purpose, an instrument wheel was designed and built which allows the wheel load, the draft and lateral force at a tractor rear wheel to be measured reliably and with sufficient precision. Experiments were carried out including driving over a ramp and axle loads were recorded.

The Institute of Agricultural Machinery of Technical University Munich [2] in the year 1999 did some studies to investigate the loads on the tractor body. The goals was to calculate the fatigue life of the tractor components. A tractor was equipped to measure the loads on the tractor axles and the loads on the three point hitch. The loads while working on the field and driving on the ISO-tracks were measured. A load spectrum was processed from the load time history by using rainfall cycle method. Finally, the total fatigue damage for the tractor components was calculated, considering damage accumulation hypothesis defined by Miner [3].

Yahya [4] developed an instrument and sensor system for an agricultural tractor. The system was constructed on the tractor and was capable of recording information for drawbar pull force, drive wheel torque and both vertical and horizontal forces at the 3-point hitches of the tractor implement. The data acquisition system utilized a designed draw bar pull transducer to measure horizontal pull at tractor drawbar point, wheel torque transducers to measure the torque at both tractor rear wheels, and a 3-point auto hitch dynamometer to measure the horizontal and vertical forces on the implement behind the tractor.

Al – Janobi et al. [5] developed a data acquisition system to measure various tractor performance parameters such as wheel forces, three-point linkage forces, drawbar force, PTO torque and ground speed. An electronic circuit was designed and fabricated to provide exact angular position measurement of the clevis bolts in the tractor wheels at any travel speed and scanning rate of the data logger used. This measurement was found to be more accurate and reliable as compared to the measurement by the marker pulse of the optical shaft encoder, which is dependent on the data logger scanning time. The force measurement from the clevis bolts combined with the angular position measurement gives the total horizontal and the vertical components of forces on the revolving wheel. An onboard data logger was used to sample signals from the various transducers as well as the angular position measurement circuit in the system. The system was field tested for its performance and found to be accurate and reliable in measurement of tractor performance parameters.

Balasubramanian et al. [6] developed a data acquisition system to measure vibration data from an agricultural tractor. Accelerometers were mounted on the front and rear axle and on the driver’s seat. An ultra sound ground speed sensor was used to monitor velocity. These sensors were connected to a National Instruments Modular Signal Conditioning Carrier (MSCC). Lab VIEW was employed to provide Graphical User Interface (GUI) to the data acquisition system; manage data acquisition and store ground speed and accelerometer signals. An obstacle course was constructed to stimulate rough field terrain. The tractor was driven through the course at prescribed velocities while data was created to mark the obstacles and provide a ground of old obstacles. Correlations could then be made between the known obstacle of varying magnitude and the measured shock and vibration events as a function of ground speed.

Redkar [7] did a study to prove that a good correlation exists between Virtual Simulation and Physical behaviour of machine components. A nonlinear behaviour of Tractor Front Axle was simulated using Abacus CAE, well known software for nonlinear simulation. Non linearity was simulated by putting the material nonlinear data in the material stress strain curve.

Mahanty et al. [8] in their study at Tata Industries employed CAE approach to analyze a new design of the front axle of an agricultural tractor. The geometric models for the existing design and proposed designs were created and imported into ANSYS. The proposed designs were evaluated for selected worst case scenarios of the tractor. Based on the finite element analysis results, redesign was carried out for the front axle.

Koyuncu [9] in his study did analysis of the FE model of the front axle support. He implemented for the load cases that are considered during the design of the front axle support. A set of load cases for the agricultural tractor front axle supports was selected by surveying the literature and consulting the designers of Erkunt Agricultural Machinery.

In this study, the cause of fracture in front axle support of tractors was detected using Computer Aided Engineering. Pro-Engineer was used to model the component; MatLab was used to analyse the forces and finally

Pro-Mechanica was used to perform FEA simulation of the fracture. This research work is focused on the Analysis of the Front Axle support of Ursus 3512 60hp agricultural tractor.

2. Methodology

2.1 *Component Modelling*

The dimensions of Ursus 3512 60hp tractor front axle support were measured and recorded. The dimensions were used in modelling the component in a virtual environment. Pro-Engineer software was used to achieve this purpose. The solid subtype – part type modelling environment was used. The component was therefore modelled as a .prt file.

2.2 *Component Properties*

The front axle support design was manufactured from GG-25, gray cast iron. Some analysis has been carried on the said material by Koyuncu [9]. GG-25 cast iron is a brittle material. The evaluations were done considering its ultimate tensile strength (uniaxial tension strength) value, which is 250MPa [9].

Table 1 The mechanical properties of GG-25 Cast iron

Material Properties	**Value**
Modulus of Elasticity	120000MPa
Poisson's Ratio	0.23
Ultimate Tensile Strength	250MPa

Figure 1 The Failed Tractor Front Axle Support

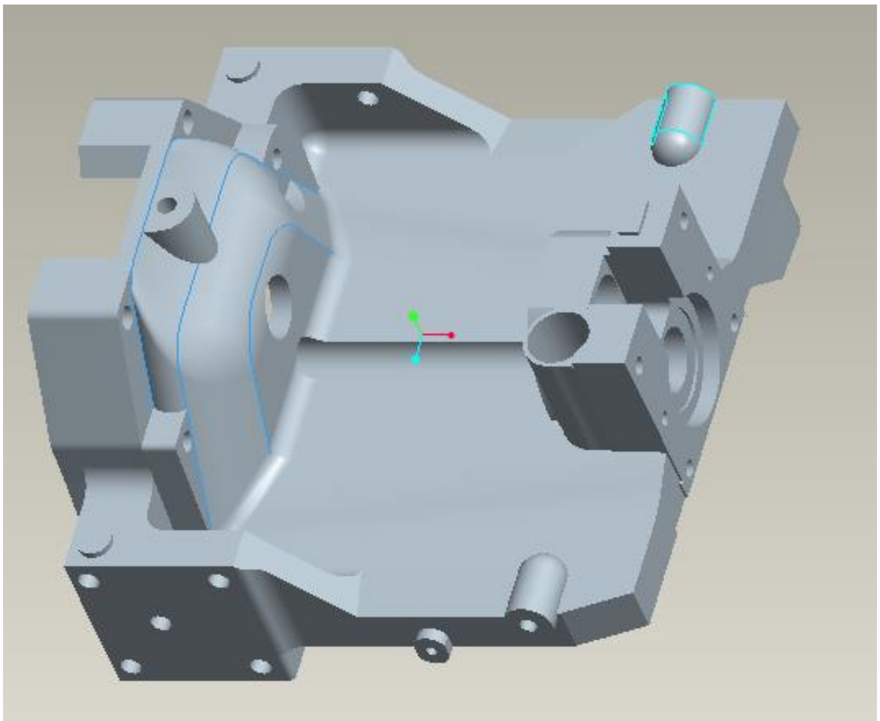

Figure 2 The modelled Tractor front Axle Support

2.3 *Analysis of Forces*

The design of the Tractor support by tractor companies has been done by bearing in mind the worst case scenario. The worst case scenario used in former designs was the load case during fatigue (3g loading case). This work therefore considers the possibility of a worst case that occurs when the front axle and front axle support make contact as a result of the 12^0 maximum rotation of the front axle during fatigue (contact load). A new free body diagram was developed from the initial diagram. This free body diagram includes the rotation of the front axle during the fatigue loads. Therefore a contact force (Fc) is applied to the front axle support on the contact location. The force components on the contact location and pivot shaft holes was calculated from the free body diagram of the front axle as presented in figure 3. The effect of the angular placement of the front axle support was evaluated by considering 36 load cases. The contact force and the reactions were calculated. The static loads and fatigue loads were analyzed as 1g load case and 3g load case. The front axle makes contact with the front axle support at 210mm length from the pivoting shaft and 400mm to the wheel. The design of the axle placement was done such that it is not being constrained by front axle support.

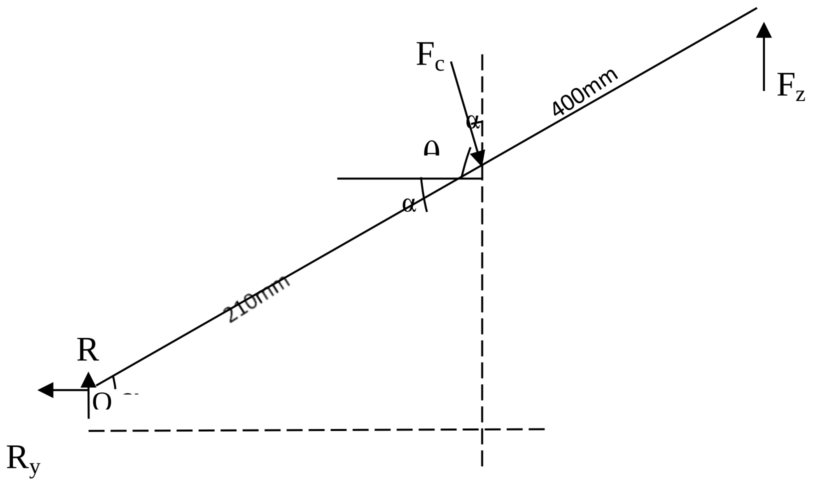

Figure 3 Free body diagram for Axle-Rotating Load Case

R_y = Reaction force acting on the front axle pivot shaft in the horizontal direction

R_z = Reaction force acting on the front axle pivot shaft in the vertical direction

F_{HR} = Force acting on the right front axle support pivot hole in the vertical direction

F_{HL}= Force acting on the left front axle support pivot hole in the vertical direction

F_c = Contact Force

F_z = total wheel force=force transferred from the front wheels to the front axle which in turn is transferred to the pivot shaft

$\theta=90-\alpha$

Resolution of forces along the vertical axis

Upward force = downward force

$R_z + F_z = F_c \text{ Cos } \propto$ (1)

Resolution of forces along the horizontal axis

$R_y = F_c \text{ Cos } \emptyset$ (2)

$\sum$ moment at point 0

$R_y=R_z=0$

$F_c \sin\propto(210 \sin \propto) + F_c \cos\propto (210 \cos\propto) = F_z (610 \cos\propto)$

$210 F_c \sin^2\alpha + 210 F_c\cos^2\alpha = 610F_z\text{Cos}\alpha$

$210F_c (\sin^2\alpha + \cos^2\alpha) = 610F_z \text{ Cos}\alpha$

Recall $\sin^2\alpha + \cos^2\alpha= 1$

$210F_c = 610F_z \text{ Cos}\alpha$ (3)

$$F_c = \frac{610F_z\text{Cos}\alpha}{210}$$

The total reaction on the axle pivot shaft is the summation of the forces on the right and left support pivot holes

$F_{HL} + F_{HR} = R_z$ (4)

But $F_{HL} = F_{HR}$

$R_z = 2 F_{HR}$ (5)

2.4 *Components Virtual Analysis*

Virtual analysis of the components was performed in this research work to provide solution to the problem of the tractor front axle failure using Matlab, Pro-Engineer and pro-Mechanica. Matlab was used to calculate and resolve the forces acting on the front axle support when the axle tilts at the angle of 1 to the maximum angle of 12 degrees. This is done to actually get the worst load case scenario.

Pro-Mechanica was used to simulate the stress variation, displacement and strains across the component. The software uses a finite element method in finding solution to problems. The properties of GG cast iron were assigned to the virtual component. The properties fed include the young modulus, the poisons ratio, the yield strength and density of the component.

In virtual analysis, attempts were made carefully to simulate the component to near real life case. Pro-Mechanica was very useful especially in the area of constraint definition. Since the component was bolted to other components/body the component constraint tool was used to simulate the bolt and nut connection experienced by the component at all necessary points.

2.5 *Load Cases*

The component was simulated under various load cases to determine the maximum stress and displacements that occur at various load cases of interest. The load cases of interests include:

i. Wheel reaction load case---------------9810N
ii. The 3g load case(fatigue)-----------------29430N
iii. The Axle-Rotating-3g---------------------56057N

2.6 *Finite Element Analysis of the Component*

Meshing is a crucial part in the finite element analysis. The component was meshed to give the following

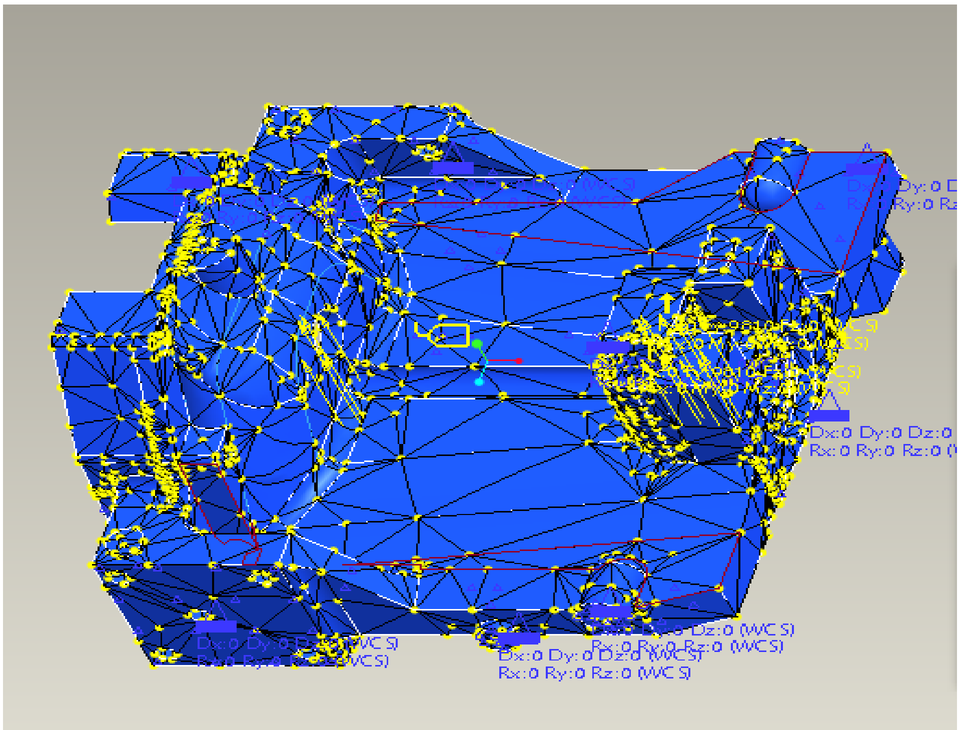

Figure 4 The meshed component using autogem tool.

Edges:	*28054*
Faces:	*39318*
Springs:	*0*

Masses:	*0*
Beams:	*0*
Shells:	*0*
Solids:	*17078*
Nodes:	*5801*
Elements:	*17078*

The convergence method used is Multi-pass Adaptive method with a polynomial order of 9 and percentage convergence limit of 10%. The analysis is also designed to converge on local displacement, Global Strain energy and global RMS stress.

2.7 *Factor of Safety*

For cast iron like other brittle materials, the yield point is not well defined as for ductile materials. Therefore the factor of safety for brittle materials is based on ultimate tensile stress. For this case a factor of Safety of 5-6 is recommended [10].

$$\text{Factor of safety} = \frac{\text{ultimate tensile stress}}{\text{Working or design stress}} \qquad (6)$$

3. Results and Discussions

3.1 Computer Aided Analysis of Forces

Table 2 shows the effect of angular variation on the forces and reactions on the axle support for static load case. For θ = 1, the minimum value for Fc and Ry as 28,500N and 497.39N.For θ =12, the maximum value for Fc and Ry as 29,132N and 6,057N.For θ=1-12, there was consistent value of 18,686N and force components F_{HL} and F_{HR} as 9342.9N.The initial value for Fg is equal to 9810N, Fc equals to 28500N, Ry equal to 497.39N and Rz equal to 18686N. The variations of the angle affected only Fc and Ry with Rz being constant all through the angular variation. An increase in θ leads to an increase in Fc and Ry.

Table 3 shows the effect of angular variation on the forces and reactions on the axle support for a load case of 2g. For θ = 1, the minimum value for Fc and Ry as 57,000N and 994.79N.For θ =12, the maximum value for Fc and Ry as 58,265N and 12,114N.For θ=1-12, there was consistent value of Rz as 37,371N and force components F_{HL} and F_{HR} as 18,686N.The initial value for Fg is equal to 19620N, Fc equals to 57000N, Ry equal to 994.79N and Rz equal to 37371N. The variation of the angle affected only Fc and Ry with Rz being constant all through the angular variation. An increase in θ leads to an increase in Fc and Ry.

Table 4 shows the effect of angular variation on the forces and reactions on the axle support for dynamic load case. For θ = 1, the minimum value for Fc and Ry as 85,500N and 1,492.2N. For θ=12, the maximum value for Fc and Ry as 87,397N and 18,171N. For θ=1-12, there was consistent value of 6057N and force components F_{HL} and F_{HR} as 28029N. The initial value for Fg is equal to 29430N, Fc equals to 85,500N, Ry equal to 1492.2N and Rz equal to 56057N. The variation of the angle affected only Fc and Ry with Rz being constant all through the angular variation. An increase in θ leads to an increase in Fc and Ry. A total of 36 load cases have been considered in this research work and the worst case scenario is the load case of 3g with wheel force of 9810N axle angle of 12^0. This worst case has Ry value of 18,171N and Rz value of 56057N.

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Table 2 Load case of 1g with angular variation of 1 to 12 degrees

Load Case = 1g, Wheel Force = 9810N, α = 1-12^0						
A	F_g (N)	F_c (N)	R_y (N)	R_z (N)	F_{HR} (N)	F_{HL} (N)
1	9810	28500	497.39	18686	9342.9	9342.9
2	9810	28513	995.09	18686	9342.9	9342.9
3	9810	28535	1493.4	18686	9342.9	9342.9
4	9810	28565	1992.6	18686	9342.9	9342.9
5	9810	28605	2493.1	18686	9342.9	9342.9
6	9810	28653	2995	18686	9342.9	9342.9
7	9810	28710	3498.8	18686	9342.9	9342.9
8	9810	28776	4004.8	18686	9342.9	9342.9
9	9810	28851	4513.3	18686	9342.9	9342.9
10	9810	28935	5024.6	18686	9342.9	9342.9
11	9810	29029	5539	18686	9342.9	9342.9
12	9810	29132	6057	18686	9342.9	9342.9

Table 3 Load case of 2g with angular variation of 1 to 12 degrees

Load case =2g , Wheel Force = 9810, α = 1-12^0						
A	F_g (N)	F_c (N)	R_y (N)	R_z (N)	F_{HR} (N)	F_{HL} (N)
1	19620	57000	994.79	37371	18686	18686
2	19620	57026	1990.2	37371	18686	18686
3	19620	57070	2986.8	37371	18686	18686
4	19620	57131	3985.2	37371	18686	18686
5	19620	57209	4986.1	37371	18686	18686
6	19620	57305	5990	37371	18686	18686
7	19620	57419	6997.7	37371	18686	18686
8	19620	57552	8009.6	37371	18686	18686
9	19620	57702	9026.6	37371	18686	18686
10	19620	57871	10049	37371	18686	18686
11	19620	58058	11078	37371	18686	18686
12	19620	58265	12114	37371	18686	18686

Table 4 Load case of 3g with angular variation of 1 to 12 degrees

Load Case = 3g, Wheel force =9810N, α = 1-12^0						
A	F_g (N)	F_c (N)	R_y (N)	R_z (N)	F_{HR} (N)	F_{HL} (N)
1	29430	85500	1492.2	56057	28029	28029
2	29430	85539	2985.3	56057	28029	28029
3	29430	85604	4480.2	56057	28029	28029
4	29430	85696	5977.8	56057	28029	28029
5	29430	85814	7479.2	56057	28029	28029
6	29430	85958	8985.1	56057	28029	28029
7	29430	86129	10497	56057	28029	28029
8	29430	86327	12014	56057	28029	28029
9	29430	86553	13540	56057	28029	28029
10	29430	86806	15074	56057	28029	28029
11	29430	87087	16617	56057	28029	28029
12	29430	87397	18171	56057	28029	28029

3.2 Computer Aided Structural Analysis of the Component

Figure 5 shows the stress variation across the component 1g Load case with maximum value of 15.41N/mm^2. Figure 6 shows the displacement variation across the component for Axle-Rotating- 3g Load case with a maximum value of 0.00456mm.The convergence of the critical measures (local displacement and Strain energy) occurs at a polynomial order of 6 which is below the set limit of 9(Figures 7, 8, 9). The convergence of the measures occurs below the set limit of 10% (Table 4). These ascertain that the results are accurate. The maximum stress value corresponds to the Analysis results (15N/mm^2) of Erukunt Agricultural machinery design using ANSYS software [9]. The tolerance is $\pm$0.41N/mm^2.

Figure 10 shows the stress variation across the component for 3g Load case with maximum value of 46.22N/mm^2. Figure 11 shows the displacement variation across the component for 3g Load case with a maximum value of 0.0137mm.The convergence of the critical measures (local displacement and Strain energy) occurs at a polynomial order of 7 which is below the set limit of 9 (Figures 12, 13, 14). The convergence of the measures occurs below the set limit of 10% (Table 5). These ascertain that the results are accurate. The maximum stress value corresponds to the Analysis results (45N/mm2) of Erukunt Agricultural machinery design using ANSYS software [9]. The tolerance is $\pm$1.22N/mm^2.

Figure 15 shows the stress variation across the component for Axle-Rotating-3g Load case with maximum value of 88N/mm^2. Figure 16 shows the displacement variation across the component for Axle-Rotating-3g Load case with a maximum value of 0.026mm.The convergence of the critical measures (local displacement and Strain energy) occurs at a polynomial order of 7 which is below the set limit of 9 (Figures 17, 18,19). The convergence of the measures occurs below the set limit of 10% (Table 6). These ascertain that the results are accurate.

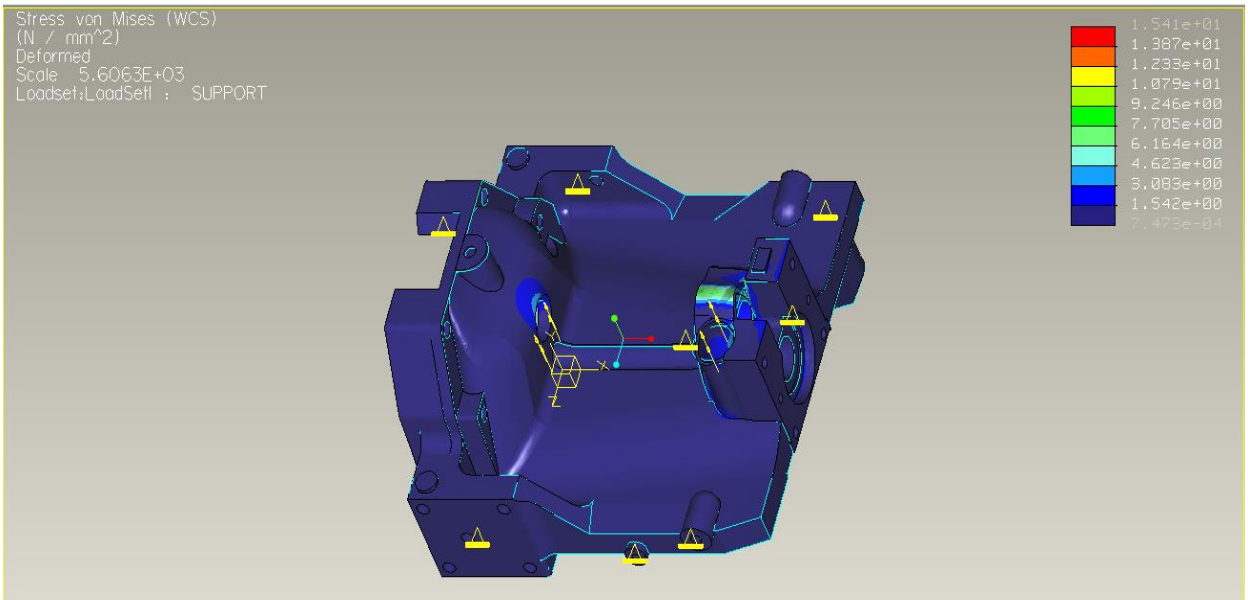

Figure 5 Stress Variation across the component for 1g load case

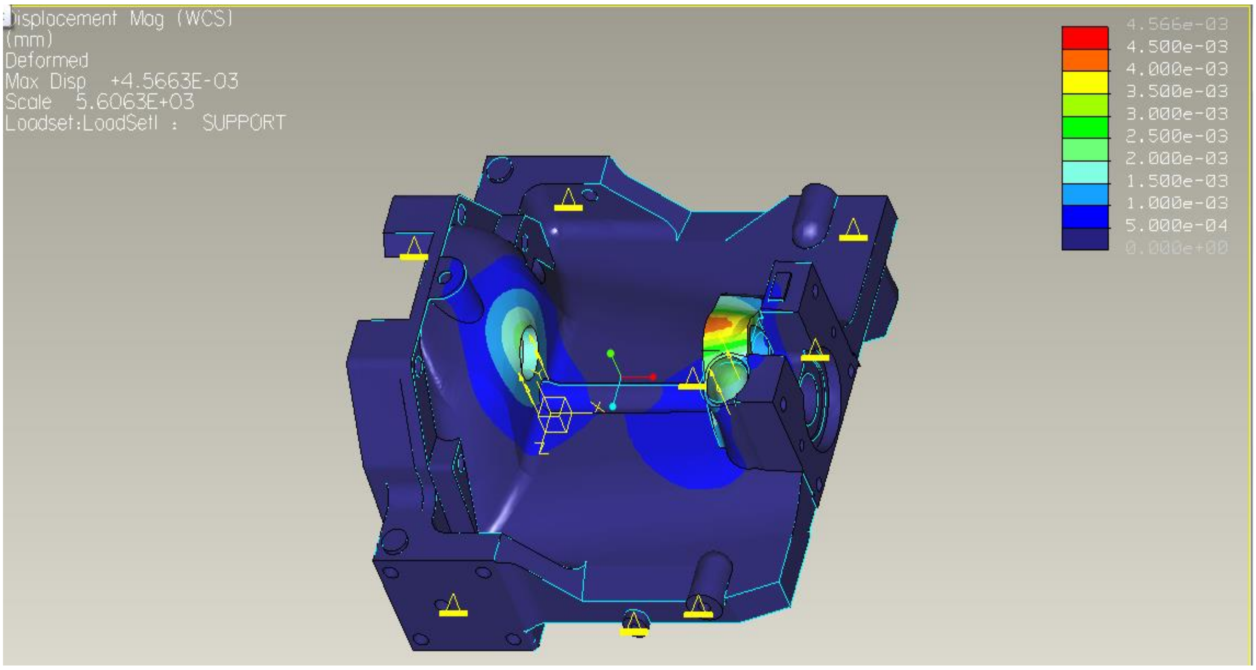

Figure 6 Displacement Variation across the component for 1g load case.

Table 5 The measures and percentage convergence for 1g Load case

Name	Value	Convergence (%)
max_disp_mag:	4.566321e-03	0.3
max_disp_x:	1.563678e-03	0.0
max_disp_y:	4.512505e-03	0.3
max_disp_z:	-1.125791e-03	0.1
max_prin_mag*:	1.685350e+01	8.0
max_stress_prin*:	1.685350e+01	8.0
max_stress_vm*:	1.540977e+01	8.7
max_stress_xx*:	6.391320e+00	0.1
max_stress_xy*:	-6.978182e+00	8.7
max_stress_xz*:	2.986170e+00	1.1
max_stress_yy*:	1.256920e+01	8.6
max_stress_yz*:	-5.232558e+00	8.6
max_stress_zz*:	9.186578e+00	4.6
min_stress_prin*:	-9.847826e+00	6.9
Strain energy:	9.247724e+00	0.1

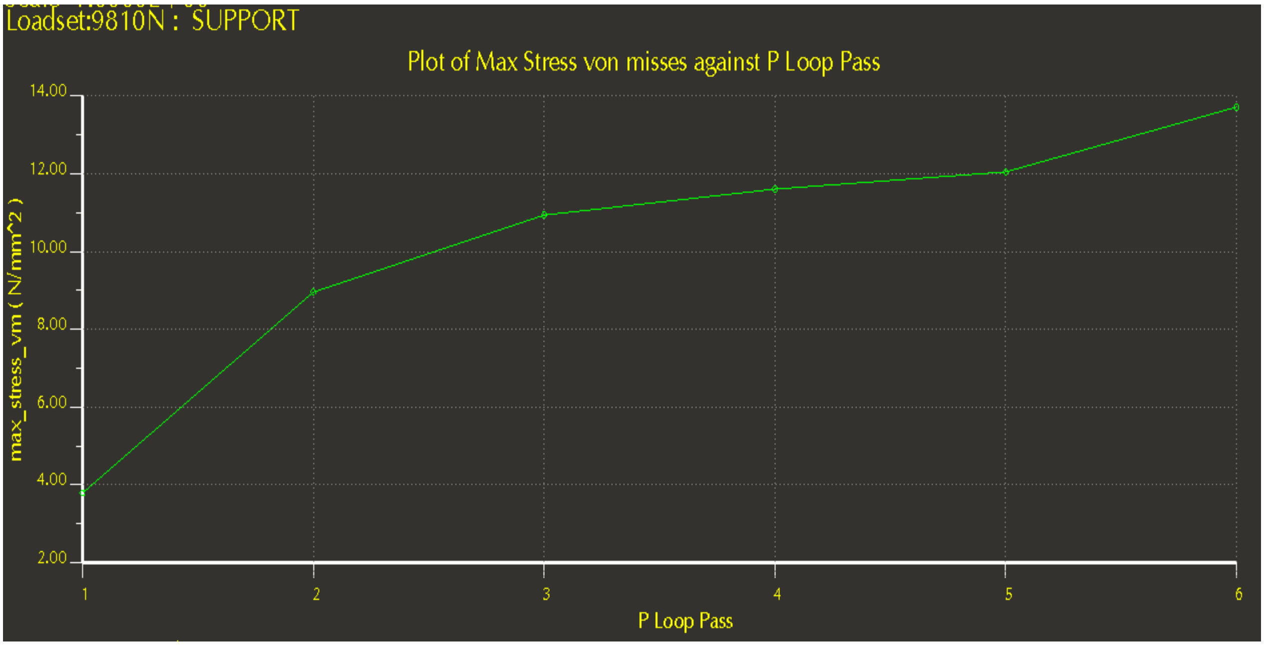

Figure 7 Plot of Max Stress against P Loop pass for 1g Load case

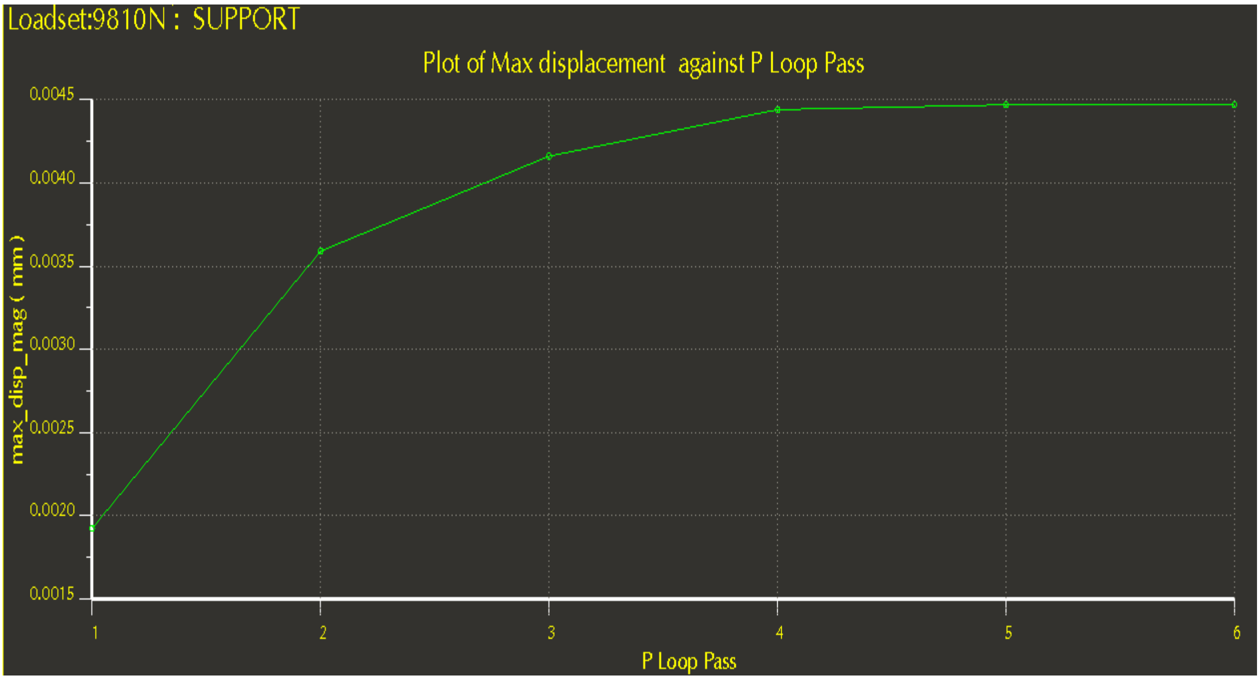

Figure 8 Plot of Max displacement against P Loop pass for 1g Load case

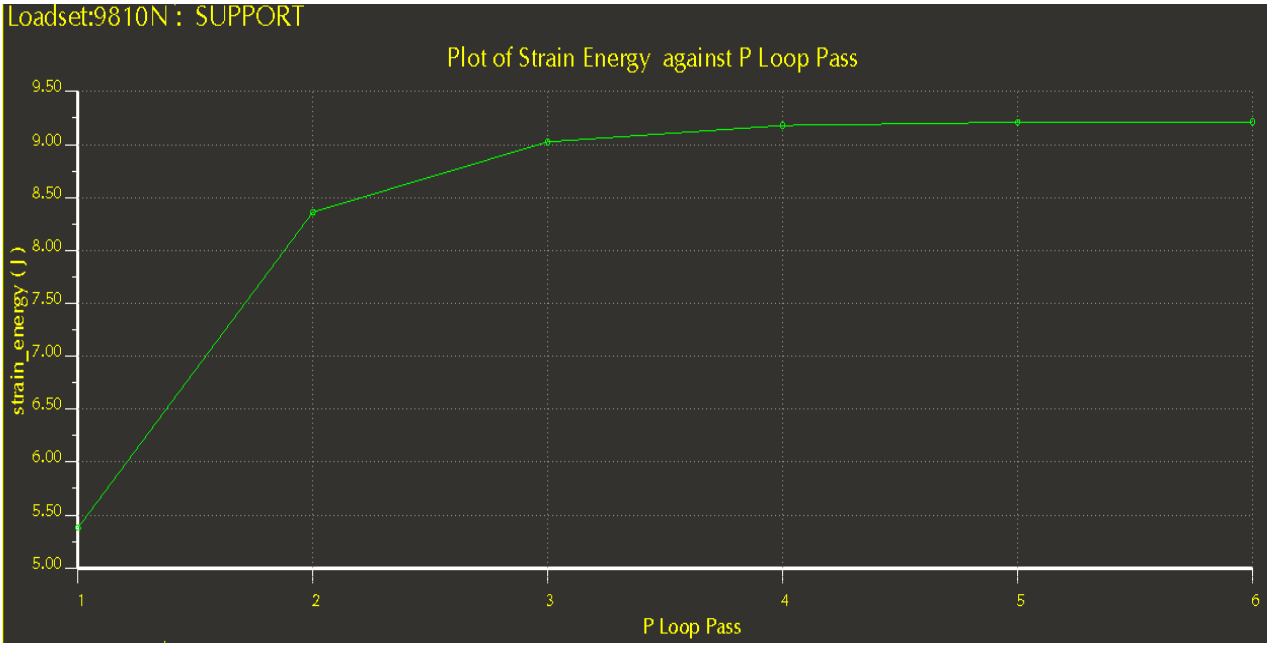

Figure 9 Plot of Strain Energy against P Loop pass for 1g Load case

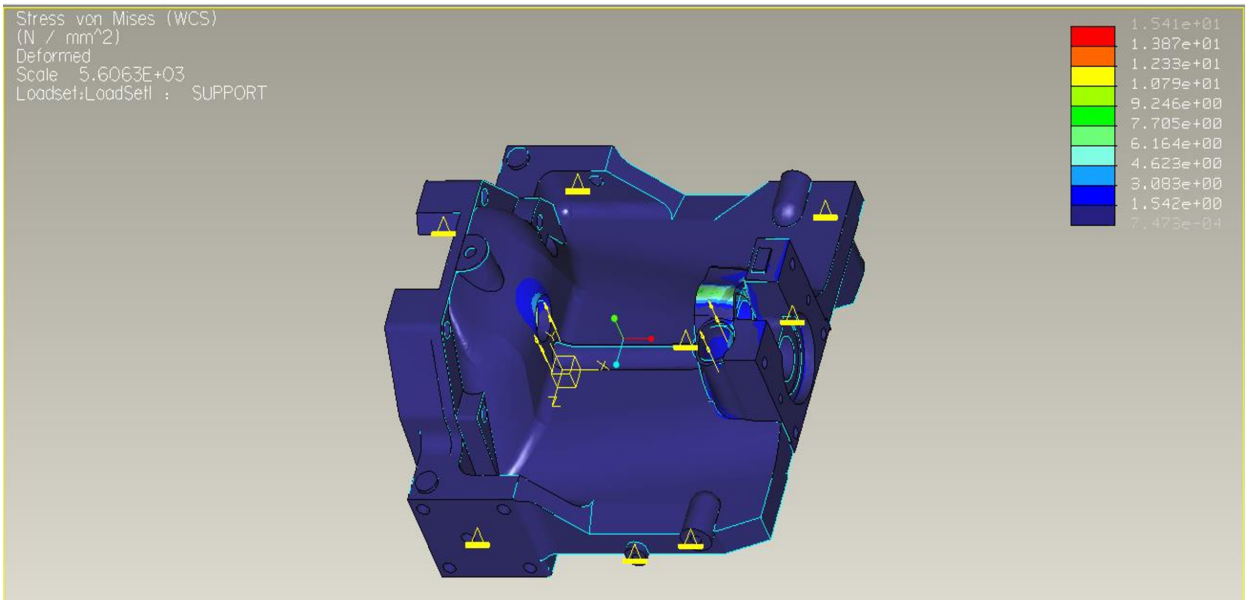

Figure 10 Stress Variation across the component for 1g load case

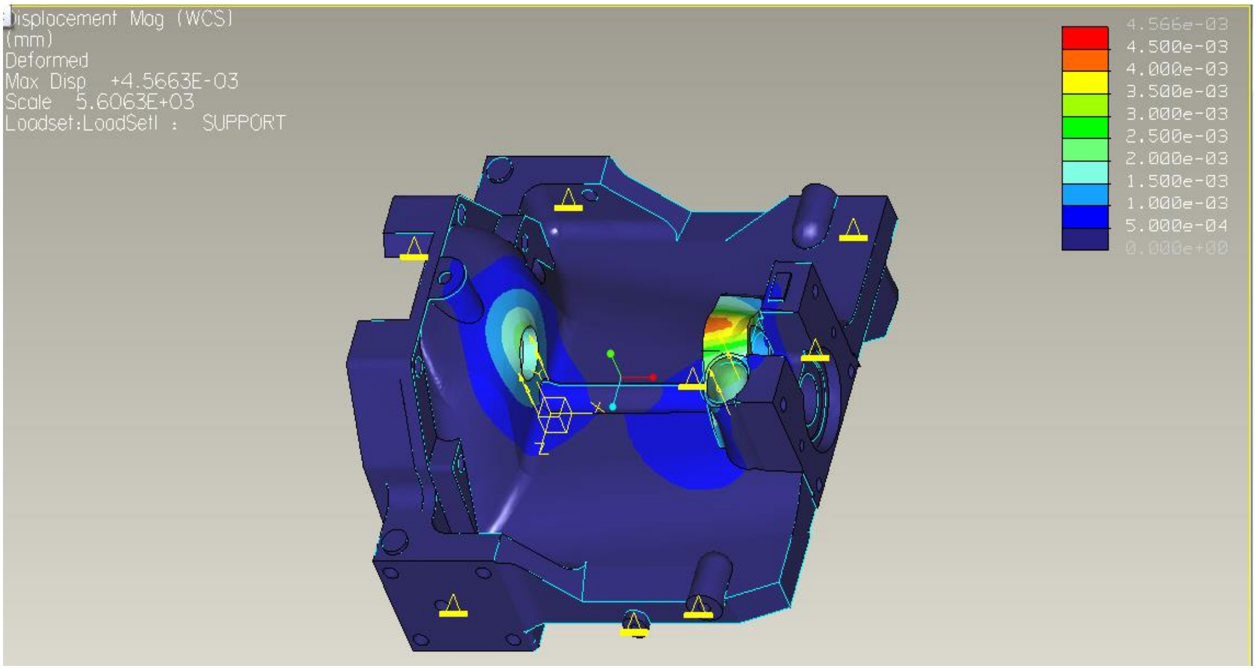

Figure 11 Displacement Variation across the component for 1g load case.

Table 6. The measures and percentage convergence for 3g Load case

Name	Value	Convergence (%)
max_disp_mag:	1.369998e-02	0.3
max_disp_x:	4.814254e-03	0.0
max_disp_y:	1.353858e-02	0.3
max_disp_z:	-3.376410e-03	0.1
max_prin_mag*:	5.061278e+01	8.0
max_stress_prin*:	5.061278e+01	8.0
max_stress_vm*:	4.622260e+01	8.7
max_stress_xx*:	1.922754e+01	0.1
max_stress_xy*:	-2.093104e+01	8.7
max_stress_xz*:	8.957982e+00	1.1
max_stress_yy*:	3.774872e+01	8.6
max_stress_yz*:	-1.569521e+01	8.6
max_stress_zz*:	2.756910e+01	4.7
min_stress_prin*:	-2.954833e+01	6.9
Strain_energy:	8.228287e+01	0.1

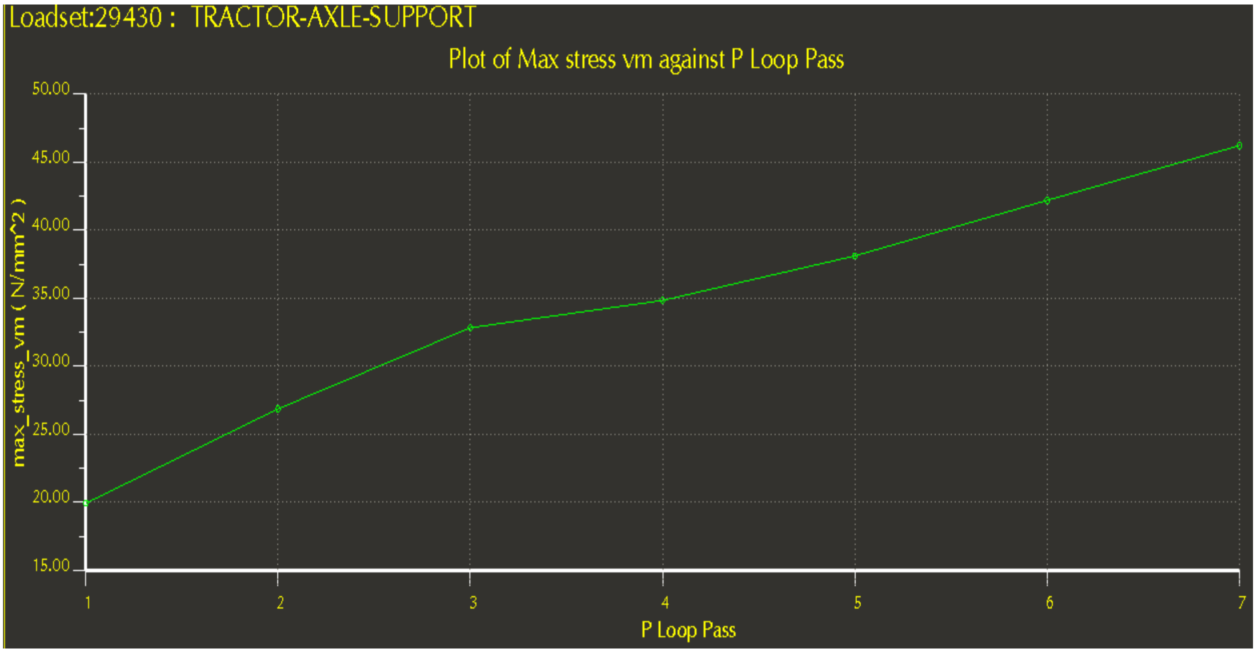

Figure 12. Plot of Max Stress against P Loop pass for 3g load case

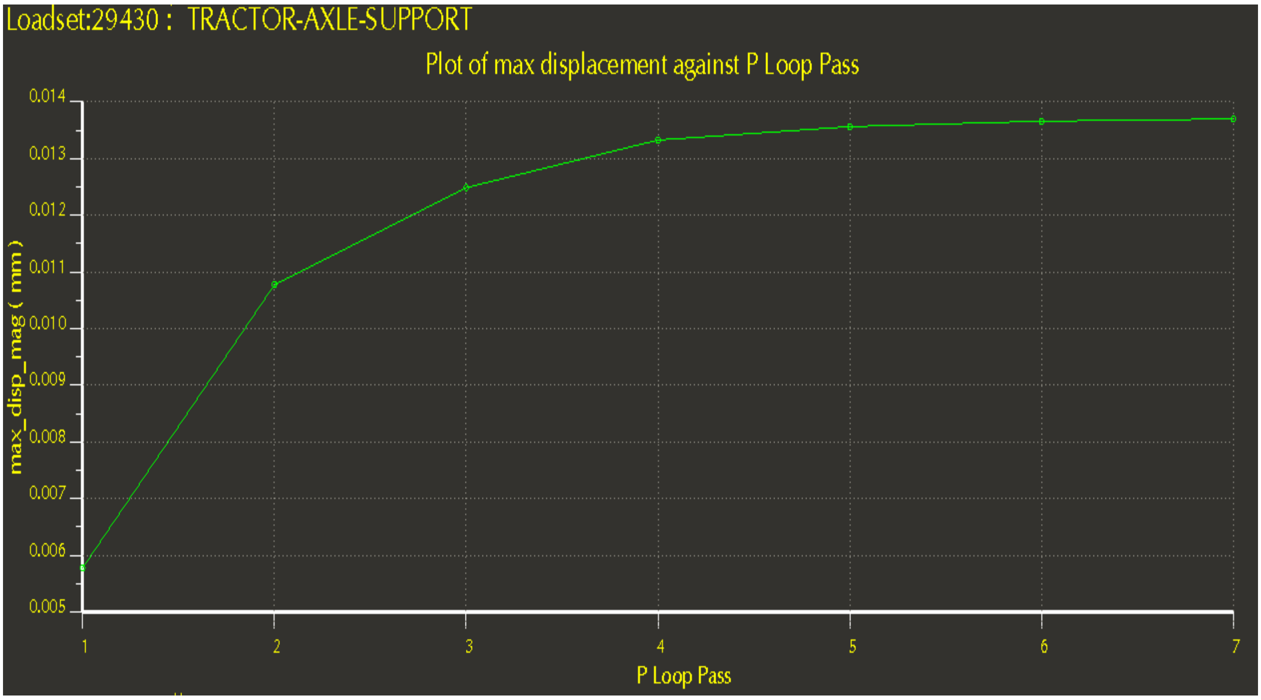

Figure 13. Plot of Max displacement against P Loop pass for 3g load case

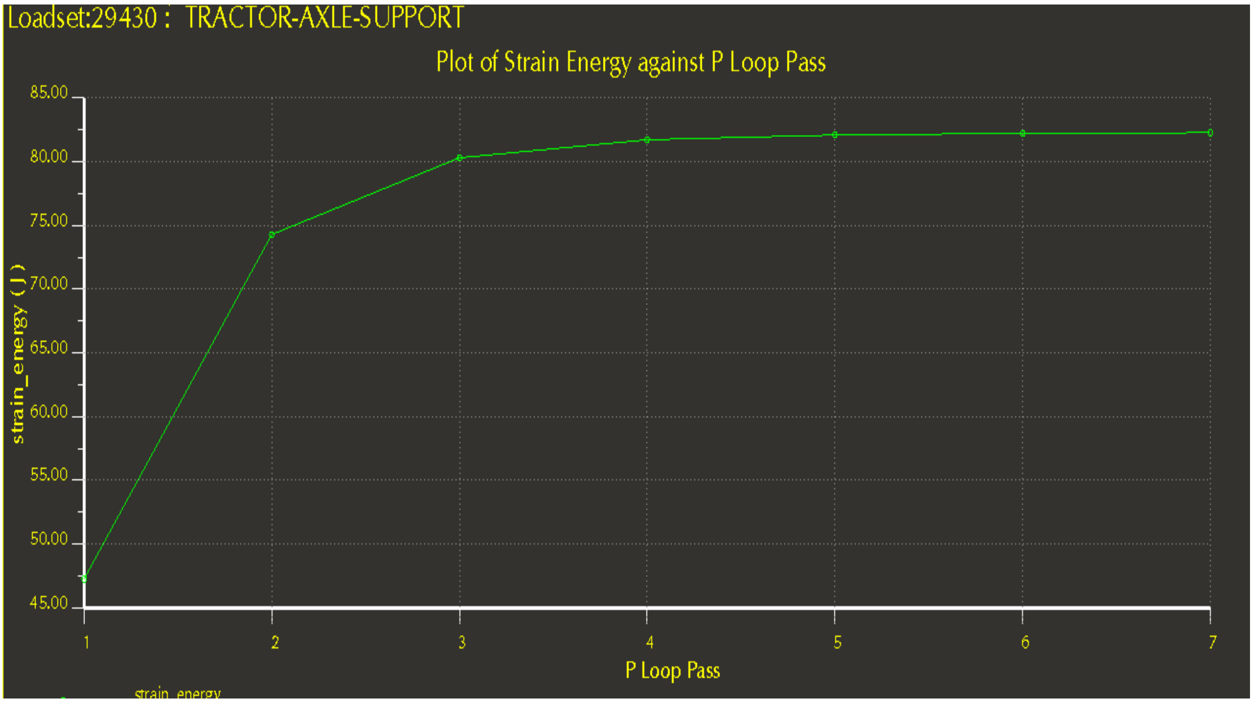

Figure 14. Plot of Max Strain Energy against P Loop pass for 3g load case

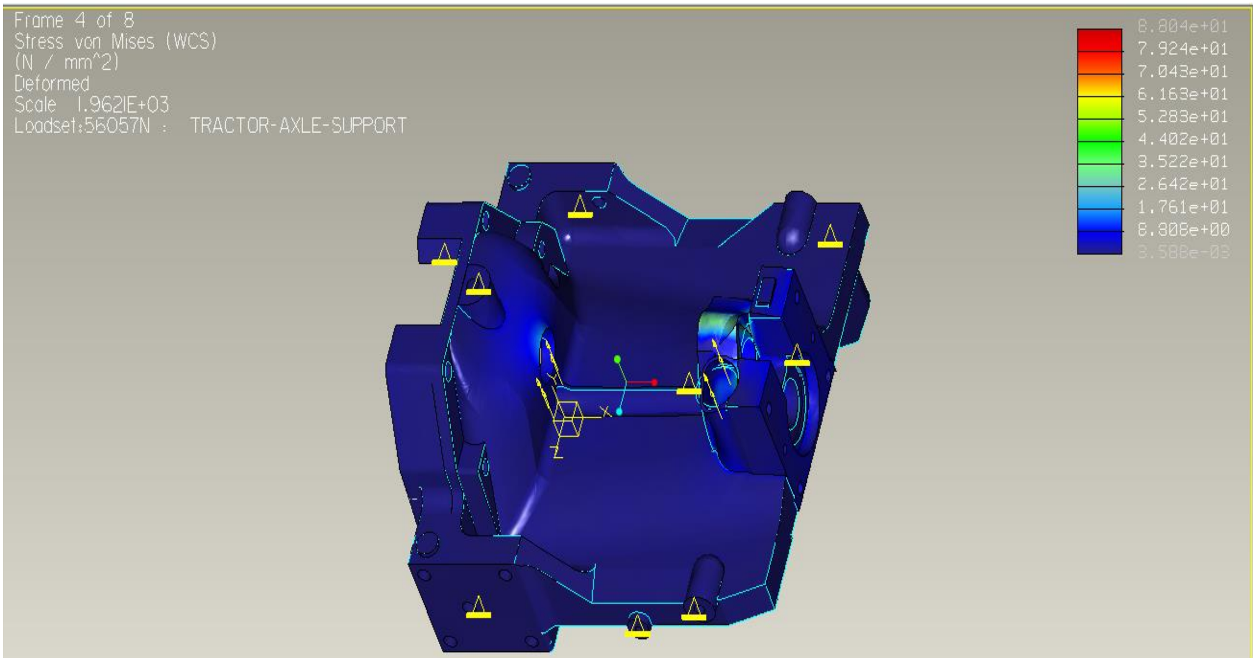

Figure 15. Stress Variation across the component for Axle-Rotating-3g Load Case

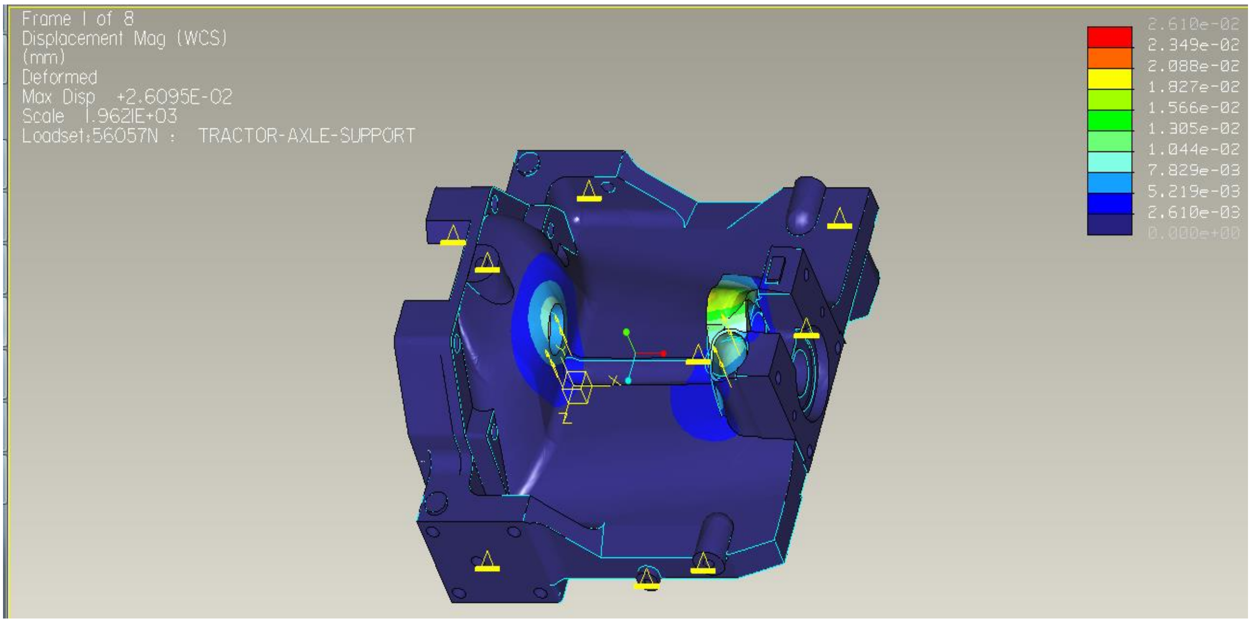

Figure 16. Displacement Variation across the component for Axle-Rotating-3g Load Case

Table 7 The measures and percentage convergence for Axle-Rotating-3g Load case

Name	**Value**	**Convergence (%)**
max_disp_mag:	2.609514e-02	0.3
max_disp_x:	9.169984e-03	0.0
max_disp_y:	2.578770e-02	0.3
max_disp_z:	-6.431240e-03	0.1
max_prin_mag*:	9.640505e+01	8.0
max_stress_prin*:	9.640505e+01	8.0
max_stress_vm*:	8.804283e+01	8.7
max_stress_xx*:	3.662379e+01	0.1
max_stress_xy*:	-3.986854e+01	8.7
max_stress_xz*:	1.706278e+01	1.1
max_stress_yy*:	7.190215e+01	8.6
max_stress_yz*:	-2.989557e+01	8.6
max_stress_zz*:	5.251243e+01	4.7
min_stress_prin*:	-5.628239e+01	6.9
Strain_energy:	2.985304e+02	0.1

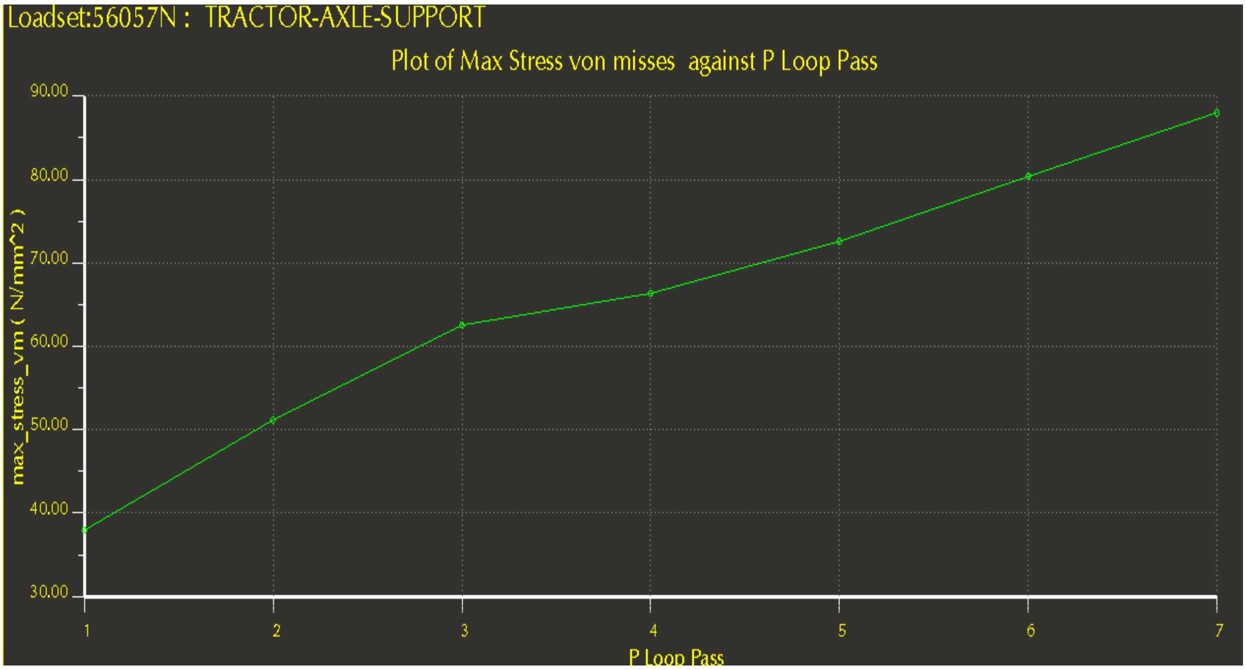

Figure 17 Plot of Max Stress against P Loop pass for Axle-Rotating-3g Load Case

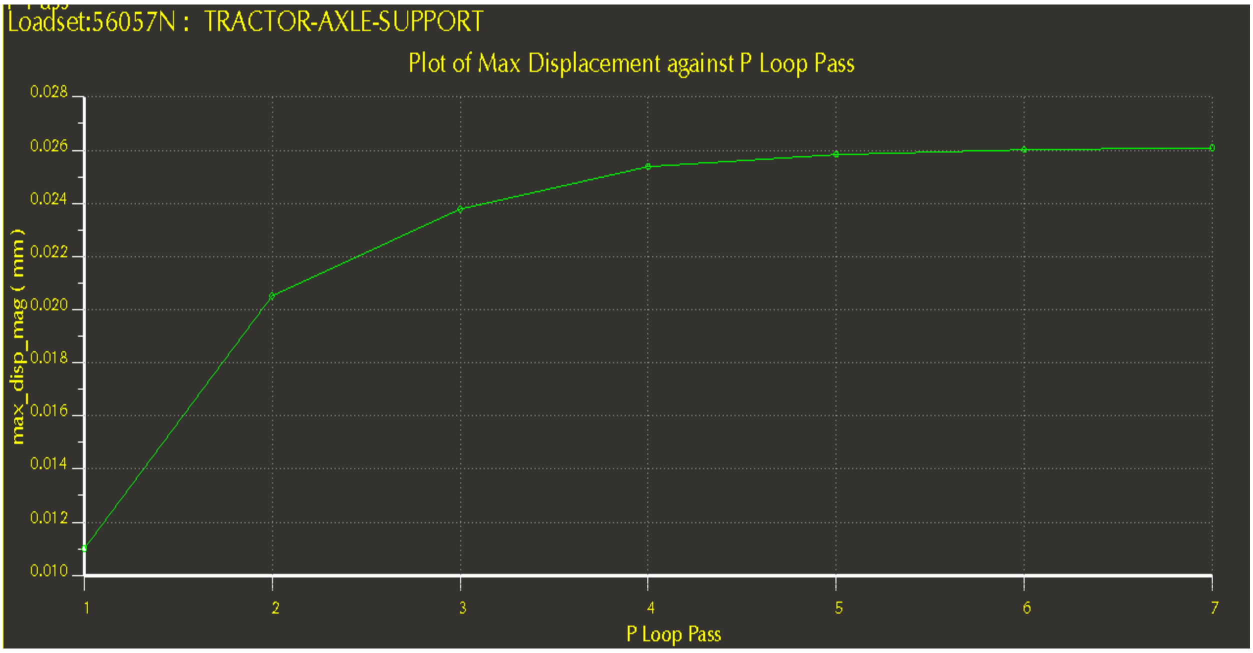

Figure 18 Plot of Max displacements against P Loop pass for Axle-Rotating- 3g Load Case

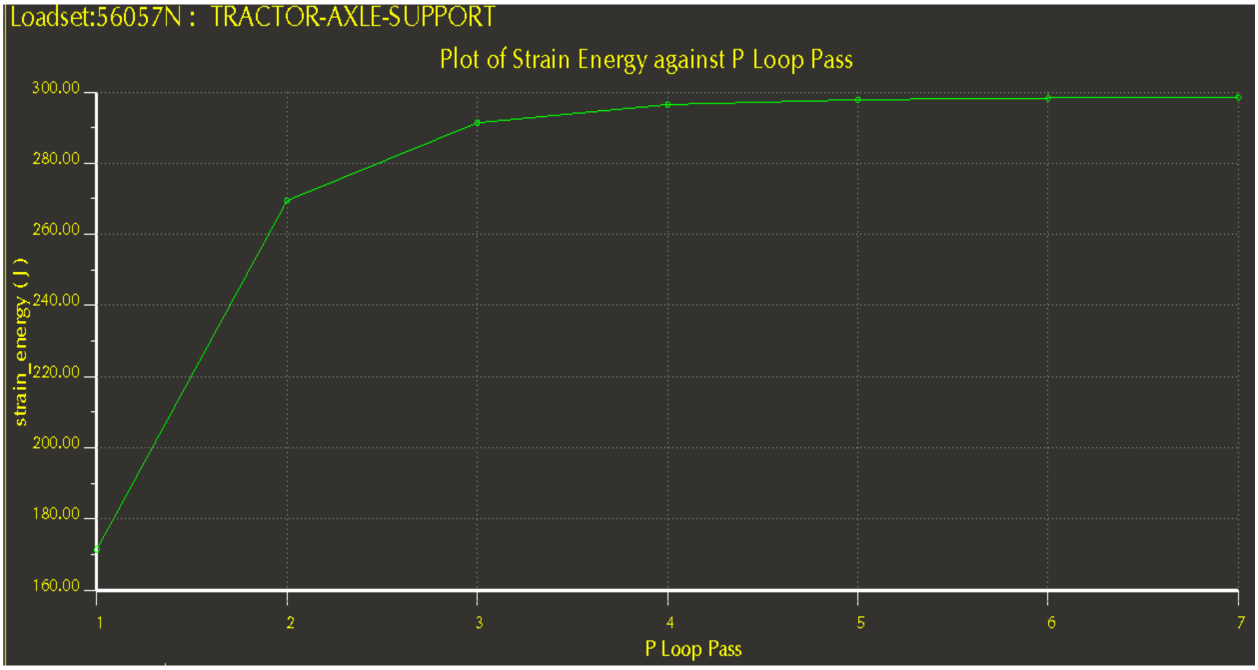

Figure 19 Plot of Strain Energy against P Loop pass for Axle-Rotating- 3g Load Case

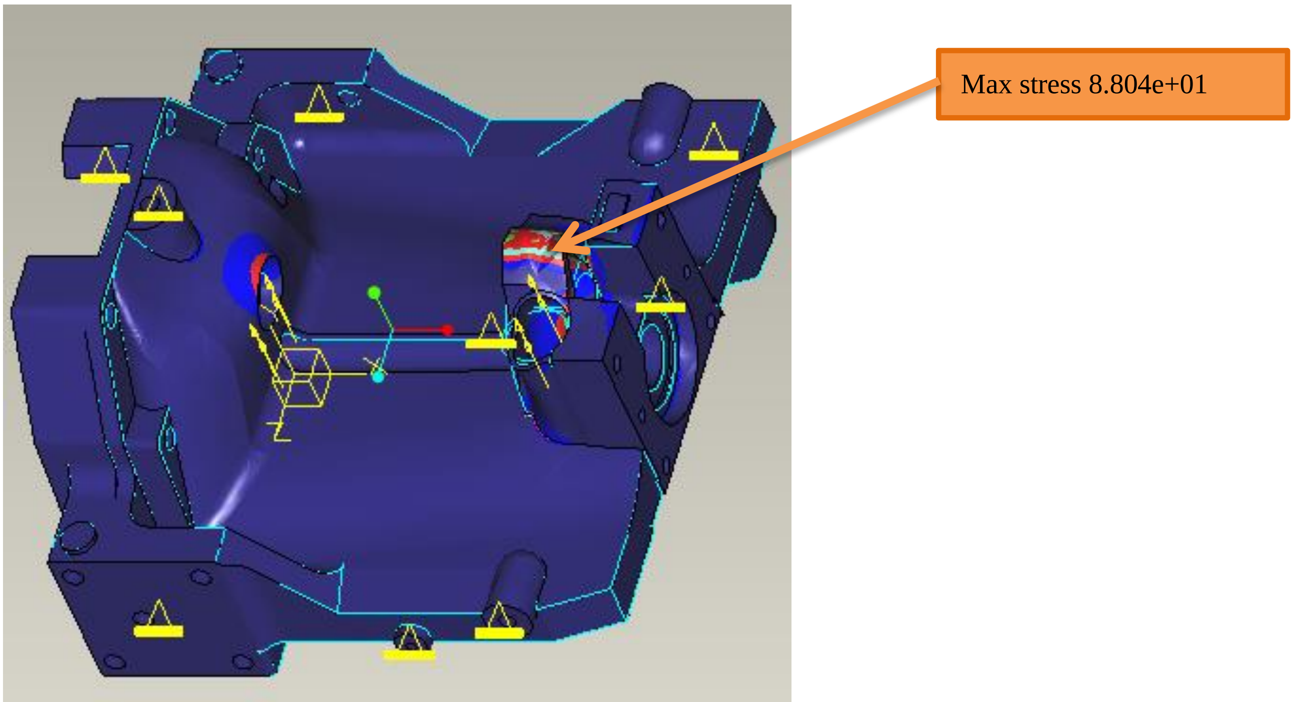

Figure 20 The Analyzed component showing the point with maximum stress

3.3 Factor of Safety

Factor of Safety = $\dfrac{\text{ultimate tensile stress}}{\text{Working or design stress}} = \dfrac{250N/mm^2}{88N/mm^2} = 2.84$

For the recommended factor of safety of 5.5 to 6.5

Minimum ultimate tensile stress = Working or design stress x Factor of safety = 484 N/mm^2

Maximum ultimate tensile stress = 572N/mm^2

For this new case, the maximum stress (Design Stress) is 88N/mm^2 (Figure 20). Since the tractor front axle support is a GG cast iron material of tensile strength of 250 N/mm^2, the safety factor for the design is 2.84. The recommended factor of safety for the component design is the minimum value of 5.5 and maximum value of 6.5. The Minimum allowable stress and Maximum allowable stress for the Component design would be 484N/mm^2 and 572N/mm^2.

4. Conclusion

The design requirement for the front axle support part is not to fracture during the lifetime of the tractor. The fracture of the component could happen in two ways; an overload causes an exceeding stress situation on the critical locations of the component or fatigue fracture caused by repeated loading on the field operations although the occurring stresses are far below the material tensile strength.

This research work analyzed 36 Load case scenarios and come up with worst Load case scenarios experienced by Ursus 3512 60hp Tractor Front axle support. The worst load case scenario was found to be the dynamic load case of 3g with wheel force of 9810N and axle angle of 12^0. The Factor of Safety for the component design was calculated to be 2.84 which were lower than the recommended value of 5.5 to 6.5. The Minimum allowable stress and Maximum allowable stress for the Component design would be 484N/mm^2 and 572N/mm^2.

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