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Optimization methodology for beam gauges of the bus body for weight reduction

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Abstract

During service, a bus is subjected to various loads that cause stresses, vibrations and noise in the different components of its structure. It requires appropriate strength, stiffness and fatigue properties of the components to be able to stand these loads. Besides, quality and optimum weight of the vehicle, for efficient energy consumption, safety and provision of the comfort to the user are highly desired. The present work proposes a methodology to minimize the bus weight by modifying its beam gauges to optimum thickness. The bus performance is evaluated by multiple iterations on the basis of parameters like frequency, distortion, stress and stiffness. The algorithm performs gauge optimization of the bus by analyzing and satisfying its structural strength through linear static analysis on a laden bus. It also performs structural stiffness analysis and vibration analysis for safety of the bus structure. This work unfolds an integrated methodology to the bus manufacturers to optimize the structural weight for improving the fuel efficiency, static and dynamic safety, and robust design. The work is implemented by creating a finite element model of the bus and optimizing in HyperWorks environment. The results are verified for a full length 11 m, 65 seats bus. The methodology helps in weight reduction along with improvement in performance parameters.

Keywords: bus body, gauge optimization, weight reduction, linear static analysis, stiffness, vibration analysis

1. Introduction

Due to recent advancements in technologies and need for a sustainable environment, the latest trend in automobile industry is towards development of light weight vehicles for reduced gas emissions and better fuel economy. It is either done by optimizing the existing auto body structure using advanced computational techniques or foraying into design improvements. Innovations in existing designs are profitable in comparison to new product designs due to existence of established manufacturing and production units, and acceptability by the market. Thus, the aim of this work is to reduce the weight of the bus structure by optimizing the beam thicknesses, achieving in process, environmental and manufacturing benefits, without compromising the structural performance.

Computer Aided Engineering (CAE) analysis, if done prior to fabrication of the product, is a time and cost saving tool in product design cycle. For engineering products, it is executed at earlier phases of design, after conceiving the concept designs in CAD environment. The CAE analysis done in this research is based on Finite Element Analysis (FEA), assisted by well-advanced HyperWorks software tools such as HyperMesh for pre-processing, Optistruct

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for structural analysis and optimization, and HyperView for post-processing and result visualization.

Optimization techniques applied for structural design is a major research area to reduce the weight of vehicles and form refinement. The performance of the bus is estimated by linear static analysis [5, 9], evaluating the strength and deformation of the bus structure when subjected to service loads while the bus is on road. Many times, the structure of the bus body is designed that is sufficiently strong yet unsatisfactory due to insufficient stiffness. Designing for acceptable stiffness [3, 11, 19] is, therefore, often more critical than designing for sufficient strength. Dynamic analysis [2, 9, 16] of the structure is a concern for the safety of the structure and it is performed to evaluate the natural frequency and mode shapes of the structure. These analysis setups facilitate an iterative optimization [12, 14, 16], proving the performance criteria for the converged design solution of the bus to be reasonable under bending, torsion, buckling and dynamic loads. The design solution satisfies all the manufacturing and weight constraints and enhances performance in terms of mass, stiffness, frequency and deformation.

The primary objective of this work is to reduce the weight of passenger bus without compromising the performance of the vehicle; and if possible enhancing the structural performance of the bus body through employing FEA, structural analysis and optimization techniques to optimize beams of the bus.

2. Methodology

To optimize the bus body, an analytical study of the requirement is done and based on that, the following methodology is proposed, Fig. 1.

From the meshed model shown in Fig. 2, the geometrical model has to be extracted. Thus, the first step is to import the geometrical model of the bus from the free domain to the desired CAD environment or model it from scratch, which is a tedious task. The geometry of the bus body is divided into six frames, i.e., top, bottom, two sides, rear and front, for the ease of handling.

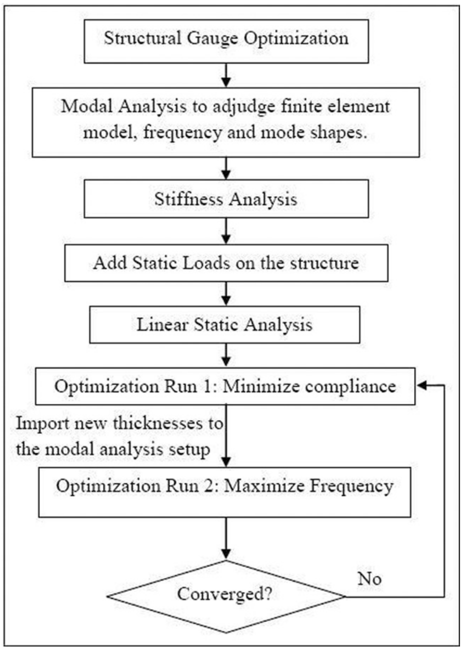


Fig. 1. Proposed algorithm

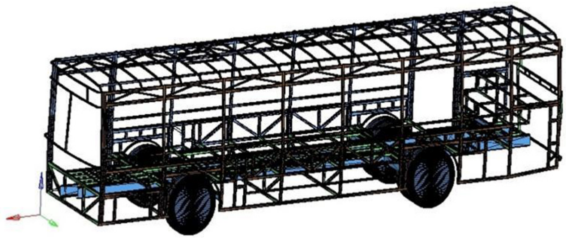


Fig. 2. Meshed bus model (from free domain)

Secondly, the median surfaces of the beams are extracted, which is followed by generation of finite element mesh on it. The meshed frames are joined replicating the weld joints between the beams. The whole meshed model is scrutinised under quality check to ensure no discontinuity or failure of elements. A finite element chassis model is prepared to support the bus structure and set up boundary conditions on the axle. Chassis model is resistant to any variation throughout the experiment.

Once the finite element model of the bus is prepared, it is subjected to (a) vibration analysis, (b) stiffness analysis and (c) linear static analysis. At first, vibration analysis is done to verify the meshed model and calculate first bending and torsion frequency. The model is verified with the established data which should be analogous to the natural frequency of a simply supported hollow beam, then the bending and torsion stiffness of the structure is calculated. This is followed by linear static analysis, for which the bus is loaded with external weights and equipment weights. The service loads are now imposed, categorized under bending, torsion, combined bending and torsion, lateral loading and longitudinal loading. These are done to analyze and visualize the cases for deformation and stress concentration, spotting the weak zones in the beams of the structure.

After procuring and fixing all the performance indices of the baseline model, the established model is to be optimized. The response parameters are frequency (to be maximized), displacement (to be minimized) and compliance (to be minimized), with an objective to reduce the mass. The design variables are the thickness of the various beams. Later, a comparison of the baseline and iterative optimized model is done. The model which complies with all the desired values of variables, responses and objective is set as optimum.

The gauge optimization of the bus body includes the following steps:

1. Modeling: The modeling of the bus body is a two-step process comprising of, first a geometric model of the bus and second, a finite element model. It would be good if both the models are developed in the same CAD environment. This helps in extraction of geometry of interconnected beams.

2. Pre-processing: This helps in defining the material, thickness and other essential properties of the bus. Pre-processor also allows the solver in the next step to predict the action of these elements and analyze the reaction to external forces and interactions.

3. Finite Element Analysis: FEA of the base model is carried out for strength, stiffness and safety. It includes linear static analysis to identify strength criteria that include maximum stresses and deflections; vibration analysis for dynamic behaviour; and bending and torsion analysis for stiffness benchmarking.

4. Vibration Analysis: Vibration analysis is done to ‘Design for Safety’. Every structure or body has its own natural frequency and when this frequency coincides, resonance occurs.

To prevent resonance, the natural frequency of the bus has to be high so that its frequency does not coincide with natural frequency of human, humps, engine vibration and other parts.

5. Stiffness analysis: The bending stiffness and torsion stiffness accounts for the overall strength of the bus structure. To evaluate the baseline stiffness of the bus, the structure is assumed to be a beam and its deformation on application of forces depends on its stiffness.

6. Optimization: In this work, structural optimization in the form of topology, shape and size are carried out. All three have a purpose in the design phase and the choice or combination depends on desirables. The purpose of parameter optimization is to find the minimum thickness of the inter-related beams to minimize the weight.

The basic idea of gauge and size optimization is to modify gauge properties so that the residual structure evolves towards an optimum solution. Gauge optimization [12] is a particular form of size optimization where design variables t_i are in the form of 2D shell elements only.

The element property p is derived as a function of design variable and constants, i.e., $p = f(DV_j, C_j)$, where DV_j is design variable and C_j is constant. The responses to be evaluated for optimum sizing of beam gauges [2, 6] would be done by iterative optimization to increase frequency on modal and decrease displacement on linear static analysis setups.

3. Implementation

In this work, the implementation of the proposed methodology is carried out with HyperWorks suite taking help from [10, 11, 15]. The HyperMesh interface facilitates both geometric modelling as well as the pre-processing of the model. FEA and gauge optimization of the bus structure is done with Radioss as solver. After solving and retrieving the results, visualization of the results of the analysis is supported by HyperView.

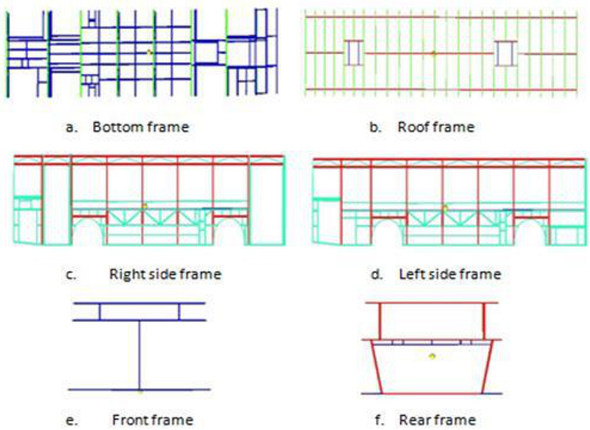


Fig. 3. Geometric models of the frames

The geometric models of various frames of the bus body are shown with the help of Fig. 3. Finite element model is represented by interconnection of a mesh of 2-D quad elements with average size 10 on the median surface of the beams. Both aluminium (Al) and steel were considered as possible materials for beams [3, 20], but in the present work, steel is preferred over Al, which in spite of being light, strong and stiff is expensive. Moreover, the results of the

Table 1. Material properties

Element type	2D shell	Beam
Material	Steel (IS:4923)	High strength steel
Young's modulus	$2.1 \cdot 10^5$ MPa	$2.1 \cdot 10^{10}$ MPa
Poisson's ratio ν	0.300	0.300
Modulus of rigidity G	$8.1 \cdot 10^4$ MPa	$8.1 \cdot 10^9$ MPa
Density	$7.9 \cdot 10^{-9}\ \mathrm{N \cdot mm^{-3}}$	$7.9 \cdot 10^{-9}\ \mathrm{N \cdot mm^{-3}}$

bus body made of steel can be validated through the published data. The material properties are defined in Table 1.

The mass of the bus structure after assigning material and thickness properties comes out to be 1.521 kg. To carry out the strength analysis of the bus structure, the bus body has to be statically loaded for the worst possible loads while on road. This is done using rigid elements (Rbe3) as shown in Fig. 4.

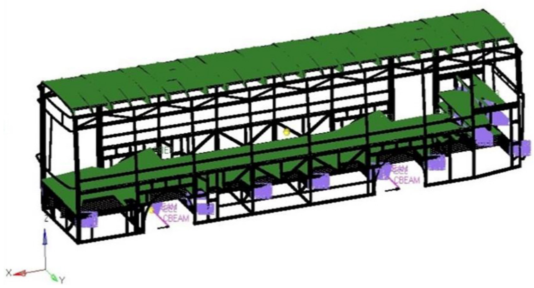


Fig. 4. Various loads on the bus body lumped using rigid (Rbe3) elements

The static loads as referred in [5, 13, 16] include the loads imposed, e.g., by passengers, engine, body coatings, windows and doors and are given in Table 2.

Table 2. Static loads

Mass category	Mass value [tonnes]
Bus structure	1.521
Chassis structure	1.153
Engine (rear mount)	0.5
Body coating	2.599
Windows and doors	0.575
Total passenger mass	6.00
Passenger mass · No. of person	$0.068 \cdot 65$
Seat and luggage	1.58
Other accessories	1.00
Total	13.416

3.1. Vibration analysis

The road induced frequencies are normally less than 5 Hz [1]; therefore, it is necessary to keep the natural frequency of the body higher than this. The natural frequency and modes are evaluated on solving the eigenvalues equations associated with the design model, and presented with the help of Table 3 and Fig. 5, respectively.

Table 3. First ten modes of free modal analysis

Modes	Frequency [Hz]	Mode shape
1	$4.26 \cdot 10^{-3}$	Rigid body mode
2	$7.79 \cdot 10^{-4}$	Rigid body mode
3	$3.26 \cdot 10^{-4}$	Rigid body mode
4	$1.22 \cdot 10^{-3}$	Rigid body mode
5	$1.25 \cdot 10^{-3}$	Rigid body mode
6	$1.40 \cdot 10^{-3}$	Rigid body mode
7	5.95	1st torsion mode
8	8.79	1st bending mode
9	10.36	2nd bending mode
10	11.74	Combined torsion and bending

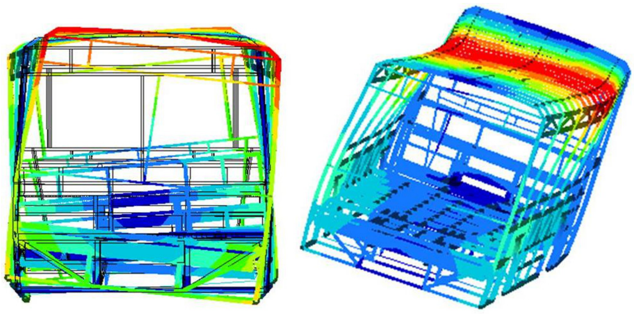


Fig. 5. First torsion and first bending mode

For free body modal analysis, the first six modes are rigid body modes. On visualisation of mode shapes, the seventh mode shows tendency of torsion, whereas the eighth mode comes out to be the first bending mode. Since the first torsion frequency is already greater than five, the objective is to enhance it further.

3.2. Stiffness analysis

The bus setup as shown in Fig. 6, where 1, 2, 3 represent the x, y, z directional constraints, respectively, is assumed to be a simply supported beam. It deforms upon application of certain force depending on the bending stiffness of the structure. Deformation is visualized along the length and for the bottom beam is presented in the form of a graph in Fig. 7.

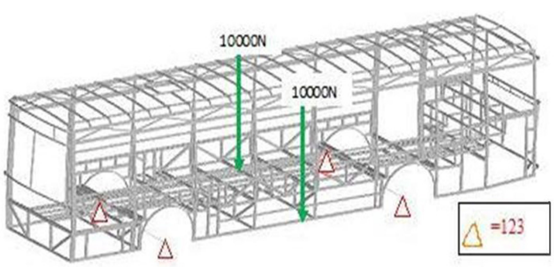


Fig. 6. FEA setup to calculate bending stiffness

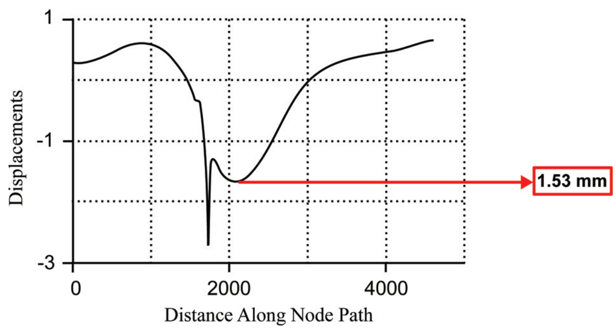


Fig. 7. Graph between the distance along the frame length and vertical displacement for the bottom beam

Torsion stiffness is a parameter to check the twist/deflection of the bus structure upon the application of torque. It should be high enough to resist the bus from twisting under various torsion loadings subjected to the vehicle. The torsional stiffness (TS) is defined with the help of the following equation:

$$\mathrm{TS} = \frac{F\,t}{\arctan\left(\frac{z_1+z_2}{t}\right)}, \tag{1}$$

where F is the force, t is the track width and z_1, z_2 are the displacements of the two spindles of the front axle. Fig. 8 shows the FEA setup to calculate the torsion stiffness, while Fig. 9 shows the displacements on the two spindles.

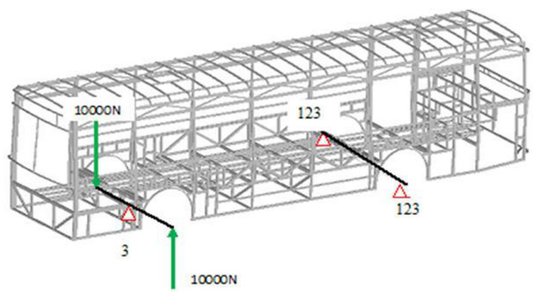


Fig. 8. FEA setup to calculate torsion stiffness

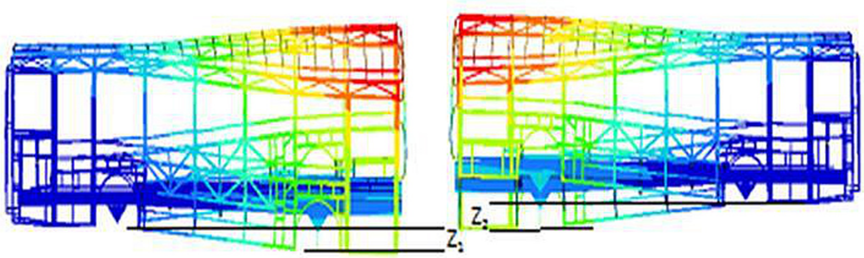


Fig. 9. Deflections z_1 and z_2

3.3. Linear static analysis

The loads imposed on the chassis and the body structure of a bus as the vehicle traverses uneven ground or when the driver performs various manoeuvres, lead to stresses and deformations. They have to be substantiated. The boundary conditions of loads are defined at the spindle locations of front and rear axles as mentioned in [7, 8, 11].

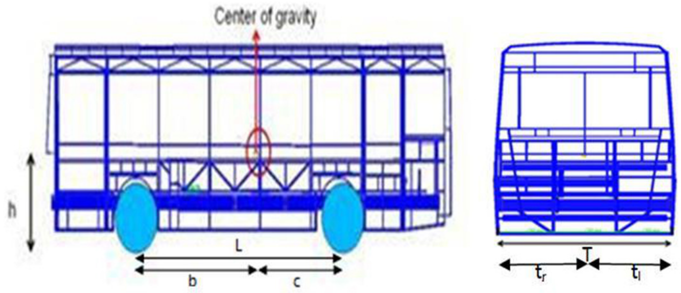


Fig. 10. Bus body dimensions

Table 4. Dimensions of the bus. Here, the abbreviation C.G. stands for the center of gravity

Variable	Definition	Measure
L	wheel base length	6 000 mm
b	distance from the C.G. to the front axle	3 566.5 mm
c	distance from the C.G. to the rear axle	2 433.5 mm
h	height from the ground to the C.G.	1 227.8 mm
T	track width of the bus	2 490 mm
t_r	lateral distance from the C.G. to the right frame	1 233 mm
t_l	lateral distance from the C.G. to the left frame	1 257 mm

Fig. 10 shows the bus model with proper dimensions, supported by its corresponding configuration in Table 4. To design for structural strength, all standard loads are grouped into five basic load categories, defined below:

(a) Bending case: It depends on the weights of the major components of the vehicle and the payload. Weight of the driver, engine, passenger, structure and chassis under an effect of gravity produces sag in the frame. Frame is assumed to act as simply supported beam and the four wheels as supports tend to produce reactions vertically upward at the axles as shown in Fig. 11. The relations for static load on the two axles and two wheels are given as

$$\begin{aligned} &\text{static loads on the front axle:} && W_{fs} = W\,\frac{c}{L}, \\ &\text{static loads on the rear axle:} && W_{rs} = W\,\frac{b}{L}, \\ &\text{static loads on the right wheel:} && W_{rw} = W_a\,\frac{l_t}{T}, \\ &\text{static loads on the left wheel:} && W_{lw} = W_a\,\frac{l_r}{T}, \end{aligned} \tag{2}$$

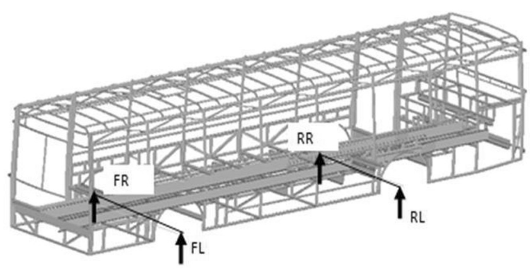


Fig. 11. Boundary conditions applied in the static case

where W is the weight of the loaded bus and W_a is the weight on axles. The percentage of weights on rear and front axles are 59.44 % and 40.56 %, respectively.

(b) Torsion: The maximum torsion is critical at the lighter loaded axle, and its value given as

$$\frac{R_f}{2}\,T_f = \frac{R_{r'}}{2}\,T_r, \tag{3}$$

where T_f and T_r are the front and rear end track widths, respectively, and R_f and $R_{r'}$ are the coupling forces on the front and rear axles, respectively, is the load on that axle multiplied by the wheel track width, Fig. 12.

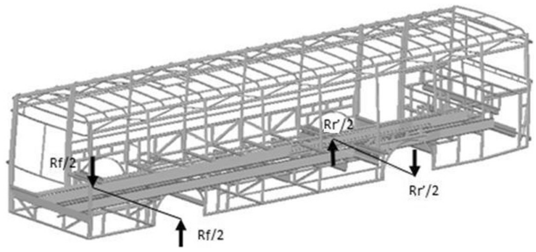


Fig. 12. Applications of boundary loads for torsion case

(c) Combine bending and torsion: The condition of pure torsion cannot exist on its own because vertical loads always exist due to gravity. Therefore, to get the realistic scenario, combined bending and torsion case is analysed as shown in Fig. 13. This figure shows a situation when one wheel of the lighter loaded axle is raised on a hump of sufficient height to cause the other wheel on the same axle to leave the ground (vertical bump case). In Fig. 13, R_r is the normal reaction force.

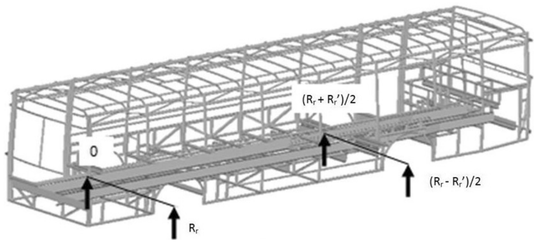


Fig. 13. Boundary loads for combine bending and torsion (vertical bump case)

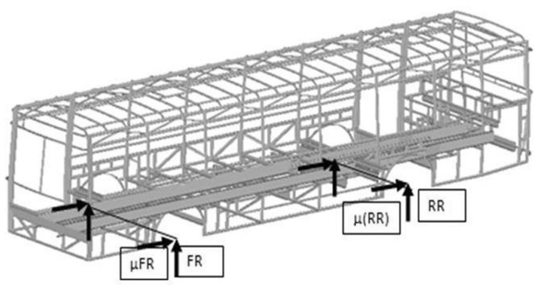


Fig. 14. Application of braking loads

(d) Longitudinal loading: When a vehicle accelerates (or decelerates), inertial forces are generated, Fig. 14. During acceleration, the weight is transferred from the front axle to the rear axle and vice versa in case of braking or decelerating conditions. The maximum load during acceleration is $A_x = 1g$ and the braking load is $D_x = 0.75g$ along the longitudinal axis. The distribution of axle loads is given as follows:

$$\begin{aligned}
&\text{load on the front axle during braking:}\\
&W_f = W\frac{c}{L} + D_x\frac{h\,W}{L\,g} = W_{\text{fs}} + W_d,\\
&\text{load on the rear axle during braking:}\\
&W_r = W\frac{b}{L} - D_x\frac{h\,W}{L\,g} = W_{\text{rs}} - W_d,\\
&\text{dynamic load transfer:}\\
&W_d = D_x\frac{h\,W}{L\,g}.
\end{aligned} \tag{4}$$

(e) Lateral loading: When the vehicle is turning, lateral loads (A_y) are generated at the tyre-ground contact patches, which are balanced by the centrifugal force, thus,

$$\begin{aligned}
&\text{lateral weight transfer:}\\
&W_{lt} = W\frac{A_y\,h}{g\,T},
\end{aligned}$$

when turning left,

$$\begin{aligned}
&\text{load on the left axis spindle:} && W_l = W\frac{t_r}{T} + W_{lt},\\
&\text{load on the right axis spindle:} && W_r = W\frac{t_l}{T} - W_{lt}.
\end{aligned} \tag{5}$$

3.4. Optimization of beam gauges

In the present work, the design variables for optimization are the thickness of the beams constrained within a defined lower and upper bound, i.e., 1.5 mm and 3 mm, respectively. Response parameters are the properties based on the design variables, like stress, displacement, frequency and mass, which are used to evaluate the performance of the structure. The objective functions, here, are three in number: to minimize mass, to minimize compliance and to maximize frequency. A few boundary conditions or constraints are used on the response and design variables to make sure that the properties of the structure are within an allowed interval. The present work includes two optimization runs, on the basis of displacement criteria and modal criteria, respectively:

(a) Displacement criteria:

When displacement is considered, a structure is considered stiffer if the force to achieve that displacement is high, thus, minimizing compliance corresponding to maximum stiffness. This means that a lower compliance means a stiffer structure and lower displacement. The constraint is to have the mass of designed beams less than 1.521 kg (as given in Table 2) and the objective is to minimize the compliance.

(b) Modal criteria:

This is a measure of the dynamic behaviour of the model, in which higher frequencies mean better ride for the passengers (to avoid resonance). Since the first two bending and torsion mode frequencies, see Table 3, contribute most to the structure stiffness, they are to be maximized with an objective to minimize the weight. The design variable properties are imported from the results of the previous (displacement) run.

In this case, the constraint is the mass taken from the previous run, i.e., mass less than 1.521 kg. The objectives are to maximize the 7^{th} and 8^{th} frequencies of the free modal analysis. Two iterative runs are consecutively executed and studied to achieve an optimum result.

4. Experimental results

4.1. Result of linear static analysis

During the linear static analysis, the highest stresses observed are those of 279 MPa and 283 MPa for combined bending and combined longitudinal and lateral loadings, respectively. The locations of these highest stresses are shown in Fig. 15.

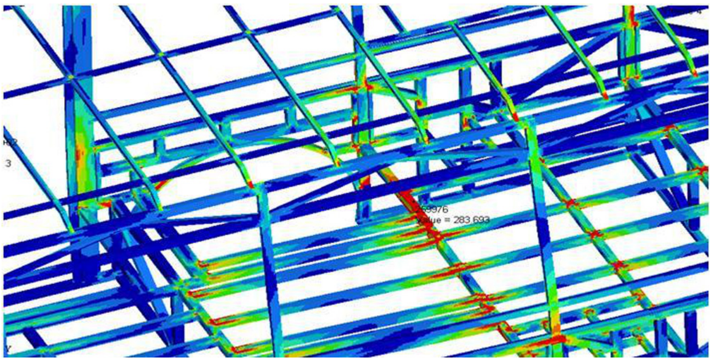
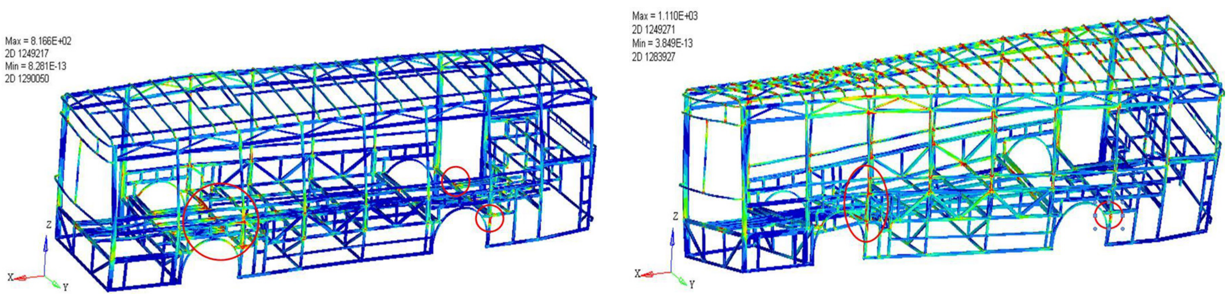


Fig. 15. Beam subjected to high stresses

When evaluating the performance of the base model, emphasis is placed on the magnitude and location of stresses and deformations occurring on the body. As circled in Fig. 16, the largest stresses are localized near the wheels and the floor adjacent to the doors.



a) b)

Fig. 16. a) Stresses for left cornering with braking; b) stresses in the torsion case

4.2. *Result of optimization*

The iterative optimization process results into multiple outcomes because of two different runs. The results in the form of varied thicknesses can be imported from one setup to another so that the optimized results are in good agreement. The beams are categorized on the basis of thickness and symmetry into small sets, denoted by $T_1, T_2, \ldots, T_{12}$. In the optimization mode, runs 1 and 2 are carried out to minimize displacement, while runs 3, 4, and 5 are carried out to maximize frequency, subjected to variation in beam gauges. The beam gauges are presented in Table 4. The analysis of the gauge variations [16, 18] during runs 1–5 led to selection of specifications of beams' thickness as per Indian market standards. On the basis of data selected for the cases A, B and C, the performance characteristics are mapped and summarized in Table 5.

The results of runs are normalized and plotted in Fig. 17. The case C is the optimum result with modified specifications.

Table 5. Parametric values after each optimization run

Beam thickness [mm]	Specifications of the base model	Product specifications as per market availability		
		A	B	C
T_1	2.0	2.0	2.0	1.50
T_2	3.0	2.5	3.0	3.00
T_3	3.0	2.5	2.5	2.50
T_4	2.0	2.0	2.0	2.00
T_5	1.0	1.5	1.5	1.50
T_6	1.5	2.0	1.5	1.75
T_7	2.0	2.0	2.0	2.00
T_8	3.0	3.0	3.0	3.00
T_9	2.0	2.5	1.5	1.50
T_{10}	2.0	1.5	1.5	2.00
T_{11}	1.5	1.5	1.5	1.50
T_{12}	2.0	2.0	2.0	1.50
		Performance characteristics		
Frequency in torsion mode [Hz]	5.950	6.180	5.980	6.050
Worst case displacement during left turning and braking [mm]	22.400	23.400	22.500	21.900
Worst case displacement during combined torsion and bending [mm]	67.300	63.830	67.260	64.800
Mass [tonnes]	1.521	1.488	1.505	1.501

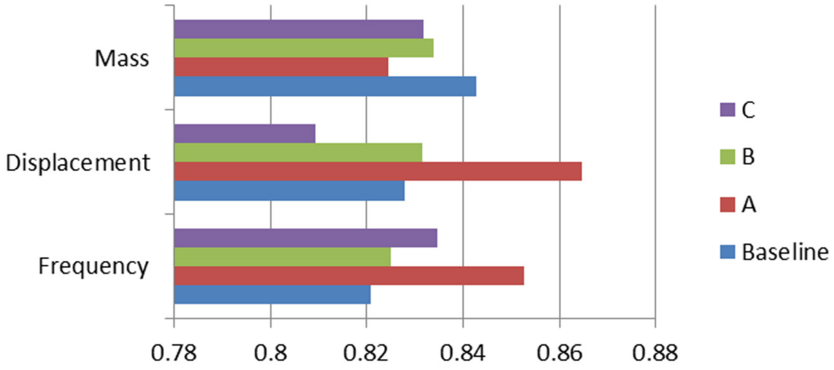


Fig. 17. Analytical chart showing performance comparison of the iterative runs

5. Discussion

5.1. Baseline model analysis

Similar to the stress raising members (e.g., holes, joints etc.), the elements at the two beams junctures of the bus body would have very high stresses, as per Neuber's criteria for elasto-plastic analysis. In the present work, these junctures are not considered due to the inherent complexity. The yield stress of the steel used in the bus structure is 400 MPa and a safety factor of 2.6 is considered so that the permissible stress [1, 5, 17] is less than 150 MPa. Further, the high stresses at specific location can be accounted to the design limitation, i.e., to the unavailability of a suspension system which acts as damper to stresses.

The bus subjected to stresses would also suffer deflection. The deflections are minimized to account for high stiffness of the structure. The maximum deflection varies according to the severity of the loading. It is highest for the torsion case, i.e., for the vertical bump when the bus rises on a hump. When deflections are not continuous and high as circled in Fig. 18, unstable and weak zones are implied. There is no limiting value to be generalized, but the objective is to minimize deflection as much as possible or to maximize stiffness/minimize compliance.

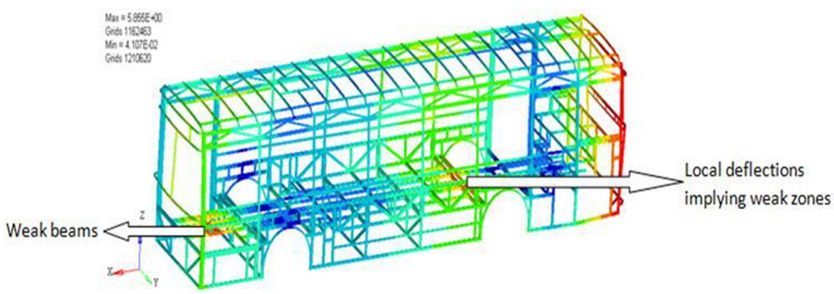


Fig. 18. Deflections under bending loads

The frequencies of the baseline model are above a limiting value of 5 Hz, which is further improved during the course of this work. The mass of the bus structure is also lower than the one reported in literature [3, 5]. Since the objective of this work is to improve the performance parameters along with weight reduction, the optimization is carried out with the minimal possible mass as a constraint.

The stiffness is an important criterion to evaluate the performance of the bus structure. In literature, the stiffness values are reported in the range of 5–15 kN/mm for a bus structure [9]. The stiffness values of baseline model are satisfactory as they lie within this range. The worst case displacement values as given in Table 5 are comparable to the results reported in literature [4, 16] and, hence, justified.

5.2. Optimized model analysis

To validate the new model, a tabulated comparison of the baseline and optimized model is made in Table 6. This table does not include the comparison of maximum stresses induced on beams due to indefinite load cases. The range of maximum stresses for the worst cases, which are turning with braking and combined torsion with bending, has reduced from a range of 250–280 MPa to the range of 200–220 MPa. Besides, for other load cases, the stresses on the beams of the optimized model are within the permissible range of 150 MPa [5]. Stresses on some of the weak beams still exceed the permissible limit and, hence, are identified for remodeling or artificial stiffeners have to be attached at identified locations.

Table 6. Comparison between baseline and optimized model

Parameters	Baseline model	Optimized model	Difference
Structure mass [tonnes]	1.521	1.500	−1.33 %
Torsion frequency [Hz]	5.950	6.050	+1.66 %
Bending frequency [Hz]	8.780	8.900	+1.43 %
Torsion stiffness [kNm/deg]	17.920	18.230	+1.7 %
Bending stiffness [kN/mm]	13.110	12.940	−1.20 %
Displacement [mm] (left turn and braking case)	22.400	21.900	−2.20 %
Displacement [mm] (vertical bump case)	67.000	64.880	−3.56 %

6. Conclusion

The gauge optimization on the bus structure is an iterative process and the mass of the final optimized model is by 1.33 % less than the baseline. The first torsion frequency has increased by 1.66 % to 6.05 Hz and the first bending frequency by 1.43 % to 8.906 Hz. The torsion stiffness is now by 1.7 % higher than that of the baseline model, whereas the bending frequency is almost the same for the two models. The deformations have moderately come down by 2.2 % for the combined turning with braking case, while it shows a decline of 3.56 % for the torsion case. Thus, one can conclude that by following the proposed algorithm, the bus body may be optimized along with minor decrease in weight by optimizing beam gauges. This has led to significant improvement of performance characteristics. The work is verified by conducting detailed analysis on a full length 11 m, 65 seats bus and helps in reducing structural mass by 20 kg along with enhanced stiffness, vibration and stress performances.

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