Topography Optimization: An Effort to Reduce the Weight of Bottom Centre Pivot

Prashant Kumar Srivastava, Simant, Sanjay Shukla

Abstract: For the growth of any nation, revenue generation is prime concern. Indian railway majorly freight movement plays a vital role in this regard. Focusing on increasing carrying capacity, weight reduction of bogie and its components become prime concern. This requires existing design improvement of critical components. In present research work topology optimization is carried out for the design alteration of bottom centrecentre pivot plate i.e. Centre pivot using ANSYS interface. Centre Bearing Plate, an integral component of three piece freight bogie, it balances and transfer various forces generated during motion of the vehicle. Pseudo-density(pf) are the design variables assigned to each finite element changes from 0 to 1. pf = 0 represents for removing of the material and pf = 1 represents for retention of the material. Further, design is modified according to the result obtained from iteration performed by FEA tool, resulting approximate weight reduction6.23%. Natural frequencies and their respective mode shapes of initial and modified designs are compared to validate the topology of the designs.

Key Words: Three piece freight bogie, Bottom Centre Bearing Plate, Pseudo-density, Weight Reduction, mode shape.

1. INTRODUCTION

Indian railway is backbone of the transportation system. Energy savings is important for production as well as transportation system, can be achieved primarily with developing light weight vehicles. Reduction of small amount of the weight of bogie components(s) results into enormous savings in terms of energy and material as well as increment to pay load. The weight and strength are always critical design parameter for rail vehicle designers. Topology optimization is being performed to modify the existing design Bottom Centre Bearing Plate concerning volume and weight reduction as objective function for visualizing most favorable material distribution inside the selected design space.

In present work, bottom centre pivot a critical bogie component of three piece freight bogie is considered for optimization. A railway freight vehicle having three piece freight bogies (shown in Fig.1a) consists of side frames, bolster and bottom centre pivot as shown in Fig.1(b). Solid model of the initial design is developed in NX-7.5 interface. Further the model is used for optimization process using Topology module of ANSYS platform applying load and boundary condition as per International Standard of Association of American Railroad (AAR) M-202[1]. The details of bogie fitted bottom CentrePivot shown in Fig. 2. The existing bottom centre pivot is designed for Indian Railways operating and loading conditions. Centre Pivot connects the rail car body and the chassis of bogie.

Fig. 1(a): Freight vehicle having three piece bogies

Fig. 1(b): Three piece freight bogie components

The development of topological optimization can be majorly credited to Bendsoe and Kikuchi [5] & [6]; They presented a homogenization based approach of Topology Optimization. Mathematical optimization procedures are applied to structures by Save etal.[7]. The state of structure is described by means of generalized loads, displacements, stress and strains. They resulted that the final structures had the minimum compliance for respective volume structures. Optimal material distribution in the shape design of structures is explained by Bendsoe [8]. Artificial density is considered as design variables and

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Prashant Kumar Srivastava, Department of Mechanical Engineering, Ambalika Institute of Management & Technology, Lucknow India
Simant, Department of Mechanical Engineering, Sagar Institute of Technology & Management, Barabanki, India
Sanjay Shukla, Research Design and Standards Organisation, Ministry of Railways, Government of India, Lucknow, India

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particular weight is assigned to these densities for filtering in between 0 to 1. The regions with dense cells having higher density numbered as 1 defined as structural shape, and those with void cells numbered as 0 having undesired material considered for removal.

Garth et al. [9] explained a 22 degree offreedom freight vehicle dynamics model which was developed by Tse and Martin. The complexity of the model was reduced by simulating while neglecting dynamic response of bolster and considering fundamental non linearity associated with centre plate separation, side bearing contact, wheel lift and friction damping. They explained the calculation of vehicle response time history for cross level variation. They also concluded that the some parameters viz. centre of gravity, height and critical speed was inversely proportional to each other.

Park et al. [10] developed a model of Korean passenger vehicle bogie frame for fatigue constraint weight optimization. Finite Element (FE) methods were used to evaluate fatigue strength of bogie frame. Further 4.7% weight of bogie frame was reduced through Genetic Algorithms (G.A.).

Shukla et al. [11] developed a Multi body dynamics model of CASNUB freight bogie to improve riding of the vehicle. They performed parametric study of suspension elements.

Kim et al. [12] performed experimental and numerical analysis of an experimental tilting train bogie bolster by means of two ways i.e. static and fatigue loading test. The safety against fatigue was checked using Goodman diagram of the material used. They also suggested removal of the material.

Li et al. [13] emphasized on the safety of bogie frame of Beijing subway vehicle under overload situation. Nominal stress method and Goodman relation were used for fatigue strength evaluation. The experiment and fatigue analysis results had suggested the weak location for failure under gear box bracket.

Tang et al. [14] evaluated static and fatigue strength of Diesel Multiple Units (DMU) bogie frame with Goodman plots. There was no high frequency vibration in running condition of trains, so modal analysis of the bogie frame were performed in free or no load conditions. Theoretical operational constraints of the design were checked. It was experienced that stress amplitudes were less than fatigue limit, which implies that bogie frame fulfill the requirements of fatigue strength.

Bubnov et al. [15] focused their work on caseted CASNUB bogie parts i.e. side frame and bolster on axle load of 245 kN. They found that side frame and bolster was most stressed part of the bogie. Structural analysis was performed on the basis of allowable stress and the safety factor of fatigue strength. Experimental investigation and theoretical analysis provide confirmation to the selected bogie for future use.

Shukla et al. [16] developed a finite element model of CASNUB three piece freight bogie frame. Transient analysis of the bogie frame was performed to calculate the fatigue strength. MATLAB platform was used to reduce the weight of the bolster fitted bottom centre bearing plate.

Prashant et al. [17] reduced the total volume of mechanical component using ANSYS software on the same von-Mises stress constraint of given load. The total volume of component was reduced by 22.5% and best design set of dimensions was acquired.

In the Hyper works Technology Conference, Varun Ahuja et al. [18] discussed about topology optimization procedure on Opti-struct software platform to reduce the weight of Engine Mounting Bracket and concluded 15% weight saving. M.V.Aditya [19] has reduced the weight of engine mounting bracket by 40% without compromising the strength by topology optimized design. Naveen et al. [20] optimized the CASNUB bogie design through approaches of size and shape optimization suggests some changes in the design so as to enhance the strength of the bogie without reducing payload carrying capacity. Various structural optimization methods have briefly explained by Prashant et al. [21] on size, shape and topology optimization.

**II. METHODOLOGY**

The weight reduction of bottom central pivot is achieved by stress constraint topology optimization [22]. The objective function is the volume of the bottom central pivot. Pseudo-Density ($\rho_e$) is the design variables assigned to each finite element ranging from 0 to 1. More elaboration of design variables can be written as $\rho_e = 0$ represents the remove of the material, $\rho_e = 0$ to 0.4 suggests removal of the material, $\rho_e = 0.4$ to 0.6 suggests marginal retention of the material, $\rho_e = 0.6$ to 1 suggests preservation of the material. The topology optimized model is obtained after performing iteration by ANSYS interface [23]. The weight of the bogie bottom pivot is optimized and initially reduced by 8.60%.

Equation (1) shows a stress based topology optimization. The problem is formulated to minimize volume as well as weight of a given design space under von-Mises stress constraint. The failure theory of this constraint optimization state that “A material will not sustain if the von-Mises stress induced in the material exceeds yield strength”. It means that for safe design and Mises stress is always less than the yield stress of the material.

$$\min V(x) = \sum_{e=1}^{n} (\rho_e)^\alpha \sigma_{e}$$

Subjected to:

$$\frac{\rho_e}{\sigma_{yield}} \leq 1$$

$$KU = F$$

$$0 < \rho_{min} \leq \rho_e \leq 1$$

Here, $V$ is the volume (objective function), $N$ is total number of elements which defines the design space, $e$ is element within the design space, $\sigma_e$ is von-Mises stress (Constraint), $\sigma_{yield}$ is maximum yield stress of each element in the design space, $P$ is penalization factor. $K$ is global stiffness matrix, $U$ is global displacement vector, $F$ is global force vector, $\rho_e$ is pseudo-density (design variable), $\rho_{min}$ is the minimum pseudo-density to
control the singularity phenomenon associated with the design variable.

III. STRUCTURAL ANALYSIS OF INITIAL MODEL

Structural analysis of a freight car bottom centre pivot has been performed using the finite element method for various loading and boundary conditions for 25 ton axle load. The drawing of the initial bottom central pivot design is shown in Fig. 3 is used to develop solid model using UGS NX-7.5 interface [24]. The model is meshed by adaptive procedure having 11578 nodes and 6651 elements. Load cases and boundary conditions proposed according to International Standard of AAR M-202 shown in Fig. 4. The casted steel recommended material properties [25] used for FE analysis [26] shown in Table 1. Critical stress zones are shown for initial design. The magnitude of stress and deformation at critical zones of initial design for applied load cases are listed in Table 2. Further Load Case (F1 & F2 i.e. maximum applied loads) on Centre Pivot is chosen for analysis and optimization work.

| S.No. | Load Cases (N) | Eqv. Von-Mises Stress (MPa) | Directional Deformation (mm) |
|-------|----------------|-----------------------------|-----------------------------|
| 1     | F1 = 666852    | 34.75                       | 0.002                       |
| 2     | F2 = 451106    | 270.44                      | 0.080                       |
| 3     | F3 = 441300    | 23.0                        | 0.001                       |
| 4     | F1 & F2       | 270.44                      | 0.080                       |

Stress variation and deformation are shown in Table 3 applying maximum forces i.e. F1 and F2 in each direction, the behavior of these values are within the permissible range.

Table 2: Stress and Deformation for Load Cases

A. Results
The design for optimum material distribution is obtained by performing Topology optimization using a module of ANSYS. For volume objective function, Von-Mises stress constraint (limited to 270.44 MPa). Density based optimization is performed for weight reduction. Iteration no. 6 suggested the best result for optimum distribution of material within the selected design space. The model is now reduced to 44.59 kg, while the initial one was 48.79 kg. Initially, the total weight saving is obtained by 8.60%. The modified design is shown in Table 4. The comparative results on the basis of mass and volumes of are also shown in Table 5. Further, according to topological investigation, the model is modified by removing undesired material also maintaining symmetry for all four sides. Structural check will performed on modified model on the basis of von-Mises stress and deformation.

Table 5: Topology optimized design

Table 1: Material Properties for Cast steel

| Property                  | Value |
|---------------------------|-------|
| Young's Modulus (GPa)     | 200   |
| Poisson's ratio           | 0.3   |
| Ultimate Tensile Strength (MPa) | 619.92 |
| Yield stress (MPa)        | 413.28 |
| Endurance Limit (MPa)     | 247.97 |

Table 4: Topology optimized design

Top View

- Remove (0.0 to 0.4)
- Marginal (0.4 to 0.6)
- Keep (0.6 to 1.0)

Bottom View

- Remove (0.0 to 0.4)
- Marginal (0.4 to 0.6)
- Keep (0.6 to 1.0)
Table 5: Comparison of original and Topology optimized design

| Models       | Volume($mm^3$) | Mass (kg) | Percentage Saving |
|--------------|----------------|-----------|-------------------|
| Original     | 6216500        | 48.79     |                   |
| Optimized    | 5680900        | 44.59     | 8.60              |

IV. MODIFIED MODEL & ANALYSIS

According to suggested material distribution, initial design is modified by removing materials uniformly along 4 sides of the original Centre Bearing Plate (i.e. Centre Pivot) of length 330 mm and radius of curvature of 680 mm. The drawing of modified design is shown in Fig. 5. The weight of modified Bottom Centre Bearing Plate is now 45.75 kg shown in Fig.6(a). The weight saving as per final modified design is obtained approximately 6.23% by initial one.

Fig. 5: Drawing of modified design

The structure analysis of the modified design is performed on to verify the strength. The meshing of design is performed by creating coarse size 6715 no. of elements and 11647 no. of nodes as shown in Fig.6(b). Load case (i.e. application of force $F_1$ & $F_2$) and boundary conditions is considered as same as the applied on initial design. The structural analysis is performed. Stress and deformation plot is obtained shown in Table 6. The comparative analysis of stress and deformation of both original and modified designs are shown in Table 7. It is observed that the parameters are within the permissible range and confirms interchangeability [27] of designs.

Table 6: Stress and Deformation plot of modified design

| Stress Plot | Deformation Plot |
|-------------|------------------|
| Von - Mises Stress = 254.40 (MPa) | Directional Deformation (X-axis) = 0.081 mm |

Table 7: Comparison of Original and Modified Design

| Design | Load Cases | Von-Mises Stress (MPa) | Directional Deformation (mm) | Vol. ($mm^3$) | Mass (kg) | Saving (%) |
|--------|------------|------------------------|-----------------------------|---------------|-----------|------------|
| Original | $F_1$ & $F_2$ | 270.44                 | 0.040                       | 6216500      | 48.79     | 6.23       |
| Modified | $F_1$ & $F_2$ | 254.40                 | 0.081                       | 5827500      | 45.75     |            |

Fig.6(a) & 6(b): Modified Solid and Mesh Model

V. MODAL ANALYSIS

The modal analysis of initial and modified design is carried out using ANSYS interface to verify the interchangeability of designs. Initial four mode shapes [28 & 29] of original and modified designs are extracted as shown in Table 8. These mode shapes are corresponding to each other satisfying outer topology of the designs.
Table 8: Mode shape of original and modified mode

| Mode | Initial Design | Modified design |
|------|---------------|-----------------|
| 1st  | 835.69 Hz     | 860.5 Hz        |
| 2nd  | 1109.1 Hz     | 1212.9 Hz       |
| 3rd  | 1602.3 Hz     | 1614.4Hz        |
| 4th  | 1831.6 Hz     | 1853.4Hz        |

VI. CONCLUSION

In present research work, existing design of 25 ton axle load bottom central pivot is verified for Indian Railway design parameters using FE platform. The design module is formulated for topological optimization subject to stress as constrained and volume of the component as objective function. The pseudo-density ($\rho_e$) is considered as design variables for each finite element range. Further an effort of topology optimization is performed by means of ANSYS interface. Initially the weight of the initial design is reduced by approximately 8.61%. The design is remodeled making an allowance for material distribution suggested by pseudo-density based optimization outcomes, and considering the symmetry of the model. The final materialsavingby performing optimization exercise is approximate 6.23%. Strength of the modified design is tested through structural investigation. The stress and deformation pattern of the modified and initial designs are in same approach. Topology of initial and modified design is verified by executing modal analysis of both designs.

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AUTHORS PROFILE

Mr. Prashant Kr.Srivastava is Asst. Professor of Department of Mechanical Engineering in Ambalika Institute of Management & Technology, Lucknow, having experience of more than 13 years in education field and a Ph.D. Scholar in the field of Design. He had completed his M.Tech in Mechanical Engineering Design from Harcourt Butler Technological Institute, Kanpur (currently known as Harcourt Butler Technological University, Kanpur), Uttar Pradesh, India in year 2011 and Bachelor of Technology from M.I.E.T., Meerut, Uttar Pradesh. He has authored two books in Mechanical Engineering and published many papers in National and International journals. His main field of research is design optimization approach applied to mechanical structure and components.

Dr. Simant, is Professor and Director of Sagar Institute of Technology & Management, Barbanki, having experience of more than 19 years in education field and two and half years Industry experience. He had completed his PhD in Mechanical Engineering from Uttar Pradesh Technical University (currently known as Dr. A. P. J. Abdul Kalam Technical University), Lucknow, Uttar Pradesh, India in year 2013 and Master of Technology from M.N.I.T., Allahabad, Uttar Pradesh, India in Computer Aided Design & Computer Aided Manufacturing (CAD/CAM). He has published three books in Mechanical Engineering and authored more than Twenty National and International papers. His main field of research is in Composite Materials, Finite Element Method and Advance Mechanics of Solids.

Dr. Shukla, is Senior Research Engineer (Design) in Government of India, Ministry of Railways, Research Designs & Standards Organization, Manak Nagar, Lucknow (India), having experience of more than 25 years in Research/Industry field. He had completed his PhD in Mechanical Engineering from Uttar Pradesh Technical University (currently known as Dr. A. P. J. Abdul Kalam Technical University), Lucknow, Uttar Pradesh, India in year 2012 and Master of Technology from V.N.I.T., Nagpur, Maharashtra, India, in Linkage Design. He has published more than Ten National and International papers. His main field of research to design & develop the railway vehicle performing Flex Multi Body Dynamics Analysis using the coupling of Finite Element Method and Multi Body Dynamics interface.