Weight Reduction of Bottom Centre Bearing Plate of freight rolling stock

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Abstract - Indian railway is designing higher axle load freight rolling stock for high speed freight dedicated tracks to improve through put. Weight and speed are always crucial parameters for high speed vehicles. Bottom Centre Bearing Plate is made of cast steel and connected between car body and bogie to transfer the loads from car body to track in static condition further it bears longitudinal traction, braking forces and centrifugal forces arising on curved track in dynamic environment. Weight of bottom centre bearing plate of freight rolling stock bogie frame has been reduced in present research work by performing parametric study of structural analysis in MSC FEA interface. Indian Railway loading and boundary conditions has been applied on finite element model of initial modified design of the bottom bearing plate. The weight of the modified centre plate has been approximately reduced by 2.72% than initial one. Natural frequencies and corresponding mode shapes of initial and modified design are compared to verify the topology of the component. The time dependent tenser values at critical zones of initial and modified design of the bearing plate have been verified on Goodman diagram.

Key Word **- Freight Rolling Stock, High Speed, Bearing Plate, Longitudinal Traction, Braking Forces, Natural Frequency.**

I. INTRODUCTION

Centre Bearing Plate is a critical structural component balance and transfer various forces generated during vehicle motion. The motion of a railway vehicle is affected by the geometry of the track, the interaction between wheels and rails, the suspension, and the inertias of various component. Suspension for a high-speed railway 'vehicle comprises between the car body and the bogie chassis arbol-'ster platform which is connected to the body by a pivot allowing the free rotation of the bogie in the curves, in combination with springs. Moreover, the bolster platform is connected to a transom of the bogie chassis by an articulated trapezium, with springs ensuring the stability. The body can thus orientate itself in the direction of the centrifugal force in the curves without loss of the stability. The articulations of the trapezium are situated very low, which reduces to a minimum the dynamic force transfers to the axles.

A railway freight vehicle having CASNUB Bogie as shown in Fig. 1 consists of bolster. The bogie along with its components has a large proportion of the total weight of a vehicle. Even if a small amount of bogie weight is reduced, it could be translated into huge amount of savings in terms of energy and material as well as it would increase the pay load of the vehicle for the given axle load.

In present work, it is endeavored to optimize the weight of bogie bottom center bearing plate by varying its three critical dimensions through a parametric study and an iterative method. UGS NX-7.5 and MSC-FEM are used for modeling and Finite Element analysis of the solid models. The Center bearing plate as shown in Fig. 2 is a vital component of railway freight vehicles and has following significant functions during operation:

1. Transmission of tractive and braking forces during running and shunting from bogie to superstructure.
2. Each centre pivot supports 5% of superstructure weight and 50% of payload.
3. Facilitates curve negotiation within Maximum moving dimensions.
4. Provides damping for superstructure with respect to track irregularities for better ride behavior.

Fig. 1 Indian Railway Freight Vehicle fitted with CASNUB Bogie

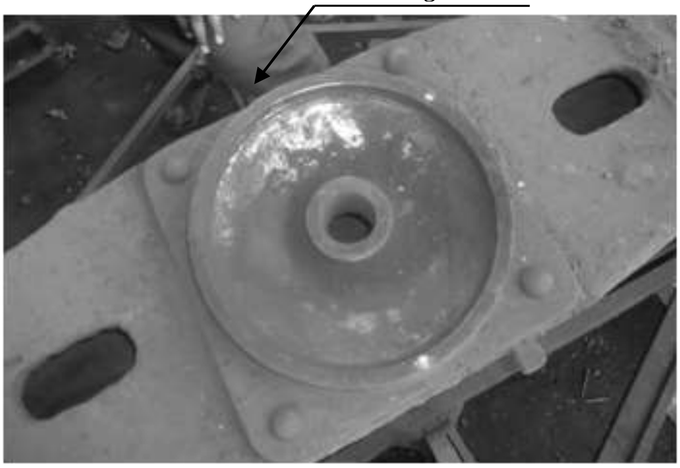

Fig. 2 Bottom Centre Bearing Plate

The bolster fitted with bottom centre bearing plate is designed for Indian railway operating conditions applying load and boundary conditions as per International Standard AAR M-202 [1] & [2]. Garg and Dukkipati [3] explained a 22-DOF freight vehicle dynamics model developed by Tse and Martin. A parametric study was performed to determine the effect of the car body's center of gravity height on the critical speed. It was found that the center of gravity height and critical speed was inversely proportional to each other.

Park et al [4] endeavored to reduce the weight of bogie frame while maintaining the fatigue constraint. A Finite Element model of the bogie frame was constructed to simulate the fatigue test. An optimization problem was framed in terms of thickness of three plates with an additional constraint of maintaining factor of safety of 2.5 for fatigue strength. Artificial neural network (ANN) was used to approximate a function for the fatigue constraint and the genetic algorithm to solve the optimization problem. The weight of Korean passenger bogie was reduced approximately up to 4.7%.

Shukla et al. [5] developed a multi body dynamics model for parametric study of suspension elements of CASNUB freight bogie to improve the ride behavior of the vehicle. Structural analysis of a Korean tilting train bogie bolster was performed by Kim et al. [6]. The model was used to investigate static main service load case and dynamic tilting loads using FE and multi body dynamics environments respectively. The fatigue strength was performed up to $10 * 10^6$ cycles under tilting load condition and the stress values were verified on Goodman diagram.

Li et al. [7] studied fatigue strength of subway vehicle bogie frame by rain-flow cycle counting method. Tang et al. [8] evaluated fatigue strength of DMU bogie frame with Goodman plots. Further modal analysis of the bogie frame was performed to determine practical operational constrains of the modal. Bubnov et al. [9] investigated casted bogie parts (bolster and side frame) strength using FE methods and modern computer technology for assessment of stress-strain state of three dimensional parts using MBD modeling, Genetic algorithm and Goodman diagram.

A finite element model of CASNUB bogie frame was developed by Shukla et al. [10] to perform transient analysis. MATLAB platform if used to reduce the weight of the bolster fitted with bottom centre bearing plate. Results are verified on Goodman diagram.

II. METHODOLOGY

A structural analysis of a freight car center pivot bottom has been performed using the finite-element method for various loading and boundary conditions as per International Standard AAR M202 [1]. The drawing and solid model of the Bottom Centre Bearing Plate having weight $49.55\ kg$ are shown in Fig. 2 & 3 respectively. The model is developed using UGS NX-7.5 interface [11]. The objective of the weight reduction is to vary the outer cup radius R_1, base length L_1 and outer-hole radius R_2 for obtaining optimized weight. The weight of the bogie center pivot bottom is optimized by suitably varying above three critical dimensions through a parametric study and an iterative method.

Structural Analysis

Structural analysis of initial design is performed by finite element (FE) methods [12] & [13] using MSC FEA interface [14]. The model is meshed by Tet 10 elements (Fig. 5) and load and boundary conditions proposed according to AAR M202 [1] as shown in Fig. 6 & 7. The details values of F_1 and F_2 are listed in Table 1.The recommended material properties for grade C material as shown in Table 2 are applied to the FE model. Critical stress zones are located on initial design. Further 3^3 modified model are developed for parametric analysis, where 3 is the number of dimensions taken for study. The behavior of stress and deflection at critical zones of these 27 cases are listed in Table 3.

Fatigue strength

The parametric study of the initial design reflects the trend weight reduction without appreciable increase in stresses. The modified values of the selected three dimensions are obtained for optimized weight performing little iteration. Dimensions of the modified design are as: $R_1 = 195mm$, $L_1 = 418mm$ and $R_2 = 40mm$. The weight of the modified design is reduced to 2.72% by initial one. The initial and modified designs are further tested for fatigue strength performing transient analysis (Figs. 8-9) applying Indian railway track excitation (Fig. 10) [15] & [16]. Tensor plots of the initial and modified designs at critical areas of selected three surfaces are shown in Fig. 11-12. Mean stress and stress amplitudes are calculated from time domain tensor values at critical areas. The mean stress and stress amplitudes obtained from max. and min. stresses values of the plots. These values are plotted on Goodman diagram (Fig. 13) [17] & [18] to verify fatigue strength of modified bottom centre bearing plate. Both designs have satisfied the safety factor of 2.5 with respect to fatigue design. It shows that the modified model is well within the factor of safety of 2.5 with respect to fatigue design.

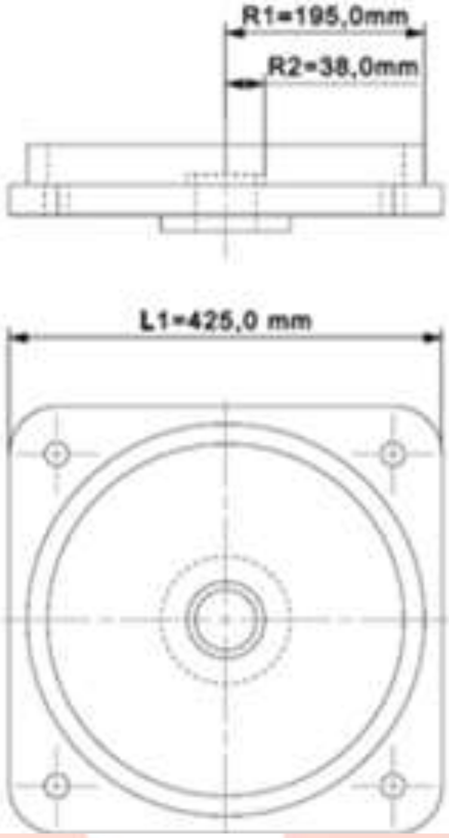

Fig. 3 Drawing of Bottom Centre Bearing Plate

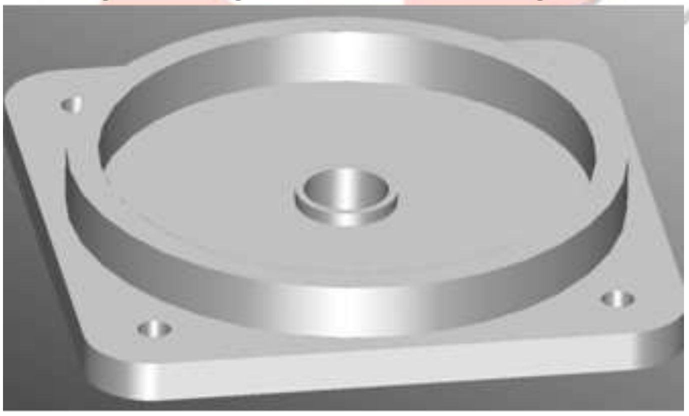

Fig. 4 Model of Bottom Centre Bearing Plate

Table-1: Details of F_1 and F_2

	Force (N)	Area (mm^2)	Pressure (MPa)
F1	666852	92106.4	7.24
F2	451106	7330	61.54

Table 2: Material properties

Grade	Ultimate Tensile Strength (MPa)	Yeild stress (MPa)	Endurance Limit (MPa)
C	619.92	413.28	247.97

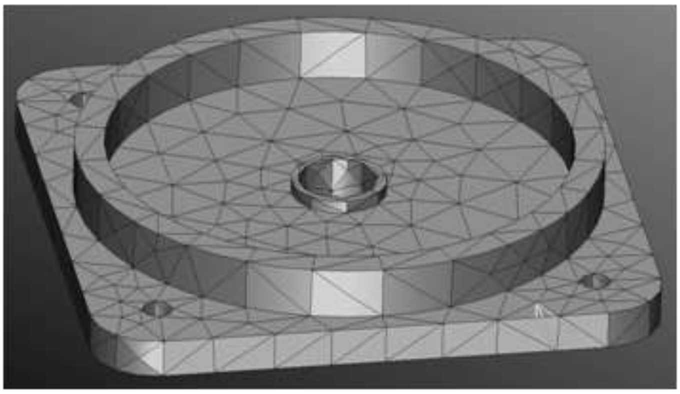

Fig. 5 Meshed model of initial design

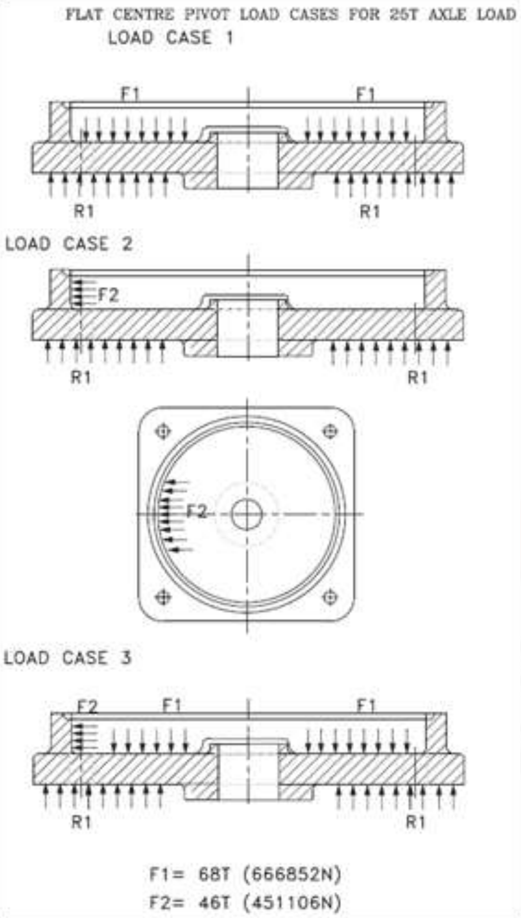

Fig. 6 Load and boundary condition developed

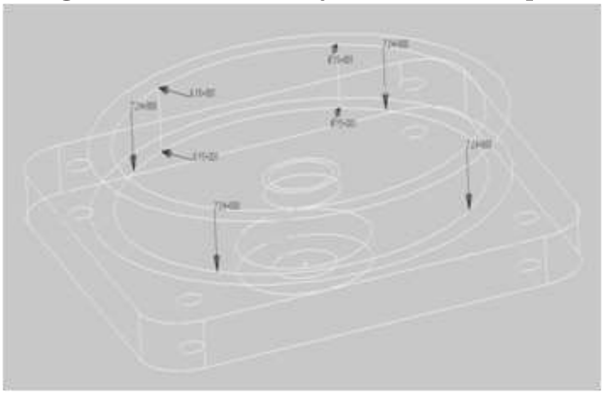

Fig. 7 Load and boundary condition applied

Table 3: Behavior of stress and deflection

SN	R_1 (mm)	L_1 (mm)	R_2 (mm)	Max. Stress (MPa)	Max. Deflection (mm)
1	197.5	430	36	286	0.047

2	197.5	430	38	287	0.041
3	197.5	430	40	276	0.048
4	197.5	425	36	282	0.049
5	197.5	425	38	296	0.047
6	197.5	425	40	318	0.047
7	197.5	420	36	298	0.049
8	197.5	420	38	272	0.049
9	197.5	420	40	255	0.043
10	195	430	36	284	0.052
11	195	430	38	267	0.045
12	195	430	40	285	0.051
13	195	425	36	304	0.053
14	195	425	38	266	0.048
15	195	425	40	281	0.048
16	195	420	36	282	0.094
17	195	420	38	279	0.053
18	195	420	40	271	0.054
19	192.5	430	36	273	0.053
20	192.5	430	38	317	0.058
21	192.5	430	40	316	0.058
22	192.5	425	36	336	0.056
23	192.5	425	38	294	0.051
24	192.5	425	40	338	0.056
25	192.5	420	36	318	0.055
26	192.5	420	38	370	0.057
27	192.5	420	40	340	0.053

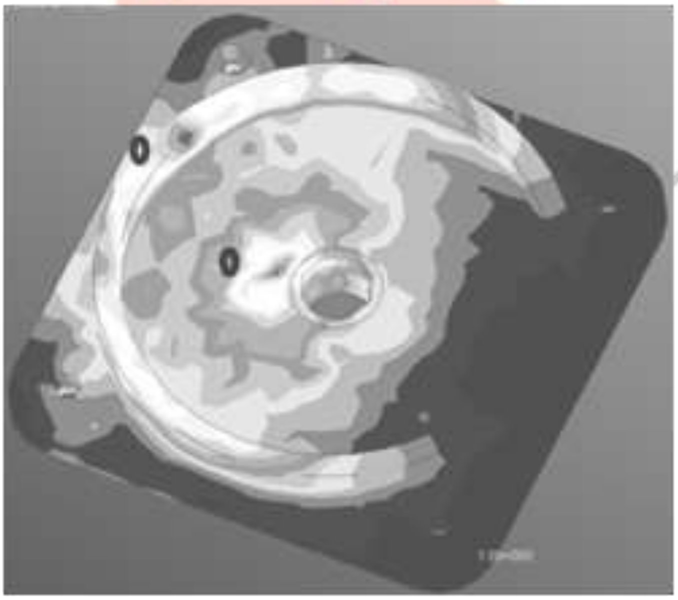

Fig-8: Transient Analysis of Initial design

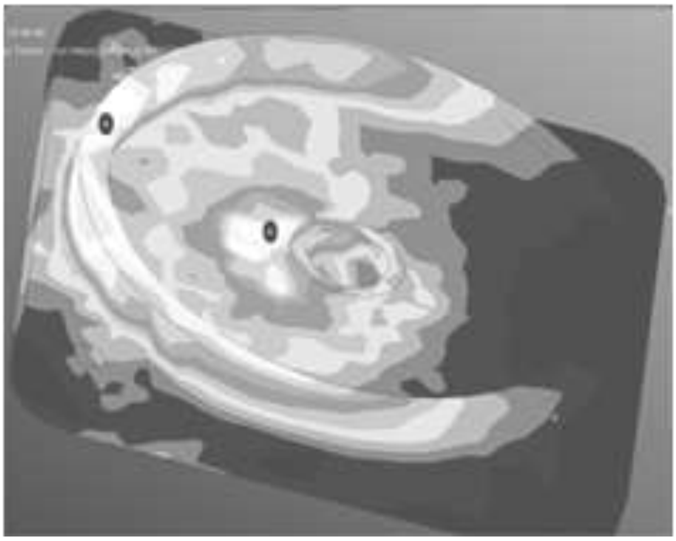

Fig-9: Transient Analysis of modified design

Modal Analysis

Modal analysis [19] & [20] of the initial and final design of bottom centre bearing plate is performed using Finite Element platform in free-free condition to verify the topology of the component. The first four mode shapes and natural frequencies are listed in Table 4. The modes of initial design are corresponding to modified design.

III.CONCLUSION

Outer cup radius R_1, base length L_1, and Outer hole radius R_2 dimensions of bottom centre bearing plate are considered for weight optimization performing structural analysis of bottom centre bearing plate using finite element methods applying Indian Railways loads and boundary conditions as directed in AAR M-202[1]. For parametric study $3^3 = 27$, models has been developed varying the above three dimensions. The finite element software MSC FEA is used to perform structural analysis of above 27 models. After doing little iteration the modified dimensions are selected satisfying the strength consideration. The weight of the modified design is reduced by 2.72% by initial one. Fatigue strength of both initial and modified design has been verified performing transient analysis considering Indian railway track signature as inputs on Goodman diagram. The mode shape of initial and modified designs are correspond to each other satisfying topological similarities

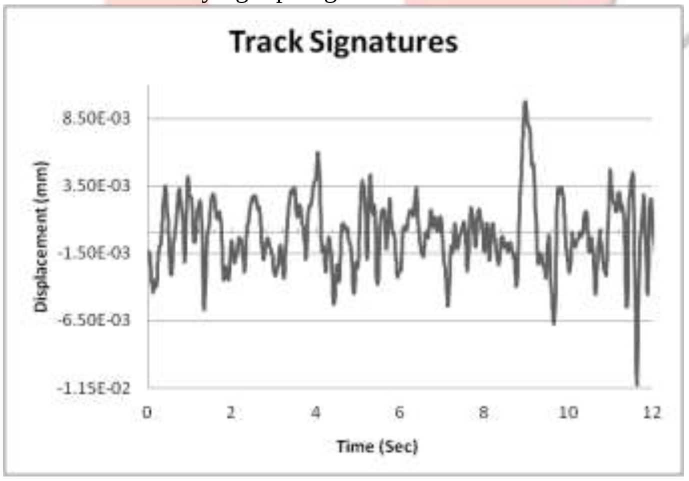

Fig-10: Indian railways track signatures

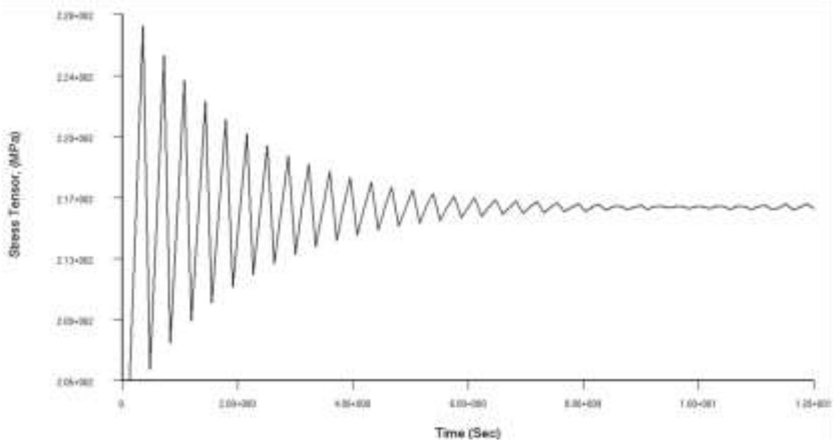

Fig-11a: Stress variation at L_1 of initial design at critical node No. 107

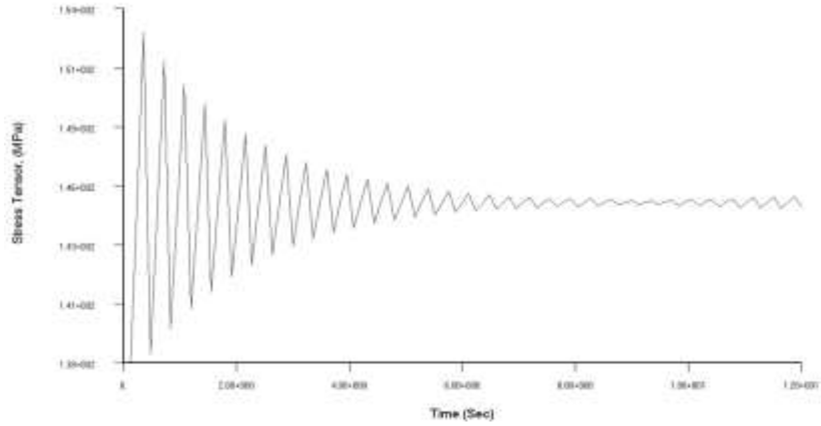

Fig-11b: Stress variation at R_1 of initial design at critical node No. 970

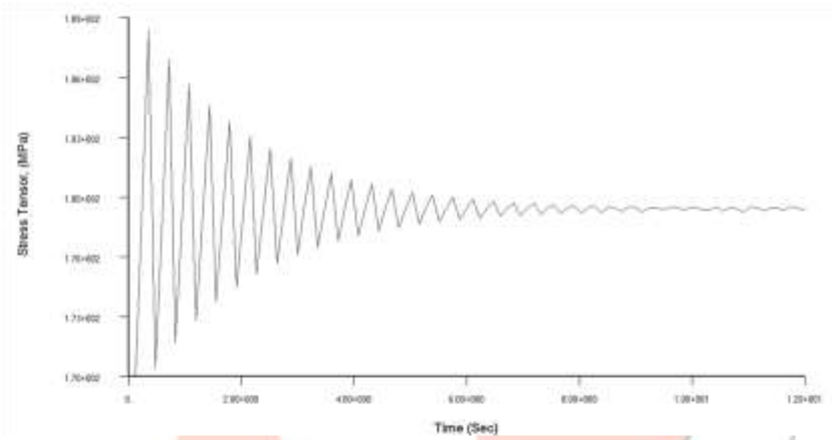

Fig-11c: Stress variation at R_2 of initial design at critical node No. 2609

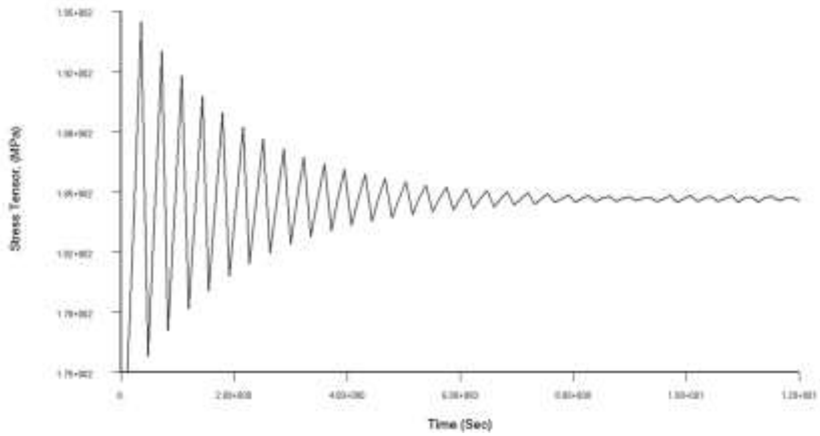

Fig-12a: Stress variation at L_1 of modified design at critical node No. 250

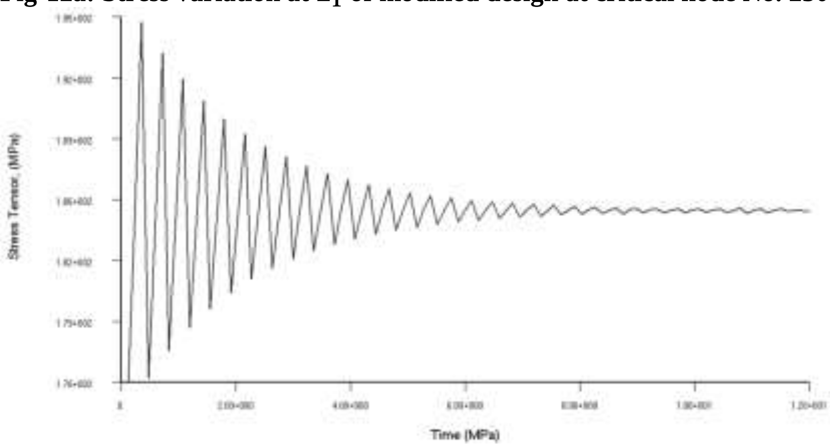

Fig-12b: Stress variation at R_1 of modified design at critical node No. 3141

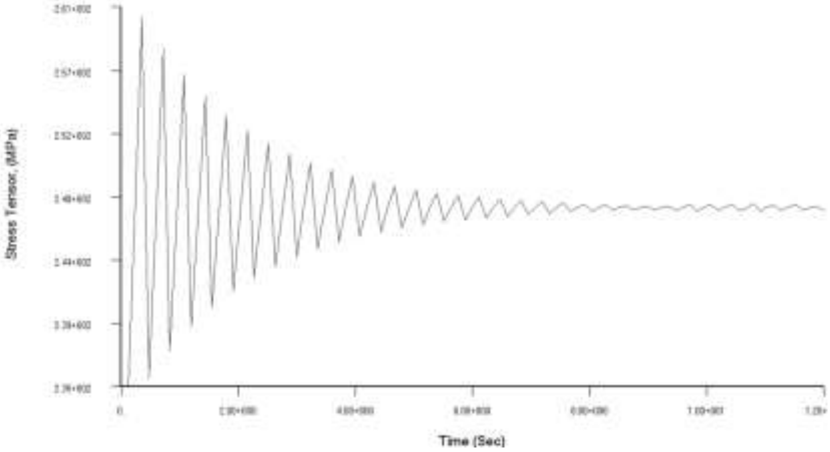

Fig-12c: Stress variation at R_2 of modified design at critical node No. 3209

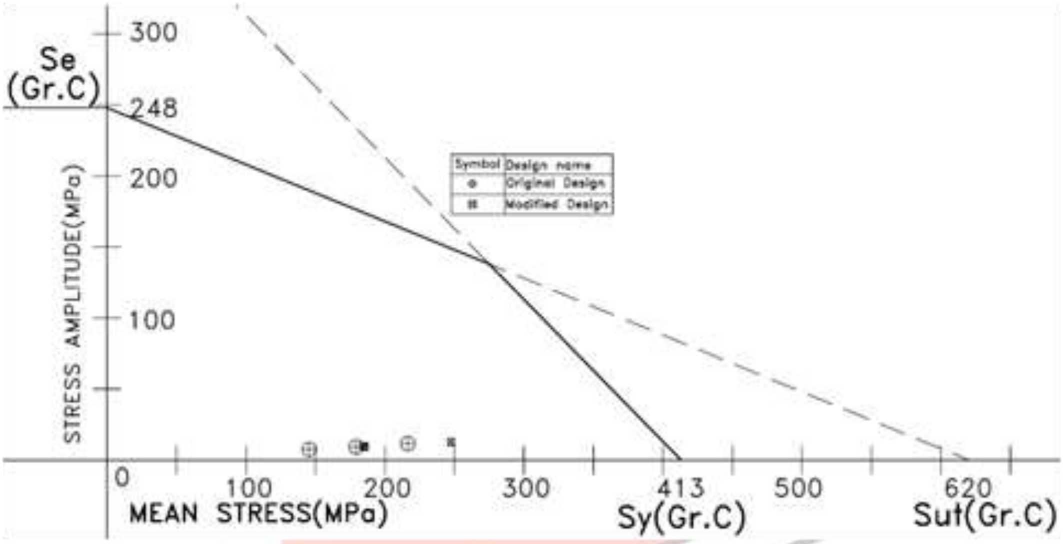

Fig-13: Goodman Diagram of initial and modified designs

IV. ACKNOWLEDGEMENTS

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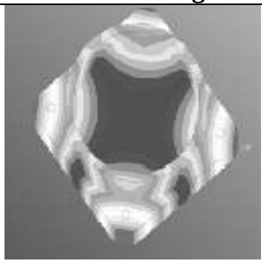
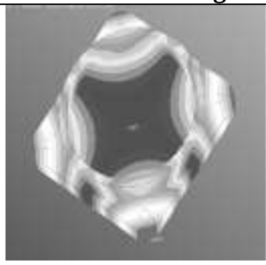
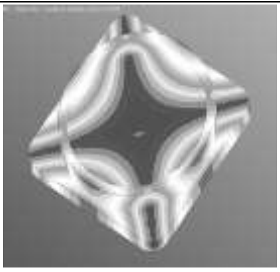
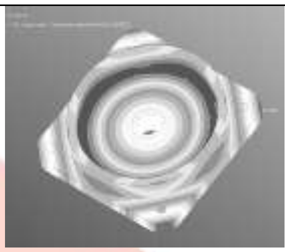
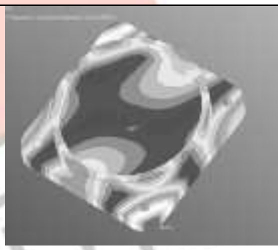
Table 4: Mode shapes of initial and modified designs

Mode	Mode shape of initial design	Mode shape of modified design
1	 731.89 Hz	 757.0 Hz
2	 994.46 Hz	 1021.1Hz
3	 1457Hz	 1503.8Hz
4	 1857.2Hz	 1929.5Hz

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