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Optimization of flatbed trailer frame using the ground beam structure approach †

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Abstract

An alternative lightweight flatbed trailer design is achieved through a multi-stage optimization procedure. Topology optimization is used to obtain the optimal layout of flatbed trailer frame beams that provide minimum compliance when subjected to bending loads and exhibits maximum torsional natural frequency. The ground structure approach is used to define the trailer frame layout by generating numerous beams connected to predefined points in the trailer. Topology optimization is formulated as a multi-objective problem subject to a mass constraint. Responses and sensitivities are evaluated using ANSYS, and the optimization problem is solved using the moving asymptotes method. The thicknesses, widths, and heights of the C-channel beams are optimized for further weight reduction while at least maintaining the structural performances of the original design. Size and shape optimizations are performed using OptiStruct. The new optimal design is approximately 13% (275 kg) lighter than and as stiff as the original design for bending loads. However, the former has 3.5 times higher torsional natural frequency than the latter. Moreover, the new optimal design has positive manufacturability because the channel beams will be made out of commercially available sheet metals. The same fabrication technology as for a conventional flatbed trailer is possibly to be used.

Keywords: Ground structure approach; Lightweight trailer; Manufacturability; Topology optimization

1. Introduction

Increasing concerns on the reduction of fossil fuel consumption and CO_2 emissions have prompted the automotive industry to develop innovations for lightweight vehicle designs. On the one hand, potential structural components can be optimally designed using lightweight or composite materials to replace conventional steel to reduce the overall vehicle weight [1-6]. On the other hand, this material replacement approach is receiving increased attention within the automotive industry, although economic issues remain a major challenge [7-9].

Lightweight vehicle design is achievable through the implementation of design optimization procedures. Topology, shape, and size optimizations have been extensively used to attain lightweight vehicle component designs, as well as enable increased stiffness and improved dynamic performance. Lee et al. [10] combined topology, size, and shape optimization techniques to determine the optimal material layout, geometry, and physical dimensions of vehicle components to reduce structural weight based on static and fatigue requirements. Jang et al. [11] presented a box-like trailer frame using topology optimization. Thickness optimization is also beneficial for reducing vehicle weight because many components are made of sheet materials. Zhang et al. [12] used the response surface method to optimize the structural sheet thicknesses of an automotive passenger car, thereby resulting in weight reduction without losing the crash energy-absorbing performance. Pan et al. [13] used a similar approach to design a lightweight B-pillar of a vehicle under roof crush and side impact requirements. Xiao et al. [14] presented a multi-objective topology optimization method to simultaneously maximize the static and dynamic performances of a steel wheel.

Commercial optimization software programs as stand-alone or add-on modules to the CAE simulation packages are available to facilitate the treatment of considerably large-scale models [15, 16]. Jang et al. [11] presented a lightweight flatbed trailer design with high stiffness using a two-step optimization implemented with the aid of commercial optimization software. Topology optimization was employed during the first step to obtain an optimum trailer frame layout different from conventional ladder-type frames. Given that the design domain was modeled as a three-dimensional continuum, the resulting optimal topology was dominated by solid continuous regions, which should be interpreted as plate members in the post-process. The plates were modeled with shell elements,

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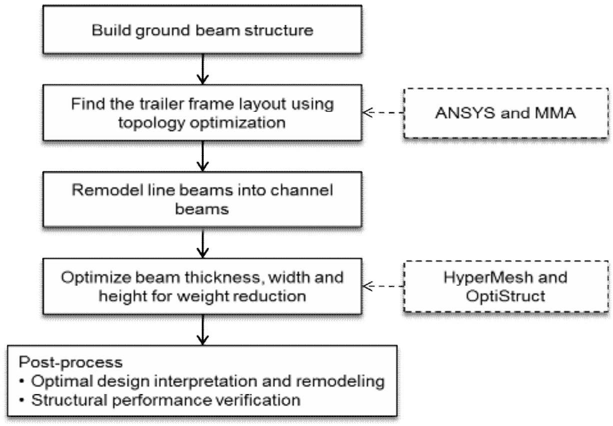


Fig. 1. Design optimization procedure.

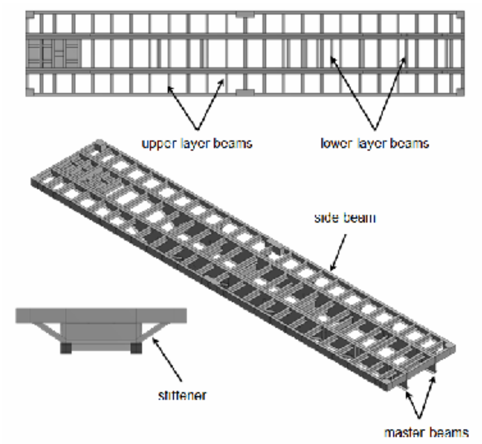


Fig. 2. Original design of the flatbed trailer frame.

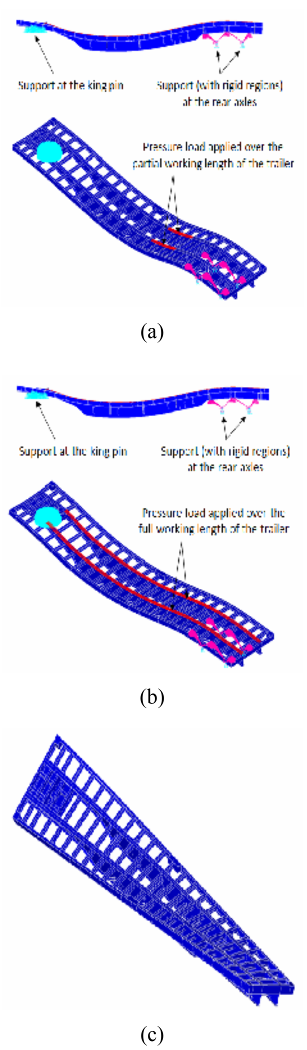


Fig. 3. Deformation under (a) load case 1; (b) load case 2; (c) torsional mode shape of the initial flatbed trailer frame design.

and thickness optimization was performed for weight reduction as the second step. The final optimization model had a 29% reduced mass with the increased static and dynamic stiffness than the original model. However, the resulting final design revealed an entirely new design concept, that is, a trailer frame with plates instead of cross beams and stiffener beams with distinct cross-sectional shapes. Thus, manufacturability is an issue because it would require additional cost for setting up the necessary fabrication and assembly infrastructure.

This study proposes an alternative multi-stage optimization approach for the design of a flatbed trailer frame structure. A trailer frame design made of channel beams can be achieved by defining the design domain with several candidate beams. The candidate beams can be constructed similar to the ground structure method (see Hagishita and Ohsaki [17] and the references therein). Topology optimization can be performed to determine the best beam layout that meets the desired structural performances under the given loading conditions and mass restriction. Once the optimal beam layout is determined, the beams can be remodeled with shell elements. Thereafter, size and shape optimizations can be performed to reduce the trailer's weight further. Consequently, the final optimal flatbed trailer frame design, which generally comprise beams with predefined cross-sectional shapes, can be achieved. Existing equipment and technology used to fabricate and assemble conventional ladder-type models can be employed to manufacture the new trailer to obtain economic benefit. The proposed trailer frame design optimization procedure is illustrated in Fig. 1.

2. Topology optimization using the ground structure approach

2.1 Original model

A flatbed semi-trailer, that is, a trailer with no front axle, is considered in this study. The original flatbed trailer frame model is shown in Fig. 2. The trailer has two layers of horizontally laid C-channel-type cross beams (20 in the upper layer and 10 in the lower layer) and 12 stiffeners subjected to optimization. The upper layer beams are connected to the master and side beams, whereas the lower layer beams are merely connected to the master beams. The two relatively large master beams located at the center and extended throughout the length of the entire trailer mainly support the bending loads imposed by the cargo. The stiffeners or reinforcement beams are connected slantwise to the side beams and near the lower flange of the master beams. Evidently, the two-layer cross beams and the slanting stiffeners of the trailer are configured to increase their torsional stiffness. The trailer is 12.2 m long, 2.55 m wide, and weighs 2164 kg. Further, this trailer is coupled to a tractor through the king pin. The allowable maximum payload of the trailer is 24 tons [18].

The original trailer was designed to operate under two possible static loading conditions. First, a 24-ton load is placed over the partial length of the trailer near the rear wheel axles. Second, the same amount of load is placed uniformly over the identified full working length of the trailer. In either condition, equally placing the loads immediately above the master beams is constantly desired to effectively support the resulting bending action. For structural optimization, structural performance defined by the strain energies resulting from the aforementioned loads and natural frequency of the first natural torsional mode are evaluated using Finite element analysis (FEA). Thus, three-structural analysis scenarios are considered. Fixed support is assumed at the coupling and the rear wheel axles for the two bending load cases, whereas unsupported condition is imposed on the same regions for the natural mode analysis. The strain energies for the first and second bending load cases are 5.364×10^{6} N-mm and 4.239×10^{6} N-mm, respectively, whereas the torsional natural frequency is 2.35 Hz. The corresponding deformations for the two bending load cases and first torsional mode shape are shown in Fig. 3.

2.2 Topology optimization formulation

Topology optimization is a method to determine the best layout of a material's limited amount over the given structure's design domain subjected to certain loading and support conditions. In this case, topology optimization is used to determine optimal location and orientation (arrangement) of the C-channel beams that constitute the design domain. The design procedure shown in Fig. 1 begins with the search for an optimal arrangement of the beams that makes the trailer as stiff as possible when subjected to the two bending load scenarios and exhibits the highest torsional natural frequency. These structural performance requirements must be fulfilled within a limited material resource. In particular, the overall weight of the trailer must not exceed that of the original design.

The design domain for topology optimization is constructed by laying out numerous C-channel beams at various locations and orientations in the region of interest in the trailer. Note that a few of these beams are coincident with the position of the original trailer beam layout. At this point, the C-channel beams belonging to the same group have the same cross-sectional dimension. For example, all the beams in the upper layer group have the same cross-section, which is the same for

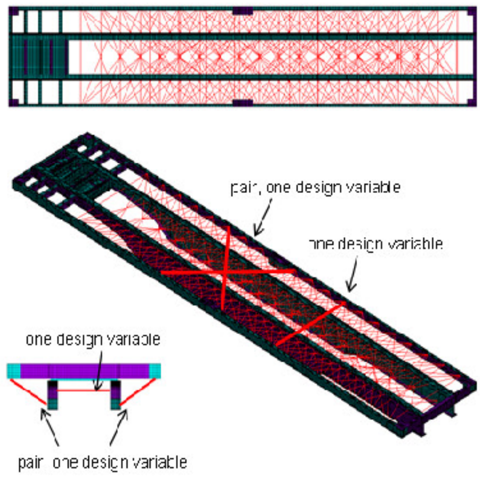


Fig. 4. Ground structure configuration of the flatbed trailer frame as a design domain for topology optimization.

the beams in the lower layer and stiffeners. This method of constructing the design domain or baseline design for topology optimization is referred to as the ground beam structure approach. The completed ground beam structure shown in Fig. 4(a) has a total of 184 C-channel beams connected to the master beams or side beams of the trailer. To ensure a symmetrical optimal trailer frame layout, one design variable is associated with one cross beam laid out normally to the master beams, a pair of diagonal cross beams, and a pair of slantwise stiffeners, as illustrated in Fig. 4(b). Finally, 108 design variables are employed for topology optimization of the ground beam structure.

The topology optimization problem is formulated as follows:

$$\underset{\boldsymbol{\rho}\in\mathbb{R}^N}{\text{Minimize}}\, F = w_1\left(\frac{U_1}{U_1^0}+\frac{U_2}{U_2^0}\right) - w_2\frac{f_t}{f_t^0} \tag{1a}$$

$$\text{subject to } G = \sum_{i=1}^{N}\rho_i v_i - M_0, \quad \varepsilon \le \rho \le 1 \tag{1b}$$

where U_1 and U_2 are the strain energies that correspond to the bending stiffness of the trailer for load case 1 and load case 2, respectively; and f_t is the torsional frequency. The superscript 0 denotes values at iteration 0. Weighting factors w_1 and w_2 are introduced to adjust the contributions of the static and dynamic performances, respectively, of the structure in the objective function. The design variable ρ_i is associated with a beam or a pair of beams with volume v_i. The mass of a beam or pair of beams is $m_i = \rho_i v_i$. During optimization, the overall mass of the trailer should not exceed the mass of the initial design M_0. In Eq. (1), $\varepsilon = 0.001$ is used.

Low energy modes are commonly known to suffer from localized modes unless particular focus is given to the penalization of the SIMP method [19, 20]. In this case, this problem occurs because the fundamental frequencies of the beams with

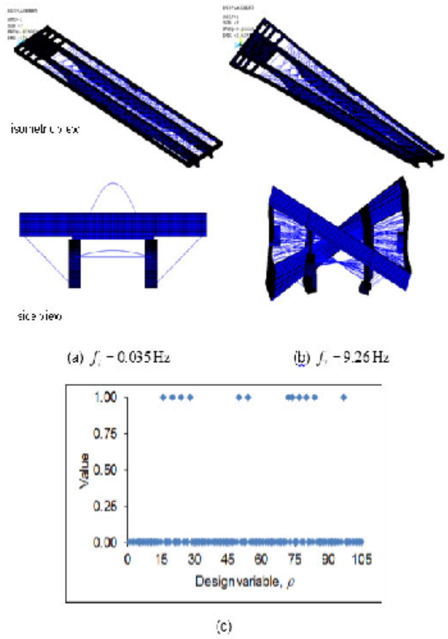


Fig. 5. Fundamental mode shape (a) using the classical SIMP approach where the local mode exists (s = 1 and p = 3 in Eq. (2)); (b) using a considerably high penalty on mass (s = 6 and p = 3 in Eq. (2); local mode eliminated); (c) the given design variables.

minimum or near the minimum densities are lower than those of the beams with considerably high or full densities. Therefore, the beams with low densities dominate the low energy modes of the entire trailer. Several approaches for avoiding such problem in topology optimization are in the Ref. [19]. In the current study, localized modes are avoided by assigning higher penalty to the mass than that for the stiffness interpolation.

$$\gamma_i = \rho_i^s \gamma_0 \ , \quad E_i = \rho_i^p E_0 \ , \quad (s > p) \ , \tag{2}$$

where γ_i and E_i are the physical density and Young's modulus of beam *i*, respectively; and γ_0 and E_0 denote those for a given material. In this case, s = 6 and p = 3 are used. Using a higher penalty to the mass than that for the stiffness interpolation, the mass of the void elements become considerably light to prevent their dominating effect on the dynamic behavior of the structure (i.e., localized modes) from the low frequency region. In Pederson (2000), using a substantially high penalty to the mass is not recommended for intermediate densities. However, noting in Eq. (1b) how the actual mass is calculated, Eq. (2) can be considered merely a relaxation to make the ratio of mass to stiffness finite in the limit of a vanishing design variable for modal analysis. Fig. 5 illustrates the elimination of local mode using the proposed mass interpolation in the first torsional mode shape of the trailer for the given distribution of design variables.

To ensure the extraction of the target mode during optimization, the Modal assurance criterion (MAC) is also implemented. In Kim and Kim [21], MAC is defined as follows:

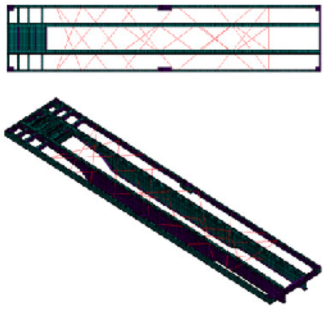
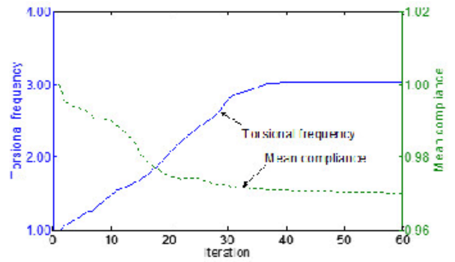
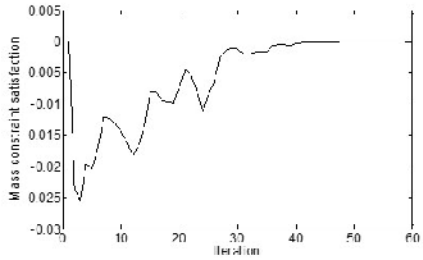


Fig. 6. Optimum topology of the trailer frame beam structure.



(a)



(b)

Fig. 7. Optimization histories of (a) the objective; (b) mass constraints.

$$MAC = \frac{\left|\Phi_a^{\mathrm{T}}\Phi_b\right|^2}{\left(\Phi_a^{\mathrm{T}}\Phi_a\right)\left(\Phi_b^{\mathrm{T}}\Phi_b\right)} \tag{3}$$

where Φ_a and Φ_b are two mode vectors of interest. Note that the MAC value considerably close to 1 means significantly high resemblance between the two modes. The target mode is selected as the mode having the highest MAC value with the torsional mode shape.

FEA and the calculations of sensitivities and MAC are performed in the ANSYS environment using the ANSYS parametric design language. In ANSYS (or any other FEA packages for this matter), the sensitivity of the objective in Eq. (1) is conveniently evaluated by expressing it in terms of the following strain and kinetic energies:

$$\frac{\partial F}{\partial \rho_i} = w_1\left(\frac{1}{U_1^0}\cdot\frac{\partial U_1}{\partial \rho_i} + \frac{1}{U_2^0}\cdot\frac{\partial U_2}{\partial \rho_i}\right) - \frac{w_2}{f_t^0}\cdot\frac{\partial f_t}{\partial \rho_i} \tag{4a}$$

where

Table 1. Structural performances of the topology-optimized trailer frame.

	Mass (kg)	Strain energy (N-mm)		Torsional frequency (Hz)
		Case 1	Case 2	
Original design	2164	1.430e6	1.130e6	2.35
Topology optimization	2097	1.409e6	1.118e6	7.42

$$\frac{\partial U_j}{\partial \rho_i} = -\frac{p}{\rho_i} \cdot \hat{U}_i^j \;\; ; \;\; j = 1, 2 \tag{4b}$$

$$\frac{\partial f_t}{\partial \rho_i} = \frac{1}{8\pi^2 f_t \rho_i} \left(p\hat{U}_i - s\hat{T}_i \right). \tag{4c}$$

In Eq. (4b), $\hat{U}_i^j$ is the strain energy of elements by load case *j*, which are associated with the design variable *i*. $\hat{U}_i$ and $\hat{T}_i$ in Eq. (4c) are the element strain energy and element kinetic energy, respectively, of the target mode associated with the design variable *i*. Deriving sensitivities is relatively standard and included in many published works (see Huang et al. [22] and Kim and Kim [21] and the references therein). The optimization problem is solved using the moving asymptotes (MMA) method [23].

2.3 Topology optimized layout

The proposed optimization employs a multi-objective formulation, such that the results can be changed depending on the weight parameters in Eq. (1). Optimization results for all the cases of weight parameters represent the so-called Pareto front solutions for the multi-objective problem. In this study, instead of investigating all the possible solutions, the optimization with more weight on the bending stiffness than the torsional stiffness is presented because the bending stiffness is the performance of the first priority for a load-carrying vehicle. Fig. 6 shows the optimal topology of the trailer frame beam structure using $w_1 = 0.75$ and $w_2 = 0.25$ in Eq. (1). During the topology optimization run, a few design variables did not completely converge to 0-1 solution. Hence, a threshold value was selected, such that the mass constraint was not violated when the design variables with intermediate values were converted to 1 (solid material) or 0 (void material). Fig. 7 shows the optimization histories for the objective and mass constraints. Table 1 shows the structural performances of the optimal design.

In Fig. 6, note that using diagonally positioned beams may increase the bending stiffness of the trailer frame. However, Table 1 shows that the bending stiffness as indicated by the strain energy values of the optimal trailer design are noticeably increased by only a small amount relative to the initial design. This increase is caused by the dominant system bending stiffness originating from the master beams excluded in the design domain for topology optimization. This increase can also explain why the stiffeners connecting the upper and lower beams did not appear in the optimal topology. As an extreme case, if the added beams are generally positioned parallel to the master beams (i.e., along the length direction of the trailer), then the bending stiffness of the trailer can be substantially enhanced. However, the torsional stiffness of the structure in this case would be lost and the mass would be increased. Compromising between the bending and torsional stiffness using the multi-objective form, the proposed optimized frame beam layout can provide considerably high torsional stiffness (substantially high torsional frequency) and significantly high bending stiffness with the same amount of mass usage as in the original model.

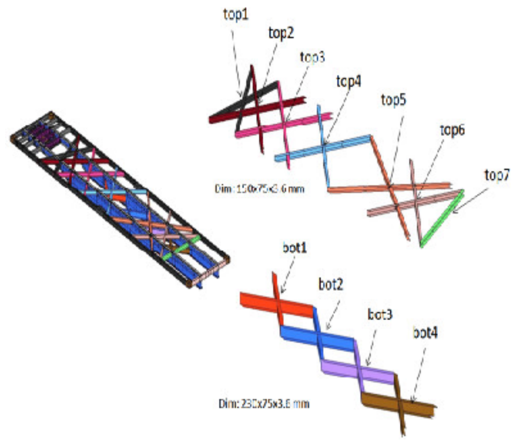


Fig. 8. Design domain configuration for size and shape optimizations.

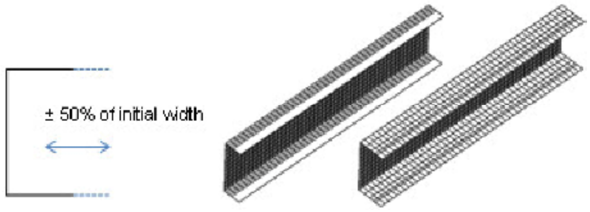
3. Size and shape optimization

3.1 Problem formulation

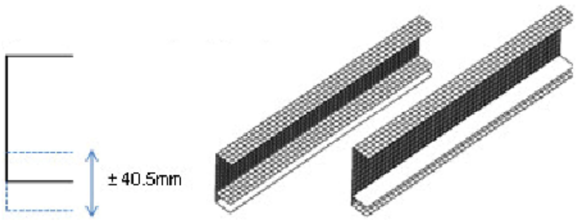
After determining the optimal beam layout, the finite element model of the optimized trailer is modified by rediscretizing the C-channel beams with shell elements replacing the beam elements. These modified trailer frame beams become the design domain for size and shape optimization, as shown in Fig. 8. The thicknesses, widths, and heights of the channel beams are optimized to reduce the trailer mass further. Simultaneously minimizing the mass constrains optimization, such that the stiffnesses of the original trailer are retained at the least. OptiStruct from Altair HyperWorks [24] are used for this optimization method.

In size optimization, the thicknesses of the plates for the C-channel beams are used as the design variables. Discrete design variables are particularly used for plate thicknesses to account for manufacturability. The discrete values are based on the thicknesses of commercially available high-strength steel, namely, ATOS 80 from POSCO [11]; 27 discrete thicknesses ranging from 3.1 mm to 14 mm are used in the optimization.

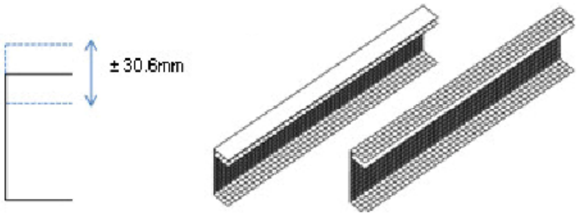
For shape optimization, the changes in beam widths and



(a) Channel beam width



(b) Channel beam height-upper layer

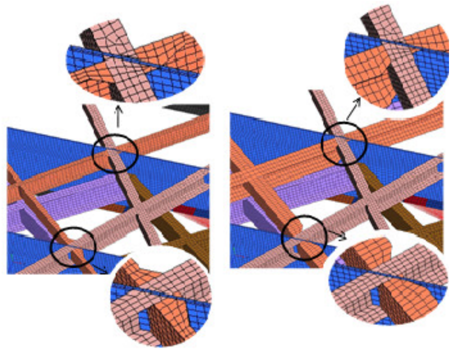


(c) Lower layer beams-lower layer

Fig. 9. Beam shape changes and corresponding morphed meshes.

heights are defined using the mesh morphing technique. After dividing a mesh model into regions or domains, the handles at the corners of these domains are used to change the model shape by moving them or changing their locations. The change in nodal locations defining a certain shape is retained in a one-dimensional array and parameterized with an adjustable parameter, that is, a design variable. Enabling the value of the design variable associated with a pre-defined shape to change during optimization can determine the best shape. One of the major advantages of mesh morphing is that model shapes can be changed without the need for remesh. In HyperWorks, the mesh morphing technique is available through the HyperMorph tool.

Consequently, the manner by which the shapes should change or the mesh should move is defined prior to the execution of shape optimization. Fig. 9 illustrates the manner by which beam shapes can be modified during shape optimization. Given that the beam widths can significantly increase or decrease (i.e., up to 50% of the initial width), the heights are enabled to change by $\pm 17.6\%$ of the initial height for upper layer beams and $\pm 20.4\%$ for the lower layer beams. These limits for geometric changes were determined based on the available spaces in the trailer that any adjacent beams can occupy without interfering or overlapping each other during shape or dimensional changes. Instead of using the shell elements and morphing technique, one may retain the use of and change the cross-sectional properties of the beam elements,



(a) (b)

Fig. 10. Simplified shape change at the following intersections: (a) Perturbation with −50% of the width; (b) perturbation with +50%.

such as bending moments of inertia and torsional moments of inertia for beam section optimization. However, this method cannot predict higher order cross-sectional deformations, such as warping, which may result from an overestimation of structural stiffness [25].

Given that the ground beam structure of the trailer frame was created using numerous candidate beams with varying lengths and orientations, the topology optimization result has resulted in a few complicated geometric regions with possibly no well-defined morphing for shape optimization. For example, even though the structural shape is changed during optimization, the beam width should be uniform throughout its entire length. However, this requirement is technically difficult in areas where one beam intersects with one or more beams. Dealing with this problem requires modifications on the morphed mesh at intersections. Fig. 10 illustrates an example of shape (width) change modification. In this case, the stiffness of the trailer will be less affected by the shape modification even though the stress levels around the modified regions may be inaccurately predicted.

The size and shape optimization problems can be simply stated as follows:

$$\begin{aligned} &\underset{\boldsymbol{\rho}\in\mathbb{R}^{N_S}}{\text{Minimize}}\, F = \text{mass} \\ &\text{subject to } G_1 = U_1 \le U_1^0 \\ &\qquad\qquad G_2 = U_2 \le U_2^0 \\ &\qquad\qquad G_3 = f_t \ge f_t^0 \end{aligned} , \tag{5}$$

where U_1^0 and U_2^0 are the strain energies of the original trailer in Fig. 2 for the first and second bending load cases, respectively; and f_t^0 is the torsional frequency of the original trailer.

3.2 Size and shape optimization results

Table 2 shows the optimal beam thicknesses, widths, and heights. The optimization history is shown in Fig. 11. Table 3 shows the corresponding structural performances of the size

Table 2. Optimal dimensions of beams resulting from the size and shape optimizations.

Beam	Dimension (mm) (H x W x t)	
	Initial	Optimal
Bot1	230 x 75 x 3.6	199.4 x 37.5 x 3.1
Bot2		
Bot3		
Bot4		
Top1	150 x 75 x 3.6	109.5 x 43.129 x 3.1
Top2		109.5 x 37.5 x 3.1
Top3		109.5 x 42.99 x 3.1
Top4		109.5 x 37.5 x 3.1
Top5		
Top6		109.5 x 43.012 x 3.1
Top7		109.5 x 43.346 x 3.1

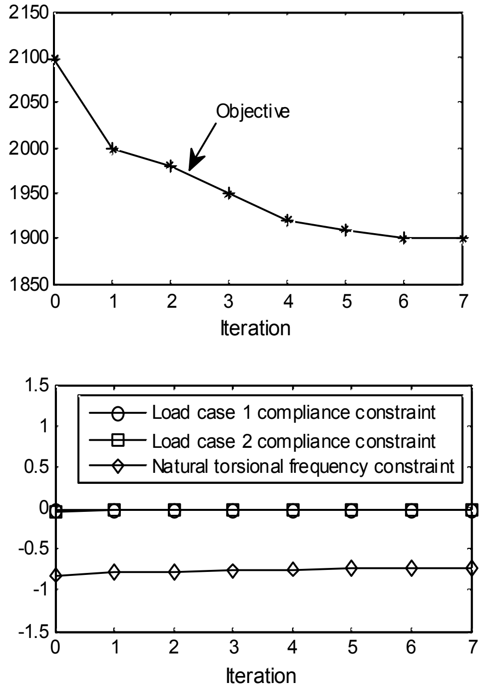


Fig. 11. Optimization histories for the size and shape optimizations.

and shape optimal design. The trailer weight is reduced significantly, whereas the bending stiffness of the optimized structure is slightly increased from that of the original model. The torsional natural frequency is also increased.

3.3 Post-process

The shape change modification at intersections shown in Fig. 10 creates non-uniform beam widths. Beams in the final design are post-processed to have uniform widths throughout their lengths. Unlike the beam thicknesses, which are specified to take discrete commercially available values, the design

Table 3. Structural performances of the trailer after the size and shape optimization.

	Mass [kg]	Strain energy [N-mm]		Torsional frequency [Hz]
		Case 1	Case 2	
Original design	2164	1.430e6	1.130e6	2.35
Topology optimization	1897	1.416e6	1.111e6	8.52

Table 4. Beam dimensions of the final trailer frame after post-process.

Beam	Dimension (mm) (H x W x t)		
	Initial	Optimal	Post-process
Bot1	230 x 75 x 3.6	199.4 x 37.5 x 3.1	199.4 x 37.5 x 3.1
Bot2			
Bot3			
Bot4			
Top1	150 x 75 x 3.6	109.5 x 43.129 x 3.1	109.5 x 43.1 x 3.1
Top2		109.5 x 37.5 x 3.1	109.5 x 37.5 x 3.1
Top3		109.5 x 42.99 x 3.1	109.5 x 43.1 x 3.1
Top4		109.5 x 37.5 x 3.1	109.5 x 37.5 x 3.1
Top5			
Top6		109.5 x 43.012 x 3.1	109.5 x 43.1 x 3.1
Top7		109.5 x 43.346 x 3.1	

variables associated with the widths and heights of beams are continuous between −1 (full reduction) and +1 (full enlargement). To improve manufacturability, the widths of beams are rounded off, such that the number of dimensional variant is reduced. Table 4 shows the dimensions of beams of the final design.

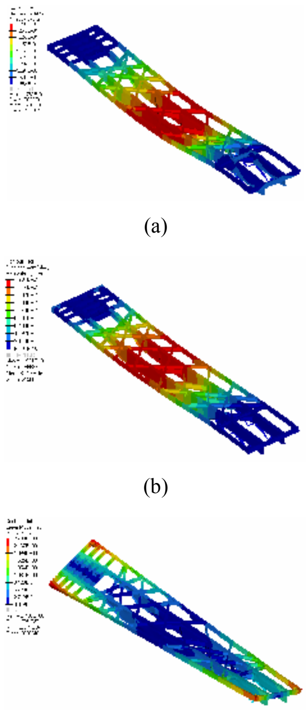
The post-processed trailer frame design is analyzed to evaluate its final structural performances. Fig. 12 illustrates the deformation of the new trailer under bending loads and the torsional mode shape. Table 5 presents the summary of the proposed optimization results. The final design is as stiff as the initial design for bending loads but 12.7% lighter and with 258% higher torsional frequency than the initial design. Although stresses are not constrained during shape and size optimization, the maximum stresses for the final design are lower than the initial design (i.e., 196.8 MPa and 187.5 MPa compared with 294.7 MPa and 462.4 MPa for the first and second load cases, respectively). For stress calculations, the converged results are obtained using fine mesh discretization around regions with high stress.

Compared with the other lightweight trailer design introduced by Jang et al. [11], which was obtained using the continuum-based topology optimization, this new trailer frame structure's major advantage lies in its capacity to generally utilize the existing manufacturing process. Accordingly, plates are cut and folded into beams with the specified sizes and assembled in a similar manner as in the original trailer.

Given the intersecting members of the new design, the con-

Table 5. Summary of the design optimization results.

	Mass (kg)	Strain energy (N-mm)		Freq. (Hz)	Max. stress (MPa)	
		Case 1	Case 2		Case 1	Case 2
Original design	2164	1.430e6	1.130e6	2.35	294.7	462.4
Topology optimization	2097	1.409e6	1.118e6	7.42	238.1	187.2
Size and shape optimization of the topology optimized design	1897	1.416e6	1.111e6	8.52	260.0	186.9
Post-processed into manufacturable design	1889	1.403e6	1.108e6	8.42	196.8	187.5



(a)

(b)

(c)

Fig. 12. Final flatbed trailer frame design: deformed shape for (a) load case 1; (b) load case 2; (c) torsional mode shape.

nection components and bolt-fastening process for intersections may be required as one additional process. By contrast, for the optimized trailer by Jang et al. [11], the manufacturing process should be significantly changed because the top and bottom flanges of the master beams are covered with large plates instead of using cross beams. Thus, the new trailer design using the proposed ground structure approach is economical and highly manufacturable in terms of the usability of existing fabrication equipment and technology.

4. Conclusions

A new lightweight flatbed trailer frame design was introduced using a multi-stage design optimization procedure that utilized CAE software packages. The ground beam structure-based topology optimization method was used to determine the optimal beam layout for the frame structure. Subsequently, the thicknesses, widths, and heights of the channel beams were optimized through size and shape optimization. The post-processing of the optimization result leads to a final design that was substantially lighter and with slightly higher bending stiffness than the original design. The increase in dynamic performance indicated by the torsional natural frequency was significant. Although the new design may require extensive effort in the assembly because of the intersecting beam configuration, this design is highly economical and manufacturable because the same equipment and technology used for production of the conventional design can be used. In case the cross-sectional shape of the ground beams is optimized simultaneously while conducting the topology optimization, an improved performance of the optimized trailer is possible. Although the formulation of the simultaneous topology, shape, and size optimizations may pose a challenge, such process significantly increases the design space. Therefore, further study on this area is worth conducting.

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