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Fatigue Analysis of Front Axle for Automobile Heavy Motor Vehicle

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Abstract: The axles in a system must bear the weight of the vehicle as well as any cargo weight. The front axle beam is one of the major parts of vehicle suspension system and it houses the steering assembly as well. About 35 to 40 percent of the total vehicle weight is taken up by the front axle. Corrosion, wear and fatigue are the main causes of failure of mechanical parts. Main failure form of front axle beam is fatigue damage. The axles serve to transmit driving torque to the wheel, as well as to maintain the position of the wheels relative to each other and to the vehicle body. Therefore, the research on the fatigue life has important value. So, proper design and optimization of front axle is extremely crucial to Fatigue strength. The paper focuses on design, analysis and optimization of front axle. The approach in this research paper has been divided into two steps. The First step involves design of front axle by Analytical method. For this, types of forces loads with the help of CAD UNIGRAPHICS NX9.Second step involved further Pre-processing using ANSYS bench work 15.0 and post processing with the help of ANSYS bench work NCODE. Also the experimentation test performed and compared with FEA results.
Keywords: Front Axle, Design, Analysis, Automobile axle, construction and working of front axle beam, Fatigue analysis, NCODE ANSYS tool.

I. INTRODUCTION

In today's competitive industrial world, there is a growing demand for more efficient and economic manufacturing process to reduce production cost, increase productivity, reduce lead time and at the same time improve product quality. During last few decades due to global economic scenario optimum vehicle design & life of different parts of vehicle, like front axle beam (FAB) are major concern. Present off-highway vehicle market demands low cost, lightweight & long life component to meet the need of cost effective vehicle with fuel efficient. This in turn gives rise to more effective use of materials and useful surface treatments that are required to increase the life of vehicle components.

During the vehicle operation, road surface irregularity causes cyclic fluctuation of stresses on the axle, which is the main load carrying member. Therefore it is important to make sure whether the axle resists against the fatigue failure for a predicted service life. Axle experiences different loads in different direction, primarily vertical beaming or bending load due to drive torque, cornering load and braking load.

In real life scenario all these loads vary with time. Vertical beaming is one of the severe and frequent loads on an axle Due to their higher loading capacity; solid axles are typically used in the heavy commercial vehicles. Due to the road surface roughness, dynamic stresses are produced, caused by dynamic forces and these forces lead to fatigue failure of axle.

Fatigue failure often occurs from cracks initiated at bottom of spring pad and notches of front axle beam. It is usually described as a sequential process consisting of three main stages, i.e. crack initiation, crack propagation and final fracture. Therefore, in order to develop durable products against fatigue as well as to access the remaining lifetime of a component or to establish maintenance procedures. Corrosion, wear and fatigue are the main causes of failure of mechanical parts. Main failure form of front axle beam is fatigue damage. The axles serve to transmit driving torque to the wheel, as well as to maintain the position of the wheels relative to each other and to the vehicle body. Therefore, the research on the fatigue life has important value.

During the vehicle life, dynamic forces caused by the road roughness produce dynamic stresses and these forces lead to fatigue failure of axle, which is the main load carrying part ofthe assembly. Therefore it is vital that the axle resist against the fatigue failure for a predicted service life. On wheeled vehicles, the axle may be fixed to the wheels, rotating with them, or fixed to its surroundings, with the wheels rotating around the axle. The axles serve to transmit driving torque to the wheel, as well as to maintain the position of the wheels relative to each other and to the vehicle body. The axles in a system must also bear the weight of the vehicle plus any cargo. The front axle beam is one of the major parts of vehicle suspension system. It houses the steering assembly as well. Hence, research on fatigue of front axle beam is very important.

II. CONSTRUCTION AND OPERATION

Live front axle: The front axles are usually dead axles since they do not rotate, in contrast to live axles that they are used in rear axle to transmit power to the rear wheels. The dead front axle has sufficient rigidity and strength to transmit the front weight of the vehicle to the front wheels through the springs.

Front axle beam is made up of alloy steel for rigidity and strength for satisfying the function of dead of axle. Generally, Front axle is the forged part and Drop forging is the process of manufacturing. Due to grain size is reduced and fiber lines are oriented in a predefined direction without breaking it unlike casting and manufacturing part, forging parts can give the extreme toughness. Forged parts provide these features with an account of additional cost due thehigh initial investment for these forging processes.

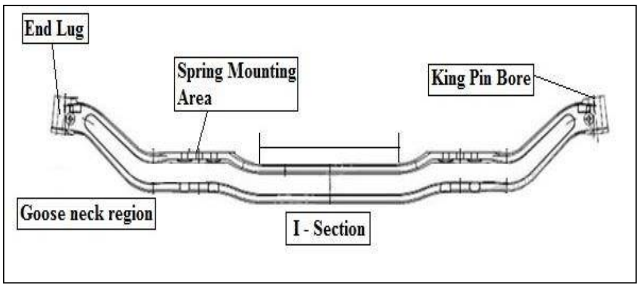
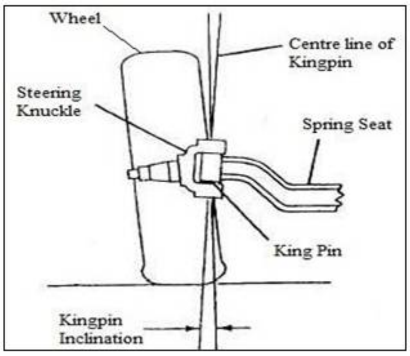


Fig1

When the vehicle goes into motion, the axle receives the twisting stresses of driving and braking. When the vehicle operator applies the brakes, the brake shoes are pressed against the moving wheel drum. When the brakes are applied suddenly, the axle twists against the springs and actually twists out of its normal upright position. In addition to twisting during braking, the front Axle also moves up and down as the wheels move over rough surfaces and also corneringforce applied as vehicle takes turn.



Components of Front axle beam

III. MATERIAL DESCRIPTION

A. Axle Details

1) *Type:* Front Non-Drive Steer Axle (Heavy Duty Trucks)
2) *Axle Rating:* 15000lbs (6803.88kg)
3) *Weight:* 105.48 kg (Forging wt.)
4) *Material:* AISI 1045

B. Chemical Composition

Sr. No.	Element	Weight Ratio
1.	Carbon, C	0.43-0.50 %
2.	Manganese, Mn	0.60-0.90 %
3.	Phosphorous, P	≤ 0.040 %
4.	Sulfur, S	≤ 0.050 %

C. Material Properties

Material property	
Yield Strength	886 Mpa
Ultimate Tensile Strength	998 Mpa
Elongation	16.5 %
% Reduction in Area	54.5%
Poisson ratio	0.3
Modulus of Elasticity	2.1e5 Mpa
Hardness	172 HB

D. Load Factor (Kc)

$$k_c = \begin{cases} 0.923 & \text{Axial Loading} \quad S<1520\text{MPa}(220\text{Kpsi}) \\ 1 & \text{Axial Loading} \quad S<1520\text{MPa}(220\text{Kpsi}) \\ 1 & \text{Bending} \\ 0.577 & \text{Torsion and shear} \end{cases}$$

IV. VEHICLE NOMENCLATURE

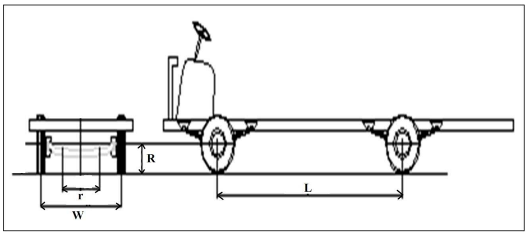


Fig 4.

1) Total weight of vehicle including carrying load = 60206.95 kg
2) Load on axle beams = 20411.65 kg
3) Gross Axle weight of FAB = 6803.88 kg ~ 7 Ton.
4) Height of C.G. of vehicle from ground = 875 mm
5) Dist. between tire centers (W) = 2135 mm
6) Pad to Pad distance (r) = 820 mm
7) Radius of wheel= 535 mm
8) Wheel base (L) = 4400 mm
9) Drop (D) = 171 mm

V. LOAD CASES

In actual working conditions the axle is subjected to dynamic forces so while doing analysis(Analytical, FEA & Experimental) of front axle beam equivalent dynamic conditions are considered as follows:-

Load Case	Vertical Load	Braking Load	Cornering
Vertical	3G = 200.25	-	-
Vertical + Braking	2.8G = 189.9	2G = 133.5	-
Vertical + Cornering	1.5G = 100.12	-	0.75G = 50.062

VI. FINITE ELEMENT METHOD

A. Pre-Processing

The FE Solid CAD model of the complete beam axle assembly is created using UNIGRAPHICS NX-9 modeling software as shown in Fig 5.

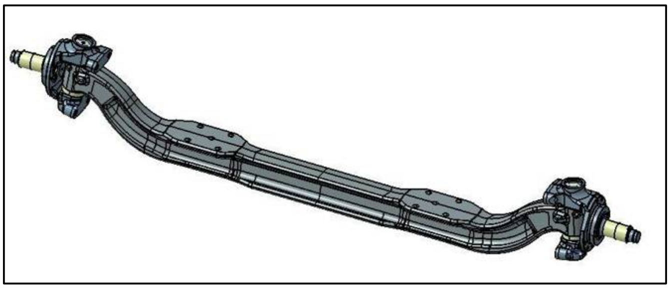
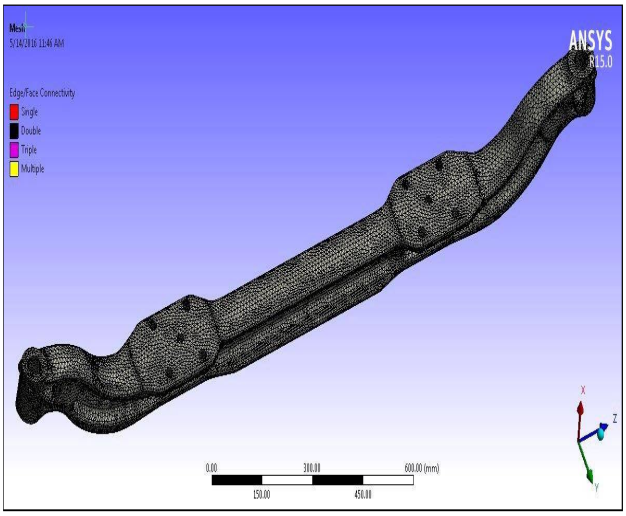


Fig 5.

B. Meshing

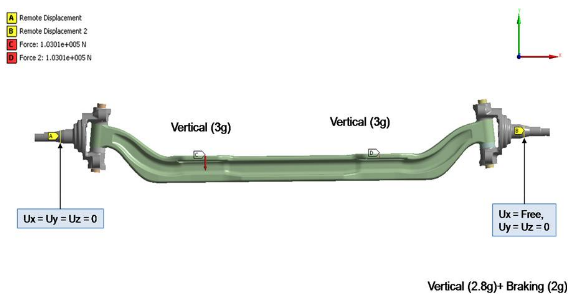
In the total assembly of front axle beam, steering knuckle and kingpin are meshed with grid size 8mm because the steering knuckle and master pin are still retained and front axle meshed with grid size as a 5mm. Tetrahedral element type is used for whole assembly. In meshing sizing option use advanced sizing function as proximity and curvature is ON with medium relevance center. Proximity Element Size is 5 and max element size is 10 as selected in meshing option. There are about 461106 Elements and 703886 Nodes in the whole assembly of Front Axle FE model.



C. Boundary Condition

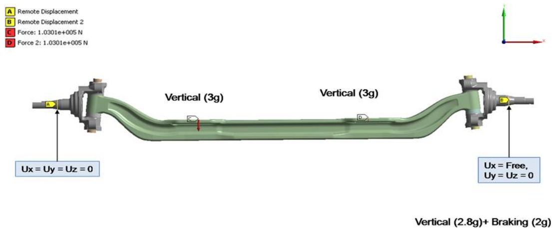
1) Vertical Boundary Condition

Movement constraints of *X*, *Y, Z* directions are applied on the left support knuckle and movement constraints of *Y*, *Z* directions are applied on the right support knuckle, as shown below



2) Vertical + Braking Boundary Condition

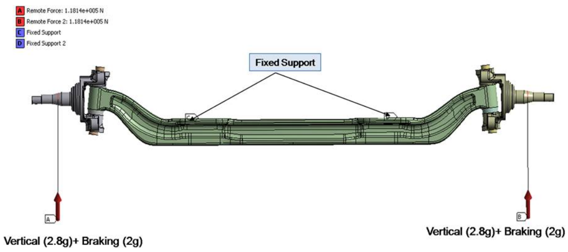
Movement constraints of *X*, *Y*, *Z* directions are applied on the left spring pad andmovement constraints of *Y*, *Z* directions are applied on the right spring pad, as shown below



D. Processing :- Loading condition

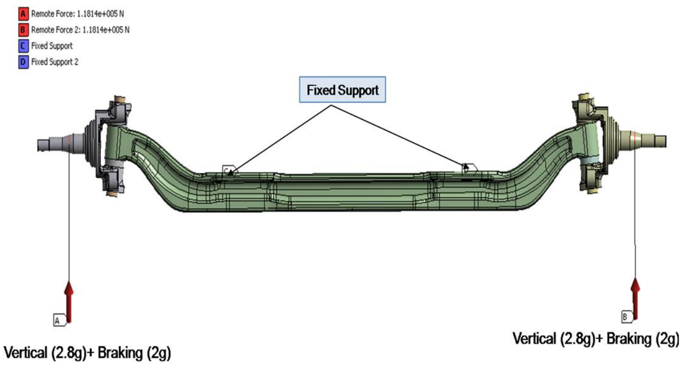
1) Vertical Loading Condition

Force applied on both left and right spring pad is 103.005 KN for vertical Loading casewhich is shown below:-



2) Vertical + Braking Loading Condition

Vertical force applied on both left and right knuckle is 94.950 KN and Braking forceapplied on both knuckle is 66.750 KN which is shown



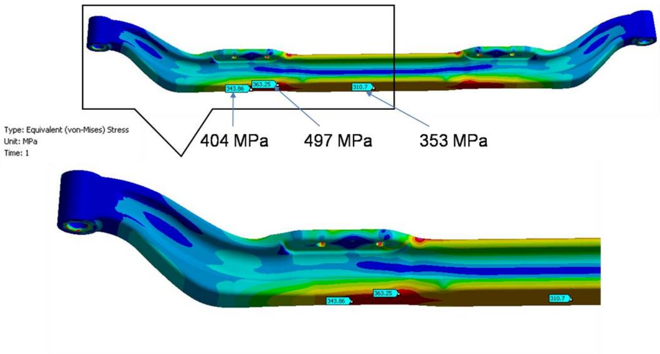
E. Post-Processing

1) Vertical Load Results

C/S	Vertical Load Fv (KN)	Stress σ N/mm²	Life No. of Cycles
1	100.12	403.99	227830
2		497.20	
3		352.88	

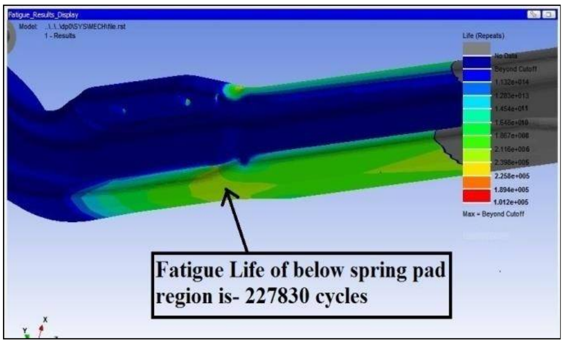
2) Stress Results

For vertical case loading, max stress region is below spring pad which is verified by FEA



3) Life result

Fatigue life for vertical loading case is 227830 cycles which is below the spring pad ofaxle, evaluated by NCODE design life software



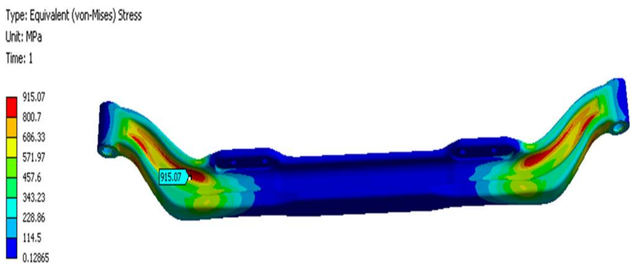
VII. RESULTS

A. Vertical + Braking Results

Cross Section	Vertical Force F (KN)	Braking Force KN	X (mm)	Y (mm)	σ (von Misses) (N/mm^2)	LIFE (No. of Cycles)
3	94.95	66.75	171	392	380.08	5197
4			255	412	572.07	
5			280	454	**915.07**	
6			386	508	796.37	

B. Stress Result

For vertical + braking loading case, max stress region is in the goose neck of axle, which isverified by FEA results



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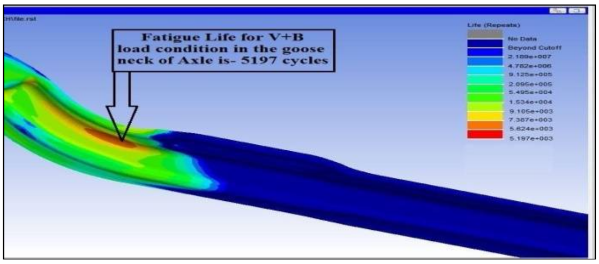
C. Life Result

Fatigue life for vertical + braking loading case in the goose neck of axle is 5197 cycles,which is evaluated by NCODE design life software.



VIII. EXPERIMENTAL INVESTIGATION

Experimental investigation is always important to test the automotive component. Experimentation test will give the proper result of running condition of vehicle. For years ago, automotive suppliers have heavily depended on the laboratory testing as compared to ground testing to validate their results and designs in their product development. The main reason for this is they have no easy methods to access the prototype vehicles for testing different kinds of products during the early development stage. Another reason is that the higher cost for ground testing as compared to laboratory testing.

Laboratory testing has been performed for three different kinds of loading conditions explained in steps as follows-

1) Vertical (Bending) Test.
2) Vertical + Braking (Bending & Torsional) Test.

A. Steps of Experimentation

1) Assembly of Front Axle Beam on a testing bed using setup fixture specified by customer.
2) Fixing the constrained region using bolts with fixture
3) Positioning and fitting of servo hydraulic actuators
4) Checking Hydraulic connections
5) Grinding the surface where the strain gauge is to be mounted
6) Cross Marking At exact location
7) Strain gauging
8) Select strain gauge form wide variety of series depending upon test conditions.
9) Soldering the output terminals to make connections.
10) Fixing the strain gauge on FAB using catalyst and M-bond.
11) Calibration of strain gauge.
12) Trial run of test setup with smaller loads.
13) Final run with stepwise increase in load (20%, 40%…100%).
14) Data acquisition using strain smart software from strain analyzer.
15) Interpretation and plotting of results.

B. Experimental Testing Equipment's

Equipment	Description	Application
Servo hydraulic Actuators & Accessories.	630KN, 250KN, INSTRON	Force
Strain gauge	CEA-06-125BZ-350 Uniaxial	Transducer (Deflection into change in Resistance)
Strain analyzer	System 6000	Data Collection
Interfacing Software	Windows based strain smart software system	Data Acquisition
Various fixtures, Dummy Wheel, Testing bed, Workstation computers	Axle Mounting, Data Interpretation, Graph Plotting Etc.	

Fig:-Fatigue Loading Testing Setup

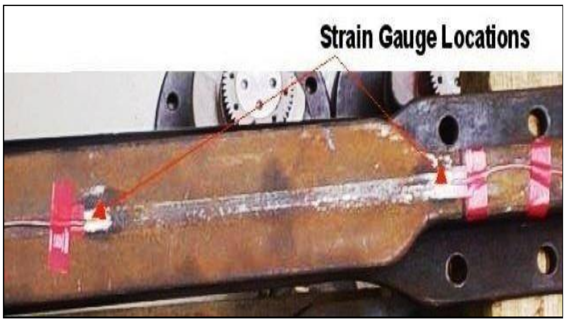


Fig: - Strain Gauge Locations for Vertical Loading Testing

C. *Experimental Results for vertical loading*

To minimize the errors caused by possible tip sliding of the dial indicators, the measurement positions and the directions was carefully selected. All test data were acquired through several repeated runs for test repeatability checking.

1) Strain Gauge Readings for different Loads:-

Load (N)		Strain (10-6)		
%	Vertical	SG1	SG2	SG3
20	40050	368	379	322
40	80100	815	826	766
60	120150	1284	1312	1086
80	160200	1638	1663	1302
100	200250	1914	2004	1484

2) Experimental Results for vertical loading:-

c/s	Vertical Force Fv (KN)	Strain (10^{-6})	Stress σ N/mm²	Life No. of Cycles
1	103.005	1914	401.94	255134
2		2004	420.86	
3		1484	311.64	

3) Experimental Results for vertical + braking loading

Load (N)			Strain (10^{-6})			
%	Vertical	Braking	SG3	SG4	SG5	SG6
20	18990	13350	356	488	709	682
40	37980	26700	980	1202	1612	1589
60	56970	40050	1312	1723	2475	2154
80	75960	53400	1690	2215	3123	2853
100	94950	66750	1885	2480	3971	3291

4) Experimental Results for vertical + braking loading

c/s	Vertical Force Fv (KN)	Braking Force Fb (KN)	Strain (10^{-6})	Stress σ N/mm²	Life No. of Cycles
3	94.95	66.75	1885	395.85	6382
4			2480	520.8	
5			3971	833.91	
6			3291	691.11	

D. Correlation of Results and Validation

Correlation of stresses is the important stage in product validation. Different combinations of strain measurements were used as targets for matching the FE results for different correlation strategies comparisons. Minimizing the difference between the calculated FE responses and the experimental measurements at the selected locations is set as optimization objective function. Three selected correlation or matching strategies are listed in the following.

IX. VERTICAL LOADING

A. Comparison of Stresses

Stresses of analytical method, experimental method and FEA method i.e. ANSYS workbench method for vertical loading case are compared. Percentage variation is up to 15% and which is acceptable. The experimental and FEA results are shows good correlation for stress, which is about 90 %.

Strain Gauge Location	Stress Analytical Von Mises (Mpa)	Corresponding Experimental Stress	% Stress Variation	Stress FEA	% Variation
1	430.38	401.94	6.6	403.99	0.5
2	439.71	420.86	4.3	497.20	18.1
3	328.95	311.64	5.3	352.88	13.2

B. Comparison of Life

Fatigue Life for vertical loading case for maximum stress is calculated using analytical, experimental and FEA method as shown in Table. In all the solutions, fatigue life is more than 2 x 10^5 cycles, which shows design is safe.

Max Stress and Life	Analytical	Experimental	FEA
Max Stress (MPa)	439.72	420.86	497.20
Life (No. of Cycles)	248656	255134	227830

C. Vertical + Braking Loading

1) Comparison of Stresses

Stresses of analytical method, experimental method and FEA method i.e. ANSYS workbench method for vertical + braking loading case are compared. Percentage variation is up to 15% and which is acceptable. The experimental and FEA results are shows good correlation for stress, which is about 90 %.

Strain Gauge Location	Stress Analytical Von Mises (Mpa)	Corresponding Experimental Stress	% Stress Variation	Stress FEA	% Variation
3	408.52	395.85	3.10	380.08	4.23
4	565.50	520.82	7.90	572.07	8.95
5	858.81	833.91	2.89	905.17	5.5
6	727.336	691.11	4.98	796.37	13.21

2) Comparison of Life

Fatigue Life for vertical + braking loading case for maximum stress is calculated using analytical, experimental and FEA method as shown in Table. In all the solutions, fatigue life is more than 4 x 10^3 cycles, which shows design is safe.

Max Stress and Life	Analytical	Experimental	FEA
Max Stress (Mpa)	858.81	833.91	905.17
Life (No. of Cycles)	5486	6382	5197

X. CONCLUSION

1) Among all analysis strategies, the experimental strain measurement is considered as the reference strain value because it gives results that are more realistic.
2) FEA SETUP is similar to experimental test setup is simulated along with same Load and boundary condition to obtain results that are more correct. Overall difference between the three different analysis types is 10-15% this indicated that the correlation results are acceptable.
3) Analytical stress calculation was rather difficult task because no formulae are available for irregular cross section. It is become possible after only simplifying the geometry. Extreme care is to be taken while simplifying geometry & applying formulae to calculate stresses and fatigue life.
4) There are various fatigue factors which are affected on life of component. But it is difficult to identify all fatigue factors which are present. So, analytical method will not give the exact result values.
5) The fatigue life is more than 2 x 105 cycles, which is the general requirement of vertical loading case. In all the above solutions, fatigue life is more than 2 x 105 cycles, which shows design is safe. Similarly, design is safe for vertical + braking loading case.
6) Under vertical loading case, maximum stress is below spring pad region. So, the life is minimum of below pad region.
7) Under vertical + braking loading case, maximum stress region is in the goose neck of axle and under vertical + cornering loading case, maximum stress region is below spring pad arm side. These are satisfied by all solution method
8) The experimental and FEA results shows good correlation for stress, which is about 90%.

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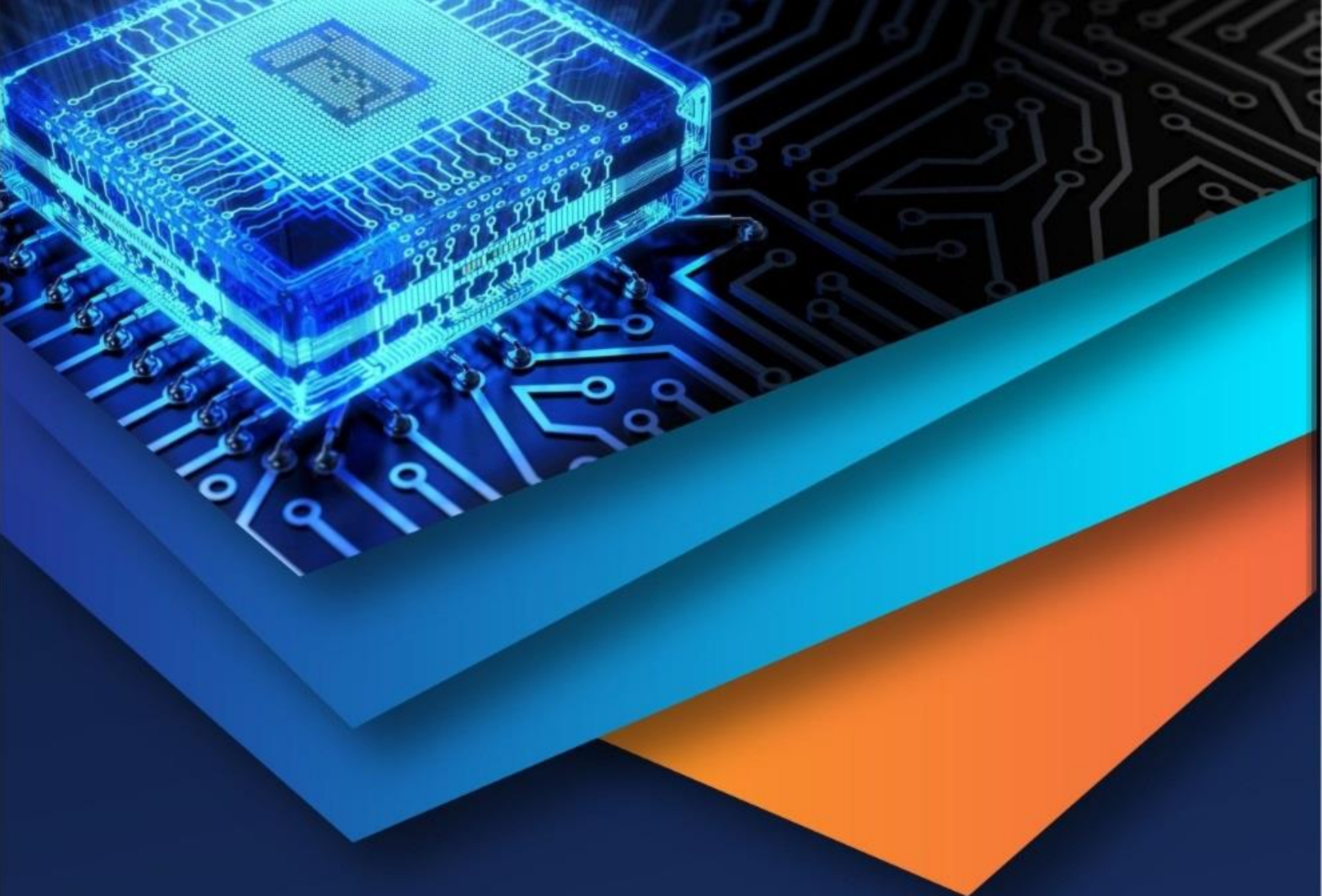

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