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 examination and development of critical components. In present research work topological optimization is carried out for the design alteration of bottom centre Bearing 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 (ρ) are the design variables assigned to each finite element changes from 0 to 1, ρ = 0 represents for removing of the material and ρ = 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 reduction 6.23%. 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 considering 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 Centre Pivot 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.

It balances and transferred various forces generated during vehicle motion. In addition, the bolster platform is connected to a transom of the bogie chassis. Centre Pivot facilitates the curve negotiation of vehicle at higher speeds. It bears 5% of superstructure weight and 50% of pay load along with various forces like braking and tractive forces.

The optimization of any engineering problem is basically the maximization or minimization of an objective function [2] subjected to given constraints. Stress constrained [3] volume minimization is our target work. Optimization of mechanical structures can easily be traced back to the start of 20th century when Michell [4] derived formulae were used for structures to minimize weight under stress constraints.

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...
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. Garret al. [9] explained a 22degree offreedom freight vehicle dynamics model which was developed by Tse and Martin. The complexity of 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 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 performing parametric study of suspension elements. Kim et al. [12] performed experimental structural analysis of Korean 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. 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. As 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. Thereal 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 casted 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 CASTNB 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 optimization 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 centre pivot. Pseudo-Density ($\rho_e$) is the design variables assigned to each finite element ranges 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 ANSYSiemens[23]. The weight of the bottom bogie 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 von-Mises stress is always less than the yield stress of the material.

Min $V(x) = \sum_{e=1}^{n} (\rho_e)^{\alpha} \sigma_{vonMises} \leq \sigma_{yield}$ \hspace{1cm} (1)

Subjected to $\rho_e \leq 1$

$KU = F$

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

Here, $V$ is the volume (objective function), $N$ is total number of elements which defines the design space, $e$ is elements within the design space, $\sigma_{vonMises}$ is von-Mises stress (Constraint), $\sigma_{yield}$ is maximum yield stress, $P$ 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.

Table 2: Stress and Deformation for Load Cases

<table>
<thead>
<tr>
<th>S.No.</th>
<th>Load Cases (N)</th>
<th>Eqv. Von-Mises Stress (MPa)</th>
<th>Directional Deformation (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>F₁ = 666852</td>
<td>34.75</td>
<td>0.002</td>
</tr>
<tr>
<td>2</td>
<td>F₂ = 451106</td>
<td>270.44</td>
<td>0.080</td>
</tr>
<tr>
<td>3</td>
<td>F₃ = 441300</td>
<td>23.0</td>
<td>0.001</td>
</tr>
<tr>
<td>4</td>
<td>F₁ &amp; F₂</td>
<td>270.44</td>
<td>0.080</td>
</tr>
</tbody>
</table>

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.

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%, 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 4: Topology optimized design

<table>
<thead>
<tr>
<th>Top View</th>
<th>Bottom View</th>
</tr>
</thead>
<tbody>
<tr>
<td><img src="image1" alt="Top View" /></td>
<td><img src="image2" alt="Bottom View" /></td>
</tr>
</tbody>
</table>

![Fig. 3: Drawing of Bottom Centre bearing plate](image3)

![Fig. 4: Various Load Cases](image4)

![Table 1: Material Properties for Cast steel](image5)

<table>
<thead>
<tr>
<th>Young's Modulus (GPa)</th>
<th>Poisson's ratio</th>
<th>Ultimate Tensile Strength (MPa)</th>
<th>Yield Stress (MPa)</th>
<th>Endurance Limit (MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>200</td>
<td>0.3</td>
<td>619.92</td>
<td>413.28</td>
<td>247.97</td>
</tr>
</tbody>
</table>
Table 5: Comparison of original and Topology optimized design

<table>
<thead>
<tr>
<th>Models</th>
<th>Volume ($m^3$)</th>
<th>Mass (kg)</th>
<th>Percentage Saving</th>
</tr>
</thead>
<tbody>
<tr>
<td>Original</td>
<td>6216500</td>
<td>48.79</td>
<td></td>
</tr>
<tr>
<td>Optimized</td>
<td>5680900</td>
<td>44.59</td>
<td>8.60</td>
</tr>
</tbody>
</table>

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 approximate 6.23% by initial one.

![Fig. 5: Drawing of modified design](image)

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 are 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.

![Fig. 6(a) & 6(b): Modified Solid and Mesh Model](image)

Table 6: Stress and Deformation plot of modified design

<table>
<thead>
<tr>
<th>Stress Plot</th>
<th>Deformation Plot</th>
</tr>
</thead>
<tbody>
<tr>
<td>Von Mises Stress = 254.40 ( MPa )</td>
<td>Directional Deformation (X-axis) = 0.081 $mm$</td>
</tr>
</tbody>
</table>

Table 7: Comparison of Original and Modified Design

<table>
<thead>
<tr>
<th>Design</th>
<th>Load Cases</th>
<th>Von Mises Stress (MPa)</th>
<th>Directional Deformation (mm)</th>
<th>Vol. ($m^3$)</th>
<th>Mass (kg)</th>
<th>Saving (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Original</td>
<td>$F_1$ &amp; $F_2$</td>
<td>270.44</td>
<td>0.080</td>
<td>6216500</td>
<td>48.79</td>
<td>6.23</td>
</tr>
<tr>
<td>Modified</td>
<td>$F_1$ &amp; $F_2$</td>
<td>254.40</td>
<td>0.081</td>
<td>5827500</td>
<td>45.75</td>
<td></td>
</tr>
</tbody>
</table>

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.
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 ($\rho_c$) 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 material saving by 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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